Conf-790631
Volume 1
(of 2 Volumes)

PROCEEDINGS

of the

Ocean Thermal Energy Conversion

6th OTEC

CONFERENCE

Ocean Thermal Energy for the 80's

WASHINGTON, D. C. June 19 - 22, 1979

Sponsored by
Ocean Systems Branch
Division of Central Solar Technology
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Gordon L. Dugger, Editor

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Ocean Systems Branch
Division of Central Solar Technology
U.S. Department of Energy

Organized by
The Johns Hopkins University
Applied Physics Laboratory
Laurel, Md. 20810

FOREWORD

The purpose of the 6th Ocean Thermal Energy Conversion (OTEC) Conference, like that of its predecessors, was to provide opportunities for information exchange and for discussion of the national OTEC program. The latest research and development work, test results, component design information, system integration efforts, and application investigations were presented. Not only the technical aspects but also the commercial, institutional, legal, environmental, and international aspects of OTEC implementation were discussed.

The reports of six working groups are featured at the beginning of Volume I of these Proceedings. Each working group held intensive discussions of a major subject area of the OTEC program. Every attendee was invited to participate in the working group of his or her choice. The reports, prepared by the executive committees of the working groups, address the technical and/or institutional status of the program; the critical-path activities needed to prepare for and achieve pilot plant design, construction and operation at early dates with acceptable risk levels; the pilot plant missions; and further work and actions needed to bring about cost-competitive, commercial OTEC plants in a timely manner. On the whole, the reports are quite positive in tone relative to readiness for the pilot plant phase.

The interesting presentations and remarks by the speakers at the plenary session, the luncheon, and the banquet follow the working group reports in Volume I.

Preprints of the majority of the technical papers were available in two volumes at the Conference. Some of the papers were represented by abstracts; none had been reviewed. Many of the authors included more current material in their oral presentations. The preprinted papers were then reviewed (but less formally than for a journal) for acceptability and clarity for use in these Proceedings. Although a few papers were declined, and two authors defaulted, these Proceedings contain, in final form, 95% of the papers that were presented. At the Conference we attempted to tape the discussions of the papers, and, where possible, the discussion follows each paper. The papers are arranged by subject matter into two volumes as shown in the Table of Contents.

The list of the 585 attendees with their addresses and, in most cases, their telephone numbers, is at the end of Volume II. The total turnout for the banquet was 600.

Two important aspects of this Conference were that 27 participating organizations aided us in organizing and publicizing the Conference, and there were 21 exhibitors, listed on page iv. The exhibits added greatly to the value of the meeting for the attendees, the press, and the general public. On the evening of the banquet, a reception was hosted by Rums of Puerto Rico, enjoyed by all.

Bob Cohen was the Department of Energy contact as sponsor for the Conference. He and Bill Richards, Chief of the Ocean Systems Branch, and the other program managers in the Branch--Sig Gronich, Bill Sherwood, Lloyd Lewis, Ken Read, and Gene Kinelski--aided in the session planning. The conference committees and session chairmen are listed on page v. The Steering Committee made the planning decisions about the content and makeup of the Conference. The Papers Committee reviewed the 170 abstracts that were submitted and accepted 125 papers for presentation at the Conference. The Session Chairmen and Co-Chairmen did fine jobs in running the sessions, and they and many of the authors joined the listed Papers Committee in reviewing the preprints for use in these Proceedings. The Exhibits and Arrangements Committee from APL/JHU headed by Bud Francis did an excellent job and were supported by many others at APL. Bob Kroll of APL handled the editorial services for both the preprints and these Proceedings, with support from many others. To all of these stalwarts I say thank you for jobs well done!

My sincere thanks for all-around help with the Conference also go to Ed MacCutcheon, Consultant. He appeared, and worked behind the scenes, in many roles and is now helping Bob Scott of Gibbs & Cox, Inc., to organize the 7th Ocean Energy Conference, which will be held at the Shoreham Hotel in Washington, D.C., June 2-5, 1980. We wish them every success.

Gordon L. Dugger
Chairman
6th OTEC Conference

6th OTEC CONFERENCE
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1. WORKING GROUP REPORTS

THE APPROACH THAT WAS SUGGESTED FOR WORKSHOP SESSIONS AND WORKING GROUP REPORTS

FOREWORD

The overall chairman for the workshop sessions and the final plenary sessions for working group reports at the 6th OTEC Conference was Frederick E. Naef of Lockheed, Washington, D.C. The Executive Committee of each working group was asked to meet on Tuesday evening, June 19, to formulate their strategy for conducting the workshop sessions on June 21 and 22 and for preparing their reports. Each chairman was given the "straw man" guidelines presented below.

SUGGESTED APPROACH

The theme of the Sixth OTEC Conference is "Ocean Thermal Energy for the '80's," and one purpose of the Conference is to establish whether OTEC plants can be built and operated in the 80's with state-of-the-art technology. It is suggested that each working group address this theme in its workshop sessions and attempt to respond with a crisp, definitive report. In this regard, we have posed three premises and two hypotheses, which are stated below with a series of questions related to each hypothesis, that may guide the working group toward responses to the Conference theme. Discussion of additional subjects during the workshop sessions is encouraged and is the prerogative of each working group chairman.

The issue is not whether there are knowns or unknowns, but whether the risk in each subsystem or area is acceptable. If further effort is needed to reduce risk to an acceptable level in a given area, your task is to recommend the steps and critical-path activities that are needed, in your judgment, to accomplish an adequate degree of risk reduction in a timely manner with a minimum allocation of resources. This is not an easy task, because most technical managers would prefer to allocate resources at each level of development until there are no risks whatsoever in their areas of responsibility.

Remember that the report of your working group may influence the pace and direction of OTEC development. Your report should feature your group's conclusions and recommendations on the course of the OTEC program. Please consider each of the questions posed under the hypotheses, and if the question is relevant to your group's area of specialization, answer it in the context of the status and needs in your area.

PREMISES

1. The scope of the OTEC system is characterized by the OTEC Work Breakdown Structure, dated May 1, 1979, as prepared by the Value Engineering Company.

2. The marginal cost of new incremental fossil generating capacity, available for synchronization with the island utility grids in the late 1980s, will be 70 mills/kWh (1980 dollars).

3. The market value (purchase price) of ammonia (FOB New Orleans) in the late 1980s, will be $280/ton (1980 dollars).

HYPOTHESIS A AND RELATED QUESTIONS

Hypothesis A

If adequate resources are allocated, a 10/40 MW_e OTEC pilot plant will be in operation at an early date.

Related Questions

1. What is the adequacy of the technology for the design, construction, and operation of a pilot plant? That is, are there any technical elements that consitutute undue risk for a governmentfunded demonstration?

2. If so, what must be done to make the risk acceptable? That is,

   a. What critical-path activities must be completed prior to the initiation of design of a pilot plant?

   b. What critical-path activities must be completed prior to the initiation of construction?

   c. What is the earliest operational date that can be achieved?

3. In the context of your working group, what is the appropriate mission of a pilot plant?

4. What changes in technical emphasis in the OTEC program do you recommend for consideration by Program Management?

HYPOTHESIS B AND RELATED QUESTIONS

Hypothesis B

With an appropriate engineering development program, OTEC plants will become commercially competitive in the late 1980s.

Related Questions

1. What engineering developments must be achieved in order to make OTEC plants cost competitive by the late 1980's?

2. Do you feel that hypothesis "B" can be achieved?

3. In what market sector?

4. What changes in technical emphasis in the OTEC program do you recommend for consideration by program management?

5. What is the most likely business structure (production, ownership, finance, government, etc.) for these first commercial plants?

6. What legislative action is required to achieve hypothesis "B"?

7. How and in what stages should the social and institutional impacts of commercial OTEC plants be examined?

REPORT OF WORKING GROUP 1: OCEAN ENGINEERING
Platform, Cold-Water Pipes, and Undersea Cables

Executive Committee		Participants	
Robert H. Douglass, Jr., Chairman	R. Abate	Larry Green	R. J. Lewandowski
Roderick A. Barr	Ned Basar	Owen Griffin	E. M. MacCutcheon
Arthur Griffin	Jon Buck	B. E. Hammett	Terence McGuinness
Robert Mast	Y. N. Chen	Don Hervey	Lou Mortaloni
Richards T. Miller	John Coyle	Duane Hove	David Price
J. Randolph Pauling, Jr.	Kent B. Davis	W. M. Howerton	R. S. Scotti
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	John Gleiter	Theodore T. Lee	Ralph Williams

OVERALL CONCLUSIONS

Adequate technologies, in terms of both engineering methods and functional hardware, exist today that can be used to design, construct, outfit, deploy, and operate OTEC plants. This working group has concluded that a pilot OTEC plant can be built within a period of four years from go-ahead and that commercial-size OTEC plants can be built by the late 1980s. These goals are readily attainable if decisive commitments are made by the federal government to realize fully the potential of ocean thermal energy as an alternative energy source.

OBJECTIVES AND HYPOTHESES ADDRESSED

The objective of this working group was to evaluate the status of key elements of the OTEC ocean system. For this study, the ocean system was restricted to four subsystems: the platform, the cold-water pipe (CWP), stationkeeping, and the electrical cable. The risks associated with each subsystem were assessed. Early reduction of the risks will accelerate the advancement of the OTEC program from research and development studies to pilot-plant explorations of the technical and economic feasibility of OTEC systems and thence to deployment of operational platforms.

This report is structured to respond to the following two hypotheses proposed by the Conference Steering Committee:

> "Hypothesis A. If adequate resources are allocated, an OTEC pilot plant will be in operation at an early date."
>
> "Hypothesis B. With an appropriate engineering development program, OTEC plants will become commercially competitive in the late 1980s."

There was also a series of questions included with each hypothesis to guide the working group toward responses to the conference theme. The hypotheses were also used by this working group to assess the risks associated with each of the ocean subsystems.

This committee did not obtain complete agreement on all of its findings. A minority report has been added to express these views.

SUMMARY AND CONCLUSIONS

Response to Hypothesis A

The OTEC pilot plant is a subscale prototype development and demonstration of a large-scale commercial plant. The pilot plant must demonstrate at significant scale (a plant size in the range of 10 to 100 MW_e) the potential economic viability of the commerical system. The plant will provide potential owners, operators, and financial investors the necessary assurance that representative low-cost electrical power can be supplied reliably at marketable costs. The pilot plant will lend the impetus to the large-scale commercialization of OTEC plants into the southeastern United States grid and the use of OTEC electrical power by energy-intensive industrial groups in sea-based installations.

This working group has concluded that the deployment and operation of the OTEC pilot plants can be achieved provided that the following program elements are properly implemented:

1. Technology Development. Acceleration of existing DOE and NOAA programs for the development and test of key OTEC subsystems hardware (for example, CWP and stationkeeping subsystems).

2. Integrated Systems Designs. The design and analysis of complete OTEC plants that demonstrate commercial viability must be initiated immediately.

Technology development includes the engineering design and analysis, and the fabrication, testing, and check-out required to reduce risks associated with moving from feasibility studies to real systems. Existing DOE/NOAA program elements (as proposed for budget request and confidence levels) are adequate but require acceleration in specific areas. Funding allocated for technology development in FY 80 is of particular importance. It does not appear to be adequate to accelerate existing technology development and could hinder the attainment of the postulated goals.

The use of existing test platforms, such as OTEC-1 and Mini-OTEC, should be vigorously exploited.

by DOE to facilitate testing of ocean subsystems. For example, alternative CWPs and electrical riser cables could be designed for the OTEC-1 platform. Testing of the attachment of a riser cable to a floating platform would provide valuable design verification data, and so would test of the cable itself.

The OTEC Program must begin transition from component and subsystem technology exploration to complete systems development. Integrated system designs represent a total systems perspective, i.e., the design of a complete plant responsive to a user-oriented specification. Specific operator requirements from electrical utilities or energy-intensive industrial groups must be included in the integrated design. The OTEC Users Council could become an effective form for the addition of OTEC operator requirements into the pilot plant systems design specification. This working group recommends that DOE establish a meaningful and vigorous dialogue with the OTEC Users Council, energy-intensive-industry groups, the financial community, and other elements of the private sector with an interest in OTEC commercialization.

The risks associated with each pilot-plant subsystem were assessed for the key phases of development: engineering design and analysis, construction and outfitting, and deployment and operations. The risks were assessed assuming that the integrated systems design and trade-off studies have been completed. It was also assumed that adequate resources would be available to support the design and analysis. The group's finding are summarized in Table 1.

Table 1

Assessment of Technology for Ocean Subsystems for Pilot Plants

	Adequate Body of Technology			
Phase	Platforms	CWP	Cables	Stationkeeping
Engineering design and analysis	yes[a]	yes[a,b]	yes[a]	yes[c]
Construction and outfitting	yes	yes	yes	yes
Deployment and operations	yes	yes	yes	yes

[a]Normal developmental and design verification test programs included.

[b]Large scale ocean testing required — estimated at one-sixth scale.

[c]Will require site specific data. (A static system can be designed for a Puerto Rico type environment.)

The group's approach was to identify the elements of the subsystems that are considered to be high risk because of their state of development. It was specified that the pilot plant must operate in sea state 6 (15 to 22 ft significant wave heights) and survive hurricane sea states. A yes answer implies that a body of technology exists that will allow the system integration contractor to build and operate the pilot plant successfully. A no answer would imply that a relevant technology base does not exist and that increased resources should be allocated for additional analysis of this subsystem. This committee, as shown by Table 1, concluded that the risks associated with normal development of the ocean subsystems are not sufficient to delay the deployment of the pilot plant beyond 1984.

Response to Hypothesis B

Commercial OTEC plants will range in size from 100 to 400 MW_e. They will operate in clusters in the Gulf of Mexico in a stationary plant mode for the production of electricity for utilities within the southeastern regional states or in the South Atlantic in a grazing plant mode for the production of electicity for an energy-intensive product. Studies indicate that both types of plants are an economic means of producing electrical energy.

Working Group 1 has concluded that commercial OTEC plants can be built by the late 1980s. The attainment of this goal is predicted on the assumption that, by 1986, a pilot plant will have been operating for two years. Technology development programs for the pilot plant must include the development of larger scale components associated with commercial plants.

The commercial plants are extensions of the pilot plant. Therefore, the pilot plant must exhibit the significant features of the commercial plant to allow the complete extrapolation of existing ocean engineering subsystem technologies.

The platform, stationkeeping, and CWP subsystems will be critical development items. The CWPs for commercial plants will be larger than those used for pilot plants (≥50-ft ID versus 30-ft ID) and may differ with respect to configuration (one or multiple pipes), construction materials (e.g., fiber-reinforced plastic, elastomer, or concrete), and fabrication. Deployment techniques may be different from the pilot plant designs. Large capacity static mooring systems have not been developed for vessels as large as the commercial plant. The stationkeeping for the commercial plant will require the advancement of existing technologies. Specifically, a promising concept requiring additional study is the use of a dynamic position system that combines effluent seawater propulsion with auxiliary thrusters.

The risks associated with each of the commercial plant subsystems were assessed for the key phases of development similar to that of the pilot plant. A summary of the group's findings is presented in Table 2. Technology exists (to lesser extent than that available for the pilot plant) to provide engineering solutions for design and analysis of the subsystems. However, the greater size of the components will necessitate the development of advanced fabrication and handling procedures in most cases.

Advanced handling procedures will be particularly important for the CWP. The CWP internal diameter will range from 50 to 100 ft, depending upon the commercial plant size. Whereas the 50-ft diameter CWPs are within current technologies, the larger diameter pipes will require significant development effort. Fiber-reinforced plastic pipes, steel pipe sections, and concrete structures have been constructed with diameters of about 30 ft. The

Table 2

Assessment of Technology for Ocean Subsystems for Commercial Plants

Phase	Adequate Body of Technology			
	Plat- forms	CWP	Cables	Station- keeping
Engineering design and analysis	yes[a]	yes[a]	yes[a]	yes[b]
Construction and outfitting	yes[c]	yes[c]	yes	yes
Deployment and operations	yes	yes[c]	yes	yes

[a]Requires normal development and design verification test programs.

[b]Requires the development of a large scale capacity stationkeeping system.

[c]Requires additional investigation of fabrication and deployment of large size components.

technology for CWP fabrication and handling exists, but it is not as robust as that for the smaller diameter pipes associated with the pilot plant. The technology developed for the pilot plant must be directly applicable to the commercial plant.

Construction and outfitting of the platforms will require facilities dedicated for OTEC plants. (For a large shipyard, this does not mean the entire yard, but clearly dedicated building positions and associated support facilities.) The facilities must have necessary water depth and large capacity handling equipment to provide the lowest construction costs. The construction should follow the principles and procedures already established for the construction of large offshore, concrete, oil-production platforms.

MINORITY REPORT

There exist deep seawater structural technologies (for example, guyed towers, tension leg platforms, and bottom-fixed platforms) being developed by the offshore oil industry that should be considered for OTEC applications. The technologies could provide solutions to the CWP and electrical cable dynamic response and attachments problems associated with the floating concepts. Investigation of the technologies should proceed in parallel with that of the pilot plant described for Hypothesis A.

Additional research and development should be concentrated on platform configurations not being studied by NOAA/DOE. Existing NOAA/DOE technology development programs are directing too much research and development funding towards floating platforms in general and two specific platform configurations in particular: the JHU/APL barge and the spar with external heat exchangers. It has not been conclusively shown that these platforms represent the best platforms for OTEC commercial plants.

REPORT OF WORKING GROUP 2: HEAT EXCHANGERS

Executive Committee

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Jim Denton	Fred Smith
Bob Heise	Leon Stabinsky
T. Kajikawa	J. Yampolsky
Kiroshi Kamogawa	Dave Yung
James Keirsey	

IMPROVEMENTS IN PERFORMANCE AND/OR COST

Rapid advances have been made in the heat exchanger art for closed-cycle OTEC plants. Several distinct possibilities exist which may offer reduced cost. Testing programs are planned or in progress to evaluate these concepts. Tests at the Argonne National Laboratory in Argonne, Illinois (ANL) have shown that enhanced surfaces may be used effectively on the shell side (power-fluid side) of shell-and-tube heat exchangers. Several innovations have been advanced in shell-less tubular exchanger designs, which do not require the use of an expensive confining shell. Tests at ANL of a folded-aluminum-tube, shell-less heat exchanger with ammonia inside the tubes have demonstrated evaporator performance exceeding predictions. In addition, exchangers of the corrugated plate type and the plate-fin type with extended surface on the power-fluid side promise high performance.

BIOFOULING CONTROL AND MATERIAL SELECTION

Distinct progress has been made in microfouling control (see also the report of Working Group 5). On-line mechanical cleaning systems (brushes and balls) are leading candidates for microfouling control in circular tubes. Recent test results suggest that the water-side fouling factor may be maintained at 0.0002 to 0.0003 hr-ft^2-°F/Btu. However, long-term performance data are needed to establish the required cleaning frequency. Chemical and mechanical cleaning are viewed as possibilities for the corrugated plate exchanger. Cleaning possibilities are yet to be evaluated for plate-fin exchangers, which have smooth noncircular channels on the water side.

Two possibilities are being considered for macrofouling control: chlorination and the use of copper alloys or antifoulant claddings on the tube sheet. If necessary, water boxes may be manually cleaned.

Material selection remains a dominant question in heat exchanger design. The antifoulant copper alloys may not be compatible with all candidate heat-exchanger materials. Further work is required on the control of macrofouling. The range of possible heat transfer tube materials has increased. In addition to titanium, several high chrome-moly stainless steel alloys offer high corrosion resistance. Copper-nickel claddings, copper alloys, and special aluminum claddings have shown promise for continued corrosion evaluation. Trade-off studies have shown that less expensive materials, of shorter life, may be economically feasible. However, the life of such materials is yet to be established.

DEVELOPMENT EFFORTS NEEDED

Further development effort is needed to qualify or determine:

1. Lower heat exchanger material costs.
2. The required frequency of mechanical tube cleaning.
3. The benefits and costs of chlorination within the limits imposed by environmental regulation.
4. Cleaning methods for corrugated plates or noncircular water channels.
5. The effectiveness of brush or ball cleaning systems in water-side-enhanced tubes. Some enhanced waterside surface geometries may be more cleanable than others.
6. The effectiveness of ultrasonic cleaning and its applicability to other heat exchanger types in addition to the folded-tube type.
7. Vertical tube, thin-film evaporators under conditions of ship motion.
8. Ammonia-water chemistry requirements versus material type, and the effects of corrosion caused by ammonia leakage.
9. Expected heat exchanger failure modes.

Proposals for new heat exchanger concepts should be evaluated by heat-transfer specialists, who may pinpoint problem areas and recommend design changes required to satisfy OTEC requirements. Heat exchanger development proposals should consciously establish maintenance plans, which propose means of leak detection and repair.

RELATIONSHIP TO PILOT-PLANT DESIGN NEEDS

Sufficient knowledge exists today to design conservatively based shell-and-tube heat exchangers for pilot-plant application. Such state-of-the-art exchangers may use enhanced surface geometries on the power-fluid side and presently defined corrosion-resistant materials. However, cost goals have the best opportunity of being met using lower-cost materials and/or new exchanger design concepts that are now being tested or will be tested next year. Although the compact plate-fin designs suggest high performance, their serious consideration will require use of corrosion-resistant materials and proof of effective cleaning methods. By 1981, the choice may be broadened to other candidate heat exchangers and materials. At that time, more than one candidate heat exchanger design type and lower-cost materials may exist as distinct possibilities. Therefore, it may be advisable to incorporate more than one heat exchanger design type in the pilot plant. The pilot-plant program should verify heat exchanger performance, evaluate fouling control, and validate an operation and maintenance plan for the heat exchangers. It should provide a basis for establishing heat exchanger material life.

RELATIONSHIP TO COMMERCIAL PLANT GOALS

A commercial-size OTEC plant should use cost-effective materials, exchanger configurations, and enhanced surfaces where feasible. Considerable development work remains to be completed if this goal is to be met. It is doubtful that all necessary work on concepts that may prove most cost-effective for commercial plants of various types (e.g., cruising and moored) will have been completed prior to initiation of the pilot-plant program. Therefore, we recommend strong support of an ongoing heat exchanger test and development program. The program should be structured such that exchanger design concepts, which are shown to be effective, can be integrated in a commercial-size plant design. We are optimistic that heat exchangers will evolve that will meet cost and performance goals for commercial OTEC plants.

DISCUSSION FROM FLOOR

G. L. Dugger, JHU/APL: I wish to reinforce the working group's statements relative to integration of heat exchangers into appropriate platform systems. The various candidate heat exchangers for both OTEC pilot plants and commercial OTEC plants should be compared on a total system basis not only with regard to feasibility and projected performance but also with regard to overall system cost. The details of required arrangements and volumes of required warm and cold water delivery and exhaust systems; headers and piping for working fluid systems including demisters, working fluid surge tanks, clean-up systems, etc.; and the heat-exchanger cleaning system affect platform size and buoyancy requirements and total system cost. Comparisons of heat exchanger core costs in $/kW$_e$ and/or on the basis of overall heat transfer coefficients (U) can be grossly misleading if this is not done. The cost goal is minimum overall OTEC system cost in $/kW$_e$ and O&M cost, not simply maximum U or minimum core $/kW$_e$.

R. Webb, Penn. State U: This point was discussed in our working group, and it raises the question of philosophy of the pilot plant — is it to be a test bed, or are we asking that it be an integrated design and that the exchanger designers be informed what kind of package and requirements they have to meet? In my opinion, it should be some of each. It should have some additional things that make testing and try-outs of other things possible. It also should show, to the extent possible, everything that would be related to the use of that heat exchanger system on a commercial plant and what it will cost. Bob Douglass made a good point on this side: the pilot plant designers must consider what needs to be demonstrated on the pilot plant from the user's viewpoint and get his input on everything including what you can do best to make sure you are using true modular components that can be scaled up to permit cost extrapolation with some confidence.

R. Scotti, NOAA: The members of the Ocean Engineering Working Group in particular, and perhaps the rest of the groups, shared this with you. And we have a lot of heartburn about the OTEC-1 experiment, primarily because it is not a total system. I bring that up to illustrate my further point: in your presentation, the purpose of the pilot plant, and also some of the questions as to where you would demonstrate alternate concepts, left me a little confused considering the ultimate purpose we understand for the OTEC-1 experiment. Our understanding is that it is primarily for testing heat exchangers, whereas we feel that it should have been a total system, i.e., a modular plant. Could you comment on that please?

R. Webb: The working group felt that any candidate exchanger probably should be subjected to a test comparable to what has been done by the Argonne Lab using 30-kW$_e$ core units and fresh water. Then the more promising candidates should be tested in representative seawater environments, and probably they were thinking of OTEC-1. Having passed those two tests and met material corrosion requirements, we could consider any heat exchanger to be a candidate for pilot-plant demonstration. The group did not feel that you should design a full-scale plant based on a candidate heat exchanger without having tested on the 10/40-MW$_e$ pilot plant a size larger than the OTEC-1 will take.

REPORT OF WORKING GROUP 3: POWER SYSTEMS/ ALTERNATIVE CYCLES

Executive Committee

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CLOSED-CYCLE POWER SYSTEMS

The working group evaluated each closed-cycle power system component with regard to risk in its performance, cost, and reliability/availability/maintainability (R/A/M) focused on the environmental issues. The risk categories were defined as:

Level 1: The technology status is at a point in development that allows normal commercial warranties by manufacturers of the component.

Level 2: The technology status leads to a risk level that is beyond normal component warranties but the risk level is acceptable if current programs in the component area are pursued vigorously.

Level 3: The risk level is at a level that requires an effort beyond the current level of activity.

Table 1 presents the consensus ratings of the working group.

Summarizing, the consensus of the group was that the conventional closed-cycle power system can be built now and that current test programs should be pursued vigorously.

ALTERNATIVE POWER CYCLES

The four alternative cycles were evaluated as to their states of maturity. The criticality of technology for various subsystems was then evaluated. The following comments address the overall state of each concept.

1. Open Cycle. A mature concept that was demonstrated by the French in the 1920s. Design work has been pursued to the conceptual level. The possibility of fresh-water production should be considered in economic evaluations.

2. Hybrid Cycle. At the conceptual level, it appears less cost effective than the open cycle although it also has water production capabilities.

3. Foam Cycle. Requires proof of scientific feasibility with surfactant being the chief problem.

4. Mist Cycle. Requires proof of scientific feasibility.

5. Criticality Ratings. Table 2 presents the criticality of technology for the major subsystems of each alternative power cycle. The lowest number indicates the greatest criticality to technical success.

Table 1

Risk Levels for Closed-Cycle Components
(1 is lowest risk or highest confidence level)

Subsystem	Performance	Cost	R/A/M
Heat exchangers by type[a]			
Shell-and-tube	1	1	2-
Plate[b]	1	1	2
Plate-fin	1	1+	2
Trombone (Folded Tube)	1	2	2
Tubular-plate[c]	2-	2	2
Turbine-generator[d]	1	1	1+
Working fluids	1	1	2
Auxiliaries	1	1	1
NH_3/Freon purge-fill	1	1	1
NH_3/Freon clean-up	1	1	1
System dynamics, controls and instrumentation[e]	2	2	-
Seawater pumps[f]	1	1	1
NH_3 freon pumps	1	1	1
Screens and trash racks[g]	2	2	3
Heat-exchanger cleaning[g]	2	2	2
Internal electrical systems	1	1	1

[a] The reliability and maintainability of all heat exchangers are strongly material-dependent.

[b] No long-term data on cleaning are available for plate exchangers.

[c] Test data are not yet available with ammonia.

[d] Stress-corrosion cracking is the concern in ammonia turbines.

[e] Control concepts vary between concepts and contractors.

[f] The seawater pump system is strongly platform-dependent.

[g] Screen and cleaning-system performance are strongly dependent on platform design, site selection, and heat exchanger design.

Table 2

Criticality Ratings for Alternative-Cycle Components

Alternative Power Cycle	Criticality Number
Open	
Turbine	1
Flash Evaporator	1
Deaeration	2
Hull Dynamics	2
Vacuum Containment	3
Direct Contact Condenser	3
Conventional Condenser (Cold Seawater inside Tubes, Fresh Water outside)	4
Hybrid	
Economically Questionable	
Foam	
Surfactant	1
Foam Transport	2
Deaeration	2
Direct Contact Condenser	3
Drag	4
Mist	
Liquid Transport	1
Mist Generator Fouling	2
Deaeration	2
Direct Contact Condenser	3

In summary it was proposed that the development of alternative cycles should be pursued for possible intermediate range application to effect economic improvements over closed-cycle power systems.

INNOVATIVE CYCLES

Innovative cycles discussed were the direct-contact, the air-working-fluid, the thermal-electric, and the gravity-opposed heat pipe. The group thought that innovative power cycles should be pursued as long-range research projects to the point at which their technical feasibility and any potential economic benefits could be credibly evaluated. The problems of such long-range R&D as perceived by the group are the lack of a central evaluation authority and the difficulty experienced by universities and other R&D activities in developing coherent programs when faced with the usual short-term funding patterns for such programs. The group recommended that a council of industry, university and DOE representatives be established to evaluate innovative cycle research programs and to assist in the long-range direction of this activity.

DISCUSSION FROM FLOOR

G. L. Dugger, JHU/APL: I wish to reinforce the working group's definition for the cost ratings in Table 1: the #2 risk level rating given to folded-tube and tubular-plate heat exchangers does not mean they are higher in cost; it simply means that the working group considers their costs to be known with less confidence than the costs of shell-and-tube heat exchangers. We at APL are convinced that folded-tube, aluminum heat exchangers will be among a few types offering lowest overall system cost.

REPORT OF WORKING GROUP 4: ENVIRONMENT, RESOURCES, AND SITING

Executive Committee

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Participants

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Alan Blumberg	Gerhard Jirka	Mack Sullivan
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James Churgin	Peter Johnson	Anitra Thorhaug
Marcie Commins	Lloyd Lewis	Laurence Tilly
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J. R. DePalma	Bob Paddock	Robert Waid
Jack Ditmars	L. Pierre	Keith Waite
Doug Evans	Avenkata Ramiah	Mike Watson
David Fry	John Reitzel	Pat Wilde
Steve Giannino	Walker Retzsch	Arthur Wilson
Juan Gonzalez	Richard Sheffield	

SCOPE OF DELIBERATIONS

Some initial discussion was devoted to interpretation of the premises, hypotheses, and questions suggested by Gordon Dugger and Fred Naef, as they related to this group. Because of the lack of time, only hypothesis A was discussed.

Hypothesis A. If adequate resources are allocated, a 10/40 MW_e OTEC pilot plant will be in operation at an early date.

The group's responses to the questions relative to this hypothesis were based on a platform of 10- to 40-MW_e size that would be moored near an island and had at least a 10-year operational lifetime. The responses are presented in the order that the questions were posed.

TECHNICAL ELEMENTS THAT CONSTITUTE A RISK

What is the adequacy of the technology for the design, construction, and operation of a pilot plant? That is, are there any technical elements that constitute undue risk for a government-funded demonstration? At least the following two general areas will constitute risks to the pilot plant if not resolved before completion of detailed design for construction.

Identification of Optimal Platform Location

The candidate locations for moored OTEC plants must be investigated early through historical and field data collection programs in order to select the optimal location. The concerns to be addressed in evaluating candidate locations are:

1. Physical Concerns
   a. Adequacy of the annual average ΔT and the seasonal variation in ΔT
   b. Current speeds and directions
   c. Wave spectra and wave traces
   d. Frequencies and magnitudes of extreme weather events
   e. Bottom topography
   f. Visibility
   g. Proximity to electricity market including required length of undersea cable
   h. Proximity to other site uses (i.e., shipping lanes and designated mineral oil exploration regions)
   i. Earthquake potential (primarily for land-based plants)
2. Biological Impact and Biofouling Concerns
   a. Commercial and recreational fishing grounds
   b. Fish spawning grounds and migration paths
   c. Known rare and endangered species habitats
   d. Other unique biological habitats or marine sanctuaries
3. Economic Potential — Realistic long-term siting plan that will maximize ocean resource potential

Federal and State Permitting Cycle

Based on the OTEC-1 permit application cycle, there are at least two federal agencies (EPA, Corps of Engineers) involved in issuing various types of platform permits. The OTEC-1 site is beyond the 3 mile State jurisdiction, but logistic support on the coast may require a permit. The number of agencies involved is expected to be greater with a demonstration plant because of its longer life time.

The appropriate regional office of the Environmental Protection Agency will require that a point-source outfall permit be obtained as specified under the National Pollutant Discharge Elimination System (NPDES). These permits take approximately 6 to 12 months to obtain and require routine effluent monitoring as a condition of the permit. This permit is required for platform operation. The U.S. Coast Guard will also be involved in issuing a permit for a 10- to 40-MW_e plant.

Other Federal agencies or departments that may play a role in licensing are:

1. Bureau of Land Management

2. U.S. Geological Survey

3. National Oceanic and Atmospheric Administration

4. Fish and Wildlife Service

MEASURES REQUIRED TO REDUCE RISKS

What must be done to make the risk acceptable? That is,

1. What critical-path activities must be completed prior to the initiation of design of a pilot plant?

2. What critical-path activities must be completed prior to the initiation of construction?

3. What is the earliest operational date that can be achieved?

The reduction of risks to an acceptable level will require the preparation of an environmental assessment (EA) based on the National Environmental Policy Act of 1969 and will be dependent on site-specific oceanographic data as well as other experimental studies conducted in the field and laboratory to evaluate potential impacts. These data will be used to assess the potential for adverse environmental impacts resulting from plant operation. Secondly, the EA/EIS will provide guidance to mitigate or reduce impacts through design modifications. Both field and laboratory data will be used by the regulatory agencies in setting effluent discharge guidelines. The working group divided into three subgroups to discuss the key issues associated with platform operation. The person(s) named in each subheading led that subgroup.

Intake and Discharge (Arthur Barnett/John Morse)

The primary issues relative to intakes are entrainment, impingement, and displacement of organisms. Lower priority issues specific to the cold water intake include the presence of reactive particulate matter as it may act as a surface area for bacteria to colonize and increase or accelerate biofouling. The reactive matter may also absorb trace elements from the condenser and be released to the receiving waters after mechanical cleaning. Another issue is the effect of fish attraction to the pipe. At the warm water intake, it is important to determine how to dispose of impinged organisms.

At the discharge, there are three primary areas to be evaluated:

1. Toxicity

   a. Biocides and other chlorination products

   b. Metallic corrosion products

   c. Variations in toxicity to organisms of different sizes and life stages and different species

   d. Effect of temperature on toxicity

2. Suspended Load

   a. Influence of the discharge on ambient dissolved oxygen concentrations

   b. Material composition and its ultimate fate in the environment

   c. Attraction to materials as a food source

3. Ammonia Leakage Effects — effects caused by low, intermediate, or major leaks during normal operation and when the flow through the plant is stopped or reduced

Items of possible importance include biostimulation, fish attraction to the plume, thermal effects, and carbon dioxide release to the atmosphere. It is important first to establish what is known and then to classify those studies that can be done prior to plant operation (toxicity) and those that must be done during operation (fish attraction, biostimulation).

Platform Consideration (George Maul/Robert Munier)

In addition to the siting concerns noted, the key platform issues to be evaluated are:

1. Attraction

   a. Concentration of various tropic levels around the platform

   b. Effect of light on attracted organisms

   c. Shape and coating material of hull

   d. Hull cleaning effect on the environment

2. Accidental releases of stored working fluid or chlorine

3. Platform health and safety

Mooring, Transmission Cable, and Shore Support Issues (Charles Bretschneider)

The mooring features of a candidate site must be known in order to identify the anchoring option. It is important to perform a detailed bathymetric and geotechnical survey of the anchor location by boring, performing sub-bottom profiling, and collecting sidescan sonar records. These data used in conjunction with information on the substrate properties are essential to selecting the optimal mooring location.

Wave and sea-state data, including directional wave spectra, directional sea state, wave period and frequency, and amplitude, are needed to define the design limits of the mooring legs. Additionally, data on the design storm conditions should be prepared for each candidate mooring site from hindcast data and meteorological information. It is important for the prediction of extreme wave heights and period that wave spectra data and joint probability distributions of heights and period be collected.

The required cable information is similar to that required for pipelines and outfalls and includes site surveys, borings, reef damage assessment, substrate properties, sea states, storm conditions, and frequency.

Obviously, not all the issues, potential impacts, or considerations can be resolved prior to the design or construction of an OTEC-10/40 MW_e platform. In fact, an early start on design to establish probable intake and discharge characteristics is desirable. However, it will be prudent to complete the permit approval cycle prior to construction to assure that the platform, as designed, will be allowed to operate. Much can be learned in the coming years through field and laboratory studies, OTEC-1, monitoring, and impact assessment studies.

The critical elements to be completed prior to design, construction, and deployment are illustrated in Fig. 1, which indicates at least four years are required to complete the studies required prior to operation.

MISSION OF THE 10/40 MW_e PILOT PLANT

In the context of this working group, what is the appropriate mission of a pilot plant? The group's response is threefold:

1. Verify near-field and far-field physical models covering the range of site conditions. The results of these observations should be compared to results obtained from coastal power plant studies.

2. Collect in situ data on toxicity of the discharge to phytoplankton, zooplankton, micronekton, and fish (larvae, juvenile, adult). These data can be extended to food chain dynamics to assess the long-term effects of OTEC operation.

3. Examine effects resulting from platform attraction and platform colonization, evaluate the entrainment mortality, and determine whether mortality can be reduced through intake design. Examine the utilization by the biological community of discharged materials to determine whether, and to what extent, they are being bioaccumulated.

PROGRAM CHANGES IN TECHNICAL EMPHASIS

What changes in technical emphasis in the OTEC program do you recommend for consideration by Program Management? The group's response is:

1. Improve information transfer and communication between the OTEC contractors, particularly between the engineering and oceanographic communities.

2. Select, classify, and designate the sites for OTEC use in thermal resource regions.

3. Collect predeployment oceanographic and biological data that will determine the temporal and spatial variability of the site/region where the platform will be located.

4. Standardize oceanographic field collection techniques and laboratory methods throughout the OTEC program.

5. Compare the shore support facilities required to those available to service an OTEC 10/40 MW_e platform for each candidate site.

6. Prepare a risk assessment model for each site that will evaluate the safety and health risk on the platform as well as to the adjacent population centers.

The accomplishment of these environmental and siting studies will require significant budget allocations.

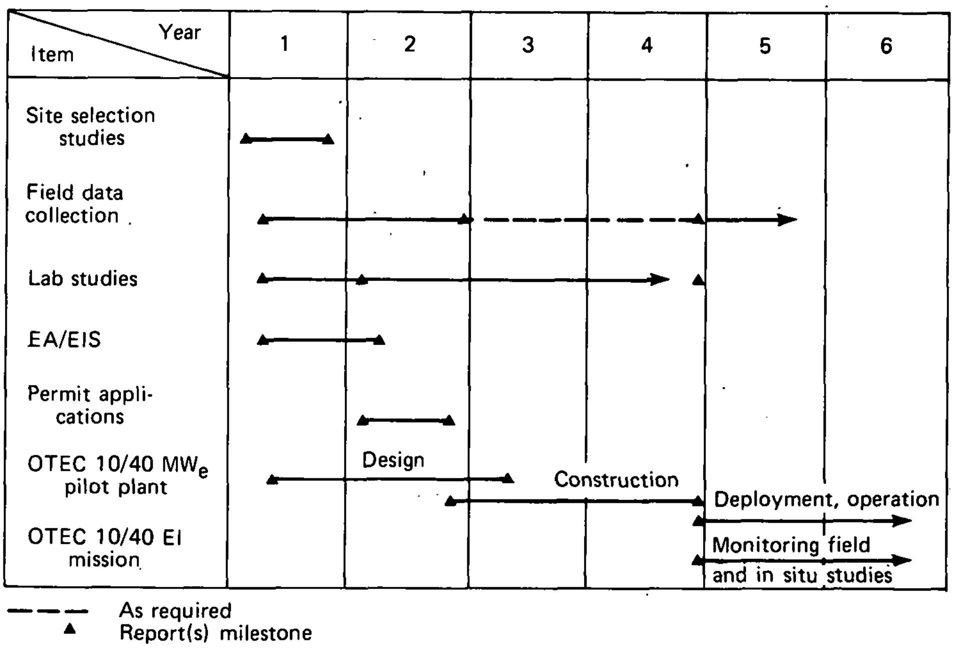

Fig. 1 Time schedule for environmental studies-hypothesis A.

REPORT OF WORKING GROUP 5: BIOFOULING, CORROSION, AND MATERIALS

Executive Committee

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Jay L. Kratz
Brenda J. Little
Michael Mann
C. F. Schrieber
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John Wallace
David E. White

SUMMARY AND CONCLUSIONS

Conclusions Relative to the Pilot Plant

The conclusions of Working Group 5 with respect to the early development of the 10/40 MW_e OTEC pilot plant are:

1. There is no undue risk in the area of materials selection for heat exchangers, pumps, ammonia system, platform, or cold water pipe.

2. There is no undue risk in engineering cleaning systems to maintain a fouling resistance of R_f = 0.0002/0.0003 hr-ft^2-°F/Btu (or less) in production-type heat exchangers.

3. The mission of the pilot plant is to verify that the design, material selection, and cleaning methods will achieve acceptable efficiency.

4. There is a strong need to improve the integration between research and engineering organizations working in the area of biofouling control and measurement, corrosion, materials, and related aspects of heat-exchanger design and performance. There does not appear to be an overall plan with specific target dates to qualify materials and cleaning methods.

5. There should be a central test facility that integrates all materials, corrosion, and biofouling work.

Conclusions Relative to Commercial Plants

The following engineering developments are required to make OTEC commercially competitive:

1. A low-cost heat exchanger material must be qualified through available test data or through the generation of the data at the above commended central test facility.

2. The maximum cleaning interval required to obtain acceptable biofouling control must be determined through at-sea testing.

3. Fatigue data for cold-water-pipe materials are needed to design a 30-year-life system.

INTRODUCTION

The Executive Committee met on Tuesday, June 19, to organize the agenda for the Working Group 5 sessions. The objectives of defining what areas need risk reduction for early 10/40 pilot-plant development were outlined as were the engineering developments needed to make OTEC commercially competitive in the late 1980's. Each member was asked to comment on the Biofouling, Corrosion, and Materials (BCM) Program as they see it and to offer their recommendations. The comments/recommendations were as follows:

1. BCM work has been piecemeal and should be contracted through one organization so that all work would be centralized.

2. There is a need to build small pilot plants so that the entire system can be evaluated.

3. There do not seem to be programmatic goals for the near or long term.

4. Biofouling and its control need more effort. Results to date are minimal and questionable. An overall plan is needed for biofouling control evaluation.

5. A matrix system for evaluation of materials and cleaning methods must be designed for use by management to select materials and cleaning methods.

6. A priority sequence for test work must be established starting with the lowest cost material to determine what must be done to qualify or disqualify that material.

7. Titanium should be used as a reference standard to compare other materials from the standpoint of corrosion, erosion, biofouling, and cleaning methods.

8. All cleaning methods should be tested on titanium to provide values for comparison when testing cleaning methods on lower cost materials.

9. The overall objective is to combine the lowest cost material with the most efficient cleaning method(s).

10. Establish a committee from materials personnel of subcontractors to meet periodi-

cally to designate desired materials test programs.

The Executive Committee decided that the working group meeting agenda would include materials for heat exchangers, seawater pumps, ammonia systems, platform, and cold-water pipe; and biofouling and its control.

DISCUSSION — WORKING GROUP

Materials

Heat exchanger materials. It was agreed that titanium and AL-6X stainless steel were qualified for OTEC heat exchangers and no further work is needed for those materials. Other stainless steels such as the 26, 28, and 29% chromium alloys having 2 to 4% molybdenum and 2% nickel were suggested as lower cost equally corrosion resistant materials but, it was agreed that none of these stainless steels have the production and field service experience to consider them qualified. It was recommended that F. L. LaQue compile a list of tests that would qualify these newer stainless steels.

On copper-nickel (Cu-Ni) it was agreed that it can not be used for ammonia service based on the latest test results from Dow (Chuck Schrieber) at Freeport, Texas, and the Coast Guard restriction of its use with ammonia. The use of copper-nickel-clad steel may offer a solution to the problem of using ammonia, but for the plate-type heat exchanger designs, there is presently no metal producer who manufactures thin-sheet, Cu-Ni-clad steel. Clad product is state of the art, but widths greater than 24 in. may not be readily available because it is not a product line of any of the metal producers. Copper-nickel will microbiofoul but at a much slower rate than other materials.

The life of aluminum alloys under OTEC conditions of frequent mechanical cleaning by M.A.N. brushes or Amertap balls is an unknown that can be resolved through the tests at F. L. LaQue Corrosion Lab, Panama City, and Mini-OTEC. In plate-type heat exchangers, the potential for crevice corrosion makes aluminum a questionable candidate, although the aluminum industry is of the opinion that the Alclad products, particularly Alclad 3004, should provide sufficient cathodic protection to prevent crevice corrosion at the faying surfaces of the plate design. Representatives of the aluminum industry offered to provide material from any phase of the OTEC test program, and they also suggested that the composition of Alclad might be altered to increase its life. It was felt that development of an improved cladding fell into the same category as qualification of the high chromium stainless steels, i.e., it is the responsibility of the industry to develop and qualify any material recommended for OTEC.

Seawater pumps. Present-day production seawater pumps are highly reliable components, and they present little or no risk to OTEC.

Ammonia system components. Piping, valves, pumps, and pressure-vessel materials for handling ammonia were considered low-risk in that low-carbon steel and stainless steel have given excellent service for many decades. The use of 0.2% water and/or stress relieving to prevent stress corrosion cracking has been well documented as a reliable procedure to control this problem.

Platform. The present state of the art for designing and building large offshore platforms of steel and concrete indicates that this is a very low or no risk area for OTEC.

Cold-water pipe. Cold-water pipes of the sizes required for the 10/40 pilot plant were considered a low risk area, but there is a question in the area of pipe joints as to whether present material candidates can provide acceptable fatigue life, especially for the large pipes required for commercial plants. It was recommended that life-cycle fatigue data of the pipe joint designs be obtained through testing in seawater where such data are not available.

Fouling and Countermeasures

The recommendations of the biofouling session of the OTEC Biofouling, Corrosion, and Materials Workshop held on January 10, 1979, at Rosslyn, Virginia, were reviewed to determine whether these recommendations could simply be endorsed by this working group as still appropriate. The general consensus was as follows:

1. Continue to monitor tests for dissolved organic carbon, particulate organic carbon, dissolved oxygen, salinity, temperature and pH, total nitrogen and phosphorus, heavy metals, and ammonia since these measurements bear on biofouling activity and are of low cost.
2. Continue film-thickness, ATP, TOC, and SEM (limited) analyses at all test sites and initiate E.D.S. and x-ray diffraction analysis of biofouling films.
3. Initiate studies to gain understanding of conditioning (or first) film.
4. Standardize specimen preparation and evaluation methods. Use only millproduct finishes (no special surface preparation) that will be purchased for OTEC heat exchangers since high quality surface finishes such as polishing, would be cost prohibitive for production units.
5. Test the sample materials at all locations.
6. Conduct continuous tests of 1- to 2-year duration to determine the most cost-effective cleaning method for the lowest cost heat exchanger material.
7. Centralize test work at one location representative of an OTEC site.

In the area of macrofouling control, the use of antifouling paints, antifouling concrete, 90-10 copper nickel cladding, and antifouling rubber sheet (No-Foul) were suggested as control methods for structural components such as the platform and seawater piping. There is adequate experience and success wtih these control methods to categorize macrofouling as a moderate risk over total plant life.

F. L. LaQue, Consultant: During the discussion among the members of Working Group 5, I was asked to set down my recommendations for criteria and tests for qualification of stainless steel alloys for OTEC heat exchangers. As stated in my paper,[1] I consider AL-6X and closely related stainless steel alloys to be qualified. For new or appreciably different stainless steels, the following information would be desired.

Essential Properties and Related Tests

Resistance to intergranular corrosion. Results of tests as per ASTM A-262-77 or G28-72. The latter is preferred for so-called "nickel-rich" alloys.

Resistance to stress corrosion cracking. Results of tests as per G-30-72, G-38-73, or G-49-76. Resistance to cracking under the extreme conditions of exposure to boiling magnesium chloride is not required. Resistance to cracking in seawater or NaCl solutions at temperatures $\geq 30°C$ would be reassuring but is not essential.

Resistance to pitting and crevice corrosion. This is the most critical property for qualification. Accelerated tests appropriate for preliminary screening include ASTM G-46-76 and G-48-76. Data on effects of temperature on incidence of crevice corrosion in $FeCl_3$ at pH 1, as illustrated by the data in Table 6,[1] and results of tests with impressed anodic currents to establish a so-called "break-through potential" (as in Table 7[1]), would be useful. The preferred "potentiostat" techniques are as per ASTM G-3-74 and G-5-72.

Corrosion tests in general. Per ASTM G-52-76.

Tests in natural seawater. Alloys qualified by screening tests such as described should be qualified further by exposure to natural, unpolluted seawater reasonably characteristic of that at potential OTEC sites. Such conditions of exposure would be expected to result in extensive fouling by barnacles and other marine organisms that could promote pitting of vulnerable alloys. While such fouling could not be permitted in OTEC heat exchangers, a demonstration of resistance to pitting under fouling organisms would be indicative of resistance to pitting under deposits of various kinds that might be encountered in OTEC service. These tests have durations of at least a year and should include multiple crevice test specimens as per the Anderson technique (Ref. 29[1]).

Comparison with other alloys. Interpretation of results of tests, and particularly accelerated and electrochemical tests used for screening, would benefit from inclusion in the test of alloys for which a relation of the test results to service experience has been established. Appropriate alloys for such comparisons would be AISI type 316 and AL-6X.

Miscellaneous Data

Galvanic behavior. Results of measurements of open-circuit corrosion potentials in seawater that would permit placing a candidate alloy in its proper place in a galvanic series, as per Table 7,[1] should be furnished.

Resistance to ammonia. It is expected that any of the iron or nickel base alloys likely to be proposed for OTEC heat exchangers will have adequate resistance to corrosion by ammonia and mixtures of ammonia and seawater. Any data that would confirm this expectation would be welcomed.

Resistance to chlorination and chemical cleaning. Available data that would support the candidate alloy's adequate resistance to corrosion by seawater treated with the low concentrations of chlorine that might be used in OTEC service would be welcomed. Likewise, any available data on resistance to attack by acids, alkalis, and corrosive salts that might be used in cleaning systems would be of interest.

Corrosion fatigue. Any available data on corrosion fatigue strength in seawater would be of interest in connection with the remote possibility that excessive vibration could occur and lead to corrosion fatigue failure.

Fretting corrosion. Since vibration could lead to fretting corrosion of tube supports, any available data on resistance to such damage would be of interest.

Effects of seawater cavitation or extreme flow velocity. The use of ultrasonic devices to induce cavitation in the seawater to clean OTEC heat exchangers is being investigated.[2] Any data on resistance of the alloy to such cavitation or to seawater flowing at high velocity would be of interest.

Metallurgical Details

Information should be provided on the metallurgical structural features of the alloy including any undesirable features such as precipitated carbides and phases such as sigma, and how these are avoided by appropriate controls of alloy composition and/or heat treatment.

Trial and/or Full-Scale Installations

Results of trial installations in condensers or other heat exchangers operating with seawater reasonably related to OTEC seawater should be furnished including characteristics of the water, operating temperature and range, operating velocity range, number and dimensions of the trial tubes, duration of the trial and its continuity, any treatment of the water as by chlorination to control fouling, and any mechanical cleaning to remove fouling. The record also should include the composition of other tubes in the trial heat exchanger, the material used for tube plates and water boxes, and the presence, if any, of anodes used for cathodic protection. Results of the tests should include details of the extent, if any, of pitting, crevice corrosion, or other forms of attack that may have occurred. Records of service and extent of any full-scale installations along these same lines should be furnished.

References

1. LaQue, F. L., "Qualifying Aluminum and Stainless Steel for OTEC Heat Exchangers," Proceedings of this Conference, Vol. II.

2. Pandolfini, P. P. et al., "Experiments in Ultrasonic Cleaning of a Shell-Less, Folded-Aluminum-Tube, OTEC Heat Exchanger," Proceedings of this Conference, Vol. II.

E. Kinelski, DOE: I wish to congratulate Working Group 5 on their deliberations and their report. On some of the points made I wish to offer bits of information, clarification, or reassurance, as follows:

1. I concur in conclusions 1 and 2 relative to the pilot plant on the basis that sufficient information should be obtained between now and the time the heat-exchanger and cleaning system designs for the first 10/40-MW_e pilot plants are committed to construction in FY 1982.[1] The points raised in the third paragraph of the materials section regarding aluminum, will have to be examined carefully at that time, with the knowledge that the first pilot plant(s) probably will be used less than 10 years. However, the pilot plant(s) will provide further data, if aluminum is used for some or all of the heat exchangers, which can be extrapolated to estimate the probable life of aluminum for heat exchangers on commercial plants.

2. Regarding conclusion 1 relative to commercial plants, work toward qualifying aluminum, and possibly galvanized steel, is planned.[1]

3. Regarding items 1, 3, 5, 7, and 8 in the Introduction, the ongoing DOE program is managed by one organization — Argonne National Laboratory — and is centralized; a DOE Program Plan stating program goals should be available by the time these Proceedings are issued; our plan for biofouling control evaluation is briefly outlined in my paper;[1] a matrix of materials and cleaning methods is included in the Program Plan; and titanium will be used as a reference standard for both alternative materials and cleaning methods.

4. I do not agree with item 6 in the Introduction; I believe that all four condidate materials — Ti, Al, S.S., and Cu-Ni — should be and will be evaluated simultaneously until such time as one is clearly ruled out.

Reference

1. Kinelski, E. H., "Biofouling Corrosion and Materials Overview," Proceedings of this Conference, Vol. II.

J. Rynewicz: The majority of the Executive Committee of Working Group 5 has reviewed Mr. Kinelski's comments and is heartened by his reassurances but still holds that the report correctly expressed its conclusions and concerns. One exception is a clarification of Item 1 in the Introduction, to wit: all BCM work should, insofar as practical, be contracted to one organization which would perform (or be charged with responsibility for) all test work in one seacoast test facility.

REPORT OF WORKING GROUP 6: ECONOMIC, LEGAL AND INSTITUTIONAL ASPECTS AND COMMERCIALIZATION

Executive Committee		Participants	
Donald H. Guild, Chairman	B. J. Anderson	Bill Hale	Ved P. Nanda
John Babbitt	Molly Andrews	Larry Hargrove	Lawrence Neuman
Malcolm S. Jones, Jr.	W. H. Avery	Turner Joy	Joseph A. Oliva
Myron H. Nordquist	A. Y. Brin	John Justus	Ribent A. Rimmenman
Fernando Perez	B. C. Chan	George Kee	Gene Rosendahl
Paul Sutherland	Charles Cochran	Tomoniton Koike	Charles Sindt
Byron J. Washom	Chuck Cruit	Didi Lyon	Gene Solon
	Adam Englund	Beth Mather	Bob Stevens
	Henry Esterly	George H. Morgan	Mark Wilkin
	Steve Griffith	Albert A. Nakaji	

RECOMMENDATIONS

After much discussion and deliberation the following were endorsed by the majority present:

1. That DOE, MarAd, NOAA, and the OTEC community support the proposed amendments to the Energy Tax Act that would provide a supplemental 20% Investment Tax Credit for OTEC.

2. That the Congress, DOE, MarAd, NOAA, and the OTEC community recognize and support the concept that for risk averse industries an effective means of raising capital is through loan guarantees. The possible methods of achieving these are:

   a. Amendments to the Merchant Marine Act of 1936

   b. Amendments to the proposed Synfuels Legislation

   c. Legislation specifically drafted for OTEC funding

3. That the DOE FY 81 budget include specific line item funding provisions for:

   a. A 10/100 MW_e floating pilot plant with cable to be constructed as soon as technically feasible

   b. A 40 MW/125 TPD OTEC ammonia plantship

   c. A technical development plan leading to a 100-200 MW_e pilot plant in the Gulf of Mexico in 1988-1990

DISCUSSION FROM FLOOR

D. Hurvey, Brown & Root: How do you propose to implement the recommendations?

D. Guild, Stone and Webster: Well, if you are asking me to find the bucks, I am sorry, I am broke today.

B. Douglass, TRW: I noticed that your group recommended a 10/100 MW_e pilot plant in the FY-80/81 time frame, and a 100-200 MW_e plant in the Gulf in the late 80's. I know that we have considered significantly the interests of the investor community but to a lot of us here from the Engineer Builder Committee that timing would seem to imply significant spiking of the effort, i.e., there would be an intensive effort on the prototype followed by a long period of no activity on Gulf-based plants prior to the initiation of the 100-200 MW_e plant. Did your group consider that aspect of the schedule?

D. Guild: No, I think you may have misinterpreted what was on the slide. We said to include in the FY-80/81 budget specific funding for items 3a and 3b but also provide a dynamic plan so that you can have in place in 1988-1990 the third recommendation.

R. Douglass: Could you amend your recommendation to say one or some plants in the late 1980's?

D. Guild: I can't unilaterally. It was a group opinion. The question came up about the implementation of these recommendations. I had a very limited amount of time to speak. I believe the Ocean Energy Council is one way of implementing some of the recommendations. We urge all of you here to join the Ocean Energy Council. If you have not received one of the blue pamphlets, please see me or one of the other people involved.

E. Francis, JHU/APL: I just got a copy of the 12-page release on the President's message establishing a new program of emphasis in solar energy. And other than the fact that he did not specifically mention any solar energy development programs or any amounts of money for ocean thermal energy, I would like to point out that there are only four lines in the whole 12 pages that touch on OTEC. "The oceans are another potential source of solar energy. We will pursue research and development efforts directed toward ocean thermal energy conversion and other aspects such as the use of salinity gradients, waves and ocean currents." My question is, is there anything we can do to get the people who staff these papers to remember that ocean thermal energy is there?

D. Guild: I believe the answer is yes. You heard in Bennett Miller's presentation at the beginning of this conference that one of the difficulties that this technology has is establishing credibility in the minds of government policy makers. To the extent that individuals and organizations will take the effort to reach appropriate representatives both in the executive branch and the congressional branch to deliver that message, the program will be helped overall.

2. PLENARY SESSION AND TALKS

WELCOMING REMARKS

Bennett Miller

Program Director for Solar, Geothermal, Electric and Storage Systems
U.S. Department of Energy
Washington, D.C. 20545

I am delighted to be here to make a few remarks to open this 6th OTEC Conference. I want to tell you a little bit about my view of OTEC, where I think the technology is now, where I think it is going, because I do get involved in some of the discussions at DOE and I do have a fair chance to put my two cents in. I hope that my views will generate some discussion and that a few of you will be able to collar me and tell me where I have gone astray, what I need to know, and help get me on the right path. Last year this group met in Miami. Since that time there have been a number of very important developments. We have OTEC-1 in the shipyard, and within a year we will have it at sea. As far as I am concerned, OTEC-1 is a key component of the program, essential to everything that follows. In addition, an industrial consortium and the State of Hawaii have put together a small version of an at-sea platform--Mini-OTEC. That is a very exciting development and is another important component in the overall OTEC program.

There have also been a number of advancements on the scientific front, particularly with respect to heat exchanger development, biofouling control, undersea cable development, and cold-water pipe dynamics, which have put OTEC on a very sound technical foundation.

But there are two problems that continue to nag the OTEC program, and I want to talk about them, because unless this group understands them, we will continue to make technical progress and essentially never get the program off and running. First is the question of credibility of the OTEC option. What I mean by that is that those of us who were trained in engineering disciplines learned that most energy-producing systems operate on large temperature differences--500-1000°F. Every degree added or subtracted was a dollar lost or gained. When you discuss a system that must operate on a 40°F temperature difference, you lose touch with people's experience, and they become skeptical. An anecdote that comes out of my personal experience involved Bob Cohen, when he got the OTEC program going at the National Science Foundation. He and I served on a review team at a University in upstate New York which had nothing to do with OTEC. I had a great deal of respect for Bob and still do, although there was this little glitch: on the plane back I asked him what he was doing. He said he was working on a power system that takes cold water from 3000-ft depth, pumps it to the top to condense ammonia, and takes the warm surface water to boil ammonia, and produces power. I was certain that Bob had flipped. I did not say anything to him, but I later told some of my colleagues at the Atomic Energy Commission. They were sure he had flipped, too. And that attitude, I assure you, still lives. What people like Bob and you have done for me in particular, and I can cite countless other examples, is to educate us, to convince us that we must rethink the conventional wisdom about the effective and economic use of a temperature difference to produce power. If you go outside your own councils, you will get the same incredulity that Bob got when he talked to me seven years ago. I urge you to get out and tell people about OTEC. I do not mean to oversell it; that does not do anybody any good. But, in my mind, virtually all of the technical components are now proven in terms of a viable OTEC system. It does not mean OTEC is going to compete instantaneously in the market place. But I believe that over the past two years we have made a great deal of progress, and the technology exists to push that progress forward to the point where we have a large-scale systems at sea producing significant amounts of energy. I urge you to try to get that story to people. It needs to be told.

The second problem has to do with the product that OTEC systems produce. Fundamentally, they are electricity producers, and that is a problem for two reasons. Number one is that electricity is not the dominant problem in this country right now. The real problem is one of transportable fuels. This morning, I had to get up at 4:00 o'clock and sit in a line of cars for 3 1/2 hours until they opened the gas station and I could get gasoline. We don't have an electricity problem. We have a liquid fuels problem, and it is occupying the minds of people. The growth of electricity in this country has slowed. Electricity is not a product that people are dying to get today. However, OTEC has a tremendous advantage in my mind in that it will produce electricity in a region of the country which is going to need it most over the next fifteen or twenty years--the Southeast. That part of the country has the lowest electricity reserve margin and the fastest electrical growth, so OTEC fits extremely well into that market. Sure, people here will argue with me that it has much broader applicaion. I grant you that it does, but in terms of the next 15-20 years, it is my own belief that it is an electricity option with a localized market. And the word "localized" has to be inserted into the presentations to people, because just electricity alone is not going to get people wildly excited.

You can produce electricity a lot of different ways and a lot more economically than by OTEC when you consider that it still is a technology that has yet to produce on a reasonably large scale.

Those two problems temper some of the very positive accomplishments over the past year. First, the credibility of using a $40^{\circ}F$ temperature difference to produce power; second, the nature of the product itself, electricity, because electricity is something you can get from lots of different options. I insert those comments here because I think this group is the key to disabusing people of those two particular myths. And you need to get out and tell that story.

After the next 15 years or so, anything can happen, and we all know that it is difficult enough to predict a year ahead or a couple of months ahead, so to try to predict even fifteen years ahead is presumptious. But that is my view.

How are we going to get OTEC going? It seems to me that we ought to target for some near term markets before we try to enter big electricity markets. What might those be? In my view, the island markets make a lot of sense. Hawaii, Puerto Rico, Micronesia, all of those island economies are essentially dependent exclusively on imported fuels. OTEC conceivably could make them independent. That would have an enormous impact, not in terms of energy savings for the United States, but in terms of visibility and proof that this technology is here to stay. So our strategy basically revolves around the development of OTEC for island markets to promote the idea of island independence and use of that demonstration and experience as a springboard for cracking the southeastern utility market in the 1990s. I have a feeling that this program can move more quickly than that. The budget has not been generous--I understand that--and I do not think it is going to be for another year. We are going to be in a very tight budget situation for 1980. I am not telling you anything that you do not know. All you have to do is read the newspapers. If OTEC-1 is as successful as I believe it will be, and if you couple that with the mini-OTEC experience that we will get over the next year or so, I think that within a year to a year and a half this program will be at a point where it can make a major claim for massive support.

In the meantime, what we have to do is look very hard at the things that lead to that jumping off point, at the kinds of things that you are working on. We need to keep our eye on the near term, to focus our attention there. I know that Bill Richards and his people in the Ocean Systems Program will do precisely that. We need to hear from you on your views. How close are we to the mark? The Federal Government is not a respository of all wisdom. We do not pretend to be. We have to make a number of value judgements and take a number of positions with respect to policy, and we will do that. It will be more or less sensible to the extent that we get more or less sensible input from you. I urge you to think through the program and to take the agressive posture that I think has been taken in the past.

I am very, very enthusiastic about OTEC. I think that within this next year and one-half we are going to see a major change in this program. I wish you a good conference.

THE NEED FOR OTEC

The Honorable Gerry E. Studds*

Chairman, Subcommittee on Oceanography
U. S. House of Representatives

I welcome this opportunity to be here with you briefly at the beginning of your 6th Annual Conference on Ocean Thermal Energy Conversion. I am not an expert on OTEC, and I shall not pretend to speak on the technical and engineering matters which are the subject of many of your detailed panels over the next few days. What I do want to do is to encourage you in your efforts to make OTEC an operational energy technology as soon as possible, and to let you know that there are those of us in the Congress of the United States who appreciate the work you are doing, and who will do what we can to support your efforts.

As a member of Congress, I hear constantly from constituents who are being hurt by the continually rising price of energy, and who are concerned about the adequacy of future energy supplies. And these are in fact things to be seriously worried about. No one at this point needs to be told about shortages of gasoline or diesel fuel. However, one fact which may not be fully realized is that the world price of oil during the last six months has risen at an annualized rate of approximately 40 percent. In addition, there is considerable conjecture that the oil producing countries may raise the price significantly more during the next few months.

What these rapidly rising prices for our traditional sources of energy mean is that alternative energy technologies such as OTEC are likely to become economically competitive with traditional sources much more quickly than was envisioned even a year or two ago. Thus, it may not be necessary to wait until all possible engineering and research improvements have been made to start building OTEC plants which can operate at a profit.

As I looked through the advance program for your panels at this conference, I was impressed by the degree to which the work on OTEC is being done by state and commonwealth governments, and by private industry. I am pleased to see that people are not waiting for the Federal Government to supply all the answers, and that they are going out and making investments of time, effort, and skill to determine some of the answers for themselves. The fact that OTEC technology is being developed in this way bodes well for its speedy transference into wide commercial application.

As Chairman of the Subcommittee on Oceanography of the House Committee on Merchant Marine and Fisheries, I have a special interest in the use of the oceans to provide renewable energy sources. I believe that OTEC has an extremely great potential as an energy technology, and that its commercial development and widespread employment will become a major energy source not only for the United States but many countries of the world.

During the next few months, my Subcommittee will hold hearings on all methods which have been proposed to use the ocean for production of energy. I have chosen OTEC as the subject of the first hearing, which will be held this Thursday morning. The purpose of the hearings will be to foster greater public awareness for the potential uses of the oceans to produce energy and energy intensive products, and to find ways in which the Congress may assist through legislation the solution of any institutional or legal problems which need to be solved. As you encounter such problems, I hope you will make us aware of them and also of any actions which we may be able to take to help solve them.

I intend to do all I can to bring OTEC to the attention of my colleagues in the Congress and of the general public. I urge you to do all you can to make OTEC a functioning, operational technology as quickly as possible.

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*This paper was presented on Representative Studds' behalf by Richard D. Norling, Staff Director, Subcommittee on Oceanography.

ENERGY FROM THE OCEANS – LESSONS FROM THE NORTH SEA

by John Derrington

Sir Robert McAlpine & Sons, London, U.K.

The United States and United Kingdom have many things in common, including a strong dependence on oil and gas for their energy supplies, an energy price that is probably too low, doubts over an increasing reliance on nuclear power and, of importance to this conference, a fast developing interest in harnessing the energy that can be recovered from the oceans arising from wave, wind, tide and thermal sources. They are also alleged to share a common language, a matter on which you may judge as you read this paper.

In 1964 it has been reported, a leading oil company geologist forecast that no resources of oil or gas under the North Sea could be exploited commercially. That geologist is unnamed - and probably now unemployed - for in the last 15 years the U.K. has become self-sufficient in oil and gas, and over 1.5 mbbd are being produced from the North Sea. Many major engineering obstacles have been overcome to achieve this, and I shall offer some comment on offshore construction work for OTEC plants based upon our experience during this initial period.

Our operations offshore have been in some of the world's most savage environmental conditions (the Ekofisk storage platform suffered three storms of 100 year return period severity in the first 20 months). Work has been carried out 100 to 150 miles from shore in water depths from 300 to 600 ft. It has required the building of offshore equipment, including barges, cranes, pipe-laying vessels and tugs of ever increasing size, and the development of advanced techniques for deep water drilling, underwater piling, heave-compensation, equipment for cranes, and other devices of considerable sophistication. Over 100 structures of varying size have been placed in the North Sea, and these include some of the largest steel structures yet made by man. Concrete gravity structures have been huge by any standard, with base areas 350 ft square and all up weights ranging around 500,000 tons, for installation in water depths of about 500 ft (Fig. 1). These have been constructed initially in basins onshore, and towed out for completion in sheltered waters close inshore. They have then been towed to a deepwater completion site for dunking - i.e. inclination, buoyancy and integrity tests - the deck structures and topside engineering modules have been added and primary hook-up carried out (Fig. 2). The latest mammoth - Statfiord B, has vital statistics as follows:

Water depth	475 ft
Base area	440 ft x 556 ft (hexagon)
Deck load	35,000 tons
Overall weight	about 750,000 tons

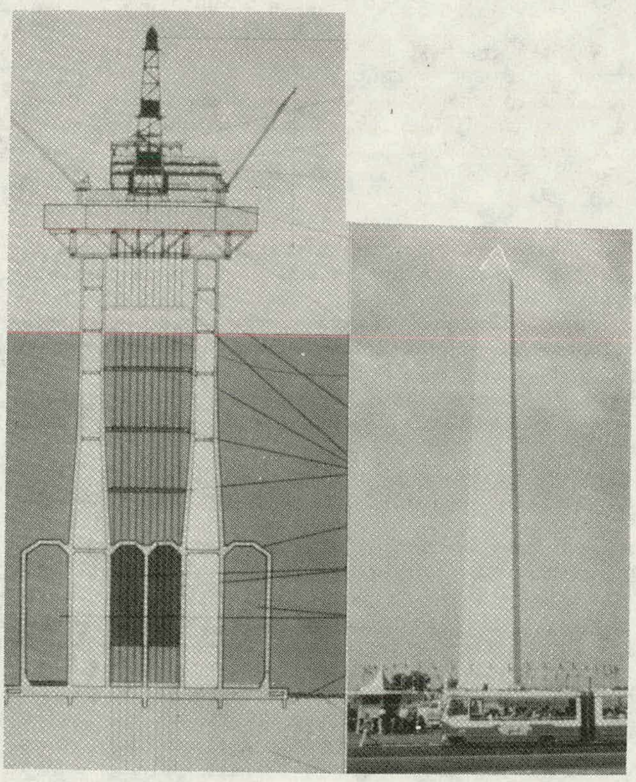

Fig. 1 Brent C Platform and the Washington Monument.

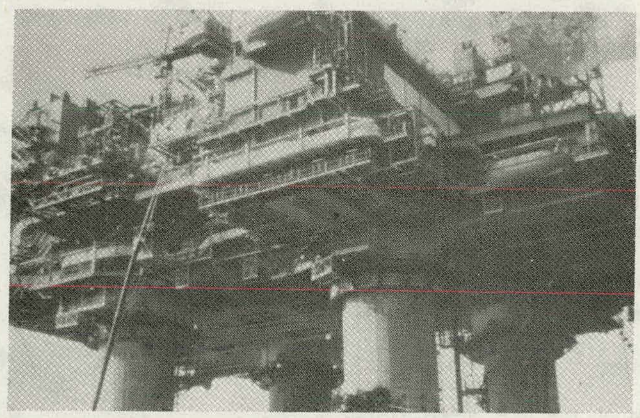

Fig. 2 Cormorant deck showing topside facilities.

The installation of these huge concrete structures has been carried out successfully using fleets of tugs with total power output approaching 100,000 IHP (Fig. 3). They have been placed accurately; in spite of doubts expressed during design, the foundation performance, under the extreme and variable weather conditions, has been as predicted.

We must conclude that, for bottom founded structures, the engineering problems have been satisfac-

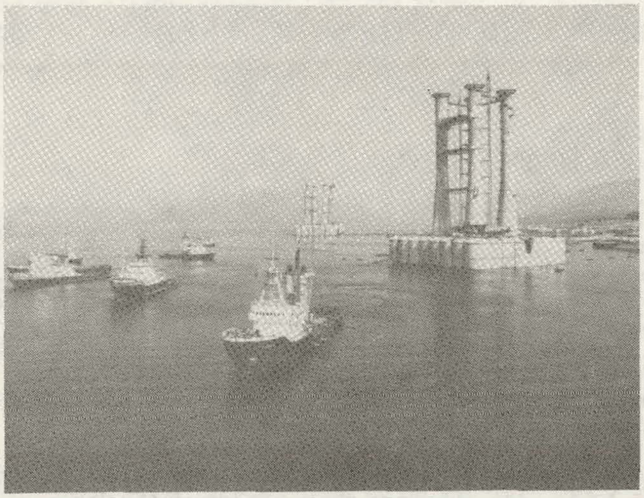

Fig. 3 Cormorant under tow.

torily solved, and that these monsters can be built to programme and within cost budgets within current engineering knowledge. The construction schedule for such a structure nowadays is for installation complete within 135 weeks of design commencing.

Our sights are now set on deeper waters where the problems of production have yet to be solved - depths over 600 ft and reaching towards 3000 ft. Here two solutions are favoured - floating structures and seabed containments. The floating structures, in our case moored over the reservoir, require consideration of buoyancy, seakeeping, high-pressure risers from the seabed, and mooring anchorages, and many of these subjects are directly relatable to OTEC structures.

The limitations of deck weight imposed by a floating solution are immediately apparent when the top-side facilities for large-scale oil production are needed. Currently floating structures with deck loadings of about 8000 tons are being considered, but when the load requirements for separation and primary treatment, gas re-injection into the reservoir, power generation, and living quarters are listed, the need for strict load limitation is apparent.

Larger hulls are possible, and floating structures of both steel and prestressed concrete have been successfully built and utilised. The ARCO barge, for the liquefaction and storage of propane, built in Tacoma, Washington for Indonesian waters, had a payload of 60,000 tons, and larger structures have been proposed for the Canadian Arctic. The responses of floating structures to the wave pattern have been the subject of considerable study, and the effect of the hull geometry, which does not follow the normal ship forms, can now be properly calculated. The advantages of a semi-submersible shape is well understood, and the mathematical means of correlating the design to the acceptable motions has now been established.

For a moored structure, the problems of anchorage can be considerable, and a proper stress history of the mooring tendons must be established. I believe the rules of the established ship classification societies for anchor tendon evaluation require updating to relate them to recently completed research. For a floating structure in the North Sea, the optimisation of deck load, hull size, wave response, and anchorage present a complex problem, as you have found for the OTEC plant concepts investigated.

Solutions for placing installations on the seabed with access by submersible vehicles are of interest in the U.K., and proposals for a complete production system (Fig. 4) in one-atmosphere containments in water depths up to 3000 ft have been developed and were presented recently to an EEC Conference in Luxembourg. Work at the U.S. Navy Civil Engineering Laboratory at Port Hueneme on concrete containments in the deep ocean has been of great value in this development.

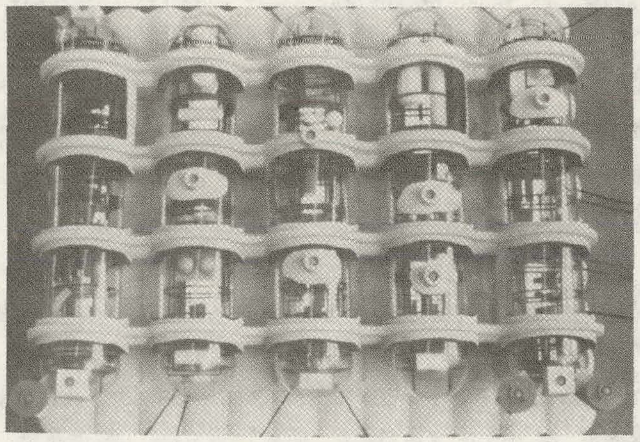

Fig. 4 Seabed oil production system.

An acceptable means for dealing safely with the disposal of high level radioactive nuclear waste on the seabed can also follow these principles.

Work in the U.K. on the benign energy sources in the oceans has concentrated, in the absence of deep and warm waters, on wave, tide and wind. These have all shown high costs relative to nuclear and fossil fuels. It has also been proved that hull or structural costs determine the overall budget of the system.

The extent to which alternative energy resources may be developed in the U.K. is reported in our Watt Committee Report No. 4. It may be seen that we have suitable conditions for wave energy stations, particularly off the Atlantic coast of Scotland, whereas high ground generally in Scotland and Wales is allocated to aerogeneration; the Severn Estuary in the Southwest is one of the world's prime sites for tidal power.

Dealing first with wind power, a full-scale prototype of a 50-m-dia., horizontal-axis aerogenerator has been commissioned. Although high ground could probably provide wind power at a reasonable cost, the favoured locations are offshore, for environmental reasons, possibly in the shallow waters of the Wash. Here, aerogenerators can be mounted on towers based on concrete caissons and coupled with pumped water storage to increase the system flexibility. However, wind power can only contribute 2 to 3% of the U.K. demand and so is of limited interest, although costs are reasonable and efficiency is high.

Wave energy systems can be argued to provide a significant proportion of U.K. requirements. Some years of government funded R & D work have been completed on several devices to date, the most promising being (a) Salter nodding ducks, (b) Cockerall articulated rafts, (c) National Engineering Laboratory oscillating water column, and (d) Hydraulic Research Laboratory rectifier.

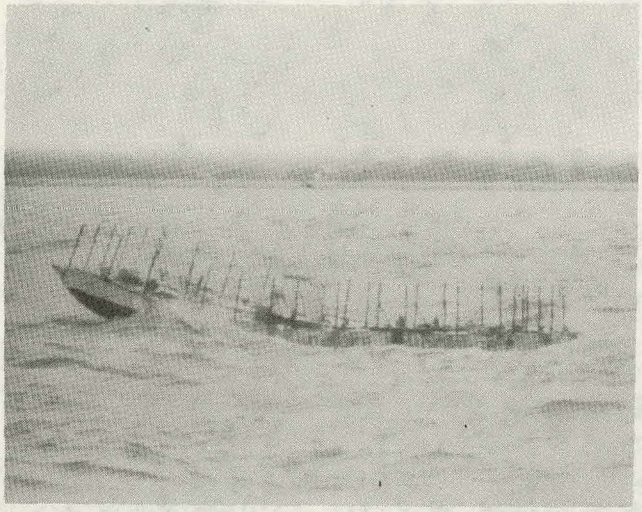

Fig. 5 Cockerell Raft 1/10 scale prototype.

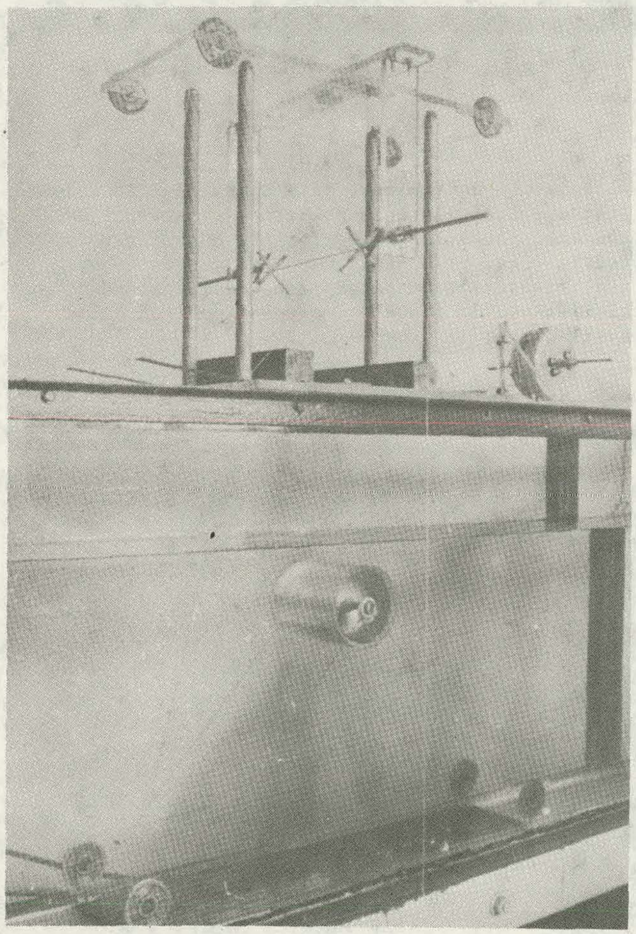

Fig. 6 Bristol University's wave energy device.

Although feasible, present designs would supply power to the National Grid at between 10 and 20 times the cost of nuclear power. Salter's system (not shown) and Cockerell's system (Fig. 5) have had 1/10th-scale trials and survived storms in excess of force 10. In normal 10 ft waves, for which the devices are tuned, the energy available is about 15 kW per ft of length, which can increase to 600 kW/ft in extreme storm conditions. This 40 to 1 ratio of working/extreme condition, the high structural cost, and integration of the mechanical and electrical work remain problem areas. The civil engineering costs are the major part of the budget, and so two new devices with low mass have been brought into the programme in search of a cheaper solution. The Bristol University submerged cylinder device (Fig. 6) shows some promise.

In Britain, the most attractive renewable energy source is tidal power from the Severn Estuary with a 12.0 m tidal range. Tidal power is totally predictable, can be used with known and accepted technology, and, using the lessons of North Sea offshore works, has been shown to be achievable within a contract schedule of 6 years for the civil works with a further 3 years for completion of the M & E supplies. Construction would consist of concrete caissons, built in three inshore basins, and towed to a fitting out facility for installation of turbines, generators, and switchgear. The small caisson which the Russians built as a prototype at Kislogubskaya in 1969 (Fig. 7) was 118 x 60 x 50 ft in size and weighed 5200 tons. It was concluded that "the float-in technique opens broad prospects of making these plants competitive with conventional plant." The technique is being used for forward projections on a 5000 MW Severn Barrage, but the problem is mainly political, which must be common worldwide for any capital investment of about 8 billion dollars.

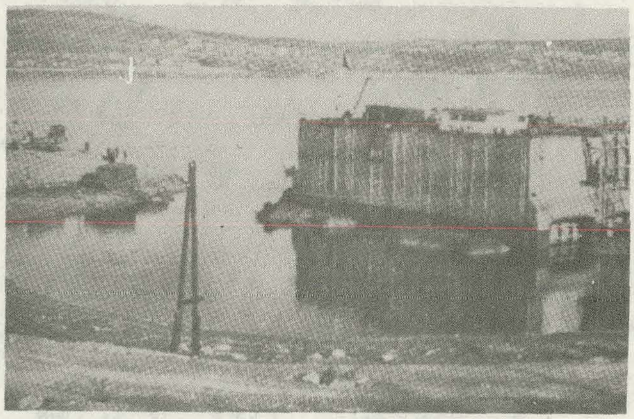

Fig. 7 Kislogubskaya caisson.

Preliminary studies are also planned for the use of North Sea techniques for undersea coal production. In the U.K., coal mines extend about 4 miles from the shore, coal quality is good and labour highly productive but haulage is expensive, and above all, ventilation is difficult. Artificial islands for improving ventilation, or for a complete offshore facility for coal production processing and export are being actively studied and look readily achieveable.

Britain has no aspirations for use of ocean thermal energy conversion, so I turn with some trepidation to the subject in the presence of so many experts. However, in referring to the use of North Sea expertise I know, as a contractor, that the knowledge of what mistakes to avoid is probably of more value than general guidance. I must apologise for the fact that with only a few days notice of this lecture, some of the data on U.S. plans for OTEC structures is outdated.

The achievement of a viable OTEC plant appears to require solutions for the following matters:

1. Suitable construction facility

2. Materials technology - hull
   - cold water pipes

3. Wave response and its effect on hull design

4. Cold water pipe installation

5. Installation of mechanical and electrical plant

6. Installation on location

In all of these, spin-off from current offshore engineering practice can be of value, not least of which is the need at an early stage to bring in the likely contractor or contractors, so that the hull design may be fully integrated with the engineering systems and the final budget and building schedule optimised.

The structural work is not formidable when considered against one medium size North Sea Platform - Cormorant 'A' and the engineering systems which that required (Fig. 8). The logistic problems of material supply are small, and provided the mechanical systems are modularised, the installation and hook-up work at the construction site can be minimised.

It will be assumed that the construction procedure follows well established lines:

1. Building a base tray about 50 ft high in a shoreside basin,

2. Flooding of the basin and removal of the sea-wall,

3. Tow-out of the tray to a mooring in deep, sheltered water close inshore, and

4. Completion of the structure and installation of the M & E systems at the mooring.

The selection of a site with these capabilities, adjacent to an area with an established infrastructure, is not easy and changes to an existing environment can be bitterly contested by various pressure groups, practically anywhere in the world today. The time taken within a fixed construction schedule to obtain the necessary approvals should not be underestimated, and the establishment of the site facilities and basin construction should then be complete in 10 to 12 weeks and ready for work to start. This may well overlap the initial engineering design and therefore is not critical to the programme.

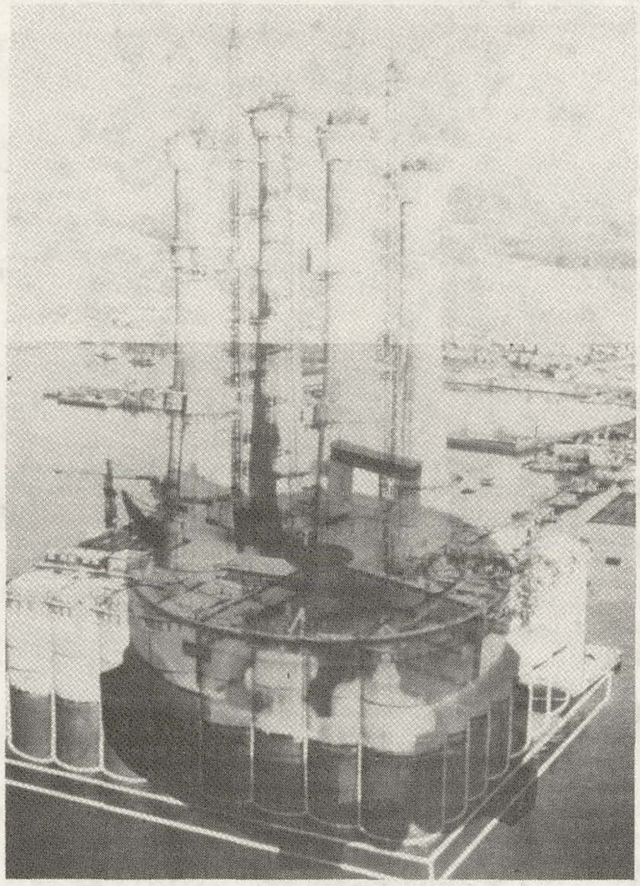

Fig. 8 OTEC/Cormorant comparison.

It may be sufficient, if a commercial is allowed, to indicate the existence in Europe of several well-established sites, at least for the building of a prototype (Fig. 9). At these sites the necessary capital investment has been made, moorings and other facilities are complete and a ready pool of labour is available.

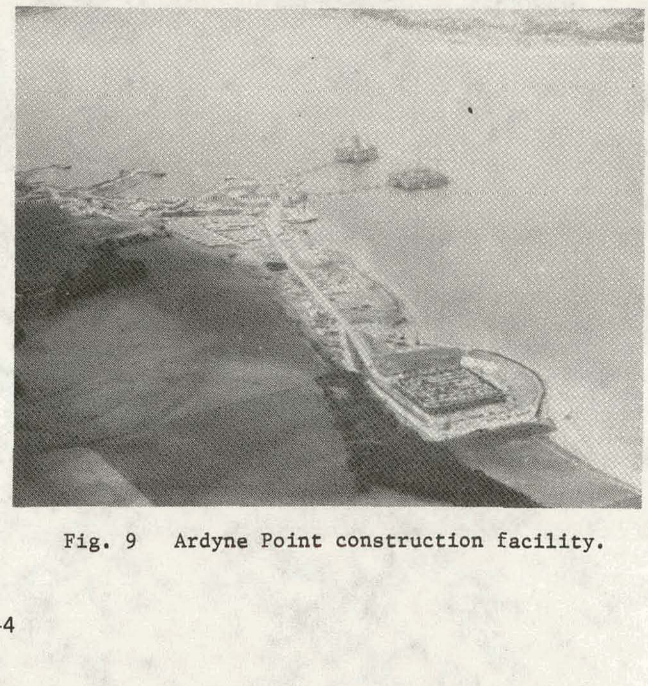

Fig. 9 Ardyne Point construction facility.

Durability and freedom from costly maintenance is of prime importance in a structure with the required life of an OTEC plant, and my natural preference for a concrete hull may already have begun to show. Concrete, either reinforced or prestressed, with a proven record of durability, has always been accepted as the natural material for the construction of offshore works for reasons of safety, economy, and durability and obviates the need for regular dry-docking and survey. Even so, the environment in which the structures remain, throughout their lives, is permanently hostile to the reinforcing bars and post-tensioning tendons, and the detailed design must be made with durability in mind.

Although considerable experience exists of the satisfactory performance of marine structures and concrete ships constructed without the benefits of modern concrete technology, concern for the durability has led to increased thicknesses of cover to the reinforcement being specified in the rules of the various ship classifying societies and national standards. F.I.P. proposals suggest 3 and 4 in. for passive reinforcement and post-stressing tendons respectively for sections over 20 in. thick in zones suffering the greatest exposure. Ship Classification Society rules propose comparable figures, but with high quality workmanship, it is possible to reduce these requirements considerably.

The use of special cements to resist the attack of chemicals present in seawater has also attracted interest, but the main requirement is a densely compacted concrete, containing good quality aggregates, a minimum cement content of 600 lb/yd^3 and a water/cement ratio not exceeding the range 0.40 to 0.42. The use of sulphate-resisting cements has been found unnecessary in the North Sea, and recent work has shown that the C3A content of the cement has little relevance to the durability of high-strength concrete. The partial replacement of cement by fly-ash or other pozzolanic materials has been satisfactory, although the reduction of the setting heat of the concrete by this means has little relevance in the structural sections normally chosen for offshore structures.

The selection of aggregates for concrete sea structures is of considerable importance in the mix design, not only from the viewpoint of adequate concrete strength, but also from consideration of concrete weight, and careful attention to the under-keel clearance during tow-out is essential.

Concrete strengths obtained with North Sea structures were originally specified in the 6000 to 7000 psi range, but in practice, much higher strengths were achieved. At Ardyne Point, average 28-day strengths were 9800 psi with a standard deviation of 530 psi. For work on seabed containments, designs have been based on 13,000 psi - three times the normal structural concrete strength, and a reduced materials safety factor is possible because of the high standards of workmanship achieved. The use of very high-grade concrete is particularly relevant to the cold water pipe construction.

The development of hull design can now be based upon recent experimental and mathematic work to limit the response under the extreme wave conditions. The response of a cylindrical hull can be reduced to less than half by changing the hull shapes to resemble a semi-submersible, and optimisation of the float design can further improve the seakeeping behaviour. Operational sites offshore Hawaii, Puerto Rico, Gulf of Mexico, or Brazil do not suffer the 100-ft-plus waves of the North Sea, but the structure cannot run before a storm and must be designed accordingly. However, sensitive parts of the machinery can be installed on specially damped foundations, although the bending forces in the cold water pipe assembly must be accurately predicted. Work at Glasgow University, funded by the British Department of Energy, has contributed considerably to the understanding and mathematical analysis of semi-submersible hulls of novel shape.

The construction and installation of the cold water pipe appears the sole aspect of the OTEC proposal, which lies outside current North Sea technology. Plans for building the pipe of concrete are based on two alternative proposals:

1. Slipforming the pipe downwards from the floating platform.

2. Prefabrication onshore and float-out for assembly and jointing offshore.

The slipform proposal is favoured from the operational viewpoint, due to the high cost of any offshore operation but the 4-in. thickness of pipe shell and the constructional accuracy needed on such a delicate structure would require very careful operation and bad weather could well delay the work. During construction, near neutral buoyancy would be maintained by positively controlled ballasting, with removable internal diaphragm seals in the pipe. Jointing of the pipe would probably be necessary for structural reasons, to allow for breaks in the slipforming operation and to provide for prestressing longitudinally in stages.

Alternative proposals for prefabrication onshore raise no new problems, apart from those discussed above, of wall thickness and tolerances. Joints would need great accuracy and steel moulds would be essential. The pipe casting operation should be straightforward, although a form of gantry similar to that used to load the 4-in. sprayed concrete roof shells for Cormorant (Fig. 10) would be needed. These shells were 40 ft square in plan, 4 in. thick and weighed up to 90 tons each. The tow to the installation site, either suitably sealed and ballasted or on barges, would not raise any new problems. The use of a large pipe laying barge similar to those used in the North Sea would simplify the installation, but a period of calm weather would be essential.

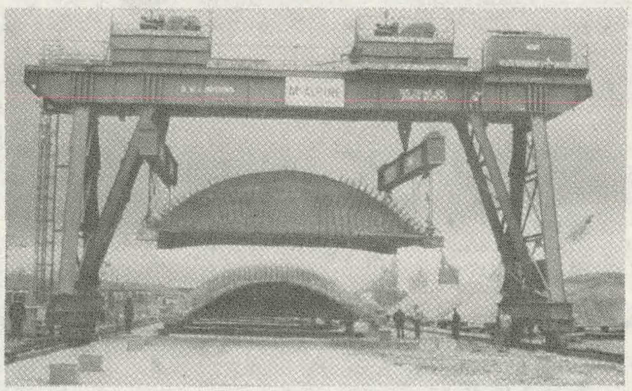

Fig. 10 Cormorant roof shells.

Connections to the OTEC hull would need diver assistance at reasonable depths, but the deep saturation diving techniques used in the North Sea would be unnecessary.

So, although the cold water pipe problem is one that would for preference be referred elsewhere, the technique for construction, handling, transport, and installation can all be based on current technology.

Turning to the mechanical and electrical systems, I speak here without any background of low-pressure ammonia-driven turbines and can refer only to the installation and hookup. The North Sea has taught us the high cost of offshore working with even a small platform requiring over 1 million man hours of work costing 35 to 50 dollars per man hour. Detailed engineering of the plant packages and comprehensive prefabrication for installation in sheltered waters are essential to successful completion. Initial reluctance to use large floating cranes and the tendency to use small 20-ton units for apparent cost reasons was counterproductive. Today, the offshore contractor relies considerably on larger lifting tools. Even with the installation of large plant packages - and these require detailed consideration at planning stage - there is always too much detailed engineering work left for fitters and electricians impacting seriously on cost estimates. So the 100-120 ton lift becomes standard practice in the construction yard, and there is increased reliance on workshop fabrication. The scope of equipment for a 100,000 bbd production platform entails a total engineering installation of 25,000 tons of pipework, compressors, generators, pumps, etc. and may give useful guidance in assessing the engineering tasks of an OTEC facility.

Some valuable work has been completed recently in the U.K. in developing the umbilical systems for electrical supplies to seabed installations, using a multi-core supply cable of 50 MW capacity, and these systems may have set a precedent for the electrical take-off from an OTEC plant. To the simple civil engineer, however, the hull calls out for a considerable size increase which would not only increase economy but improve seakeeping characteristics. An increase of 15,000 to 20,000 tons to the payload would require only a few feet extra draught and enable the installation to have on board an industrial facility which would use the power generated, reduce the demand on land utilisation and hopefully even bring a smile to the dedicated environmentalist.

The engineering of platforms for the North Sea grew from a simple extrapolation of know-how developed for the shallow water gas fields in the Gulf of Mexico in 1964 to what I believe is called an altogether different ballgame in the vicious environment of the East Shetlands Basin. It has taught us that the construction of an OTEC plant requires detailed engineering, high-quality workmanship and materials, extensive prefabrication, adequate planning, and a little luck. Structural costs of all offshore energy systems are high and full and complete optimisation is necessary for economy - not ad hoc optimisation of individual systems. Contractor involvement is, in my opinion, and I am a contractor, an essential part of the design.

An OTEC plant is similar in material and labour demands to a smallish offshore platform - say TP1 in the Frigg field (Fig. 11). The order to commence design was received at the end of January 1974 and it was placed in position exactly 164 ft from another

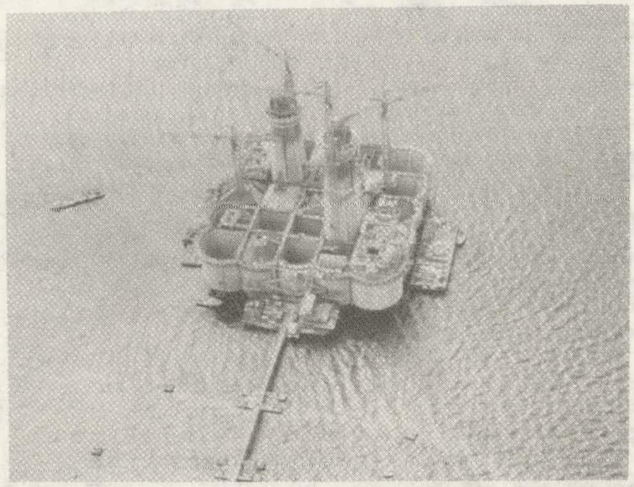

Fig. 11 TP1 under construction.

structure (Fig. 12) in 330 ft of water over 100 miles from land about 120 weeks and 30 million dollars later. At that period we were learning all about offshore engineering.

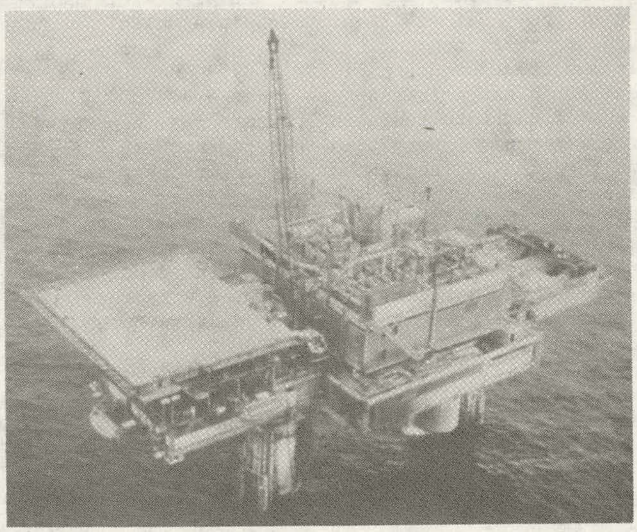

Fig. 12 TP1 in place.

And it is, in my opinion, the practical problems of offshore construction that impose the greatest obstacle to achieving an economic success on a project within the programmed time. So, I must conclude, gentlemen, by exhorting you to proceed with a prototype construction, before all the technical problems have been optimised on the drawing board, in the workshop and in the laboratory. In that way, many of the theoretical difficulties may be seen in their proper perspective, and the OTEC plants will be demonstrated as a viable source of renewable energy.

THE U. S. OCEAN ENERGY SYSTEMS PROGRAM

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Abstract

The oceans store vast amounts of solar energy, enough energy to make sizable contributions to the energy needs of the U.S. as well as those of other coastal nations. For the United States, four ocean-energy technologies offer significant promise: Ocean Thermal Energy Conversion (OTEC) (using heated surface waters), wave power, ocean currents, and salinity gradients. An overview of the DOE program for these four technologies is presented in terms of plans, concept descriptions, projected market penetration, and potential institutional barriers to implementation. The OTEC program presently receives about 95% of the total ocean energy system funds. Up to 2 Quads (2×10^{15} Btu/yr fuel equivalent or approximately 22 GW_e of average OTEC power output) is practically achievable by the year 2000, dependent only on the commercialization strategy employed after the demonstration plant.

1. Policy

The policy of the Ocean Systems Branch is to pursue programs to reduce national dependence on oil imports and other exhaustible fuels, in accordance with the National Energy Act.

2. Strategy

A three-step strategy in the development and implementation of the program is being used; the items are:

- Identify the available ocean energy resources.
- Assure technical feasibility and cost effectiveness of potential energy extraction and conversion techniques.
- Develop technology to induce industry participation leading to commercial use.

The ocean energy resources and present energy cost distributions indicate a multipronged program to effect the above strategy:

- Focus initially on markets subject to high-cost foreign fuels, interdiction and embargo such as islands (Hawaii, Puerto Rico, Guam, etc.) and the military.
- Perform further component and material research and development directed at cost reduction to penetrate the U.S. mainland market in the 1990's.
- Increase the capacity to cost-effectively distribute energy generated at sea to a larger U.S. continental area by use of energy-intensive products and development of hybrid techniques.
- Encourage and support U.S. industrial technical leadership for domestic and export production and distribution.
- Maintain close coordination with ocean-energy R&D programs of other nations.

The Department of Energy is pursuing four main technology areas for extracting energy from the ocean:

- Ocean Thermal Energy Conversion (OTEC)
- Salinity Gradients
- Waves
- Currents

Figure 1 depicts these areas and indicates the potential energy impact and the general developmental status. Of the four main technology areas, OTEC is the most advanced, and a preliminary design for a demonstration plant can be reasonably initiated in FY 1981, with subsequent construction and deployment by the end of 1985. The other three technologies require further research, technology, and systems-definition efforts to determine the more promising candidate systems for commercial-scale conceptual design.[1a] The closed-cycle OTEC system is usually designed to operate at an annual average temperature difference of greater than 37°F between the warm surface water and the cold water available from depths of 2,500 to 4,000 feet. A

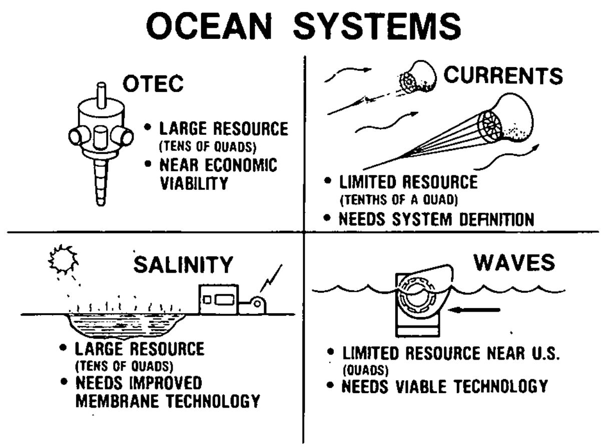

FIGURE 1. OCEAN SYSTEMS TECHNOLOGY AREAS

thermal resource map has been developed from archival surface records and subsurface data.[1b] Site-specific data are being obtained from satellite imagery, surveys, and site data. The effects on temperature profiles produced by weather and currents at specific sites are being investigated.

3. System Descriptions

3.1 Ocean Thermal Energy Conversion (OTEC)

The OTEC concept uses the temperature difference between warm surface water (80°F) and cold deep water (40°F at 3,000 feet) to run a heat engine and to generate electricity.

A fundamental advantage of OTEC over other solar energy systems is the constant availability of the resource (day and night, all year). The small available temperature difference (ΔT) results in large hardware, particularly the heat exchangers. Like other solar energy technologies, OTEC requires no fuel, so the costs of large hardware are countered by zero fuel costs. OTEC research and development has focused mainly on identifying practical ways of reducing cost. Presently four power cycles are under evaluation. Each cycle is believed to offer some cost reducing feature.

Closed Cycle. In the closed cycle a working fluid, ammonia (or possibly a Freon-type refrigerant) is used in a Rankine cycle consisting of an evaporator, a turbogenerator, a condenser and a pressurizer (feed pump). Warm surface water is pumped into the evaporator where subcooled ammonia liquid is turned into vapor. The exhaust of the turbine is wet ammonia vapor. The ammonia is condensed in a surface condenser which receives its cooling water through a deep cold-water pipe. The condensed ammonia is pressurized before it is again fed into the evaporator. To date, the closed cycle with ammonia as the working fluid has been emphasized by the OTEC R&D community (e.g., Refs. 1c-i). The DOE program has been designed (see, e.g., Ref. 1j) on the assumption that the closed cycle has the highest probability of achieving early economic viability.

Open Cycle. The simple open or Claude cycle is the forerunner of the closed cycle. It derives its name from the attempts of Georges Claude to operate an OTEC system using sea water as the working fluid. Warm sea-water is deaerated and passed into a flash evaporation chamber, where a fraction of the sea water is converted into low-pressure steam. The steam is passed through a turbine which extracts energy from it and then exited into a condenser. The condensate need not be returned to the evaporator as in the case of the closed cycle. Rather, if a surface condenser is used, the output is desalinated water. Or, if a spray, direct-contact condenser is used, the condensate is mixed with the cooling water and discharged back into the ocean.[1g-i]

A variant of the open cycle concept under evaluation is the steam lift approach.[2] In this concept, the power cycle requires a hydraulic turbine instead of the vapor turbine of the Claude cycle. The warm surface water is allowed to evaporate in such a way that it entraps substantial volumes of water with it. The condenser is located above the evaporator and it receives its cold water from the cold-water pipe. Thus, because of the pressure gradient created by the temperature difference, the vapor-liquid mixture rises from the evaporator into the condenser.

The potential energy of the warm water raised to the condenser level can then be converted into shaft power through the use of a hydraulic turbine located at the "same" elevation as the warm water intake. By rearranging the components, the plant can be submerged or floating.

Two approaches are being pursued. The first requires the addition of a surfactant to the warm water to cause foaming.[1h] The foam rises up to a condenser preceded by a foam separator. The other relies on creating a mist.[1i] In either case, the liquid is lifted upward by the vapor flow.

Hybrid Cycle. A concept combining the closed and the open cycles, the so-called hybrid cycle, has been proposed. In this concept, steam is generated by flash evaporation as in the Claude cycle. This steam then acts as the heat source for a "conventional" closed Rankine cycle using ammonia or some other working fluid.[1g]

3.2 Salinity Gradient

Theoretically, power from salinity gradients can be extracted in several ways. The two methods currently under study utilize the energy potential that exists across a selective membrane between two solutions of different salinity. The first method, osmosis, is physical and makes direct use of the flow that occurs across the membrane that permits water but not sodium, chlorine or other ions to pass through. Thus two water masses of two different salinities are required. Energy conversion is achieved by allowing the difference in hydrostatic pressure to build (because of the membrane). This difference in pressure is used to drive a hydroelectric turbine.

The second method, electrodialysis, is chemical and utilizes the electrical potential produced by combinations of selective membranes that permits one type of dissolved salt ion to pass through the membrane in one direction and the other in the opposite direction. Hence, a "battery" can be assembled that could produce electric power.

3.3 Waves

Wave energy systems extract mechanical energy from naturally occurring ocean waves and convert this energy into electricity.

3.4 Currents

These systems extract the kinetic energy available in the ocean current and convert it into electricity with a water turbine, much like a windmill. One method under study uses a rim-mounted, large-diameter, axial-flow hydroturbine. The turbine would be slightly buoyant, held below the ocean surface by tension moorings, and faced into the current by additional moorings.

4. OTEC Program Scope

The Ocean Systems Program funding is presently 95 percent OTEC, with 5 percent directed toward alternative energy sources such as salinity gradients, waves, and currents.

The OTEC Program consists of conceptual design studies, development and verification of engineering design tools, the acquisition of data through experimental test projects at laboratory and sea-based test facilities, and the resolution of legal, environmental, regulatory, and economic issues through trade-off analysis, modeling, and industry/user studies.[3-5] The program includes the evaluation of heat-exchanger performance, biofouling countermeasures, and environmental consequences at sea in the OTEC-1 test facility.[1k-m] The program is focussed on experimental verification of theoretical and numerical models, addressing principal technical unknowns during FY 1979 and FY 1980 and will initiate a preliminary pilot-plant design in FY 1981. Conceptual design studies have been completed for a 10-MW spar buoy, a 40-MW spar buoy, a 20-MW plant-ship, a moored 40-MW ship, a 10-MW landbased plant, and a 40-MW landbased plant. Conceptual and preliminary design studies on the cold-water pipe, power systems, electrical cable systems, and mooring and position keeping systems will be completed in FY 1979 and FY 1980.[1j] The major unknowns remaining which are applicable to the 40-MW pilot plant involve the systems motion and fatigue effects on the cold-water pipe and the electrical cable, and heat-exchanger cleaning effectiveness (see, e.g., Refs. 1n-q). Resolution of these unknowns and demonstration of the ability to effectively clean large-scale heat exchangers at sea shall be accomplished in FY 1980, prior to entering preliminary design.

5. Scope of the Alternative Ocean Program

The salinity-gradient program consists of basic and applied research and technology development to identify available ocean energy resources, identify potential energy extraction techniques, assess technical and economic feasibility, and to develop the technology to induce industry participation leading to commercial use. Two technologies are being pursued: (1) osmotic power and (2) dialytic power. Through FY 1979 dialytic battery experiments with a stratified solar pond will be conducted, Phase I of a preliminary design of a 50-KW osmotic power unit will be completed, and a cost analysis based on mathematical equations for a 20-MW dialytic power unit have already been completed. Preliminary results on the available and extractable resource as well as projected costs indicate salinity gradients are not as economically useful as the ocean thermal options.

The waves program has evaluated a number of wave energy extraction techniques which have been proposed. Limited model tests have been conducted both at sea and in the laboratory. These techniques which have been tested are in the categories of surface followers, pressure-activated devices and wave-focusing devices. The British wave-energy program has reported preliminary results which are unfavorable to the economic potential of wave-energy devices, particularly surface followers. The U.S. program is directed at devices which can focus the wave energy.

In FY 1980, experiments will have been conducted in wave tanks and comparative economic assessments will have been made between alternative wave-focusing approaches. On the basis of these results, a pilot experiment will be planned and initiated in FY 1981. Tests on a uniquely configured wave turbine will be continued; it is to be tested in FY 1980 as part of an international program with the Japanese, English and Canadians.

Within the U.S. currents program, a limited number of current energy extraction devices have been proposed and evaluated. These devices are in two basic categories: (1) rotary and (2) linear. A limited feasibility study has been completed on a rotary device, referred to in the literature as the Coriolis Turbine.

In FY 1980, a systems evaluation study will be completed which will consider all candidate techniques, and further design of the Coriolis turbine will be conducted.

6. OTEC Market Penetration

The preceding considerations suggest an "island strategy" is an attractive option for OTEC commercialization, whereby relatively small OTEC commercial plants can be introduced into U.S. island markets at an early date--in the course of the national program of OTEC research, development and demonstration. A projection of this possible evolution is shown as an "entering wedge" in Figure 2, which indicates, on a cumulative basis, a market penetration in the U.S. islands of several gigawatts installed by 1995 and about 10 gigawatts by 2014.

This figure shows the U.S. Gulf-Coast OTEC electricity market commencing in about 1990. Commercial production of OTEC ammonia for fertilizers could also begin by 1990. The figure further shows the possible commencement of OTEC market penetration on an international basis, starting in the 1990's. Such a "market pull" introduction of OTEC technology will allow time for the development of the submarine electrical-cable technology required for subsequent OTEC penetration of the Gulf-Coast market. The modularity and size flexibility of OTEC power plants are other factors conducive to commencing commercialization through such an island scenario, along with the relative simplicity (legal and institutional) of operation within territorial waters. Alternatively, strong interest is being shown by ammonia producers in cost sharing in the 40-MWe pilot plant to demonstrate the grazing ammonia-plant option. Cash-flow studies indicate that with suitable financing, OTEC ammonia plant ships (after 1985) could pay off equity in 4 to 5 years--returning 15 to 20 percent (compounded) on the investment. (See, e.g., Refs. 1r-t.)

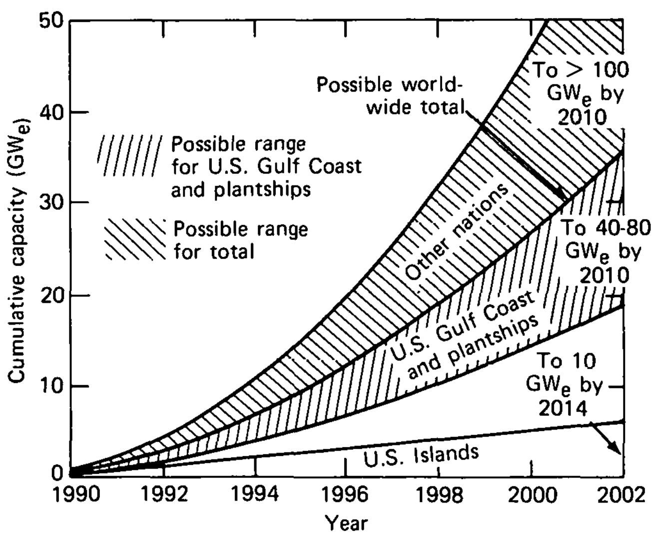

Fig. 2 Cumulative OTEC market penetration.

7. Institutional Barriers for OTEC

There are several forms of institutional constraints that might inhibit the acceptability and commercialization of OTEC. These barriers take the forms of competitive options, acceptable demonstration, financial incentives, regulatory and legal factors.[1r-v] The competitive elements have been mentioned above but need further refinement and updating. The remaining potential barriers need further study. The OTEC Commercialization Activity Plan shown in Figure 3 has been developed to help resolve and reduce the impact of the barriers and to determine the cost, targets, financial-incentives required, and demonstration requirements.

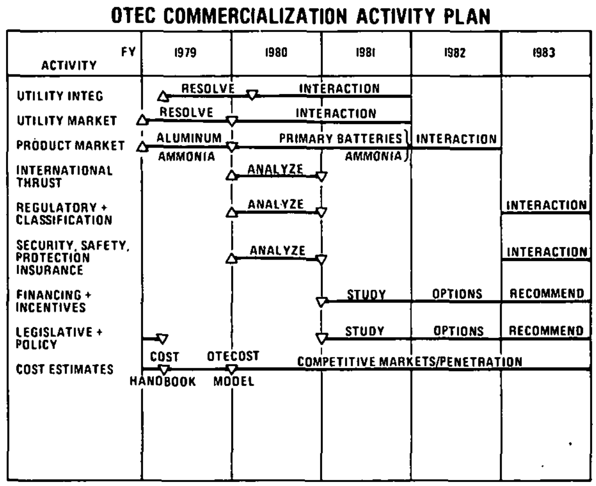

Fig. 3 The OTEC commercialization activity plan.

The OTEC pilot-plant size-requirement is of particular importance. From a technical-feasibility and scaling standpoint, a 40-MW hull and ship with an initial 10-MWe generating capacity appears to be adequate for introducing the technology. The final pilot-plant size will be defined at the end of FY 1979. That a demonstration is required by the utility/user is known today.

8. OTEC Program Schedule

The recommended schedule for this program is determined by industrial, university, and Government cost projections, which show that using today's subsystem technologies, OTEC can be cost competitive in the island environment 5 years before (1985) a commercial prototype can be fielded (1990) without crisis management or high risk of cost overrun. The total research and development and systems-acquisition costs for either accelerated or decelerated programs fall within ten percent of each other, but the options differ significantly in risk of cost overrun and amounts of OPEC-oil displaced by the year 2000. It should be noted that although the Domestic Policy Review projects 0.1 Quad (10^{14} Btu/yr, or 3.34 GW_t of fuel equivalent, or approximately 1.1 GW_e of OTEC average power output) by the year 2000, up to 2.0 Quad (2×10^{15} Btu/yr ≈ 22 GW_e of average OTEC power output) by the year 2000 is practically achievable, dependent only on the commercialization strategy employed after the demonstration plant. It should be further noted that this increased contribution incurs no increase in risk. The principal commercialization problem in OTEC is one of cash flow.

The current budget planning includes funding for the design, construction, and operation of an OTEC closed-cycle pilot plant by FY 85. This plant is vital to the OTEC program. Without the pilot plant, building of demonstration plants and subsequent commercialization are not likely. There are three factors which favor the inclusion of a pilot plant in the OTEC system acquisition strategy:

- Scaling from a small size (0.2-1 MW) to demonstration plant hardware (100-400 MW) would increase risk to a point where it could not be quantitatively calculated, nor is likely to be risked even by industrial venture capital--even if return on investment were ten to eleven percent per year.
- A pilot plant would establish the market credibility of the concept among potential users and other significant participants (e.g., banks, insurance companies, and venture-capital groups).
- A pilot plant would generate practical experience in plant operation and grid interaction phenomena for potential users.

The scaling factor centers on demonstrating the ability to maintain an acceptable level of biofouling resistance and thermal performance of large heat exchangers with thousands of passages during at-sea operation. The OTEC-1 plan includes testing of heat exchangers at the 1-MWe level. However, the scaling of heat exchangers from the small (0.2-1 MW) OTEC-1 test prototype to those sizes required for a demonstration plant and eventual commercial systems (approximately 200-400 MW) must be bridged by a pilot plant of approximately 10 to 40 MW to certify performance. The same reasoning also applies to the cold-water pipe deployment.

The second factor, "market credibility," follows from the needs to demonstrate reliability, survivability, and capacity in a relevant-sized power plant. The proposed pilot plant will test a complete power system of the same size and configuration that a small commercial OTEC plant might use as its modular building block. Cost, performance, and reliability data on an actual net-power generating plant can be produced. The pilot plant could provide the information necessary for capital investors to project the economic benefits of OTEC, including return-on-investment. Certainly, the two largest cost-uncertainty factors, i.e., systems-integration cost and O&M cost, could be estimated with much less ignorance.

Finally, the pilot plant provides the opportunity to transmit electricity to an existing grid. This feature will help to resolve two key issues of any new power system--reliability and capacity factor. With proper planning, there is a reasonable probability that some utilities or private investors would propose to the Government a financial plan that reduces the total Federal outlay for the pilot plant. Strong interest has already been expressed by Puerto Rico, Hawaii, and Guam in connecting the pilot plant to their respective utility grids and in incorporating OTEC into plant acquisition planning, as well as by an ammonia consortium in the intensive-product/grazing-plant mode of operation.

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DISCUSSION

L. Green, J. Ray McDermott: It seems to me that OTEC is putting a great number of its eggs in one basket by designing things for benign conditions. Why aren't you having people design for hurricanes? The point is being able to get power to the U.S. mainland.

W. Richards: Let me refer your question to Bill Sherwood, who manages our ocean engineering program.

W. Sherwood: I do not quite understand the question. We are designing for the hurricanes in the Gulf, Puerto Rico, and the worst 100-year storm conditions in Hawaii.

L. Green: I would say a 6-kt current or something like that would be a hurricane current. I don't believe that cold water pipes are being designed that are strong enough to withstand a 6-kt current. A 6-kt current would be a storm current for Gulf Coast conditions.

W. Sherwood: To get into knots of current and wind, let me refer you to Lloyd Lewis who has the environmental program in DOE. It is our intent that a structure built in the ocean has to withstand all the conditions that exist there; anything else is a loser.

L. Lewis: We have worked out environmental studies to support the design effort, and these studies are being updated as more information becomes available, but in essence, we are designing to the combination of the conditions of most probable hurricane, wind, wave, and currents. Charlie Bretschneider of the University of Hawaii has been supporting us in this effort. He is certainly familiar with conditions in the Gulf as well as in the Islands, and we have put together, I believe, a package that represents the conditions that would occur, at most, probably in a 100-year hurricane. These do include surface currents that are, in some sites, 4 kts or better.

Now you have to separate the wind current and the orbital water particle motion under a wave; Charlie has been trying to do that. I think he has been relatively successful, given the state of the art. We are, in fact, projecting most-probable 100-year hurricanes.

L. Green: However, many of the platform studies end with a statement such as, "the mooring system really has not been looked at," etc. Station-keeping would be a very important part of a hurricane situation. Structural integrity is also important part, too. It would seem that current studies are leading to situations where, at best, the cold water pipe would break off; at worst, the whole floater is going to come ashore. It seems to me that we are not squarely addressing the worst-case conditions.

W. Sherwood: You mean if we use the guyed tower, we will be fine! [See Pharr's paper on the subject in this volume.] I am just kidding you. Seriously, the best way to beat the ocean is not to get exposed out there in the first place, and if we can build something up from the ocean bottom, we may be better off; some day I think we will, really I do. I don't know if we are there yet in size of a commercial OTEC plant, but we are getting awfully close. A well was drilled quite recently to 4800 ft; the Cognac Tower and some other structures are at least half-way there. I truly believe that when OTEC is a viable industry, we will see OTEC plants built from the bottom of the ocean but, in the meantime it is our intent to design for the worst hurricane condition. We have to use the same criteria that the oil towers and the other structures in the Gulf of Mexico use.

REMARKS AT THE 6th OTEC CONFERENCE LUNCHEON

Edwin M. Hood

Board Chairman and President
Shipbuilders Council of America
Washington, D.C.
June 19, 1979

More than a year ago, Ed MacCutcheon invited me to say a few words at this Plenary Luncheon of the Sixth Conference on Ocean Thermal Energy Conversion (OTEC), and I have to say that in the meantime I have learned much about OTEC.

In a gathering of experts representing a broad spectrum of scientific and engineering disciplines, such as this, I would not presume to discuss the technical feasibility of ocean thermal energy, but I believe I can comment on the role which shipyards might have in bringing the imaginative OTEC concept to fruition.

As many of you may know, U.S. shipbuilders have pioneered almost every technologically advanced marine system ever developed: offshore drill ships, jack up rigs, liquefied natural gas ships, nuclear powered ships, concrete barges, submarines and semi-submersibles. The capabilities resident in U.S. shipyards and the U.S. marine component industries are equal, in my judgment, to meeting the technical requirements of OTEC.

Shipyards are strategically situated to build, assemble and launch large, heavy structures for use at sea such as envisioned with OTEC concepts. At least five U.S. shipyards, to my knowledge, are actively interested in this kind of work, as major subcontractors. One yard is already committed to the conversion of a Navy tanker which has been made available to the Department of Energy for the OTEC-1 project.

Moreover, the financial community has experience with financing ship projects under the Federal Title XI loan guarantee program administered by the U.S. Maritime Administration. Drill ships and semi-submersible drill rigs have already qualified for Title XI financing. The same kind of financing for economically feasible OTEC vessel projects could provide a needed stimulus to accelerate and establish acceptance of OTEC as an important energy source.

With typical American ingenuity, we sincerely believe that full scale commercially viable projects can be put on line through government/industry cooperation in the near future. And, the U.S. shipyard industry wants to be a part of this national endeavor. I can assure you, we need the work.

REMARKS AT THE 6th OTEC CONFERENCE LUNCHEON

Charles Matthews

President

National Ocean Industries Association
Washington, D.C.
June 19, 1979

Good afternoon. I am Charles Matthews, President of the National Ocean Industries Association, a Washington-based trade association representing some 400 companies involved in virtually every aspect of ocean resource development. NOIA is pleased to co-sponsor this 6th Annual OTEC Conference, particularly since several of our members are directly involved in various aspects of the Conference. Participating members include Lockheed, Westinghouse, TRW, Dow, Tracor Marine, Tetra Tech, Hydronautics, J. Ray McDermott, Teledyne, and Frederic R. Harris, Inc. I'm pleased to have this opportunity to welcome our members and each of the other organizations represented here, and I'd also like to take a moment to congratulate Lockheed and another NOIA member, Dillingham Corporation, for their recent success in the "mini-OTEC" project in Hawaii. Other members I haven't named will almost surely be involved in the future development of OTEC, since our members include companies with proven capabilities in heavy construction for ocean applications, shipbuilding and modifications, instrumentation, financial and management services, and an array of other service and support and supply functions.

It goes without saying that one of our greatest concerns for the future of the nation, and of the world for that matter, is the critical need for energy. Sources of energy used most commonly now and in the past, while they will continue to be available for perhaps many years yet, aren't renewable, and will eventually be exhausted or depleted to a degree which will make it absolutely necessary to find new sources. OTEC is an exciting new technology which holds great promise as one source of the energy we will need in the future, particularly since it is renewable and pollution-free. It is very appropriate that emphasis is increasingly being placed on development of renewable energy resources such as OTEC now, as the need is plainly becoming critical.

The level of attention new energy technologies are gaining internationally suggests to me that attractive new markets are beginning to exist for companies with the ability to develop those technologies, at least if the international commercial climate remains suitable. In this regard, we have been following with considerable interest and some trepidation the trend in certain international conferences such as the U.N. Conference on the Law of the Sea, toward the creation of provisions in new agreements that may require companies with the ability to develop and market advanced technologies to give access to those technologies rather freely and with only limited protections against misuse of proprietary information and trade secrets. It seems likely that the U.N. Conference on New and Renewable Sources of Energy planned for 1981, among others, will include provisions on technology transfer similar to those in current drafts of the law of the sea treaty text. I suggest that it may be appropriate for the business concerns involved to look carefully at the implications of this trend in international control of commercial development, and that perhaps our political representatives at the relevant international conferences should be made aware of the need to give a fair and just level of protection to private enterprise in the process of creating agreements which will affect much of our ability to compete effectively in world markets.

I won't take up too much of your time. Thank you for inviting me to be here today, and again welcome to Washington and to this important conference. NOIA members will be deeply involved in the future of OTEC in many ways, and we wholeheartedly endorse the efforts all of you are making to achieve commercial development of OTEC. We are here to support and serve America's ocean industries and enjoy the opportunity to get better acquainted with the people involved.

KEYNOTE ADDRESS AT THE 6th OTEC CONFERENCE LUNCHEON

Charles H. S. Eaton

Technical Consultant,
Committee on Science and Technology
U. S. House of Representatives

Good afternoon, Ladies and Gentlemen.

I would like to thank you for this opportunity to address the Sixth Annual Ocean Thermal Energy Conversion Conference this afternoon. These annual OTEC conferences have proven to be a valuable forum for the discussion of the latest developments in the Department of Energy OTEC Program.

As this conference is being held this week, the United States public is being made to realize the increasing severity of the Nation's -- indeed, the world's energy crisis. The gas lines which started a few months ago in California have spread to various parts of the country. The Nation's capitol has been hit with particular severity this summer.

If you ask the average citizen what the cause of the shortages is, he will most likely point a finger of blame at the Iranian situation, the oil companies, the government, or the OPEC Nations. If blame is to be given for the present situation, then each and every citizen of this country should shoulder some of it.

A few short years ago we received our first clear warning that the days of apparently limitless world petroleum resources were over. With the Arab Oil Embargo and the establishment of the OPEC Cartel, the industrialized Western World found itself dependent on a small number of oil exporting countries for a large proportion of its petroleum requirements. We found ourselves in a position of having the world price for oil being set and levels of production determined by the OPEC Cartel.

At the time of the oil embargo, the United States was importing approximately one-third of its oil supply. Since then, our oil imports have grown and our annual balance of payments deficit has increased to a dangerously high level. Now, almost six years later we are importing over 50% of our oil. The question to ask is not "Who is to blame?" -- but rather -- "What have we done to reduce our dependency on foreign oil and why haven't these actions had any effect?"

As far as the Federal Government's role is concerned, there has been a great deal of effort on the part of the Congress and on the part of the last three Administrations to respond to these questions. Numerous laws have been passed over the last six years dealing with energy--including the passage of the National Energy Act last fall. Most of these laws deal with various approaches to conserving energy as a means of reducing our oil dependency. Clearly, we must take every feasible step we can to conserve energy, but this only buys us time. It does little in the way of shifting us away from oil and toward other energy sources. The reason there has been no clearly identifiable national energy policy, with the exception of conservation, is that there has been no national consensus on what energy supply policy to pursue. That lack of consensus has been dangerously coupled with the denial on the part of many to even acknowledge that there actually is an energy crisis confronting us. Many interests, whether they be parochial interest, special interests, or environmental interests, influence the consideration by Congress of any major energy legislation which focuses on the supply side of the energy equation. A good example of this was the consideration in 1976 of a major federal loan guarantee and price support program for the production of synthetic fuels. This legislation was defeated before it ever reached the House Floor when the rule for its consideration was defeated by a single vote. Now, three years later the subject of establishing a major program for the development of synthetic fuels is again being seriously debated in the halls of Congress.

Today, this picture of a lack of national consensus on energy is changing. This year's very real shortages of gasoline and the possibility of electrical brown-outs in some parts of the country has caused the American public to wake up to the fact that this Nation is in a deteriorating oil supply crisis.

For this Nation to steer through the energy shortages that the next decade will bring and to put in place the non-petroleum energy resources and technologies we need will require a firm commitment on the part of the citizens of this country and strong positive leadership from the energy policymakers in Washington, and in each and every state and locality. This commitment must entail efforts to conserve energy wherever possible and it must be coupled with a willingness on the part of the government to spend a much greater amount of money on energy research and development of all promising energy technologies. But until these new technologies are technically and economically viable, we will have to rely on energy from coal, nuclear, gas and petroleum.

I have had the opportunity for the past 3 years of being a member of the staff of the Committee on Science and Technology. In 1971, the Committee recognized the potential of future energy shortages. In that year, the Committee established a Task Force on Energy which was chaired by Congressman Mike McCormack of Washington. The Task Force was the first attempt on the part of the Congress to thoroughly review the Nation's energy picture, and make conclusions and recommendations on needed Federal actions. In its report in 1972, the Task Force recognized the potential contribution that could be made by renewable energy sources when it recommended among six other key points, that:

> "Because of (solar energy's) continuous and virtually inexhaustible nature, solar energy research and development should receive greatly increased funding. Near-term applications of solar power for household uses seem likely, and central station terrestrial solar power and satellite solar power are attractive long-term possibilities."

At that time, funding for solar energy research, development and demonstration in the Federal Government was less than $2 million per year and scattered through various Federal agencies ranging from NASA to the National Science Foundation.

Early in 1973, the Committee initiated the Solar Heating and Cooling Research, Development, and Demonstration Act which became law in 1974. It established the solar heating demonstration program which has been quite a success, and under which we now have thousands of solar units on residences for space heating and hot water. The program also has hundreds of industrial facilities and commercial and public buildings on solar energy for process heat, space heating and hot water--and in a few cases--solar energy used for cooling.

Unfortunately, an economically competitive solar powered air conditioner for individual houses has not been developed, and the solar cooling demonstration program is behind schedule.

One of the most encouraging aspects of the solar demonstration program is that it has stimulated the establishment of a strong solar heating industry with several hundred industrial corporations now producing solar equipment. Even more importantly, the private sales of solar collectors and related equipment last year were more than 10 times the amount produced through all Federal programs combined. This program, when combined with the tax credits on solar equipment which became law last year, has established a strong and viable solar heating industry.

The other major piece of solar research and development legislation which also originated in the Science and Technology Committee became law in 1974, was the Solar Energy Research, Development and Demonstration Act of 1974. This act brought together solar research activity from various federal agencies, establishing the federal solar program as we now know it. When the Energy Research and Development Administration was established by Congress these research activities were transferred to the new agency.

Under the direction of ERDA, and now, the Department of Energy, solar research activity and funding has grown astronomically. The Congress has been instrumental in this growth. Each year since the establishment of ERDA, Congress has significantly increased funding well above the budget requests of first, a Republican Administration and now, a Democratic Administration.

The solar energy budget for all solar energy research development and demonstration--which was reported from the House Committee on Science and Technology only a few weeks ago totals $544 million; with an additional $57 million for bioconversion programs--$601 million in all. Of this total--$35 million is for the Ocean Thermal Energy Conversion Program with a full authorization for the OTEC-1 Test Facility.

This leads me to the subject of the OTEC technology and programs which you are discussing at this annual conference.

OTEC is one of the few solar energy options that can produce continuous rather than intermittent electricity. The concept of converting the energy stored in the thermal gradients of tropical oceans into useful energy was first produced by the French Physicist d'Arsonval in 1881 and the technical feasibility of OTEC was partially demonstrated by Claude in 1930.

The Federal effort on the development on OTEC began in earnest in 1972 when the funding for this activity was just $85,000. In the fiscal year 1980 budget this figure is now $35 million. To date, the federal government has funded approximately $130 million for OTEC.

What have we gotten for this $130 million investment to date and where is the DOE OTEC program and the technology going in the future? The DOE OTEC program is still in the component design and evaluation stage. Conceptual designs for commercial plants have been produced by several of the DOE contractors. However, no total system design has been carried out in which all the major subsystems have been integrated.

The resource that OTEC potentially could tap is vast. If you look at just the resource associated with the application of OTEC for the off-shore production of electricity which would be transmitted to shore by submarine cable in the Gulf of Mexico, estimates range from 200 to 1,000 gigawatts of electricity. There would also be additional resources directly accessible to tropical islands such as Hawaii, Puerto Rico and Guam. When you look at the application of OTEC for the production of energy intensive products with plant site not limited to nearby off-shore locations, the resource becomes almost limitless.

The Department of Energy has cautiously estimated that there will be operational, small, commercial scale OTEC plants, (40 to 100 Mwe) a size suitable for island utility application, in the early 1990's. Also, that there would be commercial large plants (250 to 400 MW_e) suitable for United States' mainland application in the mid to late 1990's. OTEC plants suitable for the onboard production of energy-intensive fuels, chemicals or materials will also be deployed.

The OTEC program has been quite successful in addressing the technical uncertainties which have confronted the program. The heat exchangers have constituted the major source of uncertainty in terms of producibility of large hardware, heat transfer performance and materials. The heat exchangers are the most critical components of the OTEC power system from the perspective of cost and performance. Depending on which of the proposed designs ultimately gets chosen, the heat exchangers may account for approximately 30% to 50% of the OTEC plant costs. Much progress has been made in this area over the past two years. Tests at Argonne National Lab on the 1 MW_t (or 30 kW_e) heat exchanger components of enhanced-surface-shell-and-tube, plate-fin, and folded-tube types have confirmed earlier single tube data. Also, biofouling rates, that have been measured, show that the rate is manageable and much lower in the open ocean than in coastal regions and that the rate is fairly site-independent. It has also been determined that titanium and aluminum heat exchangers can be cleaned with periodic brushing. With continued testing at Argonne, the seacoast test facility, and on the OTEC-1 when it becomes operational in 1980, the program can pursue more advanced heat exchangers such as a plate type heat exchanger as well as collecting data over a long period of time which will add to our understanding of the operation of heat exchangers in an ocean environment.

There are several engineering questions that still must be addressed by the program. Some of these can be answered in the deployment of the OTEC-1 test facility next year, but others can only be answered by the design and construction of a 10 MW_e modular experiment plant ship. These questions relate to the deployment, stress relief, dynamic loading and interconnection of the cold water pipe and OTEC platform. There are also questions relating to the mooring of such a large platform.

Environmental issues which still need to be addressed are those relating to the mixing of such large quantities of ocean water of differing temperatures. There is also some question as to the effects of the possible release of CO_2 from up-welled deep ocean water.

There is also uncertainty about the potential cost of an OTEC plant. Cost estimates based upon the most recent design studies generally range between $1,500 to $2,500 per kilowatt of electricity delivered to shore. Reasonable cost of power estimates fall between 30 to 70 mills per KWh for a commercial OTEC plant.

This leads us to the next phase in the program. This next phase will be the design and construction of at least one 10 MW_e modular experiment plant. Presently, the DOE program has the goal of achieving a technical and economic demonstration of a 10 MW_e scale facility in the mid-1980's.

When the technical progress made by the program is considered with the remaining technical engineering and environmental questions which only an operation pilot plant of sufficient size will answer, it is clear that we must move ahead in the near future with the modular experiment. Not just one modular experiment, which DOE is presently considering, but two modular experimental pilot plants. One for the production of electricity to be transmitted to shore and one for the production of energy intensive products.

I am not alone in this opinion. It was the opinion of the research development and demonstration panel of the Domestic Policy Review of Solar Energy that the "potential of OTEC cannot be denied on the basis of the analysis that exists to date and that a timely investment in at least one 10 MW_e modular experiment is the appropriate approach to maximize the economic impact of OTEC within the context of the entire solar program." The panel also stated that, "If successful, the 10 MW_e scale experiments will lead to commercial OTEC facilities that deliver 40 to 100 MW_e baseload electricity to tropical islands such as Hawaii, Puerto Rico, and Guam. Delay until the mid-1980's in committing to the first 10 MW_e modular experiment may lead to the dissipation of some of the industrial technical expertise that has been assembled in the OTEC program."

The construction of two 10 MW_e modular experiments will have a significant budgetary impact on the entire solar program. The cost estimates call for at least $100 million per modular experiment budgeted over a three year period. This potentially will be the largest undertaking ever taken by the solar program, but it is one that should be made to ascertain the technical and economic feasibility of large scale OTEC deployment. It will also give a clear indication as to the need of further federal involvement in the development of OTEC technology and how much of a contribution OTEC can make on this Nation's and the world's energy solutions. It will be an expensive step, but it is one that must be taken as well as those that should be taken in the areas of energy conservation, synthetic fuels production, nuclear power development, geothermal energy development, natural gas from unconventional sources and the other solar technologies. It will only be after we have successfully taken many of these steps that we will begin to see solutins to our energy problems.

Thank you.

INTRODUCTORY REMARKS AT THE 6TH OTEC CONFERENCE BANQUET

Robert Cohen

Ocean Systems Branch
Division of Central Solar Technology
U.S. Department of Energy
Washington, D.C. 20585

Welcome to this 6th OTEC Conference banquet. We have tried to choose a cross-section of the OTEC community to be our distinguished guests at the head table--including representatives of industries, utilities and other organizations interested in OTEC. Unfortunately, Jack Babbitt of the ammonia industry could not be here tonight, but we do have representatives of the aluminum industry, a utility, and some of the key organizations that have taken a great interest in OTEC power systems and plant designs.

At your right, Dr. Gordon Dugger of Applied Physics Lab of Johns Hopkins University, who organized this conference. Next, Eugene Barsness of Westinghouse; Mrs. Mary Louise Dugger; Bob Hindle of the General Electric Company; Tom Higgins of Lockheed Corporation; Louis Wilson of Middle South Services; and Bill Richards, Chief of DOE's Ocean Systems Branch. At the banquet of the 5th OTEC Conference in Miami in February 1978, we were privileged to hear Senator Matsunaga of Hawaii. He wanted to be with us this evening but could not; however, his astute assistant Tak Yoshihara, is here to represent him.

I'll skip two individuals [Governor Romero and Dr. Deutch] and introduce my colleague, José Cortez, who is here as personal representative of Dr. Bennett Miller, DOE Program Director for Solar, Geothermal,Electric and Storage Systems and José's distinguished wife, Raquel; and next, a member of the financial community from Lehman Brothers, Kuhn, Loeb: Neil Eisner. The aluminum industry is interested in OTEC as a source of electricity to produce aluminum. We have with us Capt. Malcolm Jones of Reynolds Metals. Next is my wife, Carolyn. Mrs. Dugger and she have greatly supported the OTEC program. And the TRW organization is represented tonight by Bob Douglass.

Dr. Gordon Dugger has been a meticulous organizer, both for this 6th OTEC Conference and for the 3rd OTEC Conference, which was held in Houston in 1975, just following the Offshore Technology Conference. Along with Gordon, I now recognize all the people on his team. Bud Francis arranged for the exhibits. We've never had exhibits at an OTEC Conference before, and, thanks to Bud's leadership, this unprecedented and valuable addition worked out very well. We have had help from two consultants to APL, Ed MacCutcheon, who chaired yesterday's luncheon--he has been great at enlisting all of the participating organizations, which is another first for this conference--and Joan Worden, who has beautifully handled our media relations. Bill Buchanan of APL, who also worked on media relations, is not here this evening, but his colleague, Phil Lewellen, who handled visual and audio arrangements, is present. Bill Kehoe was responsible for hotel arrangements, which have been, as far as I am concerned, impeccable. And finally, the secretarial and reception personnel, who normally work with Gordon and Bud, and have had an exceptional load for this conference--Jackie Hentgen and Joan Handiboe.

Next we turn to OTEC pioneers. As the institutional memory of this program, I will tell you about them briefly. I think the first one chronologically is Bryn Beorse, who is retired now but still working pretty hard at the University of California Seawater Conversion Laboratory. The work he did on desalinization was quite relevant to OTEC, especially to the open cycle work, and he is now a consultant to the OTEC alternate power systems program through the Oak Ridge National Laboratory.

A person most of you already know, J. Hilbert "Andy" Anderson, was another OTEC pioneer in the early 60's, together with his son, Jim. They head the company known as Sea Solar Power, Inc.

We are also honored to have with us the very distinguished scientist and OTEC pioneer who brought you the Zener Diode--I tell him he has gone from solid state to liquid state--Dr. Clarence Zener. Clarence was interested in OTEC in the early 60's when he was Chief Scientist at Westinghouse in Pittsburgh. He later went to Carnegie-Mellon University as University Professor, and is a real enthusiast for both closed-cycle OTEC and some of the advanced open-cycle variations. He has had working with him, for many of the recent years, Dr. Abe Lavi. Abe organized the First OTEC Conference held at Carnegie-Mellon six years ago this month, and he helped a great deal to organize the Fourth and Fifth OTEC Conferences in New Orleans and Miami, respectively. Abe also dedicated two years of his life to working at the Department of Energy as Power Systems Program Manager.

One pioneer who is not here tonight is Professor Bill Heronemus of the University of Massachusetts at Amherst. He and

Clarence Zener headed the two contractor teams studying OTEC systems who were on board when I came to the National Science Foundation.

And shortly after the contracts I mentioned began at NSF, Dr. Bill Avery and his colleagues at APL became very active in OTEC R&D, concentrating on the energy-intensive-product or plant-ship concept, especially for manufacturing ammonia.

The OTEC concept was invented a hundred years ago by a Frenchman, d'Arsonval. To represent the French government tonight, I recognize Philippe Marchand of CNEXO, which is the French counterpart of NOAA.

Bengt Lachmann, who heads a European consortium of about 10 nations, an industrial consortium known as EUROCEAN, which is working on OTEC, was at the Conference earlier this week.

We are pleased to have with us a large Japanese delegation of about 20 people. The senior member of that group I recognize now, Dr. Kamogawa.

Messrs. Marchand, Lachmann, and Kamogawa each described their respective OTEC development programs in papers presented yesterday morning. In all, we have 10 nations represented here, and we are very pleased by that, because the ocean thermal resource potential is large enough to serve many world energy needs, and the U.S. Department of Energy wishes to foster international cooperation in developing that resource.

There are numerous other notable individuals present from industry, academia, and government agencies. We are also pleased that a number of people from Capitol Hill have chosen to be with us tonight and on other occasions this week.

There is an experimental OTEC power system know as Mini-OTEC which is designed to produce 50 kilowatts gross power. It was scheduled to be operating now, and we had planned to announce it tonight. It is being completely funded by a consortium of private industry which includes Lockheed, Alfa-Laval, Rotoflow, Dillingham of Hawaii, and the State of Hawaii at a cost of about $3 million. Well, Mini-OTEC operation is delayed about a week, I'm told. We hope that those organizations will soon be announcing that it's working satisfactorily. We wish them success. [Editor's Note: Mini-OTEC did begin operation on Aug. 2, 1979.]

Finally, I shall introduce the DOE Program Managers in the OTEC area. Bill Richards has already been introduced. Our systems integrator, who handles putting the technology all together, is Sig Gronich. Abe Lavi's successor for power systems is Ken Read, who is returning this month to the Naval Academy in Annapolis. Bill Sherwood manages our ocean engineering and naval architecture program. Eugene Kinelski handles biofouling, corrosion, and materials. Carmine Castellano deals with projects such as OTEC-1. Lloyd Lewis handles environmental and resource assessment. And two excellent examples of interdivisional cooperation--members of the Electric Energy Systems Division--are Tom Garrity, who manages development of submarine power cables and electricity transmission systems, and Jeff Rumbaugh, who handles our relationships with electrical utility systems and technical integration into utility grids.

Now, by way of introducing the high DOE official with us, I note that Washington often looks for leadership to the Cambridge, Mass. area. Historically, we have a lot of people moving into government from Harvard, but there is another school in Cambridge. At the inception of the Department of Energy, our next speaker was called to Washington from MIT, where he was Chairman of the Chemistry Department. Previous to that he had been at Princeton University as a professor. He first served DOE in the key role of Director of the Office of Energy Research. Later, in recognition of his talents, he became Assistant Secretary for Energy Technology, and then was nominated to Congress as Under Secretary of Energy. Ladies and gentlemen, Dr. John Deutch.

REMARKS AT THE 6TH OTEC CONFERENCE BANQUET

John M. Deutch

Under Secretary
U.S. Department of Energy
Washington, D.C. 20585

It is a great pleasure to be here this evening at the 6th Annual OTEC Conference. It is a particularly propitious time because the President today dedicated a new solar hot water heating system on the White House roof and announced a major solar energy program that promises to be the largest and most ambitious program in renewable energy ever seen in this country. It will set a course and an example for the world. Today he set a goal for the percentage of our primary energy requirements that will come from renewable energy resources by the turn of the century--20 percent.

Everyone in the Department of Energy, and people who have studied energy problems, as you all have, will clearly recognize that this goal is a most ambitious one, and not a very easy one to achieve. It is going to require hard work, dedication, and a lot of vision from all parts of our society, not just the technical community, but also the business and financial communities, State and local governments, Congress and the Administration.

That hard work, that dedication, that vision are necessary if we're going to be truly independent of foreign sources of supply. There is no foreign cartel that can raise the price of solar energy, turn off the sun, or change the wind and the seas. Any of you in this room who have been in Washington, D.C. for any period of time have seen dramatic examples of our fragile dependence on overseas resources. We have gasoline lines in this town stretching around the blocks in every neighborhood, and the people are experiencing major uncertainties with regard to their daily life and work habits. It is this type of hold on us by foreign energy suppliers that we are going to break with our program for solar energy and renewable energy sources.

But I know that here I am preaching to the choir on this subject. You have been working on and discussing OTEC, a most important solar technology that is particularly well suited to the pursuit of U.S. energy independence. Our islands--Puerto Rico, Hawaii, and other tropical islands under U.S. control--are particularly attractive for early deployments. The potential for OTEC and other renewable resources for meeting the president's 20% goal is to me one of the most exciting aspects of our energy research and development efforts. The knowledge that we gain from our OTEC programs will contribute to our ability to move toward this goal.

Specifically, OTEC-1 will add substantially to our knowledge base and confirm our ability to maintain low levels of biofouling in thousands of sea water passages of heat exchangers. This demonstration is needed to establish our ability to design, build and operate units which will grow to a commercial scale. With the success of these experiments, we will look forward to having the information needed to make major new program decisions and to undertake detailed engineering design efforts for experiments large enough to be significant to prospective users.

We do not have a fixed path with respect to OTEC development. The Mini-OTEC and OTEC-1 experiences will provide scientific and engineering knowledge needed for systems that produce electricity as well as for systems that produce energy-intensive chemicals and products. The detailed design efforts that we plan to undertake will explore these various concepts.

All of this brings me to the point that the Department is optimistic about the future of OTEC. The challenge we now face with OTEC, as well as with other forms of solar energy, is to bring the technology to commercial reality at the lowest possible cost while demonstrating to the public its viability and utility in our system.

I am sure you are not here tonight to hear from a chemist. We are all honored to have with us a man who, more than anyone else, knows how vital it is to achieve energy independence; a man who has expressed his confidence in the ability of OTEC to become a commercial reality. He was mentioning to me here on the podium that Puerto Rico burns 60,000 barrels of petroleum per day for generating power, in addition to that used for transportation.

Going back to Bob Cohen's remarks, I would be remiss as a New Englander if I did not point out that the governor was educated in New England, I believe Phillips Exeter, and not at that other place, but at Yale. As a person who has taught at Princeton and at MIT, Yale is still, to my mind, better than the other place. We're delighted to have him with us, and it's my extreme pleasure to introduce the fifth elected governor of Puerto Rico, Governor Carlos Romero Barceló.

KEYNOTE ADDRESS AT THE 6th OTEC CONFERENCE BANQUET

by

The Honorable Carlos Romero-Barceló

Governor of Puerto Rico

Mister Chairman; distinguished participants in the 6th Ocean Thermal Energy Conversion (OTEC) Conference; esteemed representatives of the federal government; honored guests; ladies and gentlemen:

Allow me to begin by saying how pleased I am to be with you this evening. As proponents of rapid development of OTEC technology, the participants in this conference stand at the forefront of our nation's crusade for energy independence. And in that spirit, I am delighted to have this opportunity to meet with you, and to work alongside you to advance our cause.

Although the White House and Congress have, for several years, been talking about the need to move ahead swiftly to bring renewable energy resources on line, they have, for the most part, been doing very little about it. In the past few weeks, however, this Capital City has been afflicted with the frustration and irritation of long lines at the gasoline pumps. In that respect, I believe the timing of this conference may have been very fortuitous. We have reason to hope that the enormous personal inconvenience being experienced by almost everyone in this city will serve to rally added support to our cause, and lend additional weight to the arguments set forth by the many dedicated federal officials who share our concerns.

For at least the remainder of this century, and probably longer, petroleum is certain to remain a vitally important energy resource. But it also seems certain that it will remain expensive, scarce, and probably subject to ups and downs in availability, as the result of worldwide political circumstances. If for that reason alone, every effort must be made to replace petroleum as an energy resource wherever feasible. Environmental considerations, as well as the inevitability of the eventual exhaustion of non-renewable resources, oblige us to place a very high priority on clean, abundant alternatives: the sun, the wind, the sea, biomass, geothermal. By the year 2000, such alternatives can and should provide up to seven percent of United States power needs; by the year 2025, they may be able to yield as much as 25 percent of these requirements.

There is no question that one of the last great sources of untapped raw energy on earth is the oceans. Their tides, currents, waves, salinity, and temperature difrerences offer enormous energy potential.

Yet a great variety of other potential energy sources, systems, and technologies are vying for the public and private research and development dollar. As a consequence, it has been necessary to set priorities. Among ocean systems, ocean thermal energy conversion tops the priority list, due to the availability of ample, inexhaustible resources, and the advanced state of its attendant technology.

Currents, tides, and waves are limited resources, and presently lack viable technologies; salinity is abundant, but requires improved technology. Only OTEC combines an abundant resource with technology that is close to being economically viable.

The principal prospective applications of OTEC involve the production of base load electricity and the manufacture of energy-intensive products such as aluminum, chlorine, methane, and ammonia. In addition, OTEC could be employed in the development of a mariculture system.

Why is OTEC especially attractive to Puerto Rico? The answer is many-faceted. First and foremost, it can reduce our dependence on imported petroleum. In 1978, our expenditures for petroleum and petroleum products consumed within Puerto Rico exceeded one billion dollars in an economy whose gross domestic product came to about eight billion dollars. We currently rely almost entirely on imported oil for electricity generation. In 1978, out of the 120-million barrels we imported, we consumed over 24-million barrels for the generation of electricity. In that one year alone, this fuel for electricity generaton cost us more than 300 million dollars. Moreover, because the demand for electricity in Puerto Rico is expected to grow at a rate of 4 to 5 percent per year, that fuel bill could total about 600 million dollars by 1986 (and that estimate is predicated upon the assumption that crude oil prices will rise by only about 6.5% per year).*

Complicating our situation further is the fact that Puerto Rico is an island: this forces us to be completely self-sufficient in the generation of electricity. There is no way we can plug into someone else's power grid to tide us over in emergencies. On the contrary, we must have excess generating capacity in place and on stand-by at all times, to accommodate maintenance, unscheduled outages, and variations in demand. So it is that, among all the new sources of energy being developed, OTEC is clearly the most promising near-

*Editor's note: OPEC raised oil prices by about 50% shortly after the conference and another 10% in the fall.

term alternative for Puerto Rico, because it provides that most precious gift: a renewable base load generating capacity.

The second reason we look with such great interest to OTEC is simply that we have very few other realistic alternatives open to us. Puerto Rico has no known fossil fuel reserves--no petroleum, no natural gas, no coal. Neither do we have harvestable forests, nor any appreciable untapped sources of hydroelectric power. What we do have going for us, however, is an ideal geographic location. The waters around Puerto Rico are especially well-suited to the testing of OTEC's potential.

The temperature difference (ΔT) between the warm surface water and the cool deep water in our surrounding seas is about 22°C. Given the minimum ΔT of about 15°C required for OTEC operations, it is evident that Puerto Rico's waters represent one of the choice locations on earth for applying OTEC technology. Indeed, recent oceanographic studies have revealed that two proposed sites for sizable OTEC power plants in Puerto Rico--Punta Tuna on our southeastern coast, and Punta Vaca on our offshore island of Vieques--may be unsurpassed anywhere as prime OTEC test locations. Not only are the temperature differentials ideal, but they are found within 3 miles of the shoreline, with water depths of 3000 to 4000 feet. OTEC plants on these sites would, therefore, require undersea electrical cables of minimal length, which in turn would permit the use of alternating current for energy transmission. Clearly, then, until such time as direct current transmission cables can be developed, conditions such as those which prevail in Puerto Rico are, without doubt, the best available in the United States.

Third, in addition to being the location at which an OTEC modular experiment will be most economically viable, Puerto Rico is also the location where the need for such a project is greatest. Our unemployment rate has fallen substantially since I took office, but it nevertheless remains above 16 percent of the work force--more than twice the national average. Population per square mile is 15 times the national average. Per capita income is well below that of the poorest of the fifty states. To improve the economic well-being of our people--and thus to help us contribute our full share to the national economy--requires steadily increasing industrialization; and increased industrialization invariably implies increased electricity consumption.

I have already cited the financial hardships Puerto Rico endures as the result of our dependence upon petroleum-based electricity generation. In this regard, it is especially noteworthy that the likely optimum size of OTEC plants--around 400 megawatts--closely corresponds to the island's anticipated incremental base load power requirements.

Furthermore, a reliable, reasonably priced, local energy source would unquestionably have a positive effect on our general economy, as well as on our ability to attract and retain energy-intensive industry. Indeed, the development of indigenous energy supplies would represent a growth industry in itself. And, of course, OTEC development would also have a significant multiplier effect, creating thousands of jobs in other industries.

Given its highly promising long-term potential, where then does the national OTEC development program stand today? In spite of the indifference and resistance it has encountered in some quarters, a great deal has been accomplished. Many problems of design and application have been overcome. The basic technology for OTEC is at hand, and its operating principles are well-documented. For the most part, the remaining engineering challenges involve solutions to problems of scaling-up known construction techniques rather than of technical innovation.

The tests of biofouling rates conducted in Hawaii, Puerto Rico, and the U.S. Virgin Islands indicate that this will pose no major obstacle. Significant strides have been made with regard to the configuration and material problems of the heat exchanger. To be sure, material corrosion remains a troublesome design problem and more work is needed on the difficult problems of cold water pipe design, the mooring or station keeping of floating platforms, and the D.C. cable that will be required for higher capacity mainland plants located farther from shore. But the progress that has been made, through such initiatives as OTEC-1 and the Mini-OTEC project, and the absence of any major technological drawbacks, signal the need to shift emphasis once again--to move away from small-megawatt test platforms and to get on with the business of building a scalable working system that will deliver energy and allow us to gain systems integration and operational experience.

As a first step in making this shift, the Department of Energy is developing plans for a pilot plant project. In this regard, we feel there are two options which merit careful study.

The first is a small-scale, 40 MW(e) platform and cold water pipe, with a 10-MW(e) power module, that will provide early operational experience with an island utility, and offer the potential of penetrating a small but viable market in island communities. The United States market for a plant of this size, however, would be quite limited, because its subsequent adaptability to any large utility market on the mainland is highly uncertain, especially in view of the electric utility industry's present skepticism about the OTEC concept.

The second pilot plant option is a 100-

MW(e) platform and cold water pipe with a 10-MW(e) power module. This option has the potential to lower the cost of generation and to penetrate mainland utility markets requiring the large blocks of power presently provided by generators employing fossil and nuclear fuels.

The advantage of this latter option is that it will establish technical and economic viability in an operating environment and range which will be close to optimum size, while at the same time remaining flexible enough so that it can be scaled down to a size suitable for smaller island communities. It is this approach, then, which will most effectively serve to demonstrate OTEC's full potential: both in offshore areas and as a viable alternative for the large electric utility market on the mainland.

We in Puerto Rico are convinced that the larger 100-MW(e) modular experiment should, after careful consideration by DOE, be chosen as the next step in OTEC development. We also believe, regardless of which option is selected, that the pilot plant project, to be truly effective, must be undertaken with the active involvement of the utility industry. Ongoing involvement by the utilities will ensure OTEC's earliest possible commercial application, because once the economic and technical viability of OTEC-based power generation has been demonstrated to these end-users, we are confident that they will at last be firmly persuaded that the time has come to factor OTEC-based generation into their future plans and schedules.

Utility involvement will also ensure that the ongoing OTEC program carefully takes into account, at every stage, the needs and requirements of its end-users. Attention must be given to the utilities' acquisition criteria. An understanding of these criteria is essential: developers must comprehend the bases on which utilities evaluate designs and costs. These considerations will guide the utilities in determining whether or not they want to employ OTEC--and, if so, which system to select and which "purchase" arrangement is most appropriate--for example, outright purchase of the facility, or purchase of power from a third party, which in turn operates the facility.

We in Puerto Rico have already taken the initiative on the national level to get the utilities involved and committed. Our island's electricity production is entirely in the hands of our government-owned electric utility company. This utility, which investigated the OTEC concept as early as 1967, has been the lead utility in bringing together sixteen prime potential users, to form an OTEC Utility Users' Council. The Council's members represent utilities in Hawaii, Puerto Rico, and States in the southeast and along the Gulf Coast. These utilities, taken together, serve 18.5% of the U.S. population.

Because the Council's members have extensive operating responsibilities within their own utilities, their role will center on, first, providing advice to DOE to update the OTEC program plan from the users' standpoint, and, second, helping DOE in developing users' acquisition criteria.

Formed in March 1979, the Council will remain active at least through mid-1981, when the winners of the preliminary pilot plant design competition are selected. The Council's life, however, could be extended on through the demonstration phase, with its members participating as observers at the host utility.

In the final analysis, though, all the points I have mentioned--pressing need, available technology, identifiable markets, user participation--will not put OTEC into the utility grid without money. Money to carry the existing technology forward to commercialization and production.

Where will this money come from? So often--too often, I feel--the standard answer has been "the federal government." And indeed the federal government has increased its total OTEC budget. In fiscal year 1976, Congress appropriated $8 million, and has since increased that figure substantially, to the point where the fiscal 1978 appropriation totalled $35.3 million. But obviously, the federal government cannot be expected to shoulder the entire cost.

What then is the answer? We in Puerto Rico support the concept of risk-sharing by government and the private sector together. And we feel that risk-sharing arrangements should be based on the following general principles:

--the provision of sufficient incentives to attract the needed private capital;
--administrative simplicity and clarity, so that private investors and the government know exactly what risks they will run, and what rewards they can expect;
--a fair and equitable program, in which all parties share both the downside risks and the upside rewards;
--latitude for state and federal regulatory agencies to approve any needed rate increases;
--government participation limited to the first generation of production-sized plants using new technologies;
--and finally, provision for the withdrawal of government participation when the risks have been reduced to manageable levels.

Further, we support--indeed, advocate--matching federal government research and development funds with those of industry and state governments, to carry out energy development programs like OTEC. "Putting your money where your mouth is" commits industry and state governments to accelerating acceptance and commercialization by the private sector in a way that "standing by and waiting for success" never can.

With respect to OTEC Puerto Rico stands ready to enter into cost-sharing arrangements with the federal government. In addition to assuming a growing percentage of financial responsibility, we expect to provide site support and services in kind.

In conclusion, let me say--in all chauvinistic candor--that Puerto Rico expects to be the site of the next OTEC pilot plant. After all, we have the best test sites and we satisfy all the criteria:

--resource abundance;
--an industrial base capable of supporting the development;
--high consumer cost, due to fossil-fuel use;
--high percentage of imported fuels;
--and extensive support from state government.

We certainly do support OTEC. We took the initiative in forming the Users' Council. We provide an early market for the technology. We are willing to provide substantial financial support. And we have the mechanism, in our government-owned electric utility.

What do we need? Your moral support.

At the moment, OTEC confronts a certain inertia in Congress--and even within the Department of Energy--that might cripple development of this highly promising energy source. What we must do is to urge--no, to insist--that DOE lay out a clear program development path. And then we should push them, just as hard and as fast as we can, down that path. I am convinced that we should start the OTEC modular experiment now.

Today, Puerto Rico is far more heavily dependent upon federal transfer payments than any of us would like. More than half our population receives food stamps. Our fervent goal, as a people, is to pull our full weight as productive American citizens. If we are to achieve that goal--if all of our able-bodied workers are to be gainfully employed--the day must come when our island is blessed with abundant, reasonably priced energy. And we feel certain that OTEC can be instrumental in hastening the arrival of that day.

We need your support. By supporting us, you will be supporting your future and our nation's future.

Thank you very much.

3. OTEC PROGRAMS OF EUROCEAN, FRANCE, AND JAPAN

EUROCEAN OTEC PROJECT

Bengt A.P.L. Lachmann*

EUROCEAN

Association Européenne Océanique

Villa Richard Monaco-Ville

Principaute de MONACO

Abstract

This paper reviews the work that has been performed by the EUROCEAN OTEC group and the present state of development which has resulted in a proposal for a 10 MW_e floating OTEC (ocean thermal energy conversion) pilot plant based on the closed cycle. The site for the pilot plant has not been decided upon. This proposed programme runs over a period of 5 years and involves about 50 million US dollars. Governmental funds are required for this programme. The companies cooperating in the EUROCEAN OTEC group are: Alfa-Laval (S), Banque Européenne de Crédit (B), Hollandsche Beton Group (NL), Johnson Group (S), Kockums (S), Micoperi (I), Vattenbyggnadsbyrån (S).

Another EUROCEAN group has started a feasibility study of an aquaculture plant which is combined with a small land-based OTEC plant as the source of cold, nutrient-rich water, and a desalination plant.

Introduction

EUROCEAN is an association of European industrial companies from 9 countries: Alfa-Laval (S), Banco de Bilbao-Promotora de Recursos Naturales (E), Banque Européenne de Crédit (B), Boliden (S), Compagnie Française des Pétroles (F), Compagnie Générale d'Electricité/Alsthom-Atlantique (F), L.L.& N. De Meyer/Haecon (B), Den norske Creditbank (N), Fiat (I), Grånges (S), Hollandsche Beton Group (NL), Johnson Line-A. Johnson & Co. (S), Kockums (S), Micoperi (I), Midland Bank Group (GB), Nestlé (CH), Péchiney Ugine Kuhlmann (F), Royal Bos Kalis Westminster Group (NL), Royal Volker Stevin (NL), Saléninvest (S), Skandinaviska Enskilda Banken (S), Smit International (NL), Technital (I), Tecnomare (I), Vattenbyggnadsbyrån (S). Associate member: National Swedish Board for Technical Development (STU) (S).

The purpose of the members is to carefully develop the ocean resources: energy, living resources, raw materials and chemical substances. Technical development work is performed in cooperation by the companies, aiming at commercial operations.

EUROCEAN's secretariat, with the assistance of permanent and ad-hoc advisers and experts, initiates and coordinates studies in cooperation with the member companies, in order to evaluate the feasibility of projects. The secretariat plays a leading role in this early stage with the active participation of the personnel of the member companies.

Members interested in a specific project then form a working group and pursue the development work towards a commercial operation.

EUROCEAN adds a new dimension to international cooperation between industrial companies. Cooperation starts long before a project becomes a commercial operation. The decision to develop a project is based on relevant existing knowledge and scientific and technological data. Establishing real cooperation at such an early stage in the development of a project between companies from different countries and involved in different but sometimes overlapping fields of activities, is a difficult task. Nevertheless, EUROCEAN has demonstrated that its basic concept of pooling know-how, resources and experience in the marine field is possible and practicable.

EUROCEAN OTEC Project

The EUROCEAN OTEC work was initiated at a meeting in Venice on Ocean Energy in March 1976. Nine companies cooperated in a study which resulted in a report in November 1977.[1]

The report summarized:

I Cost estimates for future floating OTEC plants of commercial size around 100 MW_e.
   a. total capital cost = 2636 US$/kW
   b. electricity cost = 54 mills/kWh;

II The rationale for selecting closed cycle as the main alternative;

III Different forms for utilization of the energy
   a. 'local energy' = electricity to shore by cable
   b. 'local energy' to be used on the platform by integrating energy intensive industrial processes
   c. 'distant energy' by producing and transporting energy carriers such as H_2 or NH_3;

IV Combination with aquaculture using the artificial upwelling or combination with biomass production for food, fertilizer or fuel purposes.

A paper[2] was given at the 5th OTEC Conference in Miami Beach in February 1978, which relates the findings of the report.[1]

This report and other available information provided the basis for the EUROCEAN member companies' decision to continue OTEC development work along the lines suggested in the report. The continuation was formalized in an agreement reached in Monaco in January 1978, when is was decided that the immediate activities were to include detailed planning of the technical development of the OTEC system and the obtaining of funds for this development work. Component development was continued in the meantime by some of the member companies. A January 1979 meeting in Brussels led to a 10 MW_e pilot plant proposal as discussed later.

* Director General

Funding of applied development work

The funding process for this type of development work is different in Western Europe from the United States although the needs are essentially very similar. Funds are made available in the US in governmental programmes according to established practice. This does not exist in Europe for this type and size of development work but can be established on an ad-hoc basis.

The purpose of the applied development work is to decrease the techno-economic uncertainty or risk to a level which is acceptably low to the managements of the companies in order for them to decide on investing in the commercial operation. This development stage of the industrial process is non-bankable. It is however very costly since it involves design and manufacture of components, systems and ultimately a pilot plant. The process is illustrated in Fig. 1.

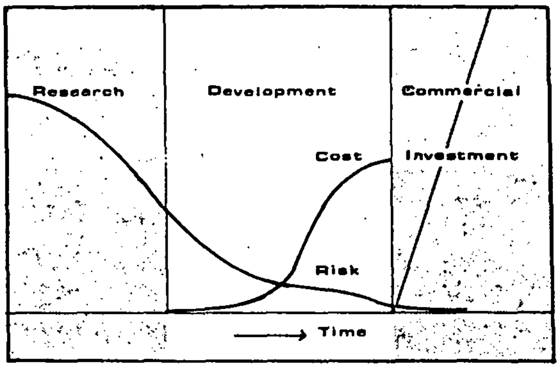

Fig. 1 Cost & risk relations in development work.

We have surveyed the different possibilities of obtaining funds for this type and size of development work, Table 1, and we have concluded that, for this specific project, government funds in the countries of the involved member companies are the most appropriate. Accordingly, we have started the laborious process to obtain such funding.

Table 1 Survey of financing institutions

1. NATIONAL INSTITUTIONS
   - government or a specific government agency
   - joint government cooperation
   - private foundations
2. INTERNATIONAL INSTITUTIONS
   A. Intergovernmental organizations with world-wide membership
      - the vast United Nations family
   B. Regional intergovernmental organizations
      - EEC, European Investment Bank, European Development fund
      - OECD
      - OPEC
      - African Development Bank
      - Asian Development Bank
      - Inter-American Development Bank
   C. Arab financial institutions
3. REGIONAL PRIVATE ORGANIZATIONS
4. NATIONAL DEVELOPMENT BANKS
5. FOUNDATIONS

OTEC-10 MW_e Pilot Plant

In Brussels in February 1979 the EUROCEAN OTEC group was reorganized and it now includes the following member companies: Alfa-Laval (S), Banque Européenne de Crédit (B), Hollandsche Beton Group (NL), Johnson Group (S), Kockums (S), Micoperi (I), Tecnomare (I), Vattenbyggnadsbyrån (S).

A decision was taken to work out a proposal for a floating OTEC 10-MW_e closed cycle pilot plant which should be used with the applications for funds to the Dutch, Italian and Swedish governments (written in April 1979).

The purpose of building a pilot plant of this size is fourfold:

1. To test the OTEC system with full-size modules and sufficiently large components.
2. To verify the cost estimates for the commercial plant and thereby reduce or eliminate the 'safety margins' in the present preliminary cost estimates.
3. To verify the possibilities of future less expensive technical solutions for instance for materials.
4. To facilitate future marketing.

Fig. 2 illustrates the forecasted influence of the pilot plant on the OTEC electricity cost and compares this with the cost of electricity produced by an oil fired power plant as a function of the price of oil.

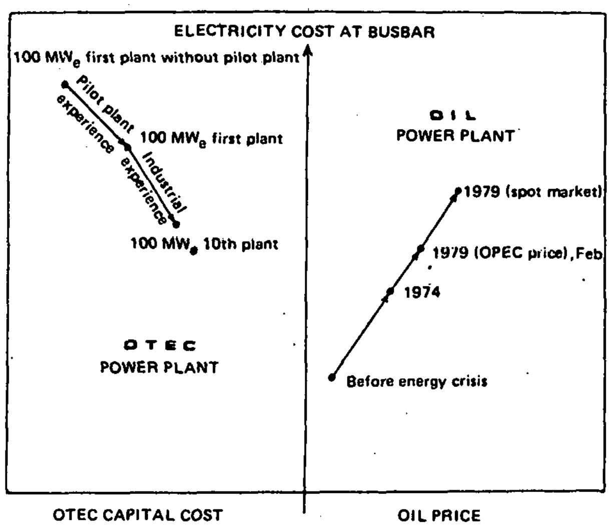

Fig. 2 Comparison of the trends in electricity costs of OTEC electricity and electricity from oil-fired power plants.

The plans for this OTEC-10 exist although they cannot be finalized until the funding is secured: An indication of the plans is given in Fig. 3. The activities extend over a 5 year period and the costs are estimated to be about 50 million US dollars.

The activities are divided into two phases starting with studies and tests which account for less than 20% of the total costs, and followed by construction, towing and installation, start-up and testing, prior to operation.

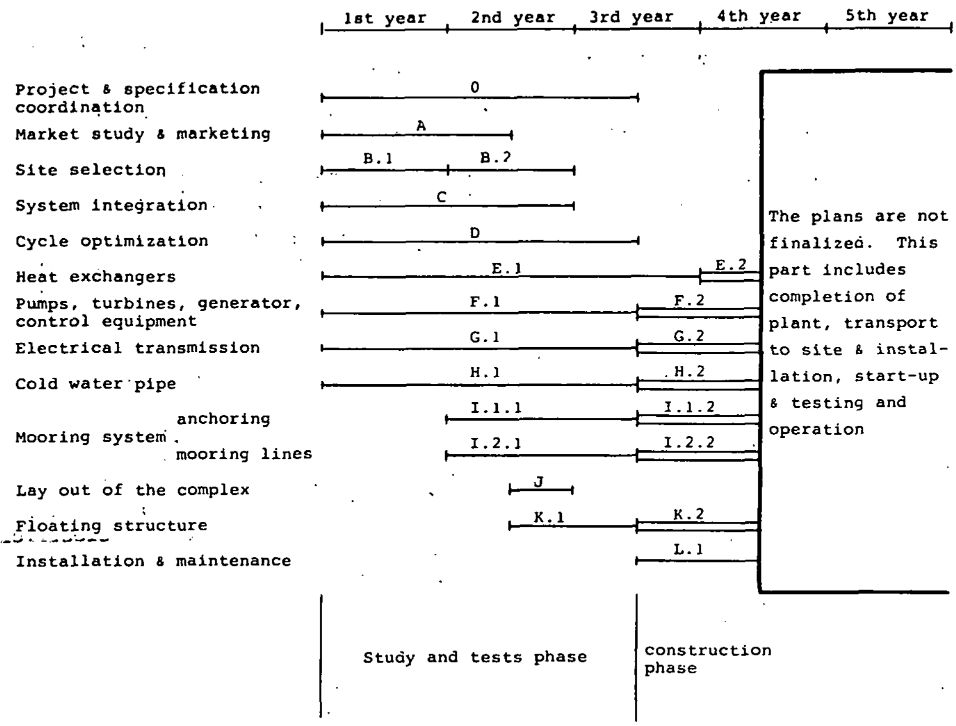

Fig. 3 OTEC-10 activities

The site selection for OTEC-10 has the following criteria: minimum temperature difference of 18°C throughout the year between surface water and water at a maximum depth of 1000 m; minimum distance to shore for these conditions; favourable meteo-oceanographic conditions (no hurricane hazards); good accessibility to the region, for instance scheduled flights; industrial activity in the area, infrastructure; political situation and stability; market situation for electricity and/or other possible products of the OTEC systems. A first list of possible OTEC sites has been compiled and analysed on the basis of these criteria in the Atlantic Ocean (Africa and Eastern America), in the Antilles, in the Pacific Ocean (Western America and Oceania), Indonesia-Philippines and in the Indian Ocean. Deeper analyses will be made before the actual site is selected.

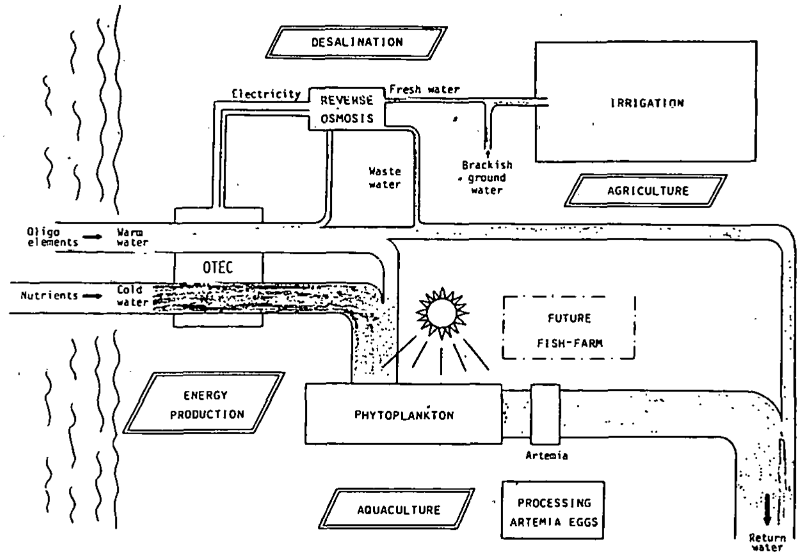

Fig. 4 Block diagram of system combining OTEC, aquaculture and desalination.

Combined Aquaculture/OTEC/Desalination Project

Another EUROCEAN project is related to the OTEC project but, from a development point of view, is an aquaculture project. A decision has been taken by eight member companies, some of which are participating in the OTEC project, to perform a six month feasibility study on a combined Aquaculture/OTEC/Desalination project, aiming at a commercial aquaculture and desalination operation with a small scale OTEC plant. Fig. 4 shows a preliminary block diagram for this combined system and Fig. 5 shows a possible first step in the development which is an aquaculture pilot plant.

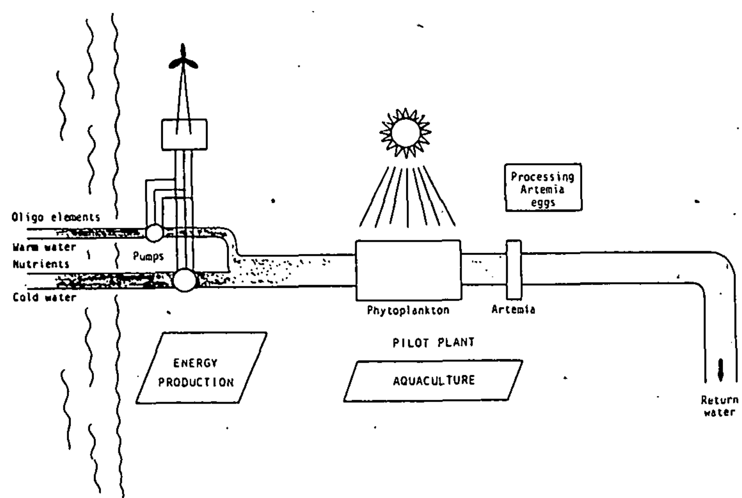

Fig. 5 Possible first stage using windmill with aquaculture system.

References (1) EUROCEAN, 1977.- *A European industrial assessment of the ocean thermal conversion system, synthesis report and annexes.*- Monaco : Association Européenne Océanique, Eurocéan.- 119 p. + 192 p.

This report is the property of the member companies of EUROCEAN and is not available for outside distribution. Reference (2) however summarizes this OTEC report.

(2) LACHMANN (B.), 1978.- EUROCEAN OTEC Project. In : *Proceedings of the 5th Ocean Thermal Energy Conversion Conference*, February 20-22, 1978, Miami Beach, Florida, vol. 1, pp. II174-191.- Coral Gables, Fla.: The Clean Energy Research Institute, University of Miami.

DISCUSSION

Question: A number of companies from different countries are involved in the EUROCEAN OTEC project. How do you manage the cooperation?

B. Lachmann: We do not have time to discuss the very interesting issue of cooperation. Suffice it to say that in our case it works. In the specific case of the OTEC project it is fairly easy because the companies from one of the countries involved (Sweden) have a major part of the systems and components. This makes it easier from a funding point of view to start establishing one national programme, to which can be added other national programmes, in this case two more.

Question: You heard Mr. Derrington point out how essential it is to have a systems concept in bringing the whole thing together. I wonder if you would comment on that.

B. Lachmann: You mean a technical system. Very early in the study work, during the period between early 1976 and 1977, we came to the conclusion that the best possibilities, taking into account the technology at large but also the specific knowledge within the companies involved, was to base the system on the closed cycle, with the plate heat exchangers. That narrowed down the total work.

Question: In the survey of financing institutions of which you showed a summary, I noticed OPEC — the Organization of the Petroleum Exporting Countries — on the list. What would be their interest in funding this type of development work?

B. Lachmann: We made this survey together with the EUROCEAN bank members. We studied the conditions for financing work of this type which is the noncommercial part of development work. It is not, for instance, what the United Nations call development work. Development work to them is building a factory in a less developed country. We had an interesting discussion with OPEC in Vienna in late 1977, and we learned that that source of funds was probably not one of the most accessible. There may be possibilities of attracting oil money from Arab sources but these possibilities are very complicated. All in all we came to the conclusion that the quickest and easiest way to fund the nonbankable development work, once the companies have established the industrial project, would be to establish corresponding government national programmes in the countries of the member companies and then to arrange an intergovernmental programme involving those specific countries.

THE FRENCH OTEC PROGRAM

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ABSTRACT

Ocean Thermal Energy Conversion (OTEC) is a French concept that was first proposed by Arsene d'Arsonval, a century ago. His friend and student Georges Claude operated the first OTEC plant in Cuba, in 1930; the first experimental floating plant, the *Tunisia*, off the coast of Brazil was a failure. A lot of work on the open cycle was done in France for the 10 MW Abidjan project in the 1940's and 50's: trials on evaporation, deaeration and successful experiments of a cold water pipe laying method. The Abidjan plant was abandoned for political and economic reasons — "the cheap oil era." A similar project in Guadeloupe did not succeed. However, there is now a new interest in OTEC. A program is under way that aims at deploying pilot plant (1 to 10 MWe) to be conducted in three phases: a feasibility study (nearly completed), a test components phase (1980 to 1982), and design and construction of a pilot plant for testing at a French tropical island (1983 to 1985). The goal of the $1.8 million feasibility study is to explore open and closed cycles for a shore-based or a floating plant. The advantages and disadvantages of the shore-based concept, the difficult problem of the cold-water pipe, deaeration in open cycle and utilization of French experience in construction of large offshore platforms are discussed.

INTRODUCTION

Ocean Thermal Energy Conversion (OTEC) is one of the most attractive forms of renewable energy for the future. In this paper, we will begin with a brief history of the work done in France. We will then present our new OTEC program which aims at building a pilot plant in the 1 to 10 MWe range at a tropical island before 1985.

According to the new 200-mile exclusive economic zone, about 8% of the areas suitable for OTEC are under French jurisdiction. Studies are managed by CNEXO (Centre National pour l'Exploitation des Oceans), the Government agency in charge of the OTEC program. Firms involved in the program are (a) for the open-cycle and floating (or beached) concept: the CGE (Compagnie Générale d'Electricité) - Alsthom-Atlantique - ETPM (Entrepose Travaux Petroliers Maritimes) and (b) for closed-cycle and shore-based concept: the Empain-Schneider group, particularly the three companies SGTE (Société Générale de Techniques et d'Etudes), Creusot-Loire and Spie Batignolles.

The OTEC program began at the end of 1978, and it is too early to present any concluding remarks on the feasibility phase that will end early in 1980. Nevertheless, we will discuss some preliminary results on shore-based, open-cycle plants and utilization of the French experience in offshore platforms for OTEC.

FRENCH EXPERIENCE IN OTEC: SOME HISTORY

In 1934, before sea tests of the first OTEC floating plant *Tunisie* off the coast of Brazil, Georges Claude was describing his project to Brazilian authorities in the following terms: "the goal of this demonstration is essentially to prove that the sea could become the cheapest and most formidable source of power in the future." Now, 50 years later, there are still no OTEC plants in operation. What have we done in the past?

The OTEC story began in 1881 when the French physicist Arsène d'Arsonval suggested operating a closed-cycle system[1]. The working fluid, sulfur dioxyde, would be vaporized by the warm (30°C) spring of Grenelle in Paris, then condensed by colder river water such as that from the Seine River. D'Arsonval noted that many places in the world could provide the necessary water temperature difference. He attested to the possibility of extracting energy from the tropical ocean by noting that a heat engine could be built to operate on the temperature difference between surface and deep seawater.

The OTEC story continued with Georges Claude, a friend and student of d'Arsonval. We can say Claude was the first OTEC'er. In March 1926, with Paul Boucherot, he suggested using seawater as the working fluid. In some very interesting papers[2,3]. Claude enthusiastically described the advantages of the open cycle: no huge heat exchangers and a minimization of biofouling. He demonstrated his idea 2 years later, operating with success a 60-kW open-cycle plant at Ougrée, along the river Meuse under a difference of temperature of 20°C. He concluded that deaeration was not a big problem[3].

Two years later, in 1930, he used the same 60-kW thermal machine in Cuba. He was successful in achieving two important feats: laying a pipe 2 km long and 1.6 m in diameter (it is important to remember that sea-work was then in its prehistoric stage); and producing 22 kW during 10 days in very bad conditions: the ΔT was only 14°C, and only 10% of the pumped cold seawater was used in the plant. Moreover, the turbine was completely maladjusted.

Claude concluded that it would be possible in the future to produce 500 kW with only 1 m^3/s of cold deep seawater.

In 1933, Claude built the first floating OTEC plant, transforming the 10,000 tonne cargo-ship, *Tunisie*. He hoped he would have no trouble with sea aggression on a vertical suspended pipe. The thermal machine was an open-cycle plant, 25 m in length and 8 m in diameter, producing 2000 kW on the turbine shaft; 800 kW were transformed in electricity and 1200 kW were used to operate a 1200 kW ammonia compressor... for ice making (see Fig. 1). The ship

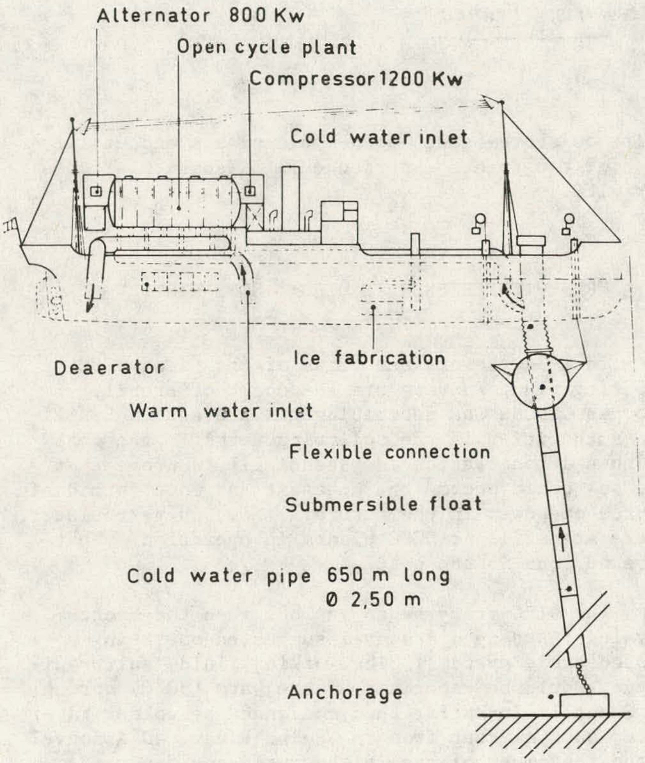

Fig. 1 Claude's floating OTEC plant, *Tunisie*.

was to be connected by a flexible tube to a semi-submersible float to which the 2.5 m diameter pipe would be suspended. Claude did not succeed in laying the pipe off the Brazil coast (120 km from Rio) because of the great heaving of the float. Finally, as the pipe sank, the ship *Tunisie* never produced any ice. After this episode, Claude was temporarily despondent, but he was not the type of man to surrender. In 1940 he proposed the 40-MW Abidjan project. It was a very daring project, without a cold-water pipe but a vertical well connected to a submarine tunnel 4 km in length! Very quickly Claude's project turned into a 10-MW, open-cycle, shore-based plant, with two units of 3.5 MWe net power and a 4 km cold water pipe (see Figs. 2 and 3).

The government company, "Energie des Mers," created in 1948 are in charge of the Abidjan project. They have done a lot of work on open-cycle power plants and pipe laying. In particular:

1. A selection of the best type of evaporator was carried out after many trials;

2. The Rateau deaerator was very efficient - its consumption was less than 10% of the gross power;

3. The construction of the 8-m-diameter turbine was achieved without major problems;

4. The final design of the plant was very good: compactness, one stage of evaporator, three stages

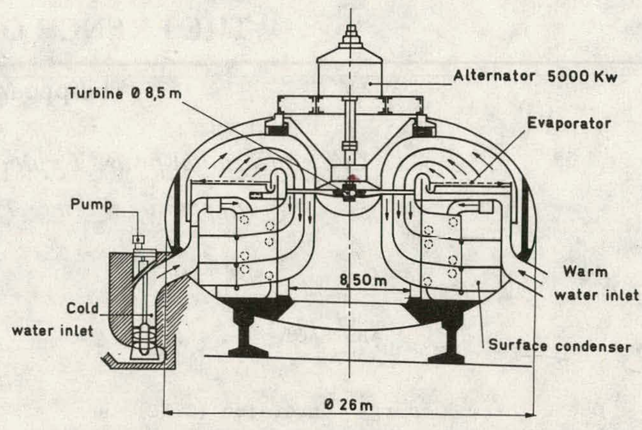

Fig. 2 One of the two 3.5-MWe units designed for use at Abidjan.

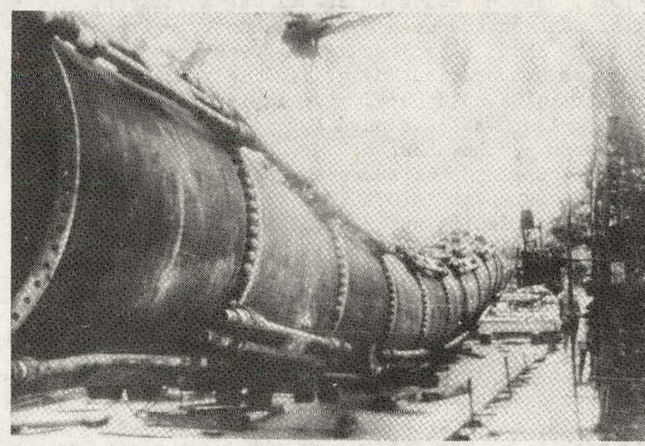

Fig. 3 The cold water pipe for the Abidjan project (flexible section).

for the falling-film condenser, hence a very low flux of cold water: 1 m^3/s for 1 MW;

5. After many trials, selection of a realistic pipe-laying method. A 150-m-long, 2-m-diameter element was laid in 1955 with success under 300 m of water.

Thus, in the middle of the 1950's it was clearly demonstrated that the Abidjan project was technically feasible and competitive. But the plant was never constructed for political reasons. After World War II, priority was given to national energy — hydropower, tidal energy with the 240-MW plant of the Rance estuary, and nuclear and oil-fired power plants.

A similar shore-based plant was studied for Guadeloupe (1959) but OTEC power was not competitive then because of the very low cost of oil.

French OTEC experience is very broad and very old. A lot of good work was done on open cycle and the shore-based concept.

THE FRENCH OTEC PROGRAM, 1978 - 1985

In France there is now a new interest in OTEC. We are directly concerned with this form of clean, concentrated, abundant, and renewable energy for our fuel-dependent French tropical islands.

At present, according to the 200-mile exclusive economic zone, about 8% of the most promising tropical ocean area for OTEC is under French jurisdiction. Most of the French Departments and Overseas Territories (DOM/TOM), such as at Réunion Island in the Indian Ocean, Martinique and Guadeloupe in the Atlantic, Polynesia and New Caledonia in the South Pacific, are located in the tropical zone.

Broadly speaking, those islands are completely fuel-dependent, and the cost per kWh is very high, more than three times the cost in continental France. Moreover, there is often a problem of fresh water, and open-cycle OTEC looks very attractive for that reason - it can produce fresh water. A range of 1 to over 10 MWe in basic power is well adapted to electrical consumption of 10,000 to 100,000 inhabitants. The major parts of those volcanic islands rise steeply from the ocean floor and have steep offshore slopes; consequently, shore-based OTEC plants could be used.

There are many isolated volcanic islands that are fuel-dependent, sometimes without fresh water, and with modest power needs in comparison with large occidental consumption. These facts led to the definition of the present French OTEC Program (Table 1).

Table 1

The French OTEC Program

Title	Description	Budget
Phase I Feasibility, 1 to 10 MWe, 1978 - 1979	Study of four combinations: open cycle or closed cycle; shore-based or floating Size effect	≃$1.8 M 2/3 government 1/3 industrial
Phase II Components, 1980 - 1982	Selection of a technology for a pilot plant Test of major components Detailed site survey	≃$5 to 6 M
Phase III Pilot plant, 1983 - 1985	Design construction of a (some) MWe pilot plant	Dependent on power level, say $20 to 30 M

The program will be managed by CNEXO and is aimed at operating a pilot OTEC plant in the 1 to 10 MWe range. It will be conducted in three phases:

I. A feasibility study comparing closed and open cycles in terrestrial or floating configurations. Scale effects will be studied, and a decision on future technical options for the pilot will be made in 1980.

II. Testing of major components such as the cold-water pipe, heat exchangers, and deaeration process. Exact contents of this phase will be established at the end of Phase I. A detailed survey of the pilot-plant's site is also planned. It will be completed in 1982.

III. Designing and building the pilot plant. Exact power, within the 1 to 10 MWe range, will be determined as a conclusion of the first and second phases and budgetary possibilities.

The $1.8 million feasibility study (including a 40% financial participation of firms selected by CNEXO) will end early in 1980. It comprises two major contracts:

1. Technical and economic feasibility of a 1 to 10 MWe open-cycle floating (or beached) plant, contracted to a group of French industrial firms: CGE (Compagnie Générale d'Electricité) - Alsthom-Atlantique and ETPM (Entrepose Travaux Petroliers Maritimes).

2. Technical and economic feasibility of a 1 to 10 MWe closed-cycle shore-based plant, contracted to Empain-Schneider Group, particularly the three companies SGTE (Société Générale de Techniques et d'Etudes), Creusot-Loire and Spie Batignolles.

In addition, a general review of possible French OTEC sites is under way by CNEXO from the standpoint of oceanographic and meteorological data and future energy needs. These studies have begun, but it is too early to present final results. Some specific points are noted in the following section.

SOME PARTIAL RESULTS

The Shore-Based Concept and the Cold Water Pipe

The shore-based concept, for powers up to 10 MWe, seems to be attractive for the following reasons: (a) no submarine cable and no anchoring problems; (b) existing experience in building a shore-based plant; (c) easy maintenance and accessibility; and (d) possible utilization of the following by-products:

1. Fresh water (for the open-cycle) is very attractive, in some cases more attractive than electricity; we estimate that a production of 200 m^3 per hour in a 5 MWe plant is feasible.

2. Aquaculture, but there is much work still to be done. Theoretically, deep water is very desirable because it is rich in nutrients and relatively germ free.

3. Use of cold seawater for air conditioning and for conservation of agriculture or fishery products.

The biggest problem is how to lay a cold water pipe of 2 to 5 m diameter and 3 km length. Some materials are under examination such as concrete with resin, iron, aluminium, fiberglass, and polyesters. A good material should have mechanical resistance, low thermal conductivity, corrosion resistance, and no aging. Various pipe laying methods are under examination such as those shown in Figs. 4 through 7.

Seawater Deaeration in Open-Cycle OTEC Plants

Open-cycle schemes for OTEC have several advantages over closed-cycle schemes using ammonia. They produce desalinated water at nominal cost. This is

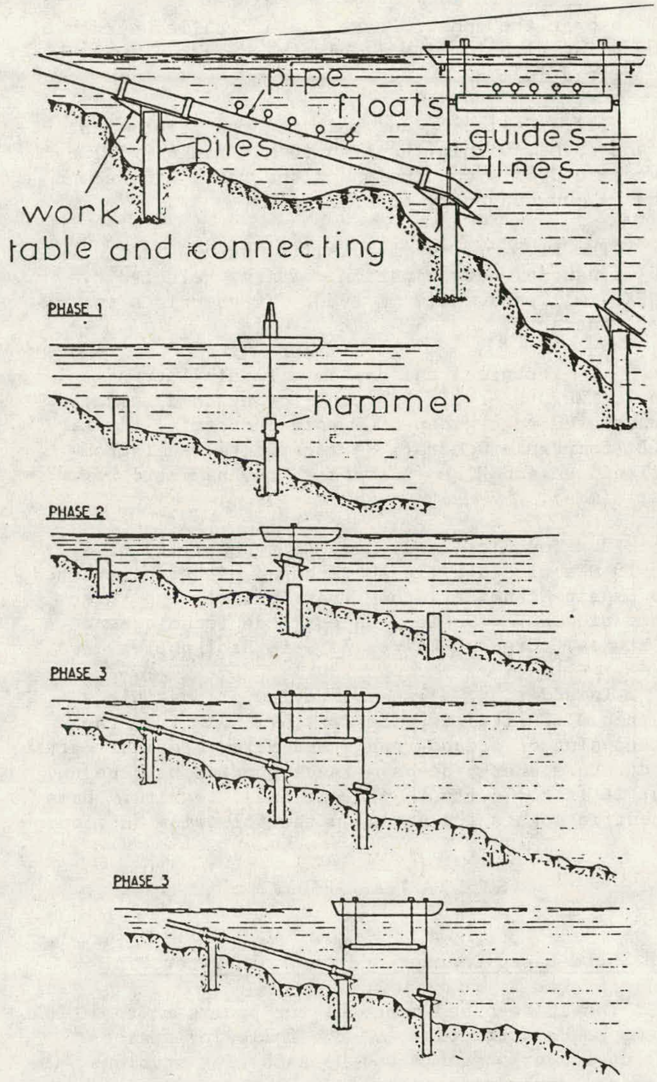

Fig. 4 The viaduct method of pipe laying.

important because tropical countries, which will be the first to operate OTEC plants of medium to low output, need water at least as much as electricity.

Surface condenser characteristics permit higher rates of heat transfer, and construction material constraints are less stringent.

For the flash operators, efficient designs are now available based on comprehensive data collected previously in France. The designs eliminate the need for heat-transfer surfaces and significantly reduce the head loss along the hot water stream as compared with typical head losses of closed-cycle schemes.

On the negative side, open-cycle turbines have to be larger for a given output to accommodate the greater specific volume of the steam. However, outputs contemplated at present by no means make turbine size prohibitive.

A major problem is the need for deaeration. Seawater from the warm surface layer is saturated with dissolved gas. A typical analysis is as follows:

N_2,	8.9 ml/l
Ar,	0.2 ml/l
O_2,	4.7 ml/l
CO_2, etc.	traces
TOTAL	13.8 ml/l.

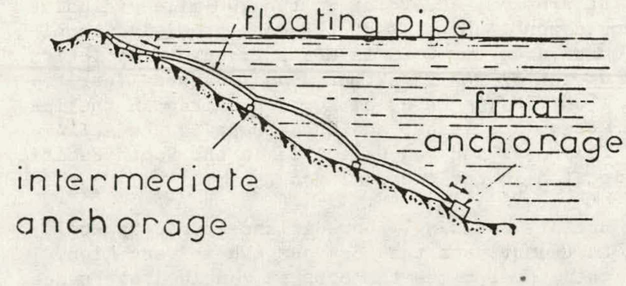

Fig. 5 The flexible catenary method of pipe laying.

The total may rise to 15 ml/l owing to diurnal variation. A 5-MWe plant uses about 30 m^3/s of hot water.

For the foregoing dissolved gas contents, the associated flow of dry air is about 600 gm/s. If all the gas had to be removed, a sieve-plate or similar deaeration column would be needed upstream of the evaporator, and a gas extractor would have to be provided to blow out the gas into the atmosphere from the pressure within the deaerator of approximately 180 millibars. The power needed to operate the sieve-plate column and the gas extractor would be at least 1200 kW, i.e., about one quarter of the total power produced by the turbine.

Georges Claude estimated that only 80% of the dissolved gas needs to be removed, but even if this figure were assumed, the power requirement for removing gas would still be about 1 MW.

Theoretical and experimental results relating to the relative rates of heat and mass transfer tend to show that only about 3% of the gas dissolved in the hot water will be effectively removed during evaporation.

Since the aforementioned, noncondensable contents in no way precludes the use of standard design procedures for the condensers, the noncondensables need not be removed prior to evaporation.

Compared with the schemes suggested by Claude and his successors, this represents a considerable saving in initial capital outlay and a considerable increase in overall plant efficiency owing to the elimination of deaerator head loss on the warm water line and to the large drop in the power needed to eject the noncondensables.

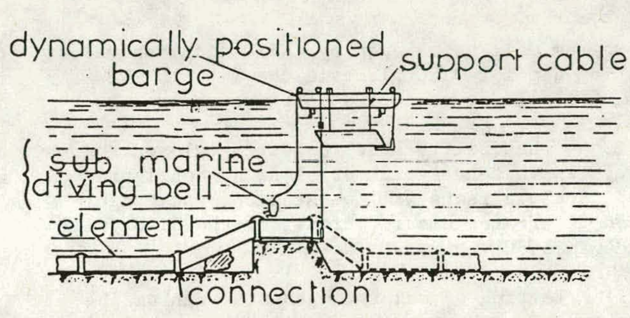

Fig. 6 The successive connections method of pipe laying.

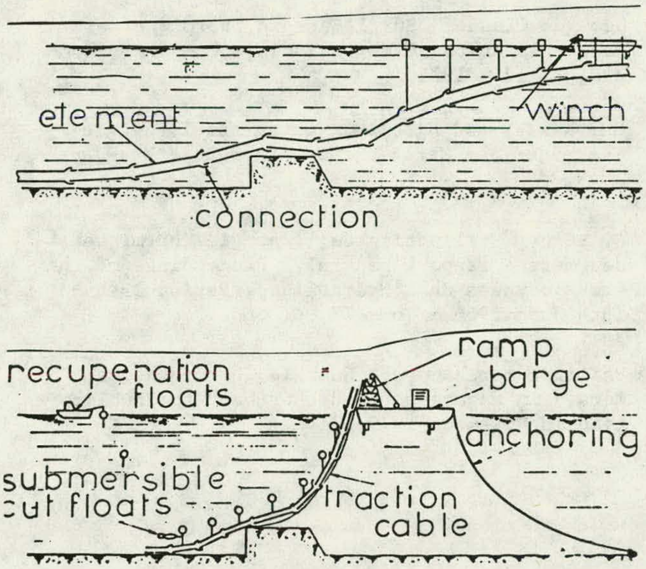

Fig. 7 The half-rigid pipe laying method (tested for Abidjan).

The net result is that the overall plant efficiency, defined as net output power divided by gross power at the turbine shaft, can exceed that of indirect, closed-cycle schemes.

The kinetics of the removal process within the evaporator thus appear to be of critical importance. This is the reason that the CGE/AA/ETPM group decided to launch a priority research and development program on the problem.

The program will include:

1. Investigations into the nucleation state and variability of the nuclei and microbubbles present in the upper layer of the sea;

2. Laboratory and field determinations of stripping kinetics in the hot-water streams and in the evaporator.

Results of the program are expected not only to vindicate a particularly attractive direct-cycle OTEC scheme but also to shed some light on the state of nucleation of the upper layers of tropical oceans.

Shore-Based versus Floating Plants

As previously noted, shore-based and floating concepts are being studied. At the end of Phase I, a comparison will be made to choose the type of implementation for the pilot plant.

It could be advantageous to use offshore technology for the design of OTEC floating units. French experience in concrete platforms is extensive. as 50% of the North Sea market has been cornered by French companies.

One of those companies, Sea Tank Co., has built three big concrete platforms for the North Sea oil fields (Frigg, Brent, and Cormorant).

"Cormorant A" is, for instance, a drilling-production platform operating in water 150 m in depth, with a storage capacity of 1 million barrels and which, with a 590,000 tonne displacement, was towed for 1150 miles.

Sea Tank Co. from the CGE/AA/ETPM OTEC group proposes a semisubmersible design, still under evaluation but very attractive because it is particularly easy to build and has a good stability in rough seas (Fig. 8). In this design, the evaporator of the open-cycle, according to the barometric concept, is below the deck, the turbine is in a column, and the surface condenser is in the semisubmersible float.

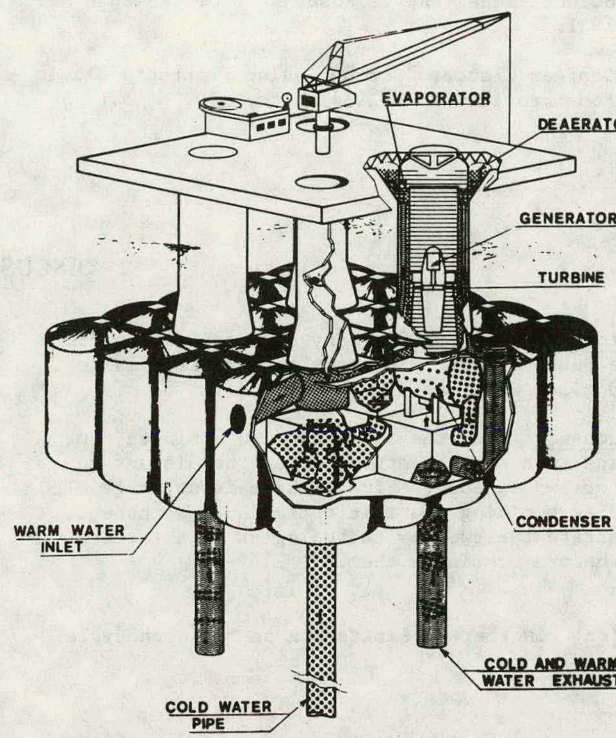

Fig. 8 Arrangement of the principal components of an open-cycle OTEC floating plant of 5 MWe gross power proposed by Sea Tank Co., from the CGE/AA/ETPM Group.

CONCLUDING REMARKS

In conclusion, the French OTEC effort is a continuation of our old but useful experience and primarily involves plants of 1 to 10 MWe adapted to our needs in the DOM/TOM. We have not eliminated the open cycle, which can produce desalinated water, a particularly attractive by-product for numerous islands. No technical choices for the pilot can be made before the different options have been carefully evaluated. We are convinced the first OTEC market will be in fuel-dependent equatorial islands, such as our DOM/TOM, and it is expected to put French industry in a position to propose in the future reliable small and middle-sized OTEC power plants producing electricity and fresh water.

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DISCUSSION

Question: Have there been any studies linking OTEC in the South Pacific with your territorial claims of 200 miles with marine mining, etc.?

P. Marchand: Yes, the idea has been explored, but we think such a combination is for the distant future for two reasons. First, there is no large OTEC or no large mining plant in operation. We have to demonstrate the two new technologies on a large scale before combining them.

Question: Is there a limitation on the open cycle?

P. Marchand: There is no more limitation than on the closed cycle. In the past, during the studies of the Abidjan Project, people thought there was a limitation to about 10 MW on the size of open-cycle plants because of the turbine size. The recent Westinghouse investigation of a 100-MW open-cycle plant, reported at this conference, indicates that a 70-m-diam. turbine is feasible now because of the availability of new technologies such as helicopter blades in new materials. In our view, a big advantage of the open cycle is a very much lower level of biofouling than for closed-cycle plants with their huge heat exchangers.

AN OVERVIEW OF THE JAPANESE OTEC DEVELOPMENT

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Abstract

Japan seems to be a country which needs OTEC. The feasibility study of OTEC power development has started in the beginning of the '70s. In these five years conceptual design of 100 MW power plants and its evaluations were made. Results show that OTEC is technically and economically feasible. Experimental studies on OTEC power cycle, and some development works have been made, especially on heat exchangers and platform structures. Also studies of application and assessment of OTEC system were conducted.

Introduction

Energy consumption in Japan is large, that is the second highest level in the free world following the U.S.A., and the import ratio amounts to 88% of the total energy consumption because of minor domestic resources.

The land area of Japan is small, but it is surrounded by a large expanse of sea, mostly warm, and "Kuroshio", one of the largest sea surrents, always brings a large amount of thermal energy from the tropical zone. Accordingly, energy production from the sea should be an important goal for Japan.

The study of ocean thermal energy conversion (OTEC) in Japan was started in 1970 by the Committee on Investigation of New Power Generation Methods.[1] A system concept was presented by Japan to the Fifth General Meeting of the Pacific Basin Economic Council in New Zealand by one of the authors, entitled "Equatorial Marine Industrial Complex", which includes an OTEC power plant and subsystems. The complex would not only produce power but would also utilize the nutrient-rich cold water discharged from the plant for the production of mariculture.[2] A conceptual design study of this system was made by the Leisure Development Center in 1973.

To guide the OTEC effort in Japan, the Agency of Industrial Science and Technology (AIST) of the Ministry of International Trade and Industry (MITI) established in April 1974 a committee for investigating the feasibility of OTEC at the Japan Heat Management Association (JHMA). The program name "Sunshine Project" was adopted.[3] The project aim was the development of new energy technologies. In the same period, experimental research on OTEC was started at the Electrotechnical Laboratory of AIST in Tokyo[4] and at the Science and Engineering Faculty of Saga University in Saga.[5]

In industrial circles, Tokyo Electric Power Service Co. (TEPSO) has started their design study of land-based OTEC power plants, some what earlier than mentioned above, based upon their expertise in power plant engineering.[6]

Conceptual Design and Economic Evaluation of OTEC for Japan

The objectives of the OTEC feasibility study by the committe in JHMA were the evolution of a conceptual design of OTEC power plants and the evaluation of technical and economic factors of the concept as a whole. As a first step, design work was conducted in 1974 on a 1.5 MW land based experimental OTEC power plant. This was followed by a system analysis of a larger ocean-based OTEC power plant. In the second year (1975), a conceptual design was prepared for an ocean-based demonstration OTEC power plant rated at 100 MW and sited at a benign tropical location. The design was based upon the current state of technology with minor technical improvements, particularly in the heat exchangers. In the third year (1976), an improved conceptual design of a 100 MW OTEC power plant, to be located at the sea near Japan, was made. This later design was based upon advanced technology. The results of the study in 1975 and 1976 were reported in the 5th conference last year.[7] In the following two years (1977 and 1978), additional work on the previous year's design was carried out. Emphases were placed especially on the optimum design of 100 MW plants, the platform structure, and station keeping, at the representative sites, structure model experiments in a wave flume, and improvements in the heat exchangers.

Results of the studies are summerized as follows. The details of the studies are being reported by other papers in this conference.[10]

Dynamic behavior of the platform for the design study on the 100 MW commercial power plant and platform structure, the locations were limited to the coast or the offshore area of the Japanese islands, such as Iriomote, Okinawa and Osami (Pacific ocean side) and Toyama Bay (Japan Sea side), where direct power transmission to the national grids is feasible. Not only the sea temperature data, oceanographic data including the wave heights and periods, the vertical distributions of currents and their horizontal patterns, but also the average and maximum wind power of these four sites were collected. Table 1 shows typical sea state data of the offshore area of Osumi island and of Toyama bay.

The dynamic analysis on the motion of the platform coupled with the cold water pipe is one of the important problems in order to optimize the OTEC plant configuration and its anchoring system, and to design the power cable arrangement. To evaluate the dynamic characteristics, it is important to

Table 1 Characteristics of typical sea states (ordinary) around the Japan

	max. significant wave height	max. significant wave period	current speed		max. wind velocity
			at surface	at the depth of 500m	
Offshore of Osumi Island	13.0m	13 sec	1.74m/sec	0.25m/sec	60m/sec
Toyama Bay	8.4m	13 sec	1.00m/sec	0m/sec	60m/sec

couple the platform and cold water pipe together regardless of which type of connection is to be used in the platform-cold water pipe juncture. The problem is indeed complicated, however, because of non-linearity on the equation of motion for the platform coupled to the cold water pipe. Also lateral vibrations due to the current should be taken into consideration.

Numerical calculations of the external forces acting on the six types of platforms were made at the above mentioned four sites. Table 2 shows an example of the results. It shows the maximum external forces acting on the ship, and the submerged disc type platforms at ordinary sea states on the offshore area of Osumi island and Toyama bay respectively. These calculations were undertaken by Mr. Suzuki of Ishikawajima Heavy Industries Co. (ship type) and Mr. Ono of Mitsubishi Heavy Industries Co. (submerged disc type). An experimental study of the dynamic behavior of the platform was undertaken using a wave flume by Prof. Nagasaki of Tokai University and members of Shimizu Construction Co., which is also being reported on in this conference.[10]

Table 2 Dimensions and displacements of ship type and submerged type platforms and maximum resultant forces to them, at Osumi Island and Toyama Bay.

Type of Platforms*		Ship		Submerged disc	
Length (ship) Diameter (disc)	(m)	230		110φ	
Beam	(m)	60			
Height (depth)	(m)	27		40	
Draft	(m)	13		55	
Displacement	(t)	145,000		409,000	
		Toyama	Osumi	Toyama	Osumi
Maximum resultant external forces to the platforms at two model sites (ordinary state)	Wave (t)	140	130	82	82
	Current (t)	90	400	143	703
	Wind (t)	30	30	10	10
	Total (t)	260	560	235	795

* CWP dimensions are sane for both cases : Intake (onetube) 500mL × 11.2mφ
Outlet (twotubes) 100mL × 8mφ

Optimization of the Plant

Sea temperatures of the Pacific Ocean side and the Japan See side show conspicuously different behavior as shown in Fig. 1. The main reason seems to be that "Kuroshio" is predominant on the Pacific Ocean side, while sea water interchange in the open ocean is not remarkable on the Japan Sea side, and also icey water flows into it from the north.

Optimum conceptual design of 100 MW OTEC power plants on board of ship type and submerged disc type platforms carried out, located in both the offshore area of Osumi island and Toyama Bay. The results are shown, as 1978 design, in Table 3.[8] Optimum depths to intake cold water to obtain the maximum busbar output are 790m in Osumi and 370m in Toyama, respectively.

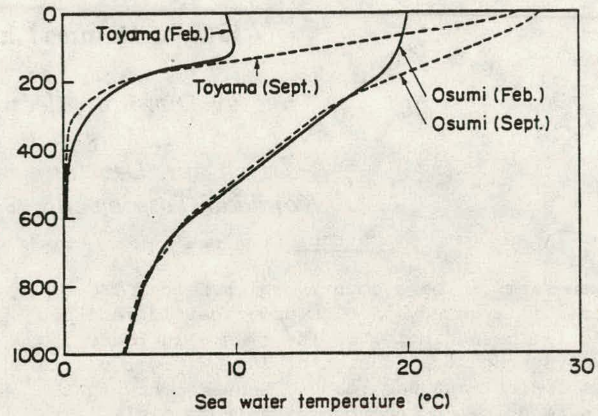

Fig. I Maximum (Sept.) and minimum (Feb.) of monthly average of sea temperature at offshore of Osumi Island (pacific ocean) Toyama Bay (Japan sea)

Unit power cost at the busbar in both sites incidentally amounts to about the same price of 12 Yen/kwh (∿60 Mills/kwh), because of the effect of a lower construction cost being cancelled by the effect of the shorter operating days in Toyama as compared with Osumi.[8]

Fig. 2 shows an artists' model of the submerged-disc type 100 MW OTEC power plant.

TEPSCO has been engaged in design work and engineering development of land-based commercial OTEC power plants for several years, through the company's investment. They developed a unique cold water intake pipe and its deployment technique. The out side and in side walls of this tube are glass fiber reinforced polyester containing glass micro balloon and polyester composits in between the two walls. Bulk density of the tube is 0.98 g/cm^3.[9]

Table 3 Major specifications and costs of 100 MW OTEC power plant

Items	1975 design	1976 design	1978 design OSUMI	1978 design TOYAMA
Gross power output (kw)	100,000	100,000	100,000	100,000
Net power output (kw)	73,940	77,210	78,770	83,100
Working fluid	Ammonia	Ammonia	Ammonia	Ammonia
W. F. Flow rate (kg/h)	1.18×10^7	1.114×10^7	0.99×10^7	0.908×10^7
Warm water temp. (°C)	28	28	28	26
Intake warm water (kg/h)	9.88×10^8	9.74×10^8	7.817×10^8	6.93×10^8
Cold water temp. (°C)	7	7	4.56	0.747
Intake cold water (kg/h)	1.01×10^9	8.09×10^8	6.156×10^8	5.69×10^8
Evap. heat transfer area (m^2) & unit	3.2×10^5;16	3.106×10^5; 8	2.62×10^5; 8	2.14×10^5; 8
Cond. heat transfer area (m^2) & unit	3.3×10^5;16	3.508×10^5; 8	2.94×10^5; 8	2.37×10^5; 8
T/G output (kw) & unit	25,000 ; 4	25,000 ; 4	25,000 ; 4	25,000 ; 4
Type of platform	Rectangular barge	Submerged cylinder	Submerged cylinder	Surface ship
Unit construction cost (yen/kw)	780,000	644,600	781,000	592,700
Unit power cost at the busbar (yen/kwh)	11.75	9.56	12.21	12.36
Power transmission cost (yen/kwh)	-	1.5(D.C)	0.66(A.C)	0.66(A.C)

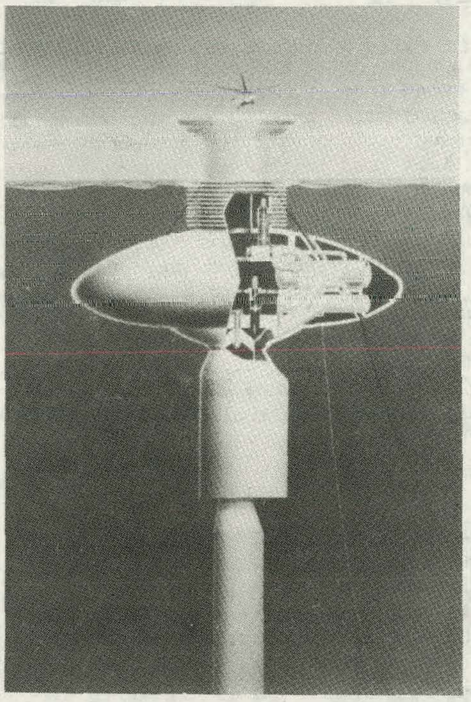

Fig. 2 An artists' model of submerged-disc type 100 MW OTEC power plant

Studies of Power System Performance

Experimental and theoretical studies on OTEC power systems and research to improve heat exchangers performance have been undertaken by two groups. At the Electrotechnical Laboratory in Tokyo affiliated with MITI, T. Kajikawa's group is conducting experimental work using ETL-OTEC-II loop as shown in Fig. 3[4]. At the Science and Engineering Faculty of Saga University, the work is being conducted by H. Uehara's group using the power loops of Siranui series, one of them is shown in Fig. 4[5]. Now Saga University is constructing a new on-shore laboratory for the experiments in actual sea conditions including biofouling of the heat exchangers.

A Government Industrial Research Institute has started evaluation of biofouling and corrosion of materials for OTEC use, setonaikai.

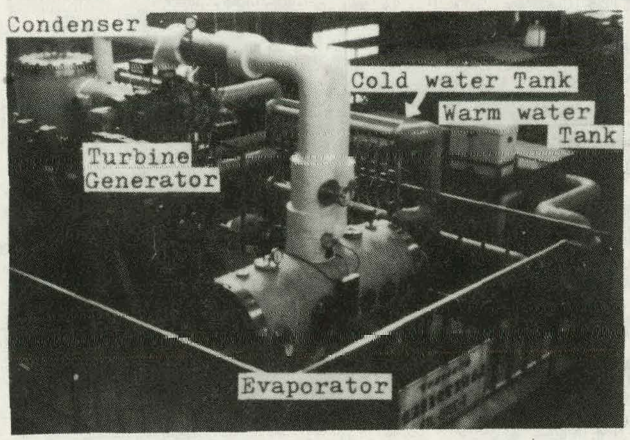

Fig. 3 ETL-OTEC-II experimental power loop

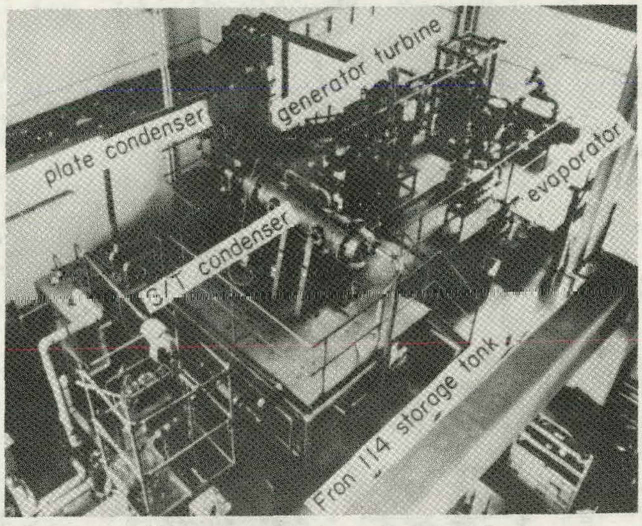

Fig. 4 Siranui 3 OTEC experimental power loop

Heat Exchanger Development

As described earlier, a great emphasis of the OTEC research program in Japan has been placed on the heat exchangers. Several types of high heat transfer heat exchanger tubes and plates, and new evaporators and condensers, which used such elements, have been developed, taking account of the effect of phase changes of the working fluids, by the manufacturers.

Some examples are as follows: Fig. 5 shows titanium condenser tubes, the upper is fluted both outside and inside, for ammonia/water use, the lower is single fluted (pitch 1.0mm, depth 0.5mm) on the outside which is the best tube for R-114/water heat exchange. These tubes were developed by Toshiba Corporation, and the lower one has been used with good performance (over all heat transfer coefficient: $K = 1800$ kcal/m^{2}h°C), for the condenser of a 1 MW binary cycle (hot water/R-114) geothermal pilot power plant by this company, under contract in the geothermal development program of the Sunshine project.

Fig. 6 shows a titanium condenser plate developed by Hisaka Warks. It shows high heat transfer per-

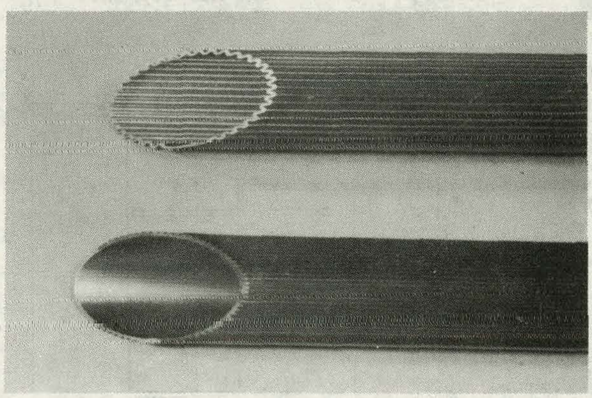

Fig. 5 Titanium fluted condenser tube (used in vertical position in practice)
upper: double fluted (for NH_3/water)
lower: single fluted (for R-114/water)

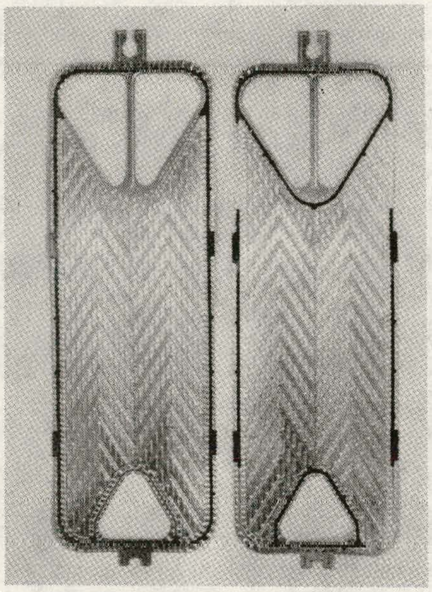

Fig. 6 Titanium condenser plates (R-11/water)

formance (K=2600 kal/m^2h°C) for R-11/water heat exchange.

A systematic study of biofouling and corrosion of OTEC heat excahnger has not started yet in Japan. Because almost all the fossil fuel and nuclear power plants are located on the sea side, biofouling studies have been made between power companies and the manufactures. Some of them have been published mostly in The Thermal and Nuclear Power (a monthly magazine). A conclusion[11] which is generally agreed to is that titanium tubes are the best against corrosion, errosion and biofouling. Even if they are contaminated more by small marine life than copper alloy tubes, this contamination is clearly eliminated by periodic washing with sponge balls.

Because Japanese fishermen are worried about the effect of cooling sea water on marine life, studies are being undertaken on this effect by fishery institutions and universities including the Marine Ecology Research Institute.

Table 4 Master program of development on OTEC power plant

PROGRAM ACTIVITY	Preparation (Phase 0) 1974-1977	1st Step (Phase I) 78,79,80,81,82,83	2nd Step (Phase II) 84,85,86,87,88,89	Commercial (Phase III) 1990....
SYSTEM ANALYSIS & ENVIRONMENTAL ASSESSMENT				
MAJOR SUBSYSTEM DEVELOPMENT				
Heat Exchanger				
Cold Water Pipe				
Platform Construction				
Station Keeping				
COMPONENT TEST & INTEGRATION				
1 MW OTEC System on a Barge				
Design, Construction, Operation				
ENGINEERING TEST FACILITY				
10-25 MW OTEC System at Sea				
Design, Construction, Operation				
COMMERCIAL DEMONSTRATION				
100 MW OTEC Plant				

Application and Assesment of OTEC Systems

Multiple applications of OTEC Systems have been studied in Japan since the beginning of OTEC research. Among them, some results of the studies on uranium exploiration by OTEC and the possibility of marine biological productivity enhancement were reported on last year in this conference.

Assessment of OTEC is being conducted by an oceanophysics group in collaboration with the marine biology group under the leadership of T. Teramoto of Ocean Research Institute of Tokyo University. The study covered the behavior of the discharged water from the plants and included the behavior of the plume from the outlet and the dependence on the current direction and velocity. The 100 MW OTEC plant was used in the study. Some results of the environmental studies were also reported on last year in this conference.

Future Plans and Conclusion

Table 4 is an OTEC power plants program proposed by the OTEC committee. The Phase 0 of the program is already finished according to this plan. The committee hopes to proceed the Phase I step involving the long term engineering test under the real ocean environment. The concept was described as the 1 MWe class power system on board a barge which could be applied by an oil tanker installed with a cold water pipe. This power plant will use ammonia as a working fluid and be equipped with test facilities to evaluate the environmental effects and station keeping device. Major specifications on this experimental facility are illustrated in Table 5.

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Table 5 Major specifications of 1 MW OTEC experimental facility

Items	Specifications
Gross electric output	1000 KW
Net electric output	624 KW
Working fluid	Ammonia
Warm water temperature	28°C
Cold water temperature	7°C
Quantity of warm water intake	12,640 m^3/h
Quantity of cold water intake	10,526 m^3/h
Total heating area of evaporator and numbers of unit	4,130 m^2 1 (plate type)
Total cooling area of condenser and numbers of unit	4,700 m^2 1 (plate type)
Numbers of turbine and rpm	1 3600 rpm
Diameter and length of cold water pipe	Dia; 1.3 m, Length; 500 m
Platform	Tanker 33,200 ton
Construction cost	3,036 million yen

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4. OTHER OCEAN ENERGY SYSTEMS

WAVES, SALINITY GRADIENTS, AND OCEAN CURRENTS - ALTERNATIVE ENERGY SOURCES

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Abstract

A brief history of three ocean energy programs - wave, salinity gradients and ocean currents - sponsored by the U. S. Department of Energy is presented. In addition to discussing the philosophical basis for choosing certain energy conversion techniques, a brief description and discussion of each technique is presented. Predictions as to the commercialization of each technique are also presented.

Introduction

In 1975 the Energy Research and Development Administration (ERDA) was formed to study all forms of energy starting from the evaluation of the resource through the energy conversion technology, environmental consequences, legal and political factors and finally to commercialization. The initial ocean energy effort was in ocean thermal energy conversion (OTEC) and under the direction of Dr. Robert Cohen. Although much interest was shown in the energy conversion of ocean waves, salinity gradients, ocean currents and tides, there were no formal programs in these areas. In mid 1975, the author joined Dr. Cohen on a part-time basis to study these other ocean energy options.

In 1976 a tidal program was formally initiated and placed under the direction of the Geothermal Branch of ERDA. Furthermore, small studies were initiated in the areas of waves, salinity gradients and ocean currents. In 1978 ERDA was absorbed in the newly created Department of Energy (DOE), and, after three years of direction by the author on a part-time basis, these three alternative ocean energy options were assigned to full-time DOE personnel.

The purpose of this paper is to describe the present DOE sponsored efforts in the energy areas of ocean waves, salinity gradients and ocean currents. The philosophical and the programatic aspects are discussed.

Ocean Wave Energy Conversion

Ocean waves have excited the imaginations of many inventive people down through the ages. There are numerous patented and referenced attempts to harness the energy of ocean waves dating back to the mid 19th century. Since very few ideas are new, most of these energy conversion ideas fall into nine categories listed by M. McCormick[5] in his 1978 paper. These are:

a. Heavying Bodies
b. Pitching or Rolling Bodies
c. Cavity Resonators
d. Wave Focusers
e. Pressure Converters
f. Surging Bodies
g. Flapping Bodies
h. Rotating Outriggers
i. Combinations of the above

Categories a. and b. have been the most popular devices used by inventors; while categories e., f., g., and h. follow closely. Cavity resonators have recently come into vogue primarily due to the excellent work of Y. Masuda.[3] See, for example, his 1978 paper. While Masuda concentrated on the experimental aspects, McCormick[4] (1976) concentrated on the theoretical aspects. Under the auspices of the International Energy Agency (IEA), Canada, Ireland, Japan, the United Kingdom and the United States have undertaken a joint full-scale wave energy conversion program to be conducted in the Sea of Japan in 1979 and 1980. Both Masuda and McCormick are directly involved in this study. The U. S. effort is being supported by the DOE. The IEA study will be conducted on a specially designed floating body called the "Kaimei", which is sketched in Figure 1. There are three air turbines positioned on the deck of Kaimei which have been designed and constructed in Japan, the United Kingdom and the United States. These turbines are excited by the air motions above the rising and falling of the internal water surface as sketched in Figure 2. Each turbo-generating system is designed to deliver an average electrical power of 125 KW in a sea having an average height of 2m and a period of 6 seconds.

The internal water column is excited by the external waves which can be in resonance with the cavity. This resonant frequency value is obtained from the equation

$$\omega_c = \sqrt{\frac{g}{L_1}} \tag{1}$$

where L_1 is the still-water length of the internal water column. The internal water column can also be excited by the ship motions of Kaimei. For example, the heaving frequency of Kaimei is obtained from

$$\omega_z = \sqrt{\frac{\gamma A_w}{m+m_w}} \tag{2}$$

where γ is the weight density of sea water, A_w is the water-plane area of Kaimei, m is the mass of Kaimei and m_w is the added mass. Obviously, the optimal design condition is $\omega_c = \omega_z$, or the water column length is

$$L_1 = \frac{\rho A_w}{m+m_w} \tag{3}$$

where ρ is the mass density of sea water. Ballast can be used to adjust L_1 and m in equation (3) for optimization.

A second DOE-sponsored effort is in the area wave

*Professor and Director, Ocean Engineering

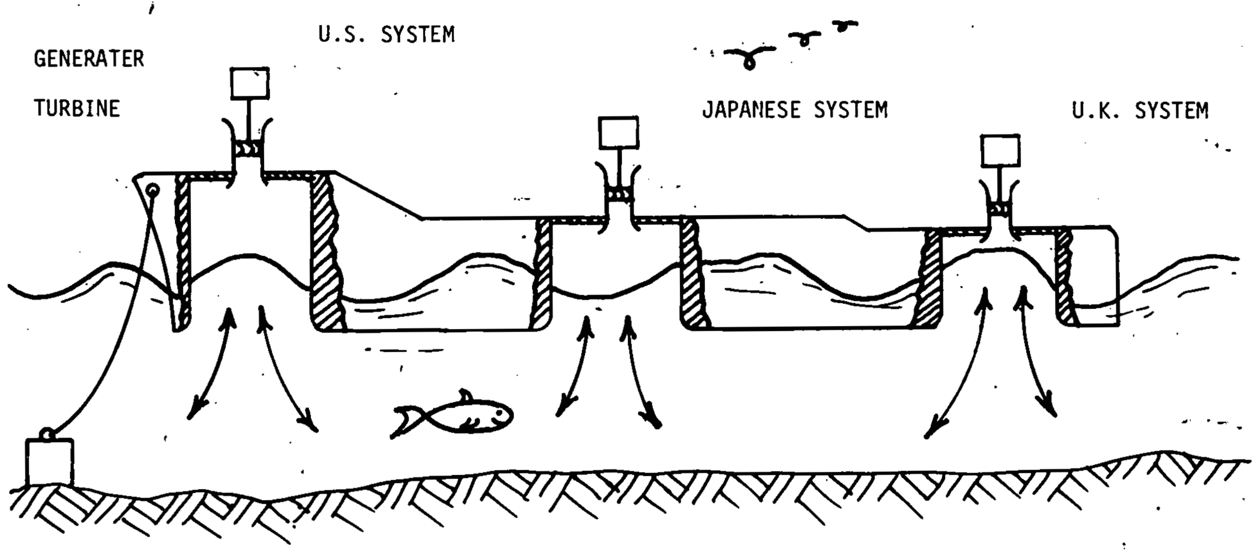


FIGURE 1. Sketch of the "Kaimei" Floating Wave Energy Conversion System.

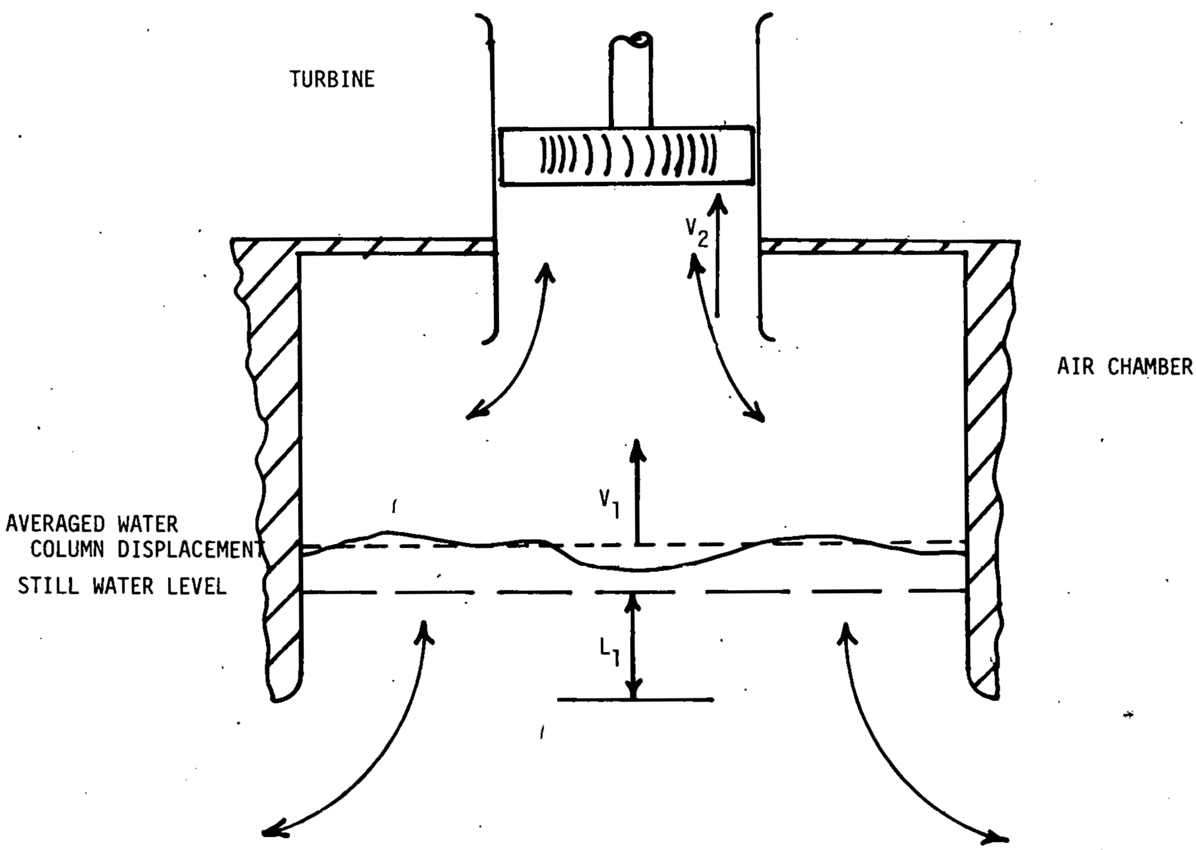


FIGURE 2. Sketch of the Water Column/Turbine System.

focusing systems. Wave focusing is accomplished by four techniques:

1. Radiant wave interaction. See Figure 3a.
2. Fresnel-type focusing. See Figure 3b.
3. Refraction. See Figure 3c.
4. Channeling. See Figure 3d.

Radiant wave interaction occurs when a body motion is in resonance with the incident wave. The motion may be heaving, pitching, rolling or cavity. Falnes and Budal[2] (1978) have shown that the wave power under a resonant condition approaches the body from a crest width of $\lambda/2\pi$ as illustrated in Figure 3a.

Fresnel-type focusing is accomplished by placing a lens type structure in the wave causing either diffraction or refraction. Referring to Figure 3b, a submerged platform, shaped like a lens, causes the incident waves to slow down as the waves pass over the straight leading edge onto the platform. When the waves leave the plane at the curved edge they are directed to a focal point. This point is found by applying Snell's law:

$$\frac{c_1}{c_2} = \frac{\sin(\alpha_1)}{\sin(\alpha_2)} \tag{4}$$

where $\underline{c_1}$ is the phase velocity in the shallow water over the platform, i.e.

$$c_1 = \sqrt{g\,h_1} \tag{5}$$

$\underline{h_1}$ being the depth of the plane of the platform, and c_2 the phase velocity in deep water, i.e.

$$c_2 = gT/2\pi \tag{6}$$

$\underline{T}$ being the wave period. The combination of equations (4) through (6) yields

$$\sin(\alpha_2) = \frac{1}{2\pi}\sqrt{\frac{gT^2}{h_1}}\,\sin(\alpha_1) \tag{7}$$

$$= \frac{1}{\sqrt{1 + (\frac{dy}{dx})^2}}$$

where $y = f(x)$ is the equation of the curvature of the lee-edge. If the lee-edge is a circular arc of radius $\underline{R}$ then the focal length, $\underline{F}$, is obtained from the lens equation

$$F = \left(\frac{1}{c_2/c_1 - 1}\right) R \tag{8}$$

For example, a 6-second wave traveling over a platform 5 meters below the still water level and having a radius of 100 meters will focus on a point 296 meters from the platform's leading edge (the chord of the circle). Thus, if the central angle of the arc, $\underline{\theta}$, is 90^0, then the focused wave energy is from a crest length (chord) of

$$\ell = 2R \sin \frac{(\theta)}{2} \tag{9}$$

$$= 141 \text{ m}$$

A refraction wave energy conversion study is being conducted at the Lockheed California Company by Mr. Les Wirt. Referring to Figure 3c, Mr. Wirt has designed an acoustically-shaped submerged dome which causes incident waves to refract and focus on a vertical axis turbine located at the center of the dome. The wave energy available for conversion by the refracting device, called DAM-ATOLL, comes from a crest width equal to the base diameter of the dome. Results of initial studies of the DAM-ATOLL concept are very promising.

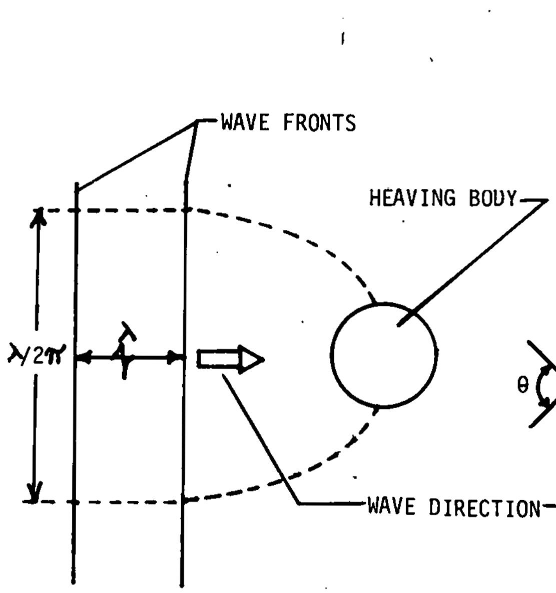
Wave energy focusing by converging channels, as sketched in Figure 3d, has been suggested by a number of individuals including an extensive experimental study performed by Bruun and Viggosson (1977).[1] Although the technique effectively focuses waves with a resulting increase in energy intensity, it appears to be feasible only in or near the surf zone where the wave energy is relatively low. Thus, DOE has not sponsored studies in this area.

The three British wave energy conversion systems known as Salter's Ducks, Cockerel's Rafts and the Russell Rectifier are being theoretically studied by Professor C. C. Mei at Massachusetts Institute of Technology under contract to DOE. It is hoped that the results of Professor Mei's study will help predict the behavior and response of the British systems under real sea conditions.

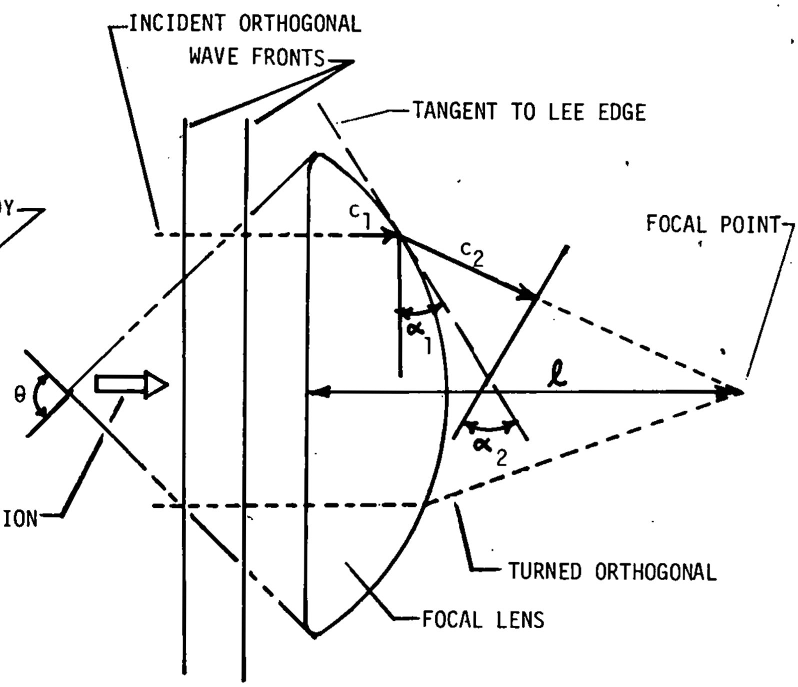
The reasons for concentrating the U. S. wave energy program on both focusing systems and the cavity resonater are the following: First of all, the capture of wave energy from wide expanses of crest length is impractical. The costs of large ocean wave energy conversion structures are enormous and for the most part, the wave energy available does not justify these costs. Furthermore, large structures close to the shore may be environmentally unacceptable. Thus, if the wave energy over a broad expanse can be focussed on one energy converter, the energy conversion might then be both cost effective and environmentally benign. Secondly, the idea of having large structures undergoing significant motions while moored at sea goes against the grain of ocean engineers. We normally design mooring and anchoring systems to minimize motions. To design otherwise is extremely expensive and, as was concluded at the Wave Energy Conversion Symposium held at Heathrow Airport (London) in November, 1978, is cost-ineffective with the British systems.

In addition, DOE is sponsoring four other small studies in wave energy conversion. Two of these involve energy conversion techniques involving protonic conduction and piezoelectricity, respectively. Protonic conduction, being studied by Professor Robert Solomon of Temple University, shows promise for the conversion of wave induced kinetic energy into electrical energy. Dr. George Taylor of Princeton Resources, Inc., has studied piezoelectric energy conversion associated with ocean waves; however, the best projected conversion efficiency is only 1%. Thus, no further work is projected in this area.

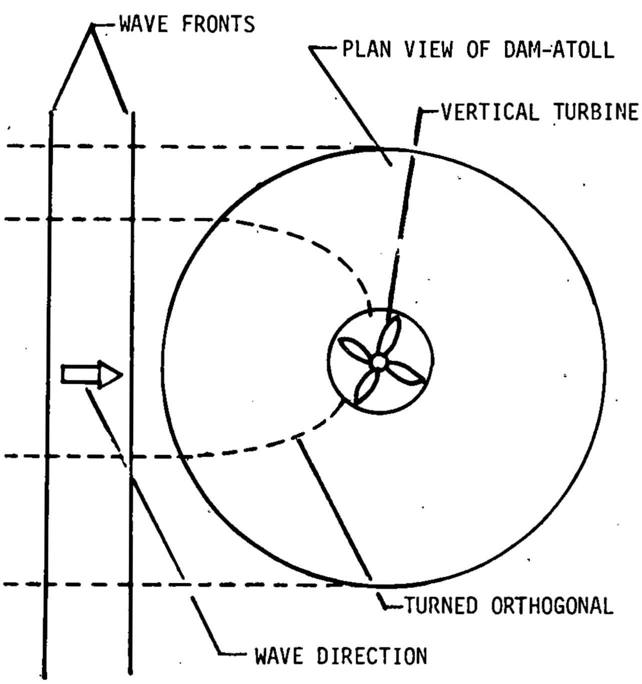
Finally, two studies are being conducted of small



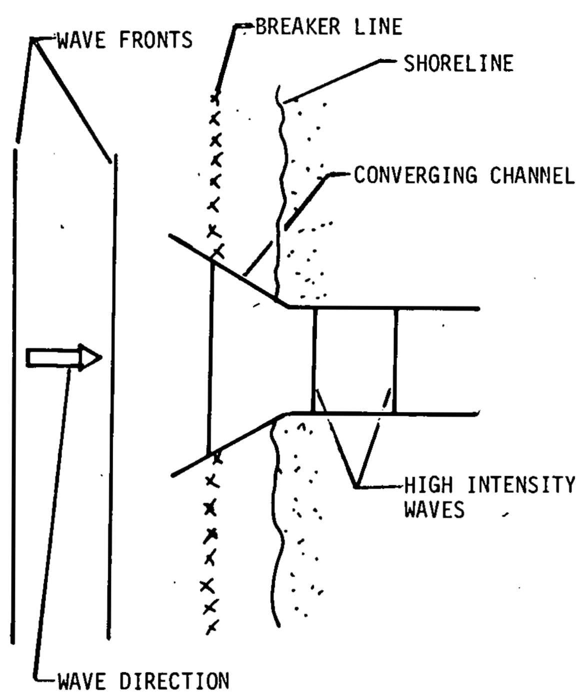
a. Radiant Wave Interaction.



b. Fresnel-Type "Lens" Focusing.



c. DAM-ATOLL Refraction Focusing.



d. Focusing with a Converging Channel.

FIGURE 3. Four Wave Focusing Schemes.

heaving systems. The first of these is by James Ringrose of the Wave Corporation of Newport, Oregon. This involves an at-sea test of a heaving float with a large vertical-axis flexible-blade propeller attached to the underside. The second study is by Dr. Larry Slotta of Corvallis, Oregon. Dr. Slotta is performing an analytical study of a semi-constrained float, suggested by Mr. Dedger Jones. These last studies are presently underway, and significant results have not yet been obtained.

The DAM-ATOLL and cavity resonance systems have an excellent chance of being commercialized on a large scale by the mid 1980's. We should then see a significant impact on energy supplies in our coastal regions by the year 2000.

Salinity Gradient Energy Conversion

Salinity gradients occur naturally where fresh water rivers meet either the oceans or the hypersaline bodies of water such as the Great Salt Lake or the Dead Sea. There are two techniques which can be used to convert the energy of mixing of high saline and low saline waters, both involving semipermeable membranes. The first, and best known, is osmosis, while the second is reverse electrodialysis.

If a semipermeable membrane separates salt water of $35^0/oo$ (35 parts per thousand by weight) salinity and fresh water an osmotic pressure gradient of 23 atmospheres will exist across the membrane. If the saline solution is saturated ($260^0/oo$) then the osmotic pressure is about 370 atmospheres. To convert the osmotic pressure, therefore, one simply allows this pressure to build up in a diluted saline solution and release the pressurized solution through a hydroturbine. To this end DOE has sponsored Inter Technology/Solar, Inc. of Warrenton, Virginia and Clarkson College of Technology to further study the concept. Bend Research of Bend, Oregon studied both commercially available and experimental membranes with the objective of energy conversion. Following these studies, DOE is now supporting Ebasco, Inc., to perform a systems design of a 50 KW osmotic power unit.

Reverse exlectrodialysis is accomplished by alternately placing anion-permeable and cation-permeable membranes in a battery type of container. Salt water is then passed between alternate membrane pairs while fresh water separates one pair from another. The salt, which is ionized in solution, looses the sodium ions to the cation-permeable membrane and the chlorine ions to the anion-permeable membrane. The positive and negative charges on the respective membranes are then transferred to electrodes located at the ends of the membrane stack. Thus, the system then acts as a battery. Under contract to DOE, Southern Research Institute undertook a parametric study of the dialytic battery, while Inter Technology/Solar, Inc., is presently studying a 100 W model of the battery.

The primary problems in both of these salinity gradient technologies lie in three areas: membrane technology, pretreatment of feed brines and waste brine disposal. Initially, most emphasis was placed on membrane technology; however, the emphasis has recently shifted to the latter two areas. Since membranes are subject to attack by organisms and other impurities the low salinity solvent and the high salinity solution must be pretreated before entering the power systems. It has been stated that this pretreatment will be critical in determining the cost of the produced energy. As to the disposal of waste brines, environmental concerns prohibit us from dumping the waste brines from the power plants. InterTechnology/Solar, Inc., is now developing a closed system technique which will unmix the waste brine solution so that it may be used again in the power system. The energy required to unmix is supplied by the sun.

Osmotic pressure conversion has by far, the greatest energy potential of the two salinity gradient energy conversion techniques. The technology required in this energy conversion is, however, in the future. Furthermore, this osmotic technique requires large volumes of fresh water and therefore, is very site specific. The reverse electrodialytic technology is essentially in existence. Although the power potential of this method is much lower than that of osmosis, reverse electrodialysis is not site dependent. Thus, an operational 50 KW system should appear by 1985.

Ocean Current Energy Conversion

The primary U. S. resource for ocean current energy conversion is the Florida Current. Presently, DOE is supporting AeroVironment, Inc. of Pasadena, California in preliminary studies of a large ducted turbine for the purpose of ocean current energy conversion.

The turbine is designed to convert 75 MW of power and AeroVironment, Inc., has proposed mooring 132 of these turbines in the Florida Current to deliver 10,000 MW to the Florida power grid. The turbine blades have a unique design, being supported around the periphery by liquid bearings.

The problems presently being studied are associated with the possible environmental effects, hydroelastic stability of the rotor blades and the mooring and anchoring systems. The preliminary results of the AeroVironment study are most promising.

Summary

Starting with no budget in 1975, the (then) Energy Research and Development Administration (ERDA) and now the Department of Energy (DOE) have made great advances in the energy conversion of ocean waves, salinity gradients and ocean currents. The budget in the present fiscal year well exceeds $1 M. Advances made in these three areas allow this writer to predict commercialization technologies in these areas before the year 2000. By the first quarter of the next century, the cumulative contribution to the U. S. energy market should be significant.

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DISCUSSION

Question: I have a simple question on the Gulf Stream turbulence: How do you take care of torque, the rotation?

M. McCormick: There are multi-rotors traveling in both directions, i.e., counter-rotating. The anchor system is composed of two mooring lines at about 45° in a beam orientation. Incidentally, we have just received the final report on June 15 that concerns the blade performance and mooring systems. The model has been on display in the Gibbs-and-Cox booth. There is also a model of a single-blade turbine, which doesn't have the counter-rotating feature, but which ran very successfully in the David Taylor Model Basin in March 1979.

Question: Does anyone have any plans to use the brine produced by the leaching of salt domes?

M. McCormick: No, actually when we found that there was a plan to replace the brine in the salt domes by petroleum, we, the technical community involved with salinity gradient conversion, did protest. In fact, in an article in Science about a year ago by Prof. John Isaacs, it was shown that we were throwing away a resource (that is, the brine) that was environmentally more acceptable, and offered greater total energy, than the energy we were storing, but wisdom does not always prevail.

Question: You mentioned that the Japanese are developing a wave machine with multiple chambers. Do you expect mutual interference between the various chambers?

M. McCormick: Yes, I do. In a test that we ran last year at the Naval Academy, one of my students tested three chambers. We found a reduction in the energy from the leading to trailing chamber. Naturally, if you are taking energy out of the wave at one point, the remaining energy must be lower. So there will be a reduction from bow to stern in wave power available. What the Japanese are hoping for — and I don't know whether this will be true — is that the wave energy from the sides will recover fast enough so that the energy drop from the bow to the stern will not be too significant. Nature really can't stand a void in energy. If you have a high energy region and a low energy region in the same neighborhood, the energy should flow along the crest to fill in the void.

Question: Is there any RFP in the U.S. on tidal energy improvements?

M. McCormick: Not that I know of. There was one a few years ago. In 1976, because of Congressional pressure, ERDA did have a tidal energy program under the geothermal group. The geothermal group then sponsored a study by Stone and Webster. A two-volume report came out by W. W. Wayne, which is an exhaustive, authoritative document, that sets tidal power in its true perspective. He found that the Red Chinese have 44 tidal power plants in operation and have another 80 planned for a total power output of 7500 MW. But he concluded that, as far as we are concerned, the Bay of Fundy is the only tidal resource in North America; it is owned by Canada. The Canadian Maritime Provinces do not need that much power, so it would be sold to the New England states. It is up to the Canadians to develop the resource with our help, of course, if it is ever going to be developed.

One last word. The Ocean Systems Branch of the Department of Energy has a Program Summary that summarizes the objectives, funding, results, and status of all active contracts, and the program plans for the various categories. You can get a copy by writing Dr. Lloyd Lewis of the Ocean Systems Branch. I hope that sometime this fall we can have a special conference on waves, currents, and salinity gradients and we may even invite the OTEC people to it.

5. CLOSED-CYCLE OTEC PLATFORMS

OTEC OCEAN ENGINEERING PROGRESS REPORT

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Abstract

Recent accomplishments in the Ocean Engineering Program are reviewed, and the development of the subsystems from conceptual design through at-sea-testing is discussed. The Ocean Engineering Program is aimed at starting preliminary design of a 10/40-MWe plant in October 1980. ~~The paper reports~~ the status of the development is reported and ~~outlines~~ the plans for continuing development are outlined. The authors judge that validated design methods will be available in time to support the preliminary design effort and that a wide range of concepts is available on which preliminary designs can be based.

Introduction

The broad objectives of the OTEC program are to extract solar energy that has been stored in the surface water of the oceans and to deliver that energy to the continental United States at a price at which the market place is willing to buy it. To achieve this objective, three areas (in an elementary sense) are under investigation in the OTEC program: first, the thermodynamic apparatus (power system) needed to extract the stored energy from the resource, second, the structure that supports and contains that apparatus and delivers the resource to it (these are the principal areas under the Ocean Engineering Program); and third, the means of delivering the extracted energy to the consumer.

Substantial progress has been made toward the solutions of the major ocean engineering problems that must be solved before an OTEC system can be designed, constructed, and deployed to produce usable power for a consumer market. This paper describes, in broad terms, the total thrust of the Ocean Engineering Program so that the purposes of the individual elements of that program will be understood. Major advances that have been made in the last year or so are summarized to illustrate and document the continuing developmental process. The interrelationships among the individual advances will be shown, as well as the way that these advances fit into a coherent engineering development plan.

Objectives and Approach

Near-Term Objectives

An engineering development program must focus on specific objectives. Therefore, the ocean engineering work discussed in this paper is based on the following specific near-term OTEC program objectives:

1. Develop, in the near future, a technology base that will make it possible to design, build, deploy, and operate an OTEC system having a capacity of 40 MWe, and to deliver the output of the system to an island market.
2. Investigate and eliminate any design features that would prevent the 40-MWe system from being generally accepted as a demonstration of the feasibility of building and operating plants with an order of magnitude greater output (a commercial plant), thus outlawing any "cute trick" that would work at the 40-MWe size, but surely not at the 400-MWe size.
3. Know enough about the possible commercial OTEC plants to provide direction to 2 above.
4. Provide the government with the technical information needed to judge industry's proposals for plant designs, performance, and costs.
5. Provide industry with demonstrated solutions of the pacing technical problems that they may or may not use, as they see fit, for commercial plants.

Approach

The approach taken by the Ocean Engineering Program is the traditional one used by development engineers consisting of the following steps:

1. Make a conceptual design of the device (system) that will perform the desired function.
2. Identify the subsystems, determine each subsystem's performance requirements, and determine the interface requirements.
3. Describe the system and its performance in analytical terms, i.e., model the real world with symbols. Use this *analytical* model to describe the performance of the subsystem.
4. Make a preliminary design of the system.
5. Build a *physical* model of the system at some appropriate size (scale) and subject it to an excitation (input). Measure the response to see if the analytical model does indeed predict the response.
6. Test the critical components at some appropriate scale under the loads predicted by the analyt-

ical model and verified by the physical model.

7. Test large-scale or full-size critical subsystems at sea.

During the period treated in this paper (May '78 — May '79), work in each of the categories outlined above has been proceeding, and some subsystems are farther along the development path than others. The following sections show the structure of the Ocean Engineering Program and how individual projects support the overall development plan and the OTEC objective.

Platform Designs

Baseline Commerical Platform for the Tampa Site

As previously stated, the 40-MWe plant to be used in the island market must not have any characteristics that would make it different in principle from a 400-MWe commerical plant (the evolutionary blind alley). But to meet this requirement, the major characteristics of commercial plants, called the top level specification, must be known. To accomplish this, a baseline commercial study was performed.

In early 1978, each of three contractors, Gibbs & Cox, Inc. (G&C), Morris Rosenblatt & Sons, Inc. (MRS), and Lockheed Missiles and Space Co. (LMSC) did conceptual designs of commercial size (400 MWe) platforms. The results of those studies have been reported in the literature.[1 2 3] The designs prepared by the three contractors were for different platform types at different sites having different design conditions. The design study served its purpose — answering the question "what do commercial plants having different platform types for different sites look like?"

This year J.J. McMullen & Associates (JJMA) conducted an analysis of those three studies and then synthesized them into a baseline design.[4] To focus the work, JJMA was asked to answer these questions:

1. Based on the information reported by the three contractors, if today we wanted to build a 400-MWe plant to be moored off the West Coast of Florida, what conceptual design should be used?

2. Give the reasons for the choice.

3. What would it cost?

The method used by McMullen in the study was to:

1. Identify differences in approach between the three contractors. Carry out trade-offs or engineering analysis, as appropriate, and resolve these differences.

2. Modify, as required, the OTEC platforms systems to suit the Tampa site.

3. Identify discrepancies in costing approaches, and modify costs, as required, to reflect the above findings.

4. Develop evaluation criteria; apply these criteria to the data generated in this study; and select a preferred concept.

The same cold water pipe (CWP) material—lightweight, reinforced concrete—and the Tampa design conditions were used for all platforms. The pipe analysis was done by Hydronautics using the NOAA/DOE/program. The pipe was hinged at the platform, with flexible joints every 112 feet.

All outfitting and subsystems were included with the exception of the power and the energy transfer subsystems. Costs are presented according to the work breakdown structure.

The final ranking is given in Table 1.

Table 1 Estimated annualized costs[a] of platforms for 8-plant, 3200-MW_e energy park off Tampa, Florida

Cost or power output	G&C ship[b]	LMSC ship[c]	LMSC spar[d]	MRS spar[e]
Capital cost, $M/yr	494	430	378	485
Operating cost, $M/yr	121	129	129	118
Total, $M/yr	615	559	507	603
Net power, MW_e	3262	3267	3316	3286
Cost, $/$kW_e$/yr	189	171	153	184
Relative cost	1.23	1.12	1.00	1.20

[a]For 12% return on investment and 20-yr writeoff.
[b]Steel ship designed for external HX.
[c]Concrete ship designed for external HX.
[d]Concrete spar designed for external HX.
[e]Concrete spar designed for internal HX.

The conclusions of the McMullen study are:

1. The most cost-effective platform for the Tampa site is the LMSC concrete spar.

2. The costs for an eight-unit 3200 MWe(net) park are:

   Capital investments $2,818 million ($850/kW_e)

   Annual operating costs $129 million

3. The ship is less attractive than the spar in a severe environment, but is a viable alternative in benign environments.

4. Compared to a spar, ship shapes suffer from higher mooring costs, higher deployment costs, and higher CWP costs.

Land-Based Plant

One of the ways of solving ocean engineering problems is to avoid them. A land-based plant would avoid all of them except the CWP problem. Although there are probably many places in the world where a land-based plant is feasible, Hawaii and Puerto Rico are two sites of interest that have deep water close enough to shore to make the scheme attractive. Conceptual designs of land-based plants, at Keahole Point, Hawaii and at Punta Yeguas, Puerto Rico were made by Deep Oil Technology (DOT) (a subsidiary of Fluor Corp.) and the results reported in Ref. 5. In general terms, the results indicate:

1. Building plants similar to the OTEC plant on the shore is done regularly, and cost estimating methods are refined and accurate. Coastal power plants and liquified natural gas terminals are examples.

2. Bringing the cold water, the hot water, and discharge pipes through the surf zone and laying them in shallow water is straightforward, as the techniques have been developed for coastal power plants and sewer outfalls.

3. The major technical problem is installing the CWP up the slope from the deep anchor at the 3000-ft depth to join the pipe laid through the shallow water.

Because of the different profiles shown in Fig. 1, two different CWPs were designed. For Hawaii, the pipe is concrete and is installed piece by piece up the slope which is quite regular. On the other hand, the slope in Puerto Rico is very rough and steep, so bouyant fiberglass pipe, anchored at many locations along its length, was designed for use here. The fiberglass portion runs from the deep water intake to the end of the surf zone concrete pipe. The fiberglass pipe is constructed ashore, joined into one piece in a harbor, towed to the site, and installed in one piece.

To the level of exactness that costs are known, the costs of these land-based plants are estimated to be about the same as for the floating options.

The idea is attractive for some applications, but it does suffer from the "blind alley" problem; i.e., it will not be the solution that will provide significant energy to the U.S.

APL Plantship

The Applied Physics Laboratory of The Johns Hopkins University has designed a plantship that offers a significant alternative to the moored plant. APL and their subcontractors have continued to advance the design of a ship that is not moored, but grazes, and instead of sending power ashore through a cable, uses the power generated aboard to produce an energy-intensive product, which is then shipped ashore. The product being considered is ammonia, but aluminum, another candidate, is under study.

The study of the grazing plantship is farther along than any of the other 40-MWe options because it started earlier and has advanced now through preliminary design. The results of the design study have been issued in a report.[6] A perspective view of the ship taken from that report is shown here as Fig. 2.

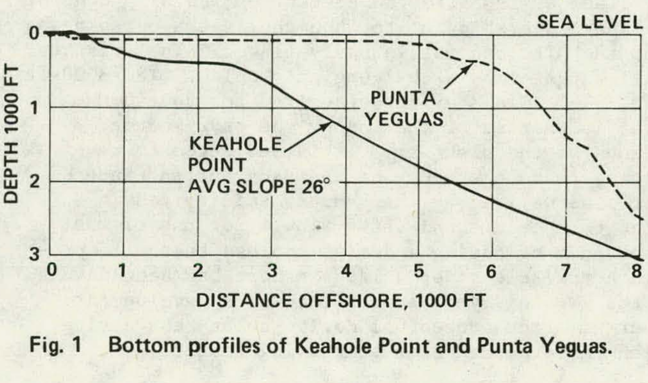

Fig. 1 Bottom profiles of Keahole Point and Punta Yeguas.

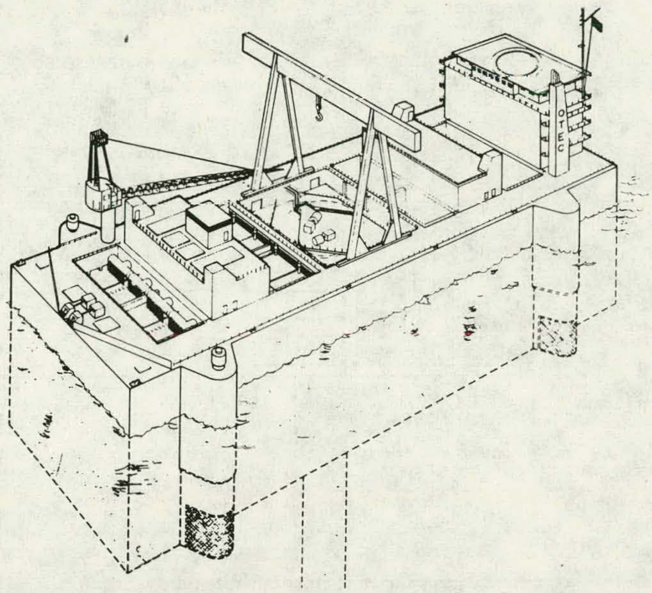

Fig. 2 OTEC pilot plantship.

The preliminary design has several interesting features. First, the absence of a shore cable is extremely attractive. As will be seen later, the riser cable interacts with the moor and cold water pipes for plants designed to deliver power to a shore grid. Avoidance of this cable reduces the complexity of the systems problem. Second, the absence of a moor reduces costs. The seakeeping calculations show that the ship can successfully ride out the 100-year storm at the site off Brazil for which she was designed.

But one cannot get something for nothing, and although it is beyond the scope of this paper to discuss the costs associated with production, transportation, and marketing of a product, one should be aware that, although the ocean engineering problems are reduced, one might be trading them for others. This is not intended to denigrate the plantship concept in any way but merely to place the situation in perspective.

Spar Platform

The third 40-MWe conceptual design was produced by Gibbs and Cox based on a spar configuration. The arrangements are shown in Fig. 3, and a photo of a model is shown in Fig. 4.

The central cylindrical hull provides the primary buoyancy and contains the power generating equipment. External, detachable spar-shaped modules contain the heat exchangers, pumps, etc. During assembly of the system, the power modules are towed to the floating platform and attached to the hull. The modules can be detached if this becomes desirable for repair or replacement. (As noted earlier, this approach was proposed by LMSC early in 1975.)

The platform is moored through the CWP, which will be discussed later. This arrangement allows

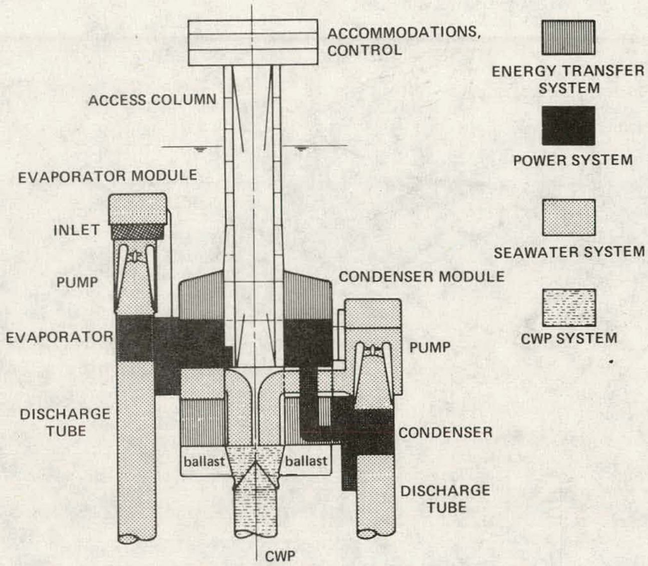

Fig. 3 Arrangement of the 40-MWe spar.

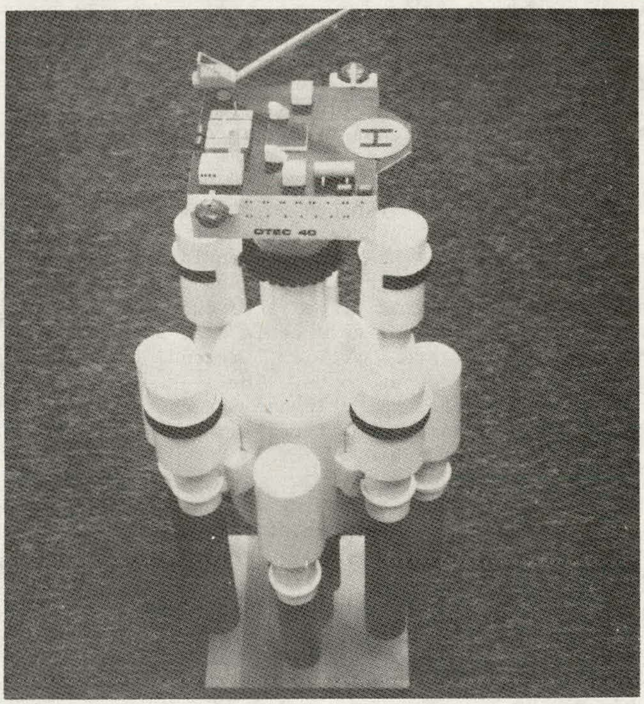

Fig. 4 Model of the 40-MWe spar configuration.

the shore cable to be integrated into the moor and the CWP, which will reduce the effects of the waves and currents on the cable.

Constructability

Deep Oil Technology (DOT) and Det Norske Veritas (DNV) had a contract to examine the feasibility of constructing the 400-MW platforms that had resulted from the earlier commerical platform studies.[1 2 3] DOT examined the steel platforms while DNV examined the concrete platforms. DNV has had extensive experience with the design and construction of the very large concrete structures that have been installed in the North Sea. They are particularly knowledgeable about the "Norwegian" method of building, in which the bottom structure is fabricated in a shallow graving dock and then floated to deep water, where the construction continues. As the draft increases, the structure is moved to deeper water. After the structure is completely built and outfitted, it is towed to its operating site.

The study concluded: (a) up to the 400-MWe size, ships and spars of either steel or concrete could be built, (b) platforms with external heat exchangers are easier to build because they are smaller, (c) the only practical way to build 400-MWe spars with internal heat exchangers is to use the Norwegian method, which requires deep, sheltered water as is found in the northwestern part of the U.S.; and (d) either 40-MWe concrete or steel OTEC plants can be built at a number of U.S. sites.

Platform Conclusions

The work on platforms has produced three concepts: the grazing plantship, the moored spar, and the land-based plant. The plantship is almost through preliminary design and the other two are ready to start that phase. The advantages and disadvantages of each have been learned, the problems discovered, and the ongoing work organized to solve them, as will be discussed later.

Station Keeping

Two contracts for the conceptual design of a station keeping subsystem have been awarded recently — one to M. Rosenblatt and Sons, Inc. and the other to Lockheed Missile and Space Co. The task is to design a moor for the APL barge and the G&C spar at the Puerto Rico site. The final objective of the tasks is to produce preliminary designs with specifications and costs of moors for these two platforms.

The work that JJMA did in the commercial baseline study shows how sensitive the mooring design, and possibly even the choice of platform, is to the environmental conditions given to the designers. For example, a design current of 5 kt is being used for the Tampa site. If that current is actually 4.5 kt instead of 5.0, it will make an enormous difference in the design, since the drag forces vary as the square of the current velocity. Good site data are essential. Statistical distributions of the currents, winds, and seas and their directions are equally important because ultimately probabilistic design techniques will have to be used to quantify risks. The efforts of the oceanographers to furnish accurate data are being encouraged and applauded. Without accurate data, the platform might be underdesigned, which might be fatal, or overdesigned, which could make the plant noneconomic.

The Cold Water Pipe

The CWP is often the element of the OTEC concept the "nay sayers" point to when they set about proving that the OTEC concept is badly flawed. There is no denying that it's a challenging problem. The 3000-ft long, 30-ft diameter pipe required for the 40-MWe plant is excited at the top by the wave-induced motions of the platform. It is subjected to wave forces, to both steady and unsteady current-induced forces as well as by the hydrodynamic forces induced by the internal water flow. Because of the importance of having a design process that will produce a reliable pipe, a large effort has been devoted to the CWP this year across the whole development spectrum, from conceptual design to at-sea testing, which has produced some gratifying results.

Feasibility and Concept Design

Two teams, TRW with Global Marine Development, Inc. (GMDI) and Science Applications, Inc. (SAI) with Brown and Root Development, Inc. (BRDI), recently completed the conceptual phase of a project that has the objective of producing preliminary designs of cold water pipes for a 10/40-MWe plant.[7 8]

The pipes were designed for two platforms, each at a different site. The first platform is the G&C spar at Puerto Rico, and the second platform is the APL plantship 200 miles off Brazil.

After examining a very large number of candidate concepts, each team presented its recommended concepts and ranked those concepts in order of merit, taking into account cost, technical risk, including construction and deployment, and operational considerations.

After a review of the conceptual designs, the contractors were authorized to proceed into the preliminary design phase for three designs each. The designs chosen are:

SAI-BRDI

1. Steel — reinforced thin shell,
2. Fiber-reinforced plastic sandwich, and
3. Bottom mounted.

TRW-GMDI

1. Fiber-reinforced plastic sandwich construction,
2. Elastomer — reinforced rubber with ring stiffener, and
3. Polyethylene — single or multiple pipes.

Botton-Mounted Pipe

A concept study of a bottom-mounted CWP was made by Ocean Resources Engineering, Inc. (ORE). ORE has experience in the design and deployment of marine risers used for drilling in very deep water by the offshore petroleum industry. The design was for a 30-ft diameter pipe for a 40-MWe plant off Puerto Rico. The material chosen was steel "... because it is the material exclusively being used by the petroleum industry in structures of this type and because it has high strength, uniform properties, and is easily fabricated using common techniques."

Figures 5 and 6 show the concept as presented in the ORE report.[9] The results of the study can be summarized as follows: (a) the bottom-mounted buoyant pipe is within the state-of-the-art, (b) the wall thickness of the pipe will be 1.0 to 1.5 in., so it can be fabricated easily, and (c) development effort will be required for the slip joint or other heave compensation that permits the ship to move with respect to the pipe and the flexible joints.

This concept is interesting and deserves to be developed in greater detail. It offers a solution to the routing of the cable. While the problem of stress in the pipe is reduced, the problem now becomes designing the heave compensation devices.

Tensioned Pipe

Gibbs & Cox in their conceptual design of a 40-MWe system using a spar configuration, has pre-

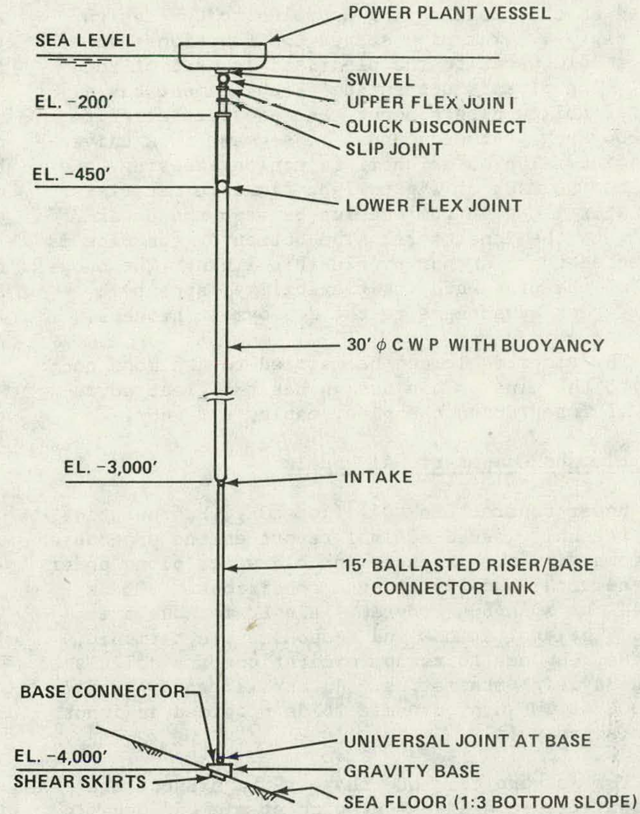

Fig. 5 General configuration of bottom-mounted CWP.

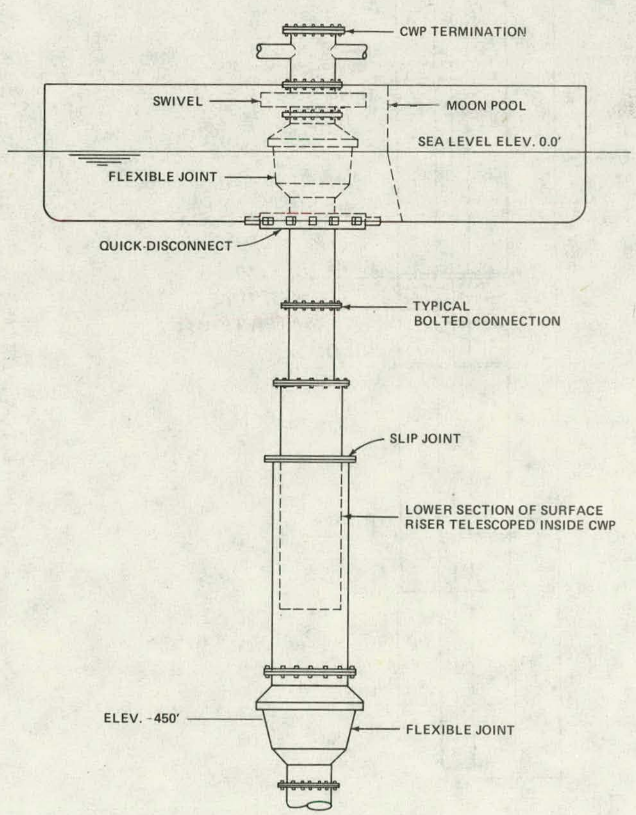

Fig. 6 Upper CWP configuration.

sented another interesting approach to CWP design (see Fig. 7). The pipe serves two functions; it brings cold water to the plant and is part of the moor. The pipe is segmented. Each segment has a central hollow pipe element that provides strength and bouyancy. At each end of a segment is a universal joint. The outer shell is nonload bearing, and thus can be made of lightweight flexible materials. The water flows in the annulus between the outer shell and the inner core. The bottom of the pipe is connected to an anchor by flexible links. The buoyancy of the pipe, and its flexibility, appear to offer great advantages to the deployment process.

The riser cable can be married to the moor and then to the pipe. This design has the great advantage of integrating the pipe, cable, and moor.

Analytic and Computational Models

Under contract to NOAA (for DOE), Hydronautics, Inc. recently issued a final report on the procedures for computing the stresses in cold water pipes under environmental and operational conditions.[10] It is called the NOAA/DOE Program. Linear methods are used. The excitations and responses are harmonic, and the response to random excitations are calculated using Rayleigh statistics. An earlier study by SAI for DOE on CWP hydrodynamic loads provided an input to this analysis.

The authors conclude that: "The methods are in generally good agreement with static and dynamic analytical solutions and available model tests and are suitable for use in CWP design studies and designs at least up to the preliminary design level."

In the conceptual CWP design studies previously discussed, TRW used a time domain solution developed by Galef of TRW which was discussed in Ref. 8. Figure 8, taken from that report, shows a comparison of the bending stresses in a concrete pipe for the APL ship computed by three different computational methods: TRW, NOAA/DOE, and Paulling. The NOAA/DOE and Pauling programs are frequency domain programs. Figure 9 shows comparisons of the bending stresses for a steel pipe.

It is encouraging, indeed, that the different computational methods based on different analytical

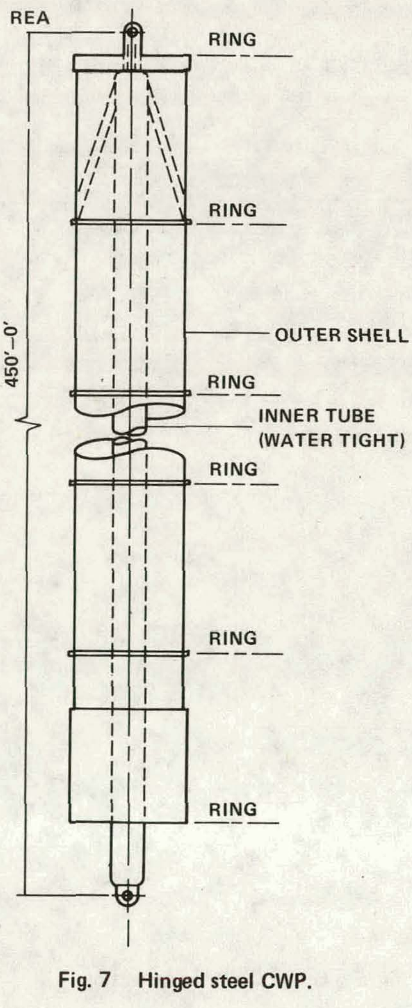

Fig. 7 Hinged steel CWP.

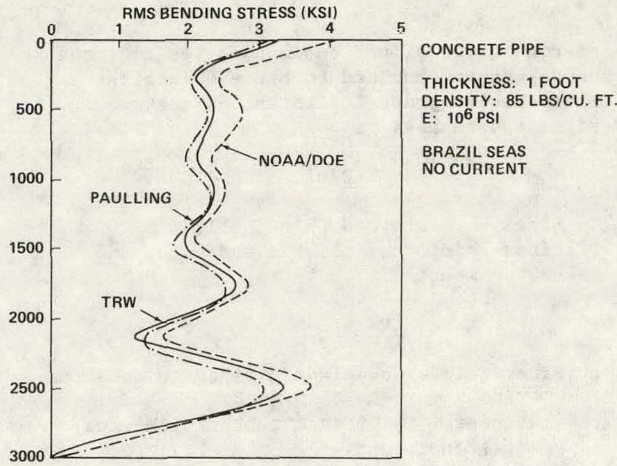

Fig. 8 Comparison of TRW, NOAA/DOE and Pauling CWP model results.

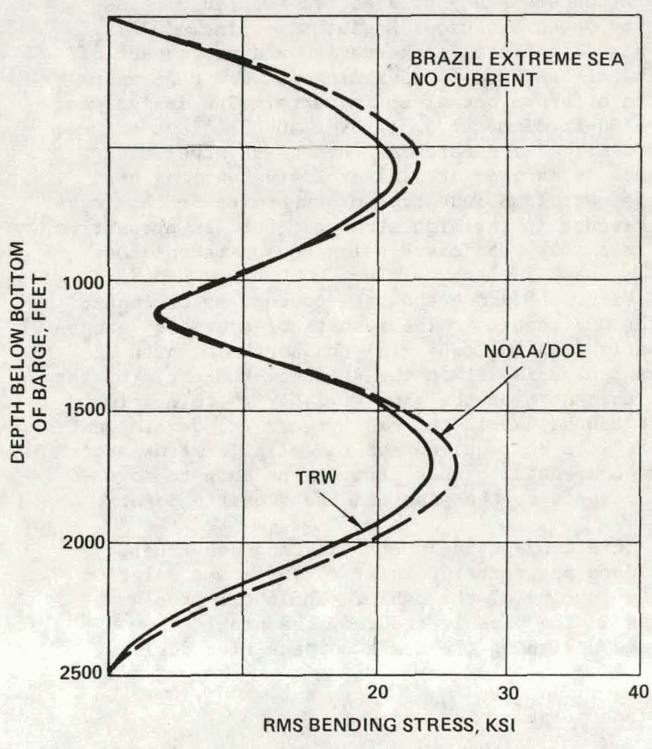

Fig. 9 Comparison of TRW NOAA/DOE CWP model results for 2-in. steel pipe.

approaches have given such similar results. It now remains to have the results verified by both model and at-sea tests.

Preliminary Design

Two CWPs have been advanced through preliminary design, one for the APL grazing plantship and one for OTEC-1 (GMDI).

The APL plantship pipe is a concrete, prestressed, segmented pipe. Each segment has a length of 50 ft and a diameter of 30 ft. The concrete is lightweight, having a weight in air of 85 lb/ft^3.

Two interesting features of the design are the pipe joints and the connection between the pipe and the ship. The pipe segments are joined by an interlocking joint made by rotating one segment of the pipe with respect to its adjoining segment. The flexibility in the joint is achieved by having the lower segment of the pipe bear on elastomeric material supported by the upper segment (see Fig. 10). A watertight seal is provided. The attachment of the pipe to the ship is accomplished by a spherical bearing surface in which the upper support ring of the pipe seats. As the pipe rotates with respect to the ship, the ring slides on the bearing surface (see Fig. 11).

The CWP designed for OTEC-1 by GMDI is a multipipe configuration using three 5-ft diameter polyethylene pipes. The pipe is designed to be cast off from the platform if the sea exceeds the design condition. The pipe has a buoyant collar and a weight at the bottom that serves as an anchor so that when it is cast off, it will go to the bottom and remain upright. The load of the anchor is taken by central cables so that the pipes are not subjected to that load (see Fig. 12). GMDI has had experience with polyethylene pipe in the sea, having used it on an earlier biomass experiment off the California coast.

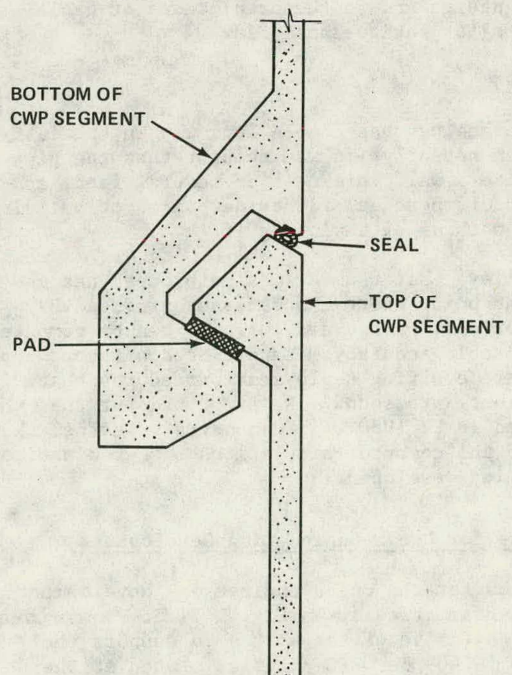

Fig. 10 APL CWP joints.

The pipe will be instrumented both during deployment and during operation of the plant. The details of the instrumentation are yet to be completed. The results of the measurements will be important to the CWP design procedure. It will be an opportunity to compare measured strains and de-

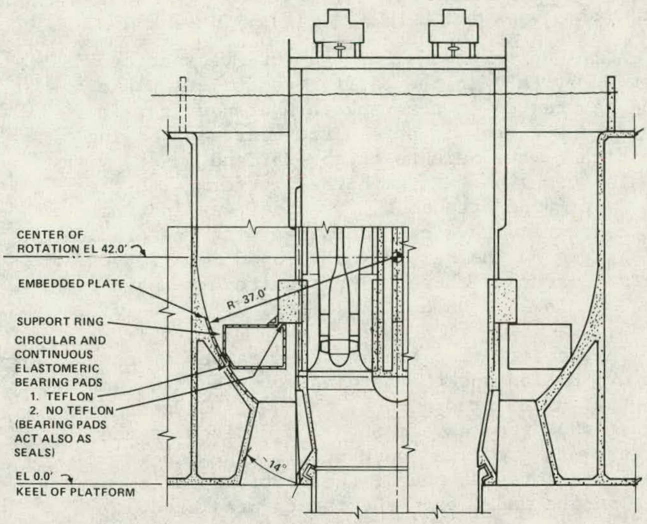

Fig. 11 CWP connection to hull of APL plantship.

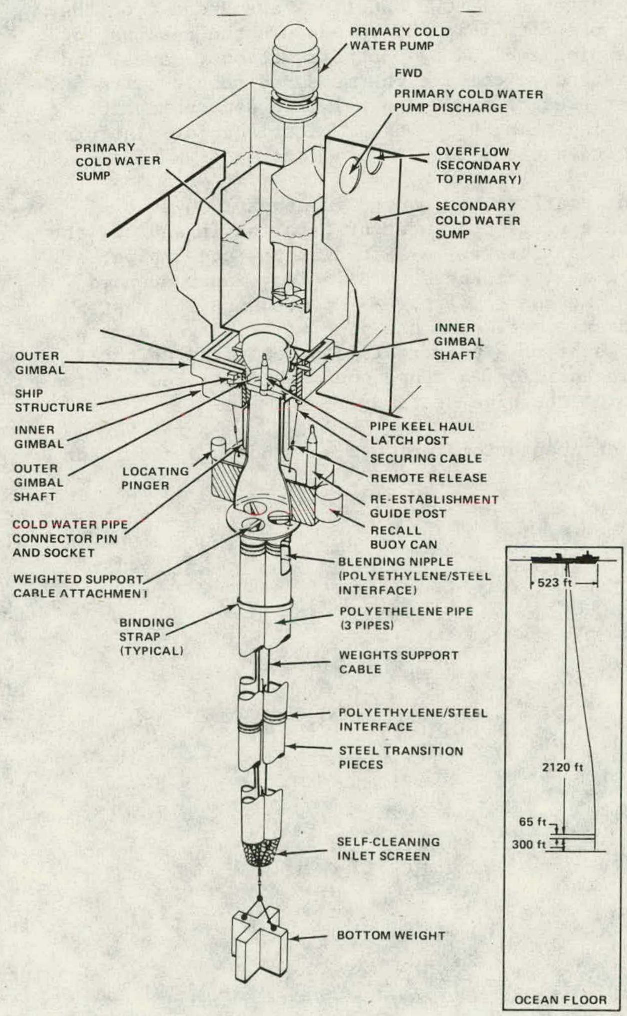

Fig. 12 CWP subsystem for OTEC-1.

flections with predicted values; but, in addition, it will furnish results on a pipe made of material with a low modulus of elasticity. These materials seem to be coming into more favor with designers. For example, the recent TRW and SAI recommendations include low mululus materials.

Model Test

A 1/100 scale model of the 400-MWe spar buoy designed by LMSC in the earlier commerical plant study was tested in the tank at Hydronautics, Inc. The tests were run in three irregular seas having significant wave heights of 15, 25, and 35 ft, with the platform alone and with the platform with the cold water pipe attached. The model was instrumented and the results have been compared with the predictions given by the recently developed NOAA/DOE CWP program discussed above. The results have been reported by Kowalyshun & Barr in Ref. 7.

They report that the measured values of the platform motion and the bending moments when compared with the values predicted by the NOAA/DOE computer program show good agreement. The motions of the platform are predicted accurately. The bending moment predicted at one of the stations is somewhat lower on the model than the theory predicted.

At-Sea Test

A larger scale CWP model was suspended from the DOT X-1 platform (Fig 13) to measure the response of a large pipe excited by platform motions, waves, and currents and to compare the measured results with computer predictions. The X-1 is a semisubmersible vessel displacing 435 tons. The triangular platform is 120 ft on each side.

The platform was moored in about 1000 ft of water in a slack moor west of Catalina Island off the coast of California. A 5-ft-diameter steel pipe, having a wall thickness of 3/16 in., was suspended through the center of the X-1 platform. The test configurations allowed the pipe to be hinged at the top by means of a universal joint or fixed at the top. In addition, a hinge could be installed between any two of the pipe sections.

Instrumentation included wave buoys and current

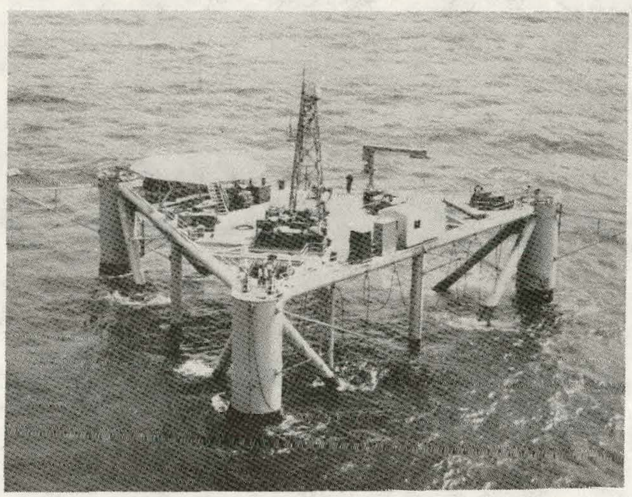

Fig. 13 DOT X-1 platform.

meters for measurment of the incident sea, accelerometers to measure platform and pipe motion, and strain gauges to measure the resulting strains in the pipe.

Data from that experiment are now being reduced and compared with the values predicted by the computer programs.

These X-1 tests have a central role in the CWP development strategy for several reasons: (a) It is the first large pipe experiment designed to test the accuracy of the computer prediction methods; (b) as will be seen later, the results can be used to check the accuracy of the predictions from the tank models; and (c) the pipe can be considered as a 1/6 scale model of the 40-MWe cold water pipe. Thus, the X-1 experiment ties the computer model, the tank model and full size pipes together. For all these reasons, the results are eagerly awaited.

Applied Support R&D

Research and development studies are aimed at answering specific fundamental questions such as:

1. Measurement of drag and lift coefficients for smooth and rough cylinders at Reynolds numbers up to 10^8;

2. Investigation of vortex shedding on OTEC pipes;

3. Investigation of the hydrodynamic loads on a cold water pipe at high Reynolds numbers in a nonuniform current;

4. Development of lightweight concrete suitable for the cold water pipe and determination of its characteristics in sea water;

5. Investigation methods of vortex suppression on cold water pipes; and

6. Prediction of the platform motions for inputs to cable design and to the predictions of cable motions for fatigue analysis.

CWP Summary

CWP technology has come a long way in the last year. Until recently, it was thought that the pipe would require a wall thickness of several feet, and anyone with offshore experience despaired of building it, to say nothing of deploying it.

We believe that the CWP design process has matured to the point where the stresses produced by platform motions, waves, and currents can be computed to an acceptable accuracy. Experienced offshore constructors are evolving deployment procedures that, in their judgment, are sound. We are confident that the work planned in FY 1980 will demonstrate further the validity of the computer simulations of CWP dynamics that are being developed.

Plans for Ocean Engineering Development

The plan for the ocean engineering development is shown diagrammatically on Fig. 14. It shows graphically the objective of the work — to support the design of a 10/40-MWe OTEC plant as stated at the start of this paper. This plan is geared to permitting the start of the preliminary design of a 10/40-MWe plant in October 1980. There is confidence

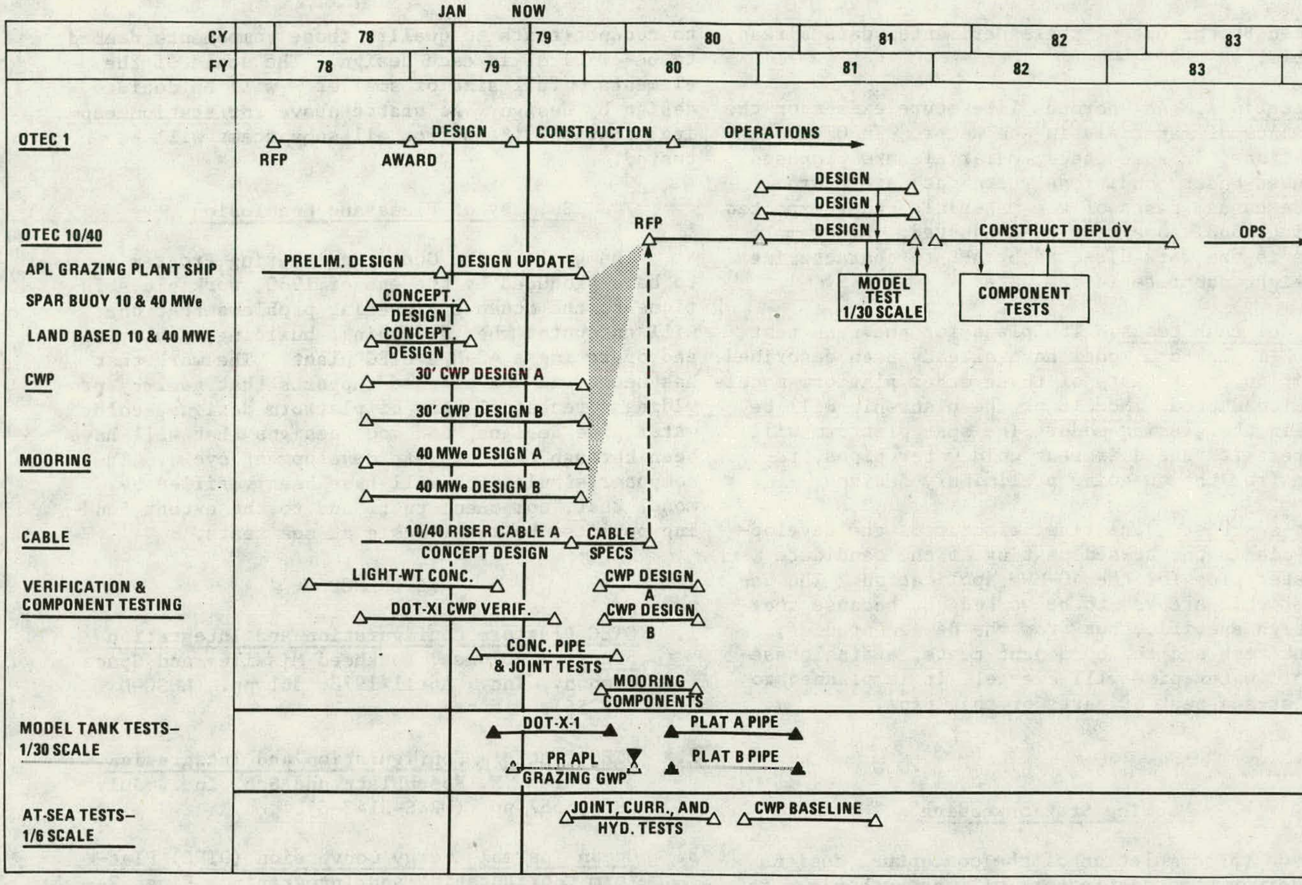

Fig. 14 OTEC Ocean Engineering Development Plan.

in the technical community that the technology base will be in hand to support that design. Conceptual design flows into preliminary design, which in turn moves to at-sea or large-scale laboratory testing. The plans for future work on the subsystems are described briefly in the next sections.

CWP Development

The tests on the X-1 platform will be extended to articulated (multiple hinges) CWP tests. A scale model of the X-1 platform and pipe will be run in a wave tank reproducing the X-1 at-sea experiment. The model will be instrumented, allowing a comparison to be made of the full-scale at-sea results and the tank measurements. When this is completed and the data analyzed, the following information will be available for the validation of designs:

1. Small scale tank models results,
2. At-sea tests of 5-ft-diameter pipe,
3. Frequency domain computer models, and
4. Time domain computer models.

It is important to note that: the tank models is 1/30 scale of the X-1 pipe and the X-1 model is 1/6 scale of the 40-MWe pipe.

Therefore, the tank model and the computer model will be used to predict the performance of the full scale 40-MWe pipe, and they will have been validated against the at-sea tests of the X-1 pipe.

Preliminary design. Upon completion of the conceptual designs of the cold water pipe by TRW & SAI, authorization was granted to proceed into preliminary design. This design process will continue, and it is expected that these preliminary designs will be completed by the end of this year.

Design validation. The next phase of the ongoing design method validation will call upon the previously described techniques, i.e., tank models, analytical models, and large scale (about 5-ft-diameter) models to validate the designs of cold water pipes resulting from the preliminary designs being carried out by TRW & SAI. It is expected that at least one of the new designs will employ materials — rubber, fiberglass, and polyethylene — having a low elastic modulus.

The result of this validation process will be two validated CWP designs, one for the barge and one for the spar shown as CWP A and CWP B on Fig. 14.

Computer-program development. The programs discussed earlier, i.e., the NOAA/DOE Program and the Paulling Program are frequency domain programs that treat the cold water pipe as a beam and treat both the excitation and resulting response (strain) statistically. There is almost unanimous agreement among experts that these models are adequate for preliminary design and can be used in comparing designs. However, for final design, a more exact method is desirable. It is intended to produce a set of programs that will treat the pipe as a shell and compute the response as a function of time. The inputs will be multiple loads whose time histories are known. These programs will be made available by assembling existing programs wherever possible, and will be validated by the use of the experimental data already available.

Materials. An enormous literature exists on the performance of materials in sea water. In OTEC applications, in some cases, materials are proposed to be used under conditions where data are sparse. In those cases, tests of the materials under expected operating conditions will be conducted. An example of this is the work directed by APL to characterize lightweight concrete in sea water.

Model tank tests. The plans for the tank test of the X-1 platform model have already been described. In addition, tank tests of three other platform models will be conducted. Models of the plantship will be tested in the grazing mode. The spar platform will be tested with the different cold water pipes, resulting from the on-going preliminary designs.

At-sea test. The final element of the development cycle is the at-sea testing of the candidate cold water pipe for the 40-MWe application. The details of this are yet to be worked out because they are design specific, but from the design studies, the tank test and the component tests, a final baseline cold water pipe will emerge. It is planned to run an at-sea test of parts of that pipe.

Mooring/Stationkeeping

Upon the completion of the conceptual designs now underway, the contractors will be authorized to proceed into the preliminary design of moors from these designs. When the designs are completed, certain critical components will be selected for testing. At the completion of this phase, baseline moor designs will be available for the candidate 40-MWe platforms.

Baseline Designs

Finally, it is planned to continue the development of the designs of the spar platform and the land-based plant up to the level of detail of the plantship. Three baseline designs will then be available.

Testing of Critical Components

To illustrate what is meant here, refer to the earlier discussion of the pipe for the APL plantship and recall the joints between the pipes and the connection between the pipe and the platform. The analytical process can predict the loads that will act through these joints, and component designers will design joints of high predicted reliability. Nevertheless, prudence dictates that those critical elements must be tested. It is planned, therefore, to conduct tests to qualify those components deemed to be critical in each design. The scale of the elements — full size or smaller — will be decided design by design. As stated above for stationkeeping, critical items from all subsystems will be tested.

Summary of Plans and Conclusion

The goal of the Ocean Engineering Program is to have produced by the end of 1980, workable solutions to the ocean engineering problems that one will encounter when designing, building, deploying, and operating a 40-MWe OTEC plant. The work that has been done and planned supports that goal by providing a technical base of platform designs, cold water pipe designs, and moor designs that will have been through the complete development cycle. The computer simulations will have been verified by model test, component test, and to the extent funding permits, by large-scale at-sea tests.

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OTEC GOES TO SEA (A REVIEW OF MINI-OTEC)

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Abstract

Mini-OTEC, ~~shown in Fig. 1,~~ is the first at-sea, closed-loop Ocean Thermal Energy Conversion (OTEC) system using surface and deep seawater to generate electric power. The Mini-OTEC cycle is installed on a moored barge incorporating the cold water pipe (CWP) in the single anchor leg. The design seawater temperature difference (ΔT) of 36°F provides thermal resource for a gross power output of 50 kW. ~~This paper presents~~ an overview of the Mini-OTEC project, including a description of the power plant, control system, instrumentation, and CWP mooring system. is presented

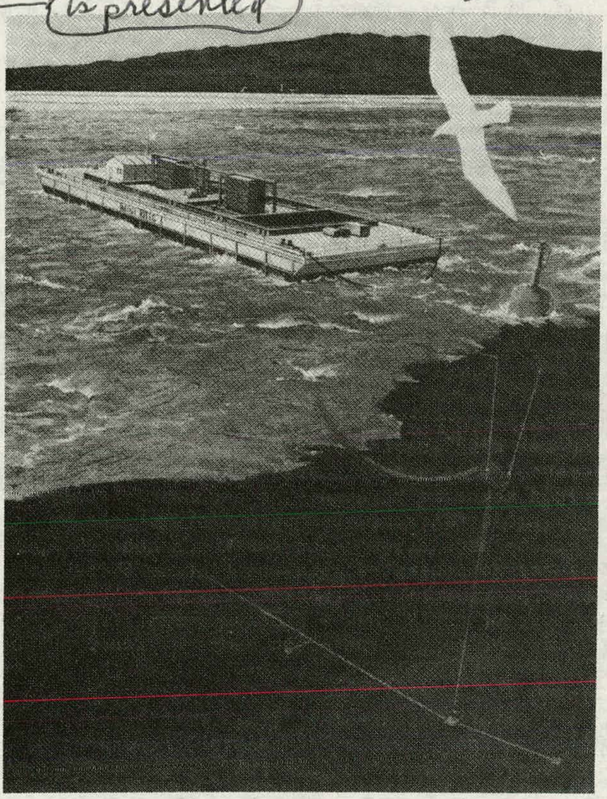

Fig. 1 Artists Concept Mini-OTEC

Introduction

The Mini-OTEC Project has resulted in the world's first floating, at-sea OTEC plant. Although it produces only 50kW(e) (gross), it contains all the essential components required for a full-scale power plant. The four major subsystems are the platform, the power plant, the mooring, and the CWP which, in this case, is an integral part of the mooring system. The CWP and mooring system dynamic analysis is treated in this paper. Current plans are to conduct experiments lasting six months*, after which most of the plant will be turned over to the State of Hawaii.

The overall project objectives are to (1) develop an operating, at-sea OTEC system, (2) gain "real world" operating experience of an OTEC system, (3) provide a subsystem, component, and technology facility that can be used as a risk-reduction test bed, and (4) expand public awareness of the OTEC potential.

The technical objectives fall into two principal areas: power system and ocean system. The power system technical objectives are (1) to expand current knowledge of plate heat exchangers, (2) evaluate biofouling effects in situ, (3) evaluate biofouling countermeasures for plate-type heat exchangers, (4) evaluate the power system control design, (5) evaluate the ammonia system dynamics, and (6) investigate the release of trapped gas from the deep cold water. The ocean system technical objectives are to (1) gain experience in the deployment and mooring of a cold water pipe, and (2) obtain qualitative data on seawater dynamics. For several reasons the technical objectives are centered on the power plant, rather than on the ocean systems. First, the project was faced with the age-old problem of time versus money; a projected financial commitment had to be set. Second, the project is of relatively short duration and planned for benign weather to avoid a strenuous test of the mooring system. And, finally, the importance of Mini-OTEC success was fully recognized at the out- set; no effort was spared to ensure a seaworthy design, but the machine <u>must</u> produce electricity.

The power system was installed and the barge outfitted in Honolulu. The barge was towed to Kawaihae, on the island of Hawaii, for deployment along with the cold water pipe, in an area approximately 7,000 ft offshore from Keahole Point, in a water depth of 3,050 ft. Table 1 summarizes the Mini-OTEC System Components.

Table 1

MINI-OTEC SYSTEM COMPONENTS

Heat exchanger	- Alfa-Laval plate type
Heat exchanger cleaning	- Chlorine; Clean-in-place
Turbine	- Rotoflow radial in-flow
Control System	- Two automatic independent systems
Cold water pipe	- Polyethylene integrated with mooring
Instrumentation	- All temperatures, pressures, flow rates
Data acquisition	- Mag tape and printout on command
Biofouling test rigs	- Separately operable
Auxiliary power	- 60 kW diesel generator

*The Mini-OTEC plant produced net power on 2 August 1979.

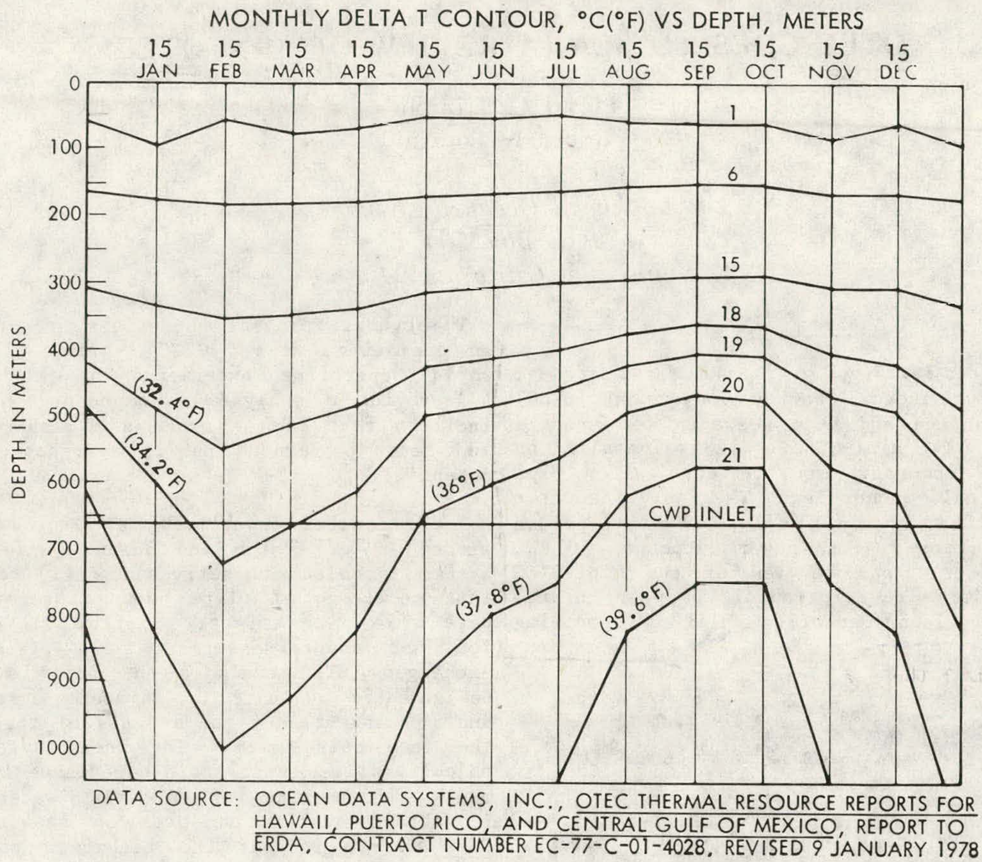

Fig. 2 Temperature Profile at Keahole Point

Power System

The design ΔT for Mini-OTEC is 36.8^oF. As shown in Fig. 2, we expect to sustain that ΔT from June through mid-November, and to be partially operational through November, and perhaps December.

Heat Exchangers

Table 2 presents a design summary of the heat exchangers. The heat-exchanger data acquired during the experiment are proprietary to Alfa-Laval. The higher ammonia flow rate in the evaporator is the result of a relatively low quality vapor (i.e., 70 percent) expected from the evaporator. The mist flows into the separator, where the gas is freed from the liquid. The liquid is returned, along with the condensate, to the evaporator. Vapor quality is one of the parameters to be determined during the experiment. Further, since this is the first in-situ experiment, we have not made an allowance for a fouling factor, but rather will determine fouling rates as a part of the experiment. The plant is designed to run 24 hours per day, 7 days per week. Current test plans include periodic tear down and inspection of plates during the operational period.

An exploded view of the heat exchanger is shown in Fig. 3. Ammonia enters the lower right-hand port and is distributed equally to the alternate plates by the gasketing arrangement. Note that the gasket in Plate 1 at the ammonia port is open to allow liquid to flow inward. The gaskets are also arranged so that vapor is free to leave the evaporator through the upper right-hand port. Seawater, for the co-flow condition, enters the lower left-hand port and is distributed to the off-alternate plates by the gasketing arrangement, through the heat exchanger, and out through the upper left-hand port. A counterflow experiment will be conducted by reversing the inlet and outlet seawater ports.

Table 2

MINI-OTEC HEAT EXCHANGER DESIGN SUMMARY

Heat Exchangers	Condenser	Evaporator
Type	Plate	Plate
Surface Area, ft^2	4390	4390
Material	Titanium	Titanium
Thermal Load, Btu/sec	2607	2693
NH_3 Flow, lb/sec	5.1	11.2 (max)
Seawater Flow, gpm	2700	2700
Seawater Temp.-in, ^oF	42.2	79
Seawater Temp.-out, ^oF	47	75
Fouling Factor, R_f	N/A*	N/A*
S.W. Pressure Drop, psi	4.4	4.4
Quality, % vapor	TBD	TBD

*No a priori fouling factor used as a design parameter.

Turbine-Generator

Table 3 presents the turbine-generator (T-G) design summary. The low efficiency in the T-G set is unique to the small size of the turbine. For an output of 10 MW(e) (net), the turbine will be

Table 3

MINI-OTEC TURBINE-GENERATOR DESIGN SUMMARY

Turbine	
Type	Radial In-Flow
Speed	28,200 rpm
Enthalpy Drop	17.2 Btu/lb
Flow	17,830 lb/hr
Output	100 hp
Efficiency	83%
Generator	
Type	Synchronous/Rotating Field
Speed, rpm	3600
Exciter	Shaft mounted, brushless
Output	50 KW, 0.8 P.F., 120/208v, 60 Hz, 3 phase
Efficiency, %	90
T-G Efficiency, %	56

directly connected to the generator. This eliminates the gearbox and the roughly 20 percent power loss associated with the gearbox.

Fluid Systems

Figure 4 is a fluid systems schematic. The plumbing and the componentry are cleaned, pressure-tested to 200 psi, and filled with dry nitrogen, purged and evacuated prior to filling the system with ammonia. The system is pumped down to a vacuum and filled with ammonia from on-board storage, using the 60-KW T-G set to drive the ammonia pumps. The condenser and evaporator seawater pumps are started to initiate ammonia flow. The bypass valve in the turbine system is partially opened at the beginning of ammonia flow and the ammonia pumps are started to circulate the ammonia. As vapor pressure builds, the turbine bypass valve is partially closed and the turbine begins to spin. Turbine speed is controlled by an internal speed control system. The automatic turbine bypass valve is included in the system to permit the heat exchangers to operate at a constant thermal load, allowing the T-G to respond to a variable load. A manual set of the bypass valve is included to permit setting vapor flow during check-out and startup procedures. The manual setting allows vapor flow without necessarily spinning the turbine. The separator, located between the evaporator and the turbine removes the liquid content, which is returned to the evaporator through the preheater. The condensate is mixed with the flow from the separator. The preheater effectiveness will be tested by varying seawater flow through it.

Control System

Two independent control systems can be used to control plant operations. Both control systems are based on a constant volume system. The first control system senses the liquid level in the separator and commands a variable speed pump in the separator return line to maintain liquid level in the evaporator. The second system, driven by a liquid level sensor in the condensate sump, controls a flow valve

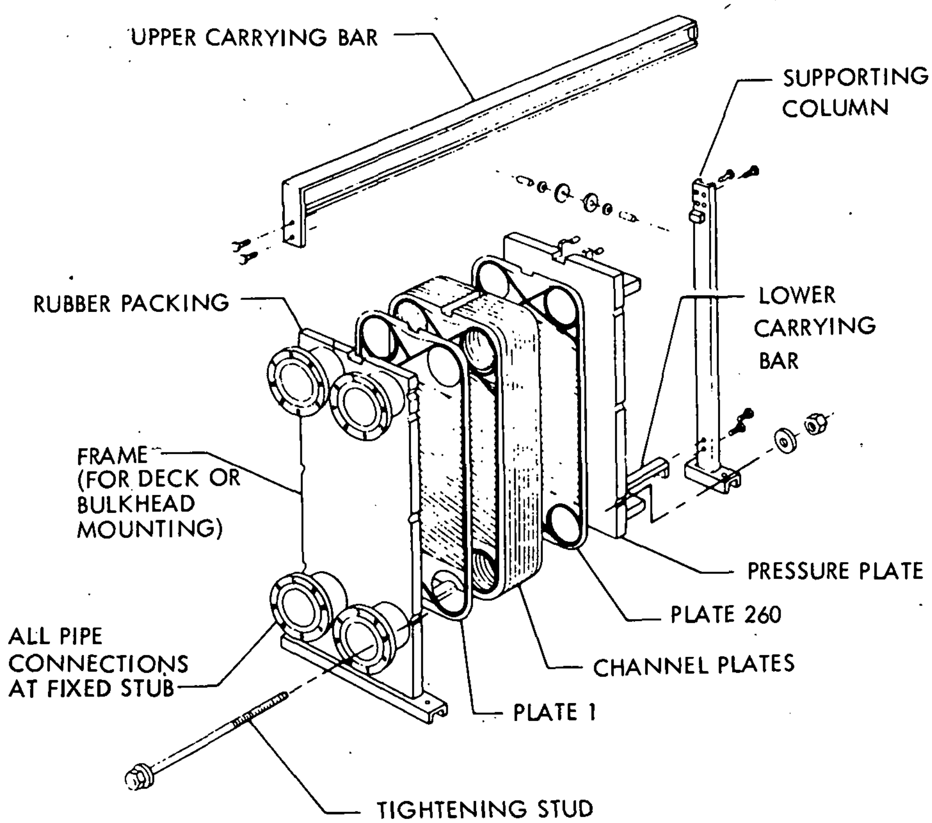

Fig. 3 Exploded View of the Plate Heat Exchanger

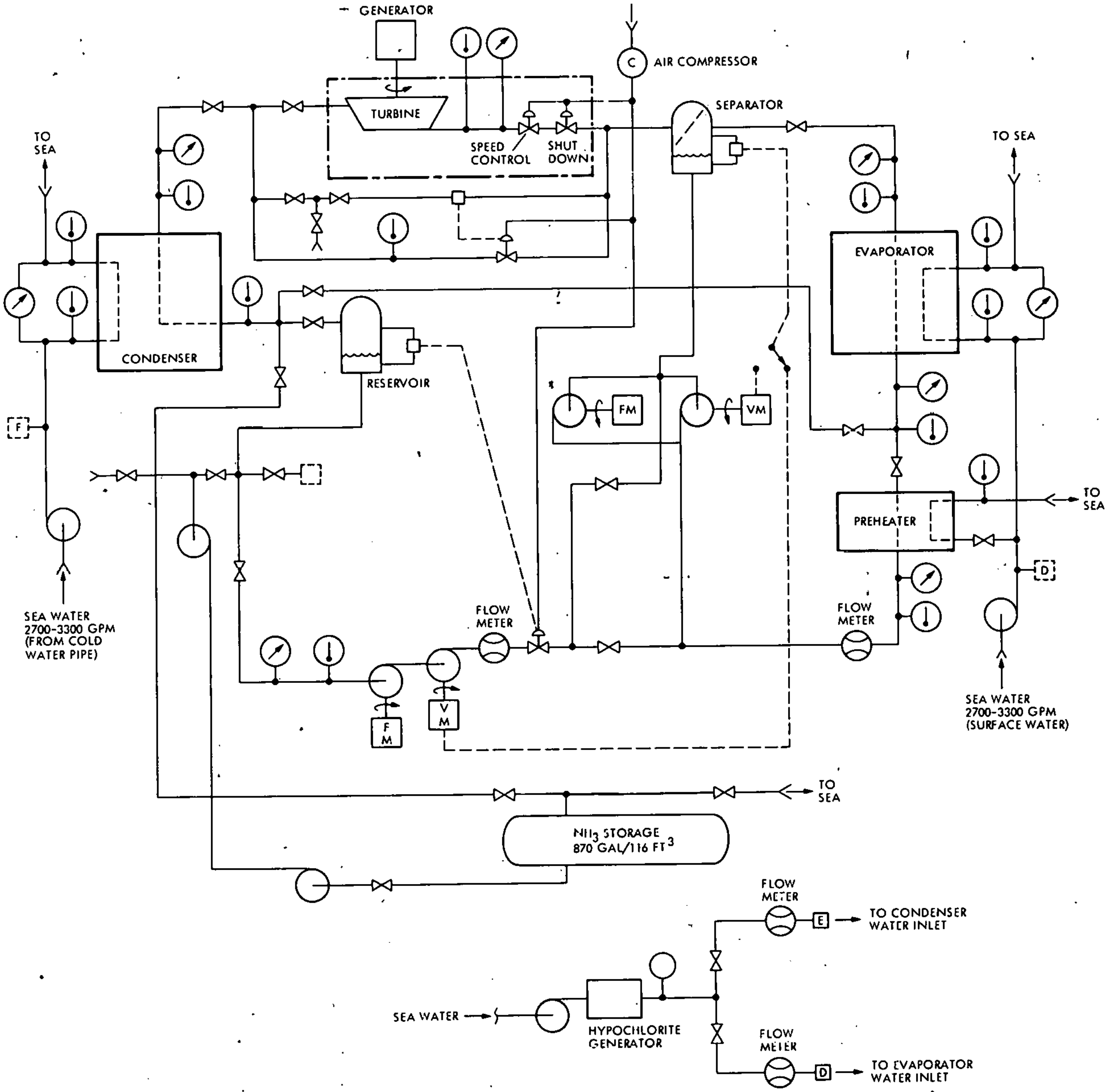

Fig. 4 Fluid Systems Schematic

in the condensate line downstream of a variable speed condensate pump. The set points on the flow rates are manually adjusted. The control systems are operated by compressed air.

The hypochlorite generator, requiring 20 gpm seawater, can supply variable amounts of chlorine to both the evaporator and the condenser. It is expected that the heat exchangers will be operating between 0.05- and 1-ppm chlorine. The T-G cooling system has its own oil-seawater heat exchanger. The fluid system includes a fire-fighting and ammonia-washdown seawater system driven by a separate pump.

Instrumentation

Figure 5 lists the power system instrumentation excluding heat exchanger sensors. Thirty-five differential pressure measurements are taken on seven

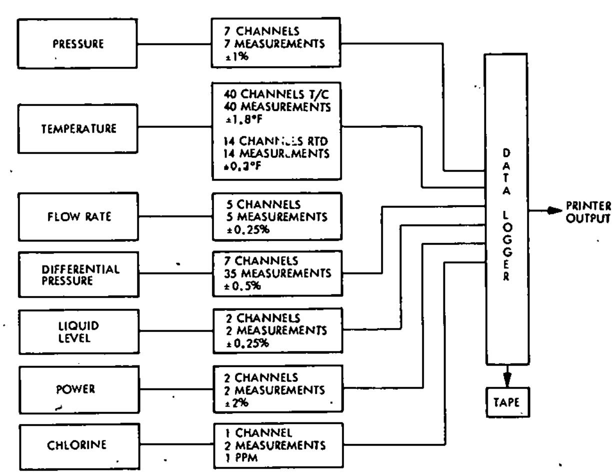

Fig. 5 Instrumentation

channels and two chlorine measurements are taken on one channel. All other measurements have their own channels. The data logger can be commanded for visual display of any single measurement. Both magnetic tape and printout can be commanded or sequenced at selective time intervals to record data in engineering units.

Power Summary

Table 4 displays the power summary. We will develop almost as much power as a VW engine at the generator terminals. However, our goal is to learn how OTEC works, not to dissipate power. The net power will be consumed by hotel load and flood lights.

Table 4

POWER SUMMARY

	KW LOADS	
	NOMINAL	MAXIMUM
COLD WATER PUMP	11.9	13.6
WARM WATER PUMP	9.4	10.7
1ST STAGE AMMONIA FEED PUMP	1.0	1.2
2ND STAGE AMMONIA FEED PUMP	5.0	6.0
SEPARATOR PUMP NO. 1	–	2.0
SEPARATOR PUMP NO. 2	1.0	3.0
CHLORINE GENERATOR	3.0	3.0
OIL COOLER PUMP	1.0	1.0
AIR COMPRESSOR	–	1.0
EMERGENCY OIL PUMP	–	3.0
TOTAL	32.3	44.5
NET USAGE	17.3 (MAX)	5.5 (MIN)

Integrated Cold Water Pipe and Mooring

Description of the mooring system with the results of a dynamic analysis performed to provide design loads and design verification are presented below.

The mooring system consists of hawser, surface buoy, hanger, crossover hose, cold water pipe, anchor line, anchor, and bottom cable (Fig. 6). The barge has the following characteristics:

Length on Waterline	105 ft
Beam	33.9 ft
Draft	4.3 ft
Operating Displacement	268 lt

The barge, moored at the bow to the buoy, is free to swing about the buoy. The mooring hawser consists of a pair of 30-ft lines attached with chain to port and starboard bow cleats and leading to a single 34-ft-long hawser. The lines are 2.1 in. in diameter, two-in-one nylon-nylon with tensile strength of 153,000 lb. The single line is 2.7 in. in diameter with strength of 257,000 lb. Additional lengths of hawser are stowed to allow the barge to be backed 150 ft from the buoy.

The hawser is shackled to the bottom of a spar below the buoy. The buoy consists of a steel cylinder 7 ft in diameter and 14 ft high, with the spar extending the buoy height to 27 ft. The buoy weighs 13,575 lb, and the displacement, when fully submerged, is 37,500 lb. Connecting the mooring buoy and CWP upper end, is a pair of 60-ft hanger lines. These lines are 2.7 in. in diameter, two-in-one nylon nylon with 257,000 lb tensile strength. The lines are attached to a steel assembly consisting of a 45-deg elbow, flange, and clamp-sleeve friction connector. A ballast and inlet assembly connector is attached at the bottom of the CWP. The 6,000-lb lead ballast maintains the required inlet depth. Seawater flows horizontally through a cage formed by a ring of bolts and then turns 90 deg upward into the CWP. These bolts also serve to mount the ballast to the connector fitting.

Cold Water Pipe

The CWP, fabricated of butt-welded 40-ft sections of high-density polyethylene, has the following mechanical properties:

Length	2,150 ft
Inside Diameter	22.1 in.
Outside Diameter	24.0 in.
Weight (in air)	28.1 lt
Ultimate Axial Load	98.2 lt
Dynamic Flexural Modulus	140,000 psi
Fatigue Strength (73°F, 10^5 cycles)	1,000 psi
Specific Gravity	0.95

The CWP with its end assemblies is supported from the surface buoy. The watch circle of the bottom of the pipe is maintained by the line to the anchor. This line is multifilament, polypropylene parallay braid, 1,520 ft in length, 2.7 in. in diameter, and having a 141,000-lb tensile strength. The line is attached to a swivel below the ballast and to a concrete clump anchor, 10 ft x 10 ft x 3 ft, and 30,000 lb in wet weight. A second line, attached to the anchor, consists of a chain, swivels, and 8,000 ft of 6 x 19 galvanized wire rope of 1.5-in. diameter. This line, resting on the steep seafloor and secured by a drag embedment anchor on shore, will prevent slippage of the clump anchor.

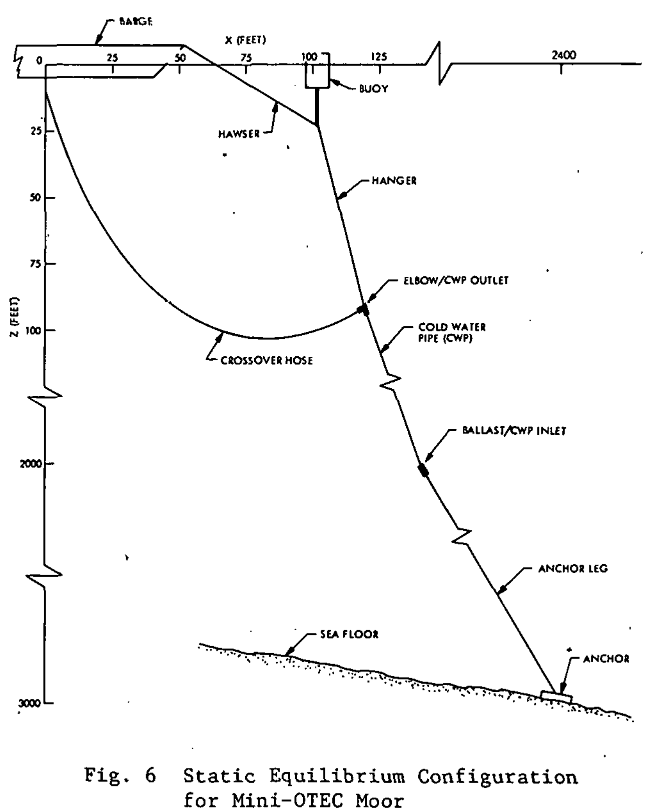

Fig. 6 Static Equilibrium Configuration for Mini-OTEC Moor

Table 5
SUMMARY OF CWP-MOORING RESPONSE

Component	Variable	Case 1: Operational Sea State, $H_{1/3}$ = 15 ft		Case 2: Stationkeeping Sea State, $H_{1/3}$ = 24 ft	
		Maximum	Minimum	Maximum	Minimum
Hawser	Tension, lb	50,684	0	27,723	0
Hanger	Tension, lb	69,391	0	49,509	7,700
Buoy	Surge, ft	21.7	-22.7	16.8	-18.9
	Heave, ft	10.7	-13.2	16.5	-10
	Pitch, deg	102.5	-90	57.3	-41.2
Barge Bow	Surge, ft	9.1	-10.5	14.6	-16.3
	Heave, ft	12.2	-12.2	17.4	-16.9
Anchor Line	Tension, lb	21,680	15,287	19,592	16,007
Crossover Hose at Barge (a)	Rotation, deg	23	10.6	114.1	36.7
	Tension, lb	8,319	-2,483	1,644	-624
	Stress, psi	269	-130	158	-152
Crossover Hose at Elbow	Rotation, deg	73.6	9.6	91.1	54.2
	Tension, lb	6,178	-3,819	9,725	-916
	Stress, psi	248	-206	187	-76
Cold Water Pipe at Elbow	Rotation, deg	24.3	15.4	26.5	17.3
	Tension, lb	67,033	-12,757	45,433	1,393
	Stress, psi	1,187	-711	938	-251
Cold Water Pipe at 215 ft above inlet	Rotation, deg	36.3	32.8	34.5	32.3
	Tension, lb	62,832	-9,958	50,287	1,967
	Stress, psi	1,521	-743	1,276	-514

(a) At Retrieval Line for Case 2.

Crossover Hose

The crossover hose, fitted to the elbow at the CWP outlet, conducts cold seawater to a surge tank situated below the barge keel, 8 ft forward of amidships. The barge end of the hose is secured by a tensioned line and connector, providing for release and lowering of the hose in high sea states. This flexible composite hose consists of six sections bolted together with overall properties as follows:

Length	200 ft
Inside Diameter	18 in.
Outside Diameter	20.3 in.
Weight (in air)	5 lt
Ultimate Axial Load	
Straight	280,000 lb
Curved	70,000 lb
Minimum Bend Radius	9 ft
External Collapse Pressure	62 psia

Design Verification

The integrated pipe and mooring system are designed to provide a continuous supply of cold seawater to the surge tank in seas of 15-ft significant wave height and to maintain barge station in seas of at least 24 ft. A static analysis was conducted to select a mooring configuration, component lengths, and anchor weight to resist barge drag and prevent CWP grounding. A dynamic analysis of the system response in a storm was conducted to select line diameters, pipe and hose wall thickness, buoy depth, and operational limits.

Dynamic Simulation

A time-domain, dynamic program was developed specifically to simulate the seaway-induced motions and loads of the coupled mooring components in the vertical plane. Barge surge, heave, and pitch response amplitude and phase operators were obtained in the frequency domain using strip theory and were assumed to be independent of the mooring. Pipe and hose were assumed to respond as tensioned beams, and were modeled by finite elements with axial, lateral, and rotational degrees of freedom at both ends of each of 20 elements. Small geometric perturbations about the static-equilibrium, catenary configuration were assumed. Element extension and variable tension were included as were the nonlinear geometric effects. Large strain and snap loadings were accounted for by nonlinear modeling of the hawser, hanger, and anchor lines. Simulation of buoy pitch, heave, and surge considered external forces and moment due to hawser and hanger tensions, large relative motion between buoy and wave surface, and full submergence of the buoy.

A total of nine configurations was examined in the iterative process of design and analysis; selected results for two cases are discussed. Environmental conditions include 3,000-ft water depth, 2.3-knot surface current with exponential decay to 0.15 knot at 3,000 ft, and 30-knot wind speed in the current plane. The irregular seaway in each case was modeled using the Bretschneider wave spectra with significant wave height and period of 15 ft, 11.2 sec and 24 ft, 13.4 sec, respectively. Recurrence intervals for storms of this severity are estimated at 1 and 5 years, respectively, for Keahole Point.

Case 1. Mooring Response in Operational Sea State

The results, summarized in Table 5, are maxima and minima for a 10-min simulation in this sea state and include the static equilibrium, calm sea condition (Fig. 6). The hawser and hanger lines experience slack load giving safety factors on tensile strength exceeding six. There is adequate separation of buoy, barge bow, and elbow at the CWP upper end to prevent collision of these components. Relatively large buoy pitch excursions are noted. The anchor line tensile excursion is small in comparison with that of the hanger as a result of lower spring constant and mitigation of wave-induced excursions with depth. Crossover-hose stress distribution peaks at the barge and elbow. Stresses are extremes of tension and compression and include axial and bending in the outer fibers. Minimum safety factor on hose ultimate compressive stress is 2.4 and on CWP yield stress is 2.1.

Response of this configuration in a seaway of 20.3-ft significant wave height (not shown) resulted in line safety factors of less than six and wall stress safety factor of less than two, which are below design criteria. Hanger line diameter and hose wall thickness were increased with no appreciable increase of safety factors. The larger line is stiffer, contributing to increased coupling of buoy and pipe. The larger resultant tensile excursions approximately equal the higher strength, leading in some cases to a reduction in safety factor. Based on these results, the 15-ft seaway was selected as the operational sea state.

Case 2. Mooring Response in Stationkeeping Sea State

The alteration of the operational configuration selected for stationkeeping at the site in higher sea states consists of extending the hawser length, and then disconnecting and lowering the crossover hose via a retrieval line. This change decouples the crossover hose and buoy from the barge.

The effectiveness of this change is illustrated in the second case, wherein the hawser length is 150 ft and retrieval line is 258 ft (Table 5). In this higher sea state, all but one of the stress levels are reduced, so that safety factors on pipe and hose are 2.5 and greater, while line safety factors are 6.7 and greater. Reduced buoy motions and nonslack hangers also indicate the effectiveness of increasing hawser length. Selected time histories illustrate 5-min. portions of this simulation (Fig. 7). The plots are of dynamic oscillation excluding static reference values.

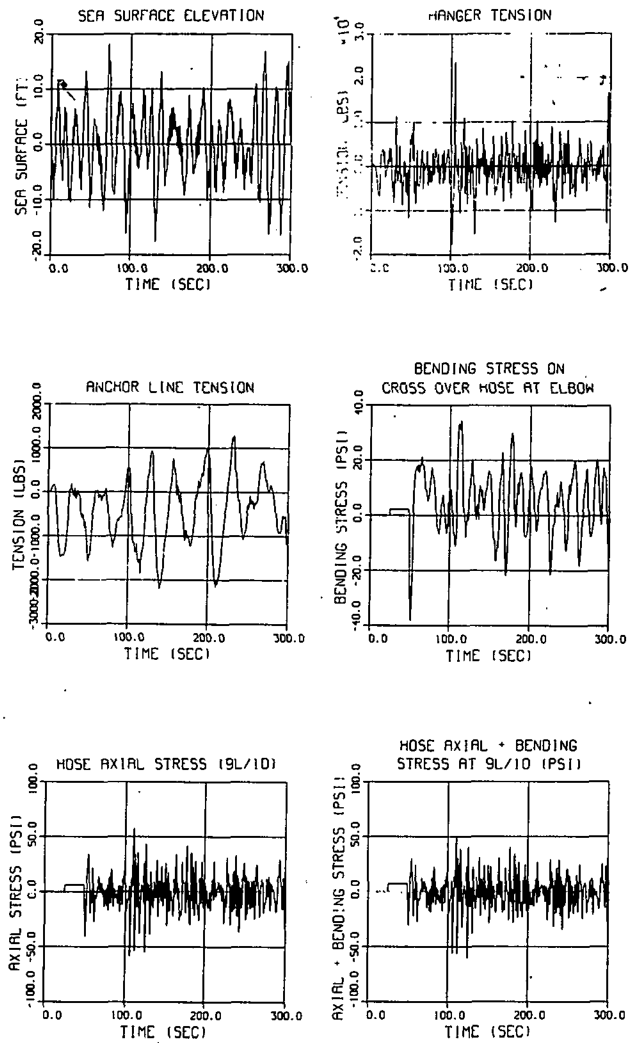

Fig. 7 Mini-OTEC CWP Mooring Seaway Response Histories, Case 2; $H_{1/3}$ = 24 ft

Acknowledgements

This project, conceived more than a year ago, is funded by a consortium of participants, including the State of Hawaii, Lockheed Missiles & Space Company, Inc. (LMSC), and Dillingham. Alfa-Laval Energy Systems is a major contributor to the project, supplying the pair of instrumented heat exchangers. The power system was designed by LMSC, in conjunction with Alfa-Laval. Argonne National Laboratories provided courtesy reviews. The hardware for the power system was procured by LMSC and shipped directly to Honolulu, where it was assembled by LMSC, with Dillingham support. The Navy barge is on loan to DOE for use by the State of Hawaii. Participants in the barge outfitting, under the direction of Dillingham, were Makai Offshore Engineering, Alfred E. Yee and Associates, and Ralph M. Parsons. LMSC provided analytical support to Dillingham for the CWP-Moor. The simulation program was developed by LMSC.

DISCUSSION

A. Galef, TRW: The film of deployment suggested high curvature and associated high bending stress. What was the stress level? Are you sure the pipe was not damaged?

R. Potash: CWP deployment followed controlled lowering of the gravity anchor. The film showing the CWP traveling below the ocean surface was taken with a telephoto lens resulting in a fore-shortened view. Hence, the radius of curvature apparent from the film is greater than the actual. To my knowledge, the actual radius and resultant bending stress were not measured and are not known. Satisfactory performance of the system would indicate that the pipe was not damaged.

T. McGuinness, NOAA: What precautionary measures were taken to minimize property damage if the Mini-OTEC mooring system fails?

R. Potash: An auxiliary anchor leg is stored on the Mini-OTEC barge for use in the event of separation of the barge from, or failure of, the primary single-anchor leg, single-point mooring. This auxiliary system includes a drag embedment anchor stowed in a launching rack on the bow. A 1700-ft length of attached nylon anchor line is stowed in a rope locker on the deck and adjacent to the anchor. The anchor is held in place by a lever.

AN OVERVIEW OF THE OTEC-1 DESIGN

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is given including

Abstract

~~This paper gives~~ an overview of the OTEC-1 system design, the vessel and deep water discharge, the power loop and associated supporting systems, the cold water system, the mooring system and the anchoring site selection.

The T-2 tanker **CHEPACHET** is now being modified in Portland, Oregon to serve as the OTEC-1 platform. The ammonia power cycle is located in a compartment provided by converting three ships centerline cargo tanks. The cold water pipe system consists of three, 48-in polyethylene pipes, upper and lower transition pieces, and a stabilizing weight at the bottom. It is attached to a roll-and-pitch gimbal under the moonpool sump. Cold water from a depth of 2250 ft is pumped through the pipe and the OTEC condenser to a discharge mixing tank. Warm surface water is pumped from a sump forward, through the plant's evaporator and into the mixing tank. The mixed discharge is then pumped aft into the deep water (240 ft) discharge hose. Power loop water piping onboard is a five-ft diameter steel pipe (discharge system: six-ft diameter). Ammonia storage and processing equipment is located on main deck forward and in the OTEC compartment.

The vessel stationkeeping system employs dual, rotatable thrusters and a single-point moor. The OTEC-1 operating site is 14 nmi northwest of Keahole Point, Hawaii in 4500 ft deep water. The seabed at this site requires a compliant mass moor provided by 262 tons of chain.

Introduction

The OTEC-1 system is being developed by Global Marine Development Inc. and its subcontractors, TRW and Northwest Marine Iron Works, under Department of Energy Contract No. ET-78-C-03-1785. This contract calls for Global Marine Development Inc. to design and install the OTEC-1 test loop on a floating platform and for the performance of tests in the vicinity of the Island of Hawaii. The OTEC-1 test facility will then operate for 3½ years on station, testing the initially installed system and other heat exchangers, as provided by DOE.

The OTEC-1 test loop is rated at a nominal 1MWe power output but will not produce any usable power in its present configuration. The facility is being built for the sole purpose of testing scaled, prototype heat exchangers and other OTEC equipment, under a variety of operating and environmental conditions. Control systems will also be studied to develop operating criteria for future, full-sized plants.

The OTEC-1 facility will comprise four different but fully integrated systems:

- The test platform including the primary (propulsion) power plant, auxiliary systems and accommodations for operating personnel.
- The OTEC-1 plant and its ancillary systems, including cold and warm water circuits.
- The cold water pipe (CWP) of 2250 ft length.
- The mooring or stationkeeping system.

This paper presents brief descriptions of these systems and their functional and physical interfaces.

OTEC-1 Test Platform

The T-2 tanker **CHEPACHET** (TAO-78, Figure 1) was selected for the OTEC-1 mission because she was available in excellent physical shape; is seaworthy and has seakindly behavior, a necessary trait in view of the operational seastate environments; has ample space for system equipment and sufficient facilities for personnel accommodations; and, her turbo-electric main power plant will provide the power necessary for operating all test systems and auxiliary equipment.

Figure 1. Chepachet leaving San Francisco Harbor.

Inactive since 1972, the **CHEPACHET** recently moved to the shipyard to be modified for the OTEC-1 Program. In the course of reactivation, she will retain her ABS classification of Maltese Cross A1E and Maltese Cross AMS, but her service will be changed from "Oil Carrier" to "Ocean Research." She will also be inspected by the U.S. Coast Guard and issued a Certificate of Inspection as a "Miscellaneous Vessel, Research Platform for OTEC-1 Service."

Vessel Arrangements

After modification and installation of OTEC-related equipment and facilities, the **CHEPACHET** will proceed under her own power to the operating area where she will be anchored with a single-point mooring system. The propeller drive motor will be secured, and the OTEC systems activated (see Figure 2). All DC electrically powered equipment will also be activated, together with two rotatable thrusters.

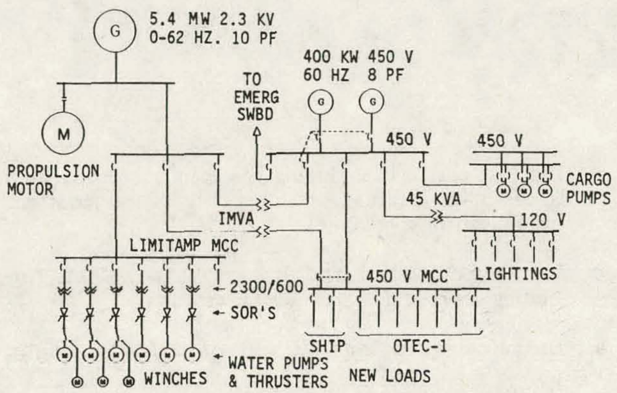

Figure 2. Electrical One-Line Diagram.

On station, the midship house will accommodate ten Test Director Contractor personnel, while all OTEC control functions will be performed from a data acquisition and processing control center located in a modified 40 ft van behind the midship house on the bridge deck level. An additional van, located directly behind the control center, will house a biofouling and corrosion laboratory.

The general arrangements of the OTEC-1 vessel are shown in Figs. 3(a), (b), and (c). The most significant structural change to the vessel involves the removal of two transverse bulkheads between the centerline cargo tanks to provide a sufficiently large compartment for heat exchangers and primary power cycle equipment.

Additional space has been provided also for transformers, SCR drives and motor controls for ammonia, cold and warm water pumps, thruster drives, and other OTEC equipment. Such equipment has been installed in new electrical compartments in centerline tank 9 and on the main deck under the midship house.

The cold water (CW) supply is provided through the Cold Water Pipe (CWP) to the moonpool or primary sump; then it is pumped into a secondary sump and from there to the ammonia condenser. The warm water (WW) supply is pumped to the ammonia evaporator from the warm

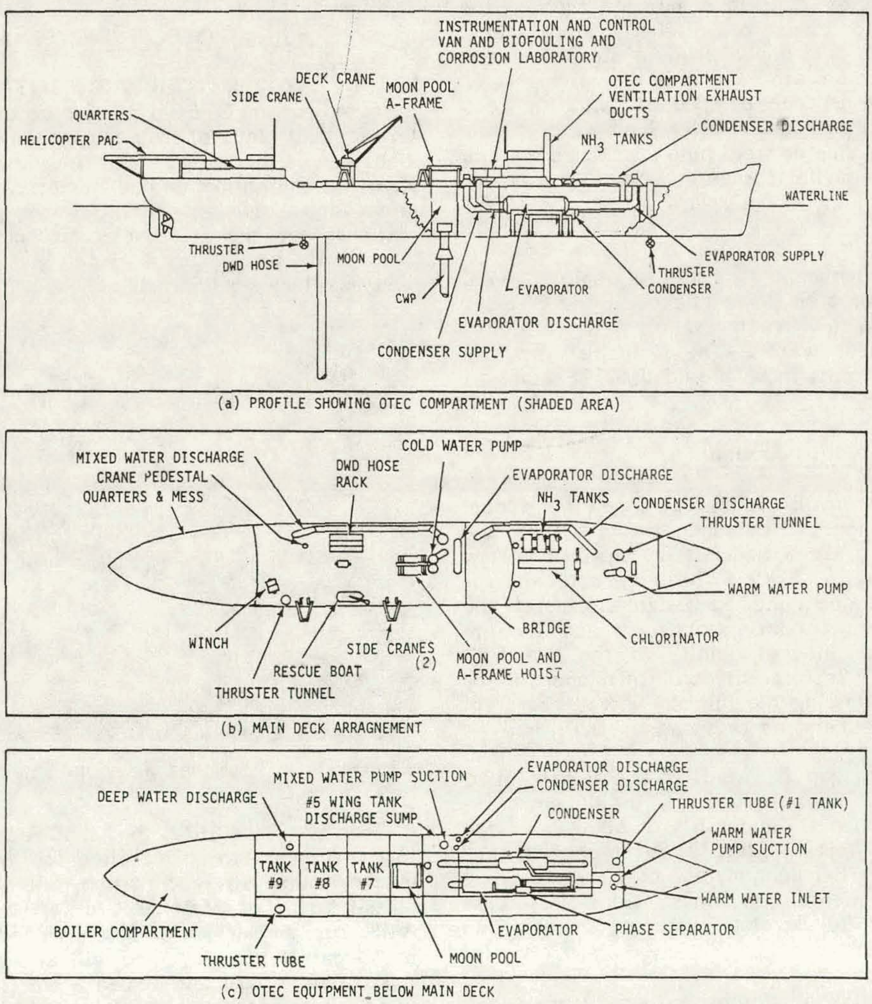

Figure 3. Views of Chepachet with OTEC Equipment Installed.

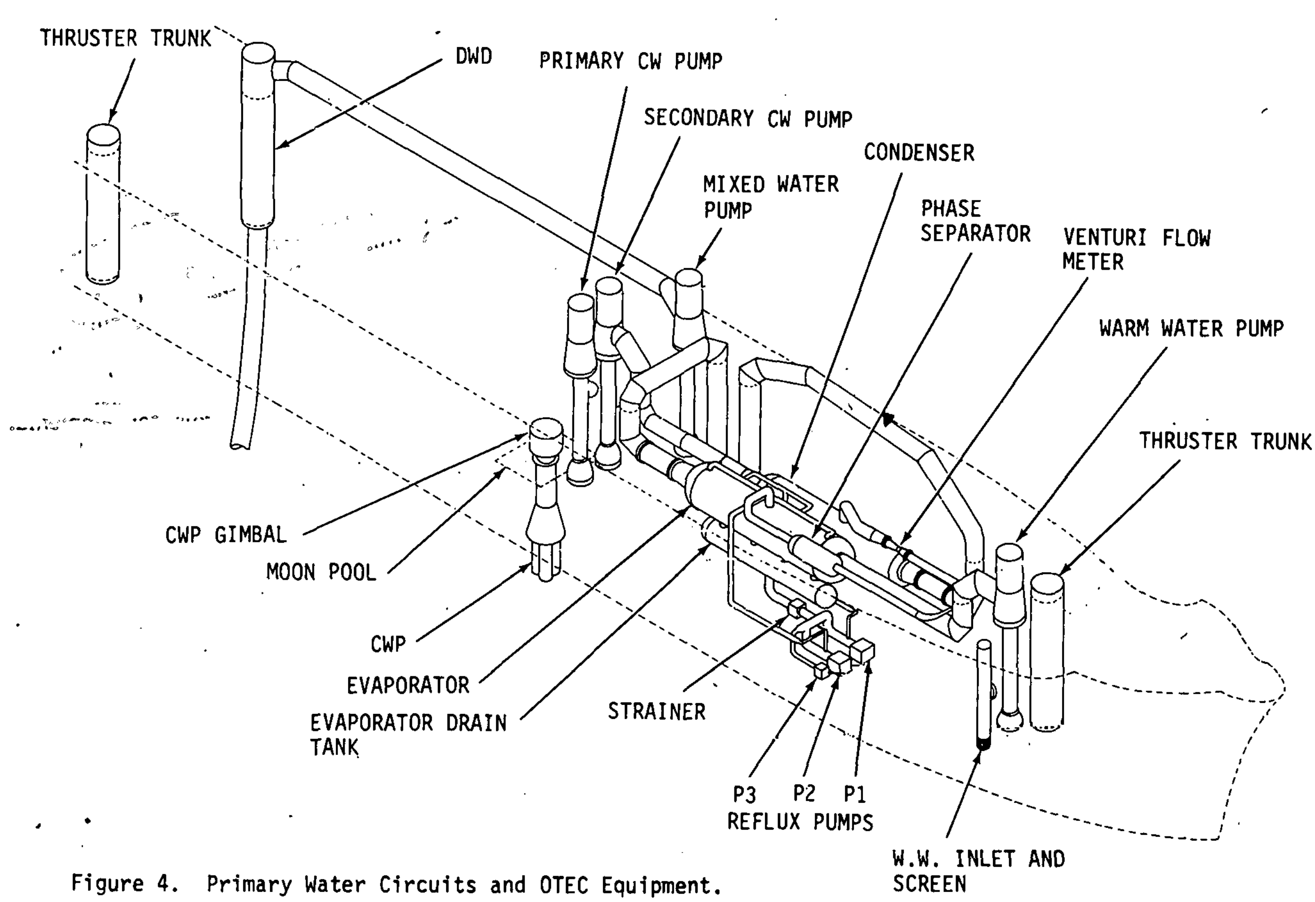

Figure 4. Primary Water Circuits and OTEC Equipment.

water sump in no. 1 starboard cargo tank. Both CW and WW are discharged from the two heat exchangers into no. 5 port wingtank, and from there the mixed discharge is pumped to the deep water discharge (DWD) system, located in no. 9 port wingtank. These primary water circuits are shown in Figure 4.

The thrusters, retractable and fully rotatable in azimuth, are located in no. 9 starboard and no. 1 port tanks in nine ft, six-in tunnels extending the full depth of the vessel through the deck and the bottom shell. A total of 2000 SHP is available for positioning the vessel with respect to weather direction (to minimize vessel motions) and for maintaining tension in the mooring lines.

The OTEC-1 power cycle auxiliary systems (ammonia supply and storage, nitrogen inerting gas for system purging, and a hypochlorite generator) are all located on the foredeck over the OTEC compartment. A 4.75 ton API designation crane is also provided for resupply activities. A similar crane, located forward of the aft house, will handle stores and ships supplies. A rescue boat is stored on the starboard side and is handled by a separate davit.

For handling and installing the CWP, two over-the-side A-frames have been provided; an A-frame and a fixed crane have also been added over the moonpool for use during CWP installation and for other tasks.

All existing ships service systems are considered adequate albeit in need of some refurbishment; they will be serviced and upgraded to meet present day regulations. A larger emergency generator system will be provided to meet the additional needs of the OTEC systems. New sanitary processing and holding facilities will be added.

The OTEC-1 vessel has been fitted with a helicopter landing platform over the stern and will accommodate a fully loaded S-61 helicopter (50,000 lbs).

Safety Aspects

While the design of the OTEC-1 vessel is fairly uncomplicated with respect to marine experience, the OTEC-1 system working fluid, ammonia, has imposed a few non-standard design requirements. The ammonia in this case is handled both as a gas and as a liquid and is both toxic and highly irritating to human eyes and respiratory systems. It is considered explosive and reacts exothermically with water. Certain steels may also become brittle when cooled by contact with liquid ammonia. For protection against these factors, the following safety measures have all been integrated into the design:

- The OTEC compartment is fumetight with external observation platforms and view ports; normal access to the compartment is through a pressurized airlock. Additional emergency escape exits are provided both fore and aft in the compartment.
- A separate compartment ventilation system is installed with exhaust 20 ft above the top of the midship deck house. The normal capacity of this system specifies 12 changes/hr (API RP500) with an emergency rate of ventilation of 20 changes/hr (52,000 cfm) per USCG rules. This system, shown in Figure 5, has three supply and two exhaust fans; the normal compartment pressure will be slightly negative. The airlock airchange rate is 20 changes/hr.
- A fresh water deluge system will be provided, using 160,000 gal of water stored in the no. 4 port wing tank. Normal deluge discharge rate will be 2000

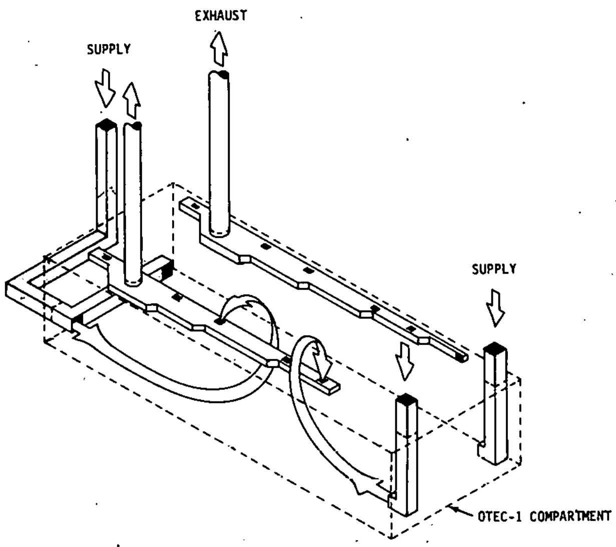

Figure 5. OTEC Compartment Ventilation

gpm (approximately ½ gpm per ft^2 of compartment deck area) through a nozzle spray system located under the main deck. In case of a catastrophic failure, the compartment can be flooded three ft deep in 45 minutes. This flooding rate will be sufficient to limit gas generation due to water-ammonia exothermic reaction (i.e. compartment gas generation is kept within the capacity of the ventilation system). The deluge system will be manually initiated and controlled in response to "High Ammonia Concentration" alarm. For local spill washdown, the ship's fire fighting system will be used directly.

In general, washdown water and diluted spills will be pumped into the discharge tank and directly overboard in harmless concentrations. (In fact, any free ammonia will be converted to nutrient and enter the food chain.)

o For general respiratory protection, filter masks will be available for all hands aboard. For extended use in the OTEC compartment, a breathing air supply manifold will be provided. Also, in all appropriate areas, self-contained breathing apparatus will be available.

Before closing our discussion of the OTEC platform, we should dwell briefly on ships motions and dynamic environments. The contract states that the vessel, the power cycle, and all onboard equipment will be fully operable in seas of 16 ft significant height waves. To meet these conditions, the design effort considered vessel motion very carefully to establish appropriate dynamic design criteria.

The OTEC-1 Test Loop

The major portion of the OTEC-1 test loop equipment is located in the spaces created from centerline tanks no. 2, 3, 4, and 5. The general arrangements of major equipment in these areas are shown in Figure 4.

The OTEC-1 facility will provide a realistic operating environment for testing the theory of producing energy from the ocean. The seawater systems will also provide the opportunity to obtain direct measurements of the long-term effects of biofouling on system performance. Both evaporator and condenser will therefore be extensively instrumented on a local zone-by-zone basis, as well as on an overall basis. Changes in performance due to biofouling/cleaning will be measured and compared to performance of clean tubes. Different methods for reducing fouling will be compared, evaluating mechanical (scrubbing) action and chemical systems.

Power Loop Arrangement

The basic power loop is shown in Figure 6. The average heat exchanger tube fouling factor is expected to run from 0.0002 to 0.0005 hr-ft^2-°F/BTU. The design of this system is based conservatively on the higher value; however, with a fouling factor on the order of 0.0002 hr-ft^2 - °F/BTU (which is expected to be the case), approximately 15% less heat exchanger tube area will be required to produce the same result. The effectiveness of the waterside cleaning system will, therefore, have a marked effect on heat exchanger cost for a given system; note that the heat exhange costs can be expected to run as high as 50% of total plant capital costs.

The condenser is supplied with a nominal CW flow of 68,000 gpm from a 450 hp pump; the design flow can be varied from 90% to 115% of this nominal flow. Similarly, the WW supply (600 hp pump) provides a nominal 82,600 gpm to the evaporator. All CW and WW piping is 60-in OD steel piping, ½-in thick wall, with internal epoxy coating. The mixed-flow discharge to the deepwater discharge (DWD) system requires a 750 hp pump and a 72-in OD pipe. All pumps are of the vertical propeller type, with vertical DC motors (see Figure 4).

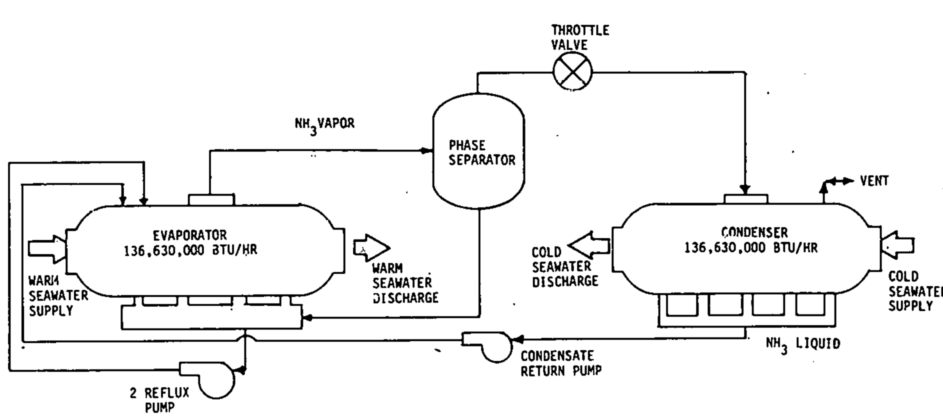

Figure 6. The Basic Power Loop Schematic.

Both condenser and evaporator are located on a foundation flat in the OTEC compartment. The condenser is located on the port side, approximately 25.5 ft above the ship's baseline; the evaporator is slightly further aft on the starboard side and about two ft higher in elevation.

Filled with ammonia and water, the condenser and evaporator weigh 240,000 lbs and 430,000 lbs respectively, a fairly hefty load to support in a dynamic environment. Despite the 16 ft significant sea state requirement, however, the dynamic design requirements were met once the foundation flat was added to our design. The CW and WW circuits also received special attention with respect to vessel dynamics. For example, the WW circuit represented a horizontal length of nearly 150 ft in which, at any given time, there was nearly 100 tons of water. While the vessel will move in six degrees of freedom (heave, pitch, roll, yaw, sway, and surge), only heave, pitch and surge will have any significant effect on the internal hydrodynamics of this configuration.

Since the vessel in a regular seaway would normally react in a sinusoidal manner (with a total period of less than ten sec), the inertia of the moving sea water suppresses the effects of motion; the "worst case" net result was only a $\pm$ 2% pressure variation of the evaporator WW head which degraded to $\pm$ 1.75% with quartering waves. The net effect of this pressure "surge" would be negligible on variations of BTU heat exchanger output.

As an interesting aside, the large surfaces of condenser and CW piping exposed to the high humidity tropical air will result in a fair amount of condensation. In the OTEC compartment, this condition will create the need for drip pans for collecting and channeling condensation away from maintenance and personnel operating areas and walkways; exposed instrumentation must also be designed accordingly. The total amount of condensation which can be expected on high-humidity days is around 300 gal/hr.

Ammonia Systems

The ammonia system for the initial OTEC-1 plant configuration is a closed loop, as shown in Figure 6 schematic. The ammonia liquid enters the evaporator and is sprayed onto the WW tube bundles in two planes. Spray headers are located lengthwise above the water tubes and also on the horizontal midplane between the two tube bundles (where they spray down over only the lower bundles). The ammonia which does not evaporate drains into a surgetank below the main evaporator shell and is recirculated or refluxed back to the top level nozzles. The reflux rate is up to four times the evaporation mass flow rate. The midplane feed level is used only to provide make-up feed to replace the evaporated ammonia.

Approximately 256,000 lbs/hr ammonia vapor exits the top of the evaporator at 72° and 133.4 psia. This vapor passes through a phase separator which removes ammonia droplets and dries the gas. The ammonia vapor then passes through the throttling valve, simulating an ammonia turbine in terms of pressure and temperature losses, and enters the top of the condenser at approximately 55°F and 85.8 psia. The vapor lines are 24-in and 36-in, into and from the 10-in throttling valve, respectively; some 12-in lines are also used.

The ammonia vapor then feeds into the top of the condenser and flows straight down across the tube bundles. The vapors condense and drain into the five zone drain pots at 48°F and 88.4 psia, enter the condensate return pump, and are pumped into the evaporator at 148.4 psia. The cycle is then repeated.

To operate efficiently, the ammonia power loop requires an ammonia support system to receive, process, and continually purify ammonia for make-up purposes. Since the cycle efficiency decreases approximately 2% for each percent of water in the working fluid, the power loop requires nearly pure ammonia.

The ammonia support system is shown in Figure 7. Its major components are the rectifying (purification) column, the reboiler and condenser, the blowdown and surge tanks (all located in the OTEC compartment) and three tanks of 4,000 gal capacity each (located on the foredeck).

The rectifying column and reboiler will remove water and other contaminants, primarily lubricating oils, from the ammonia supplied to the vessel. The system will also clean contaminated ammonia stored in the wet ammonia and heel storage tanks. The clean (dry) ammonia will be stored in the three ammonia storage tanks.

Contaminated ammonia in the ammonia loop will concentrate in the evaporator sump and will be continuously bled into the heel storage tank. The ammonia in this tank will be cleaned as needed.

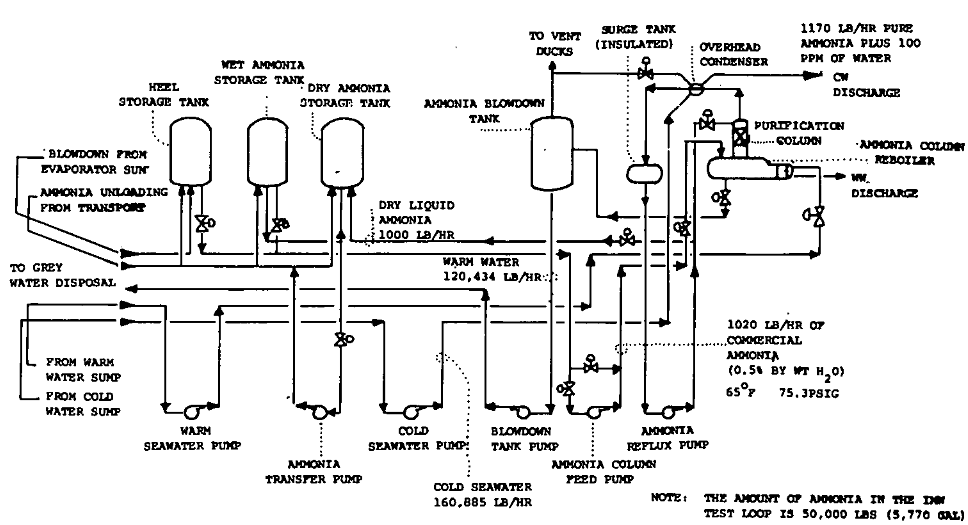

Figure 7. Ammonia Support System.

During purification, the ammonia (vaporized in the reboiler by warm seawater) enters the top of the rectifying column and exits as vapor. The vapor is then condensed to liquid in the overhead condenser (cooled with cool seawater). Most of the clean liquid ammonia is pumped into the dry ammonia storage tank and the remainder refluxed back into the rectifying column.

Ammonia waste exits from the reboiler with 25% water (by weight) and is piped to the ammonia blowdown tank. There it separates into volatile products which are discharged through OTEC compartment vent ducts during normal operations. During a total purge, approximately 2,900 lbs will be released in this manner.

Nitrogen Supply and Purging System

Nitrogen gas will be used as a flushing agent for purging the ammonia power loop and for preventing mixing inflammable mixtures of ammonia vapor and air. Also, following maintenance activities involving opening all or part of the ammonia equipment, air and water vapor must be removed (purged) from the power loop prior to filling it with the ammonia working fluid. Cavities within the system will also have to be purged with nitrogen prior to maintenance or inspection operations.

Figure 8 shows the nitrogen support subsystem design. This system will store liquid nitrogen at 250 psig. The liquid nitrogen flows through a vaporizer and is gasified. The converted gas is reduced to the desired pressure and distributed, as required. The nitrogen system has been sized to accommodate three purges of the ammonia power loop.

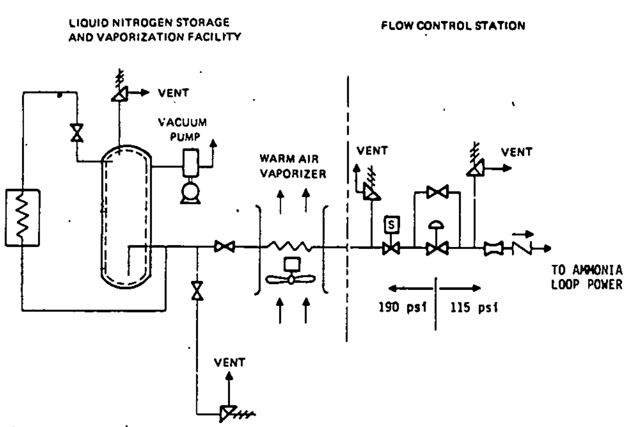

Figure 8. Nitrogen Support System.

The pressure and liquid level of the storage container will be monitored in the control center. All low temperature components and piping are designed with materials suitable for operation at -200°C. A drip pan will be provided to prevent liquid nitrogen or condensed air from dripping on deck plates and other steel structures. The unit will be refilled by means of a specially designed transporter for shuttling liquid nitrogen from onshore facilities.

Biofouling Control

For years, common power plant maintenance in this country has relied on intermittent application of relatively high doses of chlorine (2-5 ppm residual) to prevent or control fouling of intake and discharge systems and heat exchangers. While it is undeniably effective, this practice does cause some damage to the ecological system. In other countries, however, continuous, low-level chlorination (0.02-0.05 ppm residual) has been found effective in controlling biofouling organisms.

For this program, chlorine will be produced from seawater. The chlorine generator will produce sodium hypochlorite (NaOCl) and hydrogen gas electrolytically. As much as a ton of free chlorine can be produced daily. Normal injection rates will be 0.05 ppm; minimum to 1 ppm maximum will be injected, if required, every eight hours to maintain biofouling within acceptable limits, the latter as a 15-min injection three times per day.

The hydrogen gas produced as a byproduct will be collected and mixed with air, then vented safely to the atmosphere.

A system designed to remove biofouling mechanically by circulating small sponge rubber ("Nerf") balls through the heat exchanger tubes will be used separately or in conjunction with chlorine. This system, the Amertap system, is shown schematically in Figure 9. The balls are biodegradable sponge rubber, slightly greater in diameter than the tubes. They are injected into the CW and WW streams ahead of the heat exchangers and are collected after passing through the tubes.

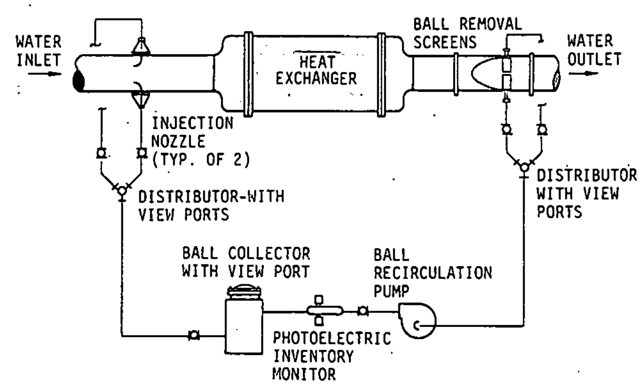

Figure 9. The Amertap Tube Cleaning System.

The balls are randomly distributed. At any given time, about 10% of the tubes are "scrubbed" by these balls, each pass providing an average tube cleaning frequency of about once every ten minutes. The number of balls may also be varied to provide different cleaning frequencies. If stubborn fouling persists, these "Nerf" balls may be replaced by abrasive, silicon carbide sponge rubber cleaning balls.

Note that both CWP intakes and inlet to the WW sump will be provided with large mesh screening (1 in x 4 in); finer mesh screens (3/16 in x 3½ in) are provided ahead of the pump suction in both sumps.

For macrofouling, screens and sumps will be cleaned thoroughly with a vacuum cleaner-type tool. Matter picked up in this manner may be discharged overboard if the quantities are reasonably small (after all, it is part of the food chain); otherwise, it will be collected and sent ashore for disposal.

Deep Water Discharge System

The deep water discharge (DWD) system conducts the mixed CW/WW outflow from the OTEC-1 vessel while causing a minimum of harm to the environment. At a temperature that is about 20° colder than the ambient surface water, this discharge may affect marine life around the vessel if it were discharged near the surface. Therefore, the DWD will be deployed to conduct the discharge plume to a water depth where it will mix with water of the same density. Since both salinity and temperature will be slightly different than the surrounding water, mixing will continue downstream for some distance.

The discharge water is pumped from wing tank no. 5 port to a discharge caisson in wing tank no. 9 port, and into the flexible discharge "hose". The "hose" is manufactured from neoprene-coated, dacron fabric with flanges at both ends. The flange-to-flange length of each of the eight hose sections is 36 ft. Outflow discharge can, therefore, be carried to variable depth down to approximately 240 ft (the bottom of the caisson is 24 ft below the waterline).

The hose sections are stowed in a rack on the aft deck and are handled by the aft pedestal crane during installation and removal. To secure these sections during transfer, the crane is fitted out with an auxiliary winch (operated from the crane cab) for handling the bottom of the hose sections under windy conditions.

The bottom section of the DWD is a weighted cylinder section of 20,000 lbs. This weight will stabilize the hose. This cylinder also has 12 orifices or openings in the side; the bottom end of the cylinder is closed, providing another 10,000 lb. downward force to further stabilize the hose. The DWD is supported at the bottom of the caisson by a gimbal, which permits the hose to swivel, and a rubber bellows to accommodate angular displacement of the hose without leaking discharge water into the surface water.

The Cold Water Pipe System

The Cold Water Pipe (CWP) Design basically consists of 7 major elements:

- o Gimbal/Diaphragm Assembly
- o Pipe Assembly
- o CWP Instrumentation
- o Material Considerations
- o Model Testing
- o CWP Handling System
- o Manufacturing and Assembly Considerations

These elements are integrated to cover the complete range of activities associated with deployment (up-ending), keelhauling and mating operations, and release and recovery. Figure 10 depicts the overall system as installed in the moonpool area.

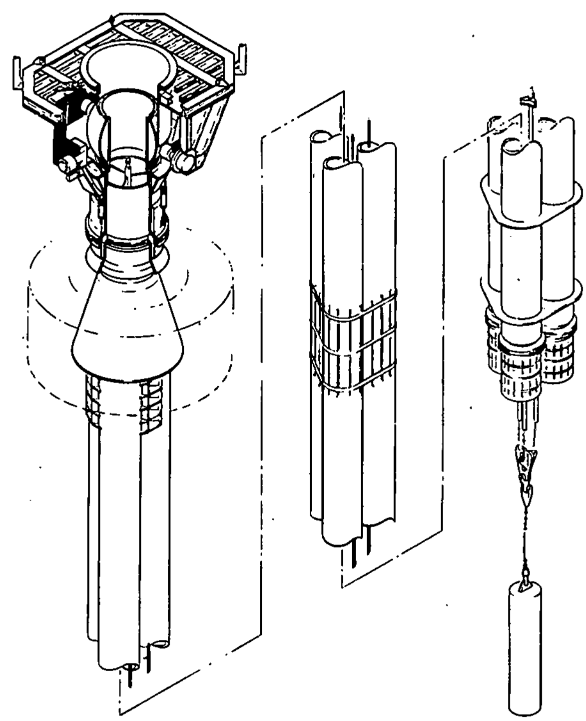

Figure 10. Cold-Water Pipe Assembled with Gimbal/Diaphragm System.

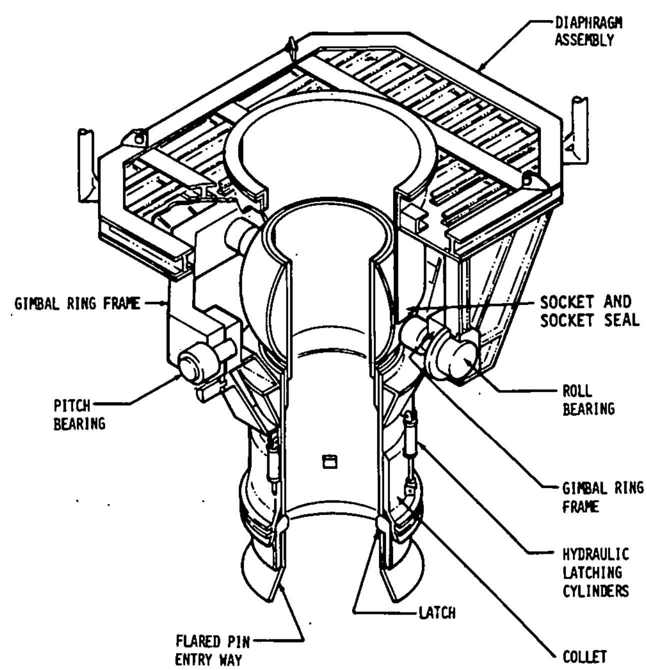

Figure 11. Gimbal/Diaphragm Assembly.

Gimbal/Diaphragm Assembly

The gimbal/diaphragm assembly provides a structural interface between the ship and pipe assembly and isolates pitch and roll motions (see Figure 11). The uppermost diaphragm subassembly acts as the bottom of moonpool sump and provides a barrier to the warmer external surface seawater.

The integral gimbal/diaphragm assembly can be raised out of the water for transit and for maintenance activities. In the operating position, the diaphragm is supported along all four peripheral sides by a support structure attached to the enclosing moonpool bulkheads. Additionally, the diaphragm lower pedestals fit into guide clamps on fore and aft moonpool bulkheads. An elastomeric gasket, approximately 2-in thick, is used as continuous seal around the diaphragm periphery. Vee-type pedestal elements, hanging beneath the diaphragm at the fore and aft ends, provide support for the gimbal roll axis bearings. The bearings are designed for the maximum vertical loads of about 430,000 lbs and longitudinal and transverse loads of 80,000 lbs. The gimbal assembly weighs approximately 50,000 lbs dry weight, 44,000 lbs in operating condition.

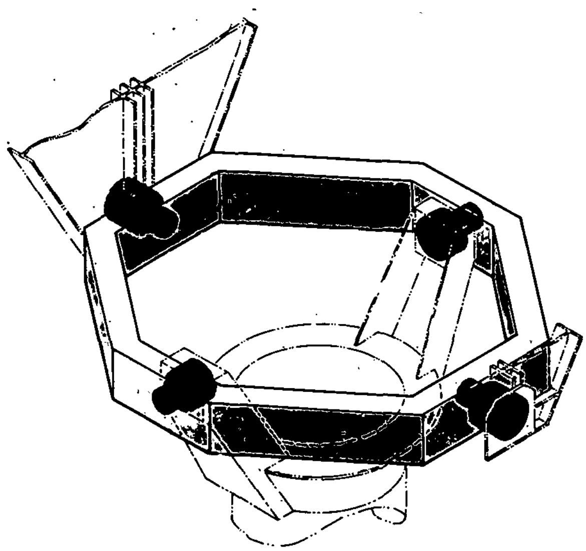

Figure 12. Gimbal Ring Structure.

An octahedron-shaped weldment (see Figure 12) makes up the gimbal ring structure surrounding the inner spherical upper socket seal. The gimbal ring transfers CWP loads to the diaphragm support structure and into the ship structure proper. The upper spherical socket provides a sealed flow path into the primary sump. The lower socket area contains the flared entry guide and is designed to mate with the pin of the pipe assembly and seal off the surrounding warmer water.

To maintain the CW temperature in the sump, close limits on allowable WW leakage has been established; a total leakage of one gallon of WW into the sump is equivalent to the loss of about 340 BTU's from the CW. There are three areas which potentially might leak: around the periphery of the diaphragm subassembly, at the spherical sockets and between the socket and pin. Special seals have been designed for each of these areas, and little leakage is expected.

Pipe Assembly

The pipe assembly is designed around three 48-in diameter polyethylene pipes for conducting the CW from a water depth of approximately 2250 ft to the OTEC-1 platform. The pipes are joined at the top with the upper transition piece and have individual (but restrained) lower transition stabilizer weights. A bottom weight is supported below these pipes by three wires (running inside each pipe) from the upper transition piece.

The pipe assembly (Figure 10) is approximately 2500 ft long, including bottom weight, when completely extended; each continuous polyethylene pipe is 2212 ft long. At the upper end there is a steel cone-shaped "pin" which mates with the gimbal socket structure shown in Figure 11. Around this pin is a buoyant, syntactic foam flotation collar. The pin structure supports the bottom stability weight through three polyethylene-encased steel cables, one cable through each 48-in pipe. The upper pin connects the three polyethylene pipes. At the bottom end of each pipe, a steel (ballast weight) collar is supported by a standard polyethylene stub joint to form the lower transition area. Hanging off the steel ballast collars are the coarse grid inlet screens which provide for horizontal inlet flow. The bottom stability weight prevents excessive displacement of the lower end of the pipe, and aids in keeping the three pipe subassemblies together.

The upper pin bolts to the top diaphragm plate which, in turn, is connected to each of three upper transition joints. For each pipe subassembly, the upper transition consists of an internal key, the polyethylene flange, and joint bolted "horn". The horn was provided to achieve a gradual transfer of load from polyethylene to steel. Figure 13 illustrates this design. For even the most severe design conditions, maximum stress levels are below 2000 psi. The rubberized elements provides a relatively "soft" bearing area that spreads the loading imposed on the polyethylene during high bending moments.

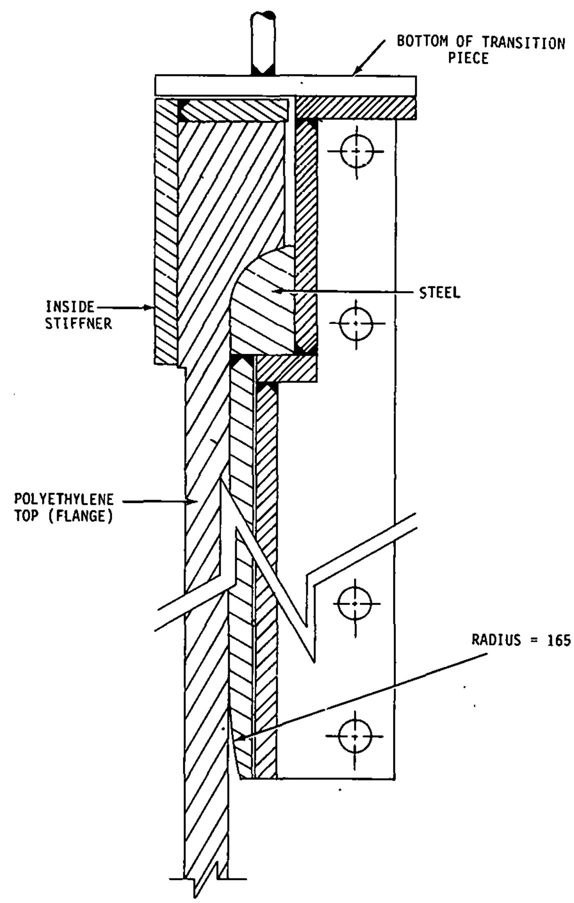

Figure 13. Section showing Attachment on Top of Polyethylene Pipe to the Steel Transition Piece.

The pipes are banded together at approximately 20 ft intervals. The bands have been designed to provide independent axial motion of the three pipes and also to act as support for the instrumentation cabling. In the interior sector formed by the three pipes, a six-in diameter polyethylene pipe provides a conduit for an additional, retrievable instrumentation package.

The polyethylene pipes terminate with the bottom ballast weights, tied together to prevent any lateral movement between pipes. These weights also counteract the pipe buoyancy (density of the polyethylene used is 0.95). They are cylindrical, closed at the bottom, with vertical slits for horizontal inflow of cold water while keeping swimming animal life from entering the OTEC-1 system.

The final subassembly is the bottom stability weight and associated fixtures and cabling. The primary function of the bottom weight is to maintain cable tension to limit bending of the respective pipe subassemblies without incurring high axial loads within the polyethylene. After running through the bottom of the intake screens, the cables are connected to a single, 300 ft long bottom weight support cable. The length of this cable precludes possible shock impact resulting from emergency release of the CWP and free fall to the seabed floor.

The weight distribution of the CWP is as follows:

	Dry Weight (lb)	Wet Weight (lb)
Upper Transition and Pin	60,000	52,000
Polyethylene Pipe Sections	726,000	(60,300)
Lower Transition Pieces and Ballast Weight	74,800	65,300
CWP Stability Weight	159,000	96,000
Buoyancy Collar	52,800	(88,000)
Total, Detached Weight of the CWP	1,073,600	65,000

CWP Instrumentation

The CWP has been instrumented to determine the position of the pipe with respect to the OTEC-1 vessel (bottom displacement and gimbal rotation), to evaluate the dynamic behavior of the pipe against analytical modeling, and for ascertaining that pipe stresses are within the structural limits of polyethylene.

The vessel has a single-point moor. To avoid fouling the CWP with the mooring line, it is important to maintain tension in the mooring line should wind and current subside. While the mooring line tension alarm would require action on the part of the ship's personnel, the location of the pipe would assist in positioning the ship.

The pipe has been designed with the aid of several, quite sophisticated computer programs. While we, and many other experts, have generated our best analyses to include every material consideration and design configuration into these programs, there are many technical constraints difficult to measure or define. Therefore, to evaluate the tools we have used to evaluate the CWP design, a confirmation and/or a refinement of our design parameters will be necessary for future pipe designs and studies. This, of course, will be measured in terms of realistically recorded and analyzed weather, ocean environments, and ship's motions.

Some instrumentation will also be used to monitor stresses in the pipe directly. Such design criteria as collapse pressures (hoop stresses) reflect the pipe behavior with respect to the pressure gradient across the thickness of the pipe wall. The axial stresses in the pipe will also be measured by specially mounted strain gages along the length of all three pipes.

A small fourth pipe will provide a channel for future instrumentation systems. Such systems would provide a flexible capability consisting of inclinometers, accelerometers, or other measuring devices.

In addition to the pipe structure and material, deep water temperature and pressure at the intake (ballast/lower transition pieces) are monitored. The temperature is recorded to determine "heat" losses in the pipe; the pressure may be used to determine the elevation of the bottom of the pipe and hence the length and (loaded) stretch of the pipe under dynamic loading.

Material Considerations

The use of high density polyethylene as a structural material for a flexing pipe under load represents an advance in technology. The OTEC-1 Program will use standard but high grade polyethylene sections which are fusion-bonded in the field to make up the CWP. The polyethylene is flexible, reducing ship-induced dynamic loads significantly. However, the high density polyethylene has certain material characteristics which make it behave differently than more "standard" engineering materials; these must be fully understood and adhered to. The properties of particular concern are a variable modulus of elasticity, and creep and its recovery. These are significant to design in a marine environment due to the cyclic nature of the loading.

Polyethylene is a viscous elastic material for which strain is neither proportional to stress nor independent of loading time. A typical relationship between strain and time is shown in Figure 14. The elastic strain follows Hooke's Law. The retarded strain (primary creep) gradually recovers in the course of time, and the viscous strain (secondary creep) is maintained after unloading. In this case, the pipe was designed for strains nominally within the limit of retarded strain range: up to 6% with a factor of safety of three to four to bring the maximum acceptable static strain to 1½ to 2%, and the maximum total static and dynamic strain to 3 to 4%.

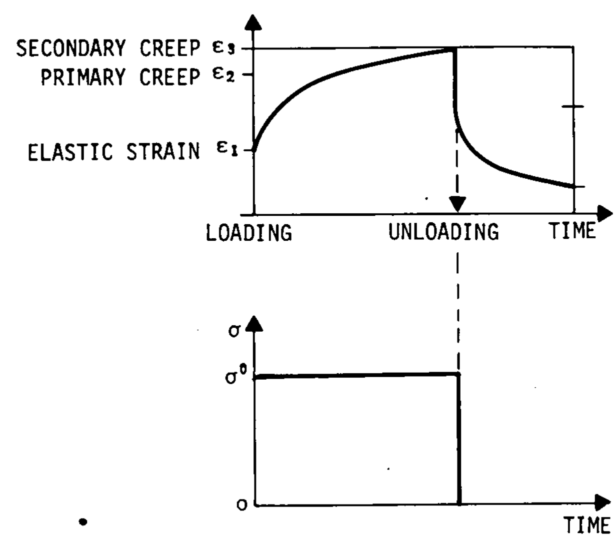

Figure 14. Typical Polyethylene Properities.

Polyethylene characteristics are also sensitive to temperature. Figure 15 shows the yield stress time relationship as a function of temperature, a factor considered in the pipe design because of the wide temperature range. Such characteristics make it difficult to model pipe behavior by computer. It should also be noted that creep modulus behaves differently in water than in air and varies with time as well.

From all our studies and analyses of the CWP, we have accounted for the differences between static and dynamic loads. This was especially important when considering the nature of the short term dynamic forces (see Figure 16). The polyethylene material, however, will recover induced short term dynamic strains as long as the relaxation periods are sufficiently longer than the duration of load application. Note that the two most critical long-term load criteria are the pressure differential across the pipe wall at the upper section and the ballast weight tension at the lower section.

For collapse pressure differential, the factor of safety is 3.5 static and 1.4 dynamic.

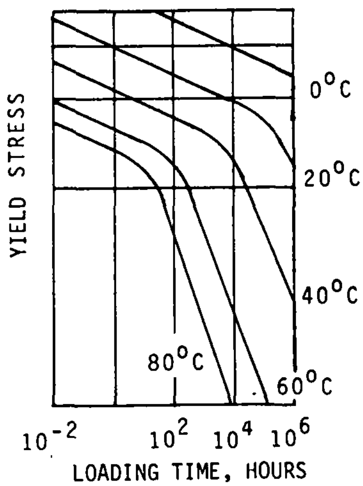

Figure 15. Typical Errors of Yield Stress vs. Loading Time and Temperature for Polyethylene.

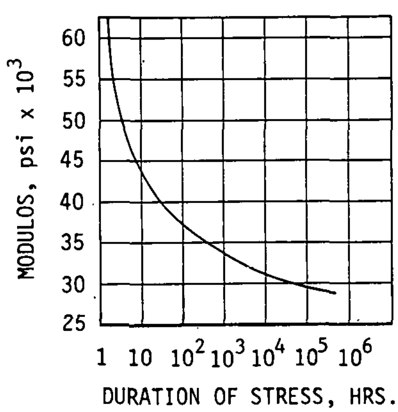

Figure 16. Modulus of Elasticity vs. Load Application Time.

During deployment and up-ending of the pipe, it is likely that the pipe will experience some of the highest stresses. Figure 17 shows some of the design considerations, including the importance of maintaining tension in the pipe while it is being lowered to an upright position. However, if stresses induced by vessel and pipe motions in waves were coupled with the steady-state bending load, the loading could exceed safe design limits. During deployment, therefore, the pipe tension will be kept above 10,000 lbs to quickly and completely submerge the pipe and alleviate the danger of damage from surface waves.

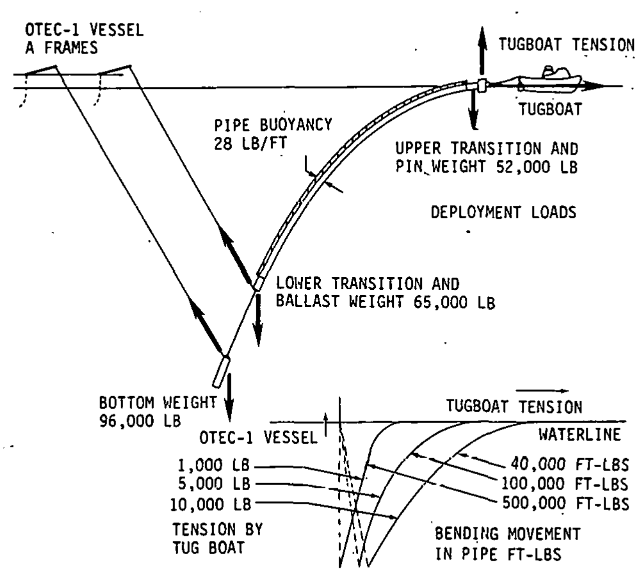

Figure 17. Forces During Up-Ending CWP Assembly.

CWP Model Testing

To determine more precisely the drag coefficient of the pipe to predict the behavior of the pipe cluster in an ocean current, the pipe configuration was tested in a model test basin. From these tests, some basic hydrodynamic characteristics of the pipe cluster were established:

- At subcritical Reynolds numbers, the drag of the pipe bundle was roughly equal to the drag of a single pipe of the same total diameter (i.e. D = 96-in full scale).

  The Reynolds Number at 1.2 kt in sea water for the pipe bundle is 1.2×10^6. For the model configuration chosen, only half of this value was attained. Above $R_e = 0.6 \times 10^6$, the drag coefficient was estimated to be 0.5 or less. Figure 18 shows the drag coefficient as a function of Reynolds Number.

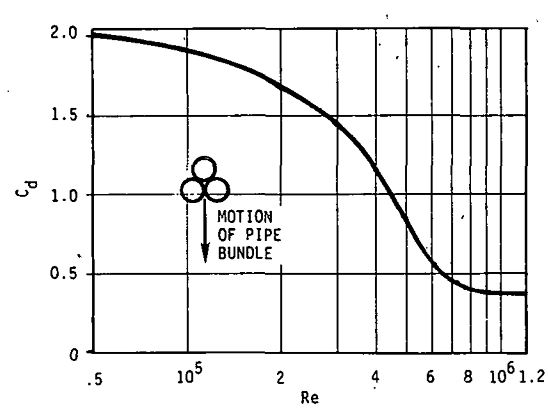

Figure 18. CWP Lateral Drag Coefficient.

- The only stable orientation of the pipe cluster (with respect to the current) was found to be two pipes facing the current with one pipe trailing. Other orientations will tend to twist the pipe bundle to this stable orientation. Since current measurements at the site show a classical Ekman Spiral to be present, the pipe bundle may twist accordingly. However since the magnitude of the current velocity vector diminishes rapidly with increasing water depth, this problem will be quite minor.

o Galloping was observed during the tests to have more effect on pipe motion than did vortex shedding. Neither appear to be of a magnitude of any significance to the pipe design.

CWP Handling System

The CWP handling system is made up of a moonpool crane structure, two over-the-side crane structures, a dual winch cable system plus a handling control system (see Figure 19). This equipment is used to:

o Upend the pipe assembly to a vertical attitude;

o Support the pipe assembly while alongside the ship;

o Keelhaul the pipe assembly to the moonpool;

o Mate the pipe assembly within the socket; and,

o Lower the pipe assembly to the seabed.

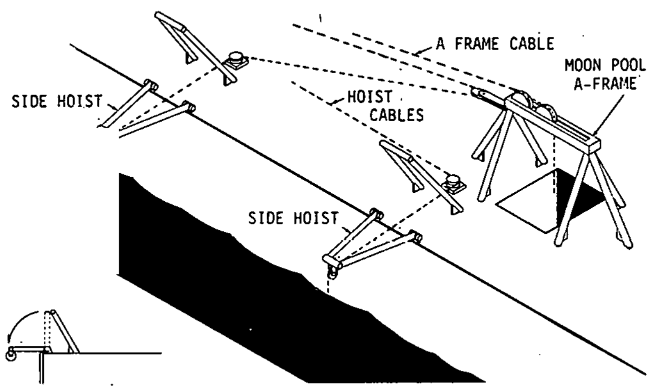

Figure 19. CWP Handling System.

In addition, the equipment can raise the gimbal/diaphragm subassembly to an above-water transit position or to a slightly higher maintenance position. All controls and instrumentation necessary to control the pipe handling equipment are located at the winch station.

Figure 20 shows the up-ending of the pipe. As already indicated, it is important that there be at least 10,000 lb tension in the CWP during this process to avoid high bending stresses. During up-ending, the over-the-side A-frames or crane structures support the lower transition piece (ballast section and the bottom stabilizing weight). Having lowered these to the full depth, the lowering lines will be released and retrieved. The keelhaul line, previously "threaded" through moonpool and gimbal to the starboard side of the ship, is attached to the inhaul guide post, and the CWP assembly is lowered until it is beneath the moonpool (see Figure 21).

The keelhaul line is then hauled to raise the CWP and let the pin enter the socket (at this time, the socket will be free to swing to permit the pin entry at the angle of

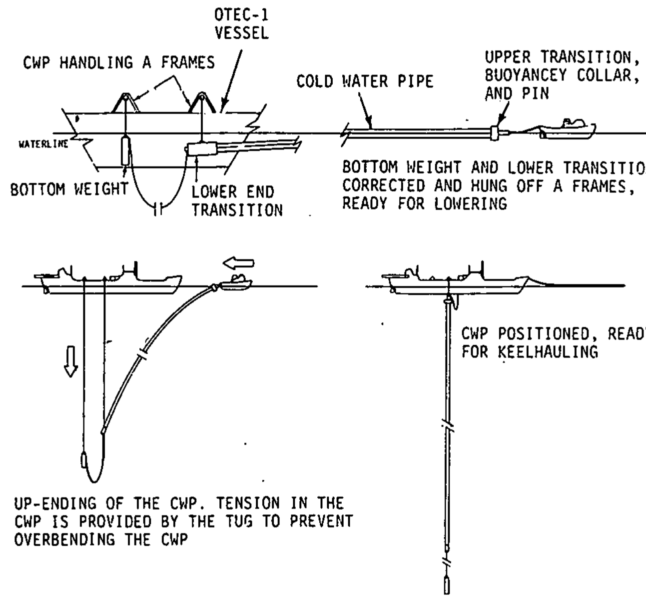

Figure 20. Upending the CWP.

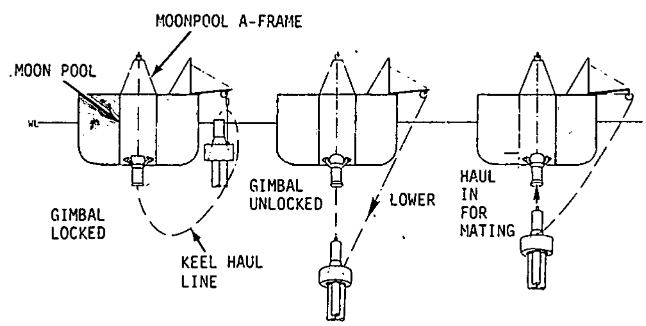

Figure 21. Keelhauling Sequence

the streaming CWP). This operation is monitored by two underwater cameras (monitored at the winch control station) and supported, as required, by one or two divers.

The pin is drawn fully into the socket and latched with a Regan type latch (see Figure 22). The keelhaul line is released, and the CWP is ready for operation.

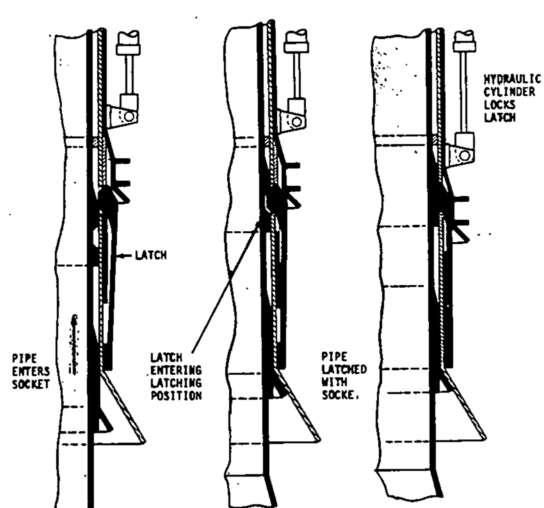

Figure 22. Latching Sequence.

During the three months period when the ship is at a shipyard for modifications to OTEC equipment and general maintenance, the pipe is "stored" on the seabed. The pipe is unlatched from the socket and lowered to the bottom and released from the lowering line. The stablizing weight acts as a mass anchor which prevents the pipe from moving about, and the buoyancy collar gives the CWP assembly buoyancy so it will stand up during the period of inactivity.

When the vessel is re-deployed to the site, she is first attached to the moor and then is moved to the CWP location (by tug and thruster power). The recall buoy (located on the buoyancy collar) is remotely released, and the recall messenger line is hauled aboard the ship. The re-attachment tool is sent down this line, fastened to the keelhaul line, and is latched to the CWP. The pipe is then hauled in the same manner as above.

Manufacturing and Assembly Considerations

The CWP System consists of two main hardware areas:

- o The Gimbal/Diaphragm assembly
- o The CWP assembly

The gimbal/diaphragm assembly will be installed while the ship is in drydock to insure proper fit with the ship's structures. The gimbal/diaphragm assembly is a series of large weldments and some specialized components. The diaphragm subassembly, gimbal ring, and yoke/socket subassembly may be built separately, but they must be mated at the shipyard. There are some critical, final installation dimensional requirements, however, which require detail attention such as gimbal bearing alignment and manufacturing control of the curved surfaces of the socket assembly.

Some critical fits and plating tolerances are involved with socket and pin assemblies. The latching equipment is also somewhat specialized due to the high strength steel and requires close tolerances. The gimbal instrumentation (designed into the bearing assemblies) will also require sophisticated installation techniques.

The need for achieving proper manufacturing control of the polyethylene pipe for the CWP is of particular importance. Our vendor, DuPont of Canada, has a good record of high quality polyethylene pipe manufacturing experience. The DuPont factory will also make up (i.e., assemble) the steel transition pieces with the flanged polyethylene pipe end sections since they will require close-fit tolerances.

The polyethylene pipe material will be shipped in 60-ft sections to the Port of Kawaihae on Hawaii where the three, 48-in pipes will be assembled. A graded area, 300 ft by 2500 ft will be required. A portable polyethylene bonding or welding machine will fuse the pipe sections into three lengths of 2112 ft flange-to-flange. The bottom stabilizing weight wires are then installed, the top and bottom transition pieces assembled to the polyethylene pipes, and the CWP instrumentation system installed. Approximately three weeks will be required for assembly, with two additional weeks required for check-out and attachment of instrumentation.

Final assembly of the 1,074,000 lb CWP System will require heavy lift equipment for handling the subassemblies. The CWP will be assembled on carts running on a track into the sea for launching. After assembly, the pipe ends (with the transition pieces) will be lifted onto barges for transport to the site; the polyethylene itself will float.

The assembled CWP will be towed to the vessel already anchored at the site. The bottom weight and lower transition pieces will be attached to the A-frame handling wire ropes and lowered. The pin/top transition piece with flotation collar will be transported by tug, which provides enough tension in the pipe to prevent overbending and buckling of the pipe. The limiting seastate for these operations is SS4.

When the pipe has been up-ended in this manner, the keelhauling wire rope line is reaved from the pin through the yoke/socket to the moonpool A-frame. The CWP is then mated to the socket female guide and latched. The pipe is now installed and ready to operate.

Station Keeping

The OTEC-1 vessel will be anchored northwest of the Island of Hawaii in 4500 ft of water. The specific point location of the anchor site is $156^{\circ}10'10''$ W and $19^{\circ}55'10''$ N.

The vessel will be permitted to swing about that point in a watch circle of 1¼ nmi radius. The moor will be required to maintain the vessel on station in winds of 30 kt, waves of 16 ft significant height, and an ocean surface current of 1.2 kt.

The OTEC-1 vessel will remain on station for nine months. For the balance of the 12-month operating cycle, she will be moved to a shore facility for maintenance and modifications. The CWP will be lowered to the seabed where it will remain upright and anchored by the bottom stabilizing weight until retrieved.

The stationkeeping system consists of three parts: the thrusters, the act of finding and selecting a suitable anchoring place, and the anchoring system.

Thrusters

The thrusters will be employed to position the vessel with respect to the directions of seas and winds and also to provide auxiliary power to back off the mooring system in slack weather. During development and/or attachment of the CWP, the vessel will also maneuver with the thruster system. The thrusters must, therefore, be rotatable in azimuth and have variable thrust (propeller rpm). Also, to provide as large a turning moment as possible, the thrusters are placed in no. 1 cargo tank, port and no. 9 wing cargo tank, starboard. This arrangement provides a turning movement of nearly 10×10^6 ft-lb at 1000 hp for each thruster - satisfactory for any contingency within the weather spectrum described above.

The thrusters will maintain the heading of the vessel into the seas from $0^{\circ} \pm 45^{\circ}$ to minimize vessel motions. It is expected that the thrusters will be used less than 10% of the time and then only at reduced power. Operation of full speed can be expected once or twice a season.

The thrusters are DC variable speed drive, constant pitch propellers, made by Propulsion Systems Inc. The thruster/drive/azimuth control package operates within a vertical, 9½ ft diameter thruster tunnel. When necessary, the entire package can be hoisted up to an upper position for stowage during transit or for major maintenance. Normal in-service maintenance on the motor drive is performed with the thruster in the "down" position.

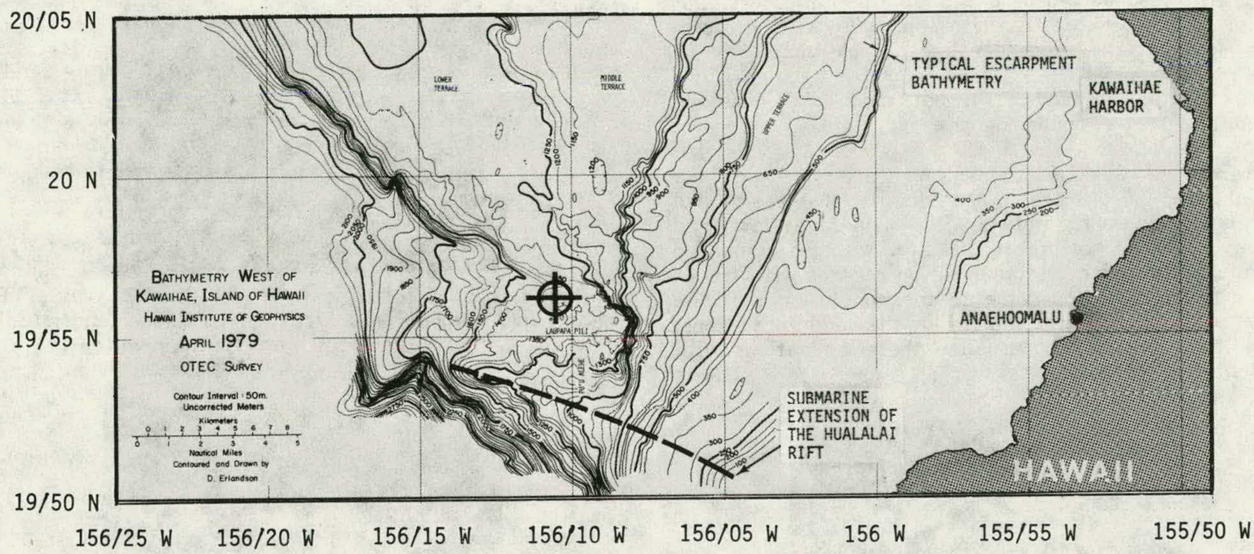

Figure 23. OTEC-1 Site Bathymetry.

The thrusters are operated by joy-sticks from a bridge console. Single stick control can be employed to control single thrusters or both. Thrusters will also be operated by an alarm command of "Slack Mooring Load" (i.e., low mooring tension) indicating that the vessel is drifting toward the moor.

Site Survey and Selection

The initial bottom physical data available from the selected site indicated a sufficient sandy overburden over a bedrock which would enable the use of virtually any kind of drag embedment anchor.

It was decided that more site data would be required, and in October 1978 Global Marine Development Inc. dispatched the R.V. KANA KEOKI under the leadership of Dr. C. E. Helsley and J.F. Campbell, from the Hawaii Institute of Geophysics (University of Hawaii) to the site to verify bathymetry and sedimentary overburden. After searching the entire Kohala Plateau and the southern tip of the Island of Hawaii no sedimentation of any consequence was found. What was found was a fairly hostile bottom formed too recently (in geological terms this meant 500,000 to 800,000 years ago, the oldest formation to the north) for any sedimentation to collect. From this and four subsequent surveys, using both the R.V. KANA KEOKI and the R.V. NOI'I, adequate bathymetric data was finally developed for the site.

Figure 23 shows the bathymetry of the general site area. Notice the series of escarpments (scarps) on the main area of the Kohala Plateau. These are thought to have been formed by faulting since each terrace is fairly flat and bears evidence of having been for a short period of time in the photic zone before the structure lowered further into the sea. A few remnants of small reef structures were visualized from seismic records and some reef material was brought up from bottom drags. As an example of what these terraces look like, Figure 24 shows the escarpment of such a faulted terrace on the eastern slopes of the Island of Hawaii.

The bathymetry also shows an area, Laupapa Pili, which may have been an old underwater lava lake formed from eruptions of the submarine extension of the Hualalai Rift, shown with a slight northward extension, called Pu'u Alehe, a 400-ft high volcanic hill. At the eastern end of Laupapa Pili, a steep, 2500 ft scarp forms a very forbidding rocky and boulderous boundary. Along the bottom of this scarp sedimentation we had hoped to find sedimentation, expecting it to have migrated down the slope. Some sedimentation was indeed found here, but it was too close to the scarp for the "comfort" of the CWP which, during periods of ships maintenance at a shore facility, would be lowered to the bottom and parked for three months.

We decided to provide the OTEC-1 vessel (and CWP) a stand-off cushion from the scarp (and from the 800 m isobath which represented the lower end of the pipe, with a small clearance below the stabilizing weight) and set the distance away from the 800 m isobath to be 2.5 nmi. This took the site out of the little sedimentation we did find during our surveys.

The "wind-shear" line is an invisible line between "good" and "bad" weather. It swings up and down from a line tangent to the northern side of the Kohala Mountain Range, most often crossing the $156^{\circ}10'$W longitude around 20°N.

As experienced first hand, the Alenuihaha Channel between Hawaii and Maui is quite rough. The wavetrain which forms north and east of the Hawaiian Archipelago enters this channel like a venturi and heaps the seas up well towards the southwest. The windshear line is the boundary line for these waves (which sometimes can reach 10 to 12 ft on an otherwise nice day). Since the OTEC-1

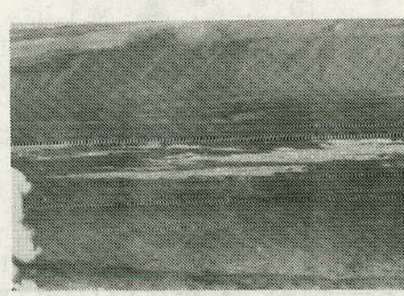

Figure 24. Faulted Terrace on the Island of Hawaii

vessel will be supported by one or more small boats, this is no place to operate. The site selected was, therefore, as far south of the windshear line as possible.

The question remained: If not sand, what was the bottom like at the anchoring site?

We studied the various aa and pahoehoe-type lava flows (Figure 25) onshore in the vicinity of the site and found great similarity between the land lava flows and pictures from the bottom at the site. We also compared our underwater seismic sedimentation records with traces on land (Figure 26) and came to the conclusion that the seabed would look fairly much like the land formations, especially if one would visualize these formations with sedimentation deposits.

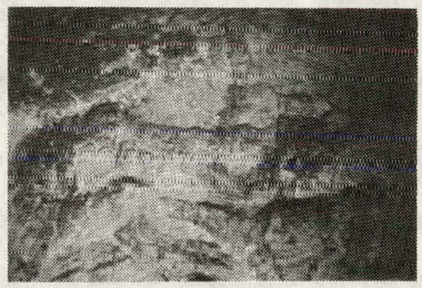

Figure 25. Typical Lava Formation at the Anchoring Site.

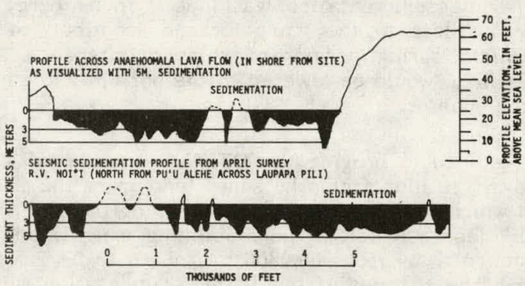

Figure 26. Comparison of Bedrock Profiles on Land and on the Ocean Floor.

For the selected site, therefore, we determined that the seabed formation was fairly flat (hills up to 30 ft over 3000 ft distance) with outcrops and pressure ridges 6 to 10 ft high of exposed, very rough solid lava, pillow lava and rocks of all sizes, and a small amount of sedimentation filling hollows (3 to 5 ft) and crevices (to 20 ft). (See Figs. 27, 28 and 29.) No evidence was found that a bottom current existed.

Accordingly, we designed a compliant mass anchor.

<u>**The Mooring System and Anchor Design**</u>

As shown in Figure 30, the mooring system is very simple, consisting of a mass moor made from chain.

The chain is made up in seven clusters of 37,000 lbs of 2 in chains, each approximately 10 ft long. These chain clusters are attached to a 2 3/4-in center chain which attaches to the mooring wire. The breaking strength of 2 3/4-in stud link chain is 578,000 lbs.

The mooring wire is 3860 ft of 2 7/8-in wire rope, encased in polyethylene for corrosion protection and supported by a submerged buoy 100 to 150 ft below the surface. The submerged buoy will have a buoyancy of approximately 20 tons.

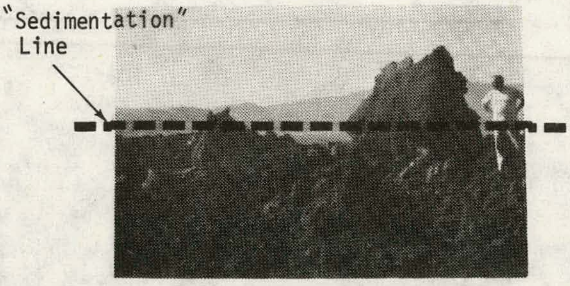

Figure 27. Lava Field Similar to Seabed Formations as Visualized with Sedimentation.

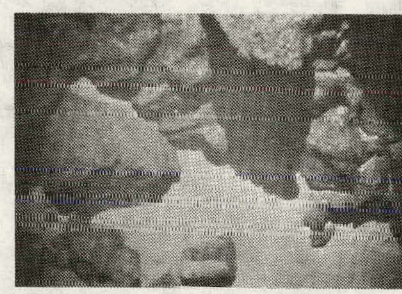

Figure 28. Seabed Rock Formation.

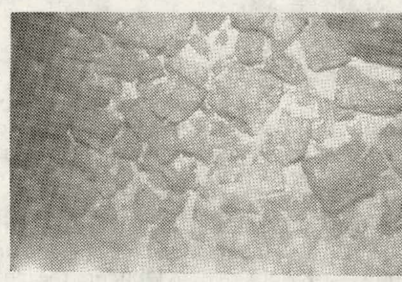

Figure 29. Typical Seabed Rubble Field. (The Viewing Field is 10 Ft Wide.)

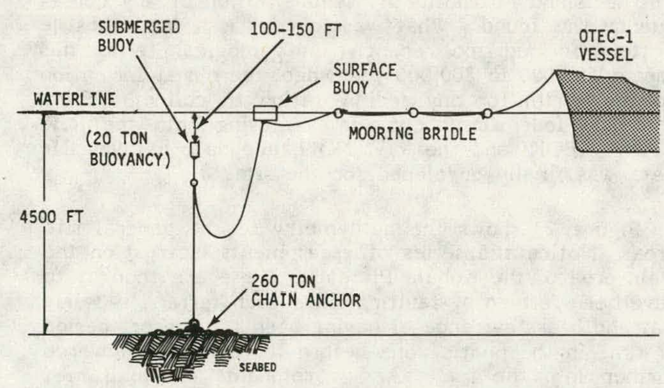

Figure 30. OTEC-1 Mooring System.

The mooring line is a 16-in diameter braided nylon rope (1445 ft long) attached to a flounder plate under the submerged buoy and to a surface buoy of 15 ft diameter. Unloaded, this buoy has an initial draft of 2 ft (buoy weight is approximately 10 tons). Full-load immersion is 5½ ft for a maximum mooring load capability of 100 tons. With an additional 3½ ft of reserve buoyance, the mooring buoy will be 11 ft high.

The hawser or mooring bridle is a 16-in braided nylon line, 1000 ft long. The hawser will have a specific gravity of 1.14 so floats will be required to keep it on the surface. These floats will be painted or wrapped with reflective tape to afford visible recognition at night to avoid small boat damage to the bridle. The mooring buoy will be provided with navigational systems as required by the U.S. Coast Guard. The bridle has a quick-connect attachment to a pelican hook, shackled to a chain, fastened to the vessel and to the loadcell which alarms low or too high mooring loads.

Since this, to date, is the largest anchor ever deployed, detailed analyses have been made to insure its safe "delivery" to the sea bed 4500 ft below.

With the entire sub-sea system stretched out (by tugs and barges) on the surface, the chain mass anchor is allowed to free fall to the bottom. Impact velocity will be approximately 30 fps which should help embedding the chain in whatever cracks and sedimentation there is. At the end of the descent, the wire rope (bottom attachment to the anchor chain) will stop 15 ft short of making contact with the seabed. Once the anchor is deployed, the mooring line, surface buoy and bridle is attached with diver support.

The OTEC-1 vessel can now be moved to the site, the pipe towed out, up ended and attached, and the OTEC-1 system will be ready to operate.

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DEVELOPMENT OF A TEST PROGRAM FOR OTEC-1

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Abstract

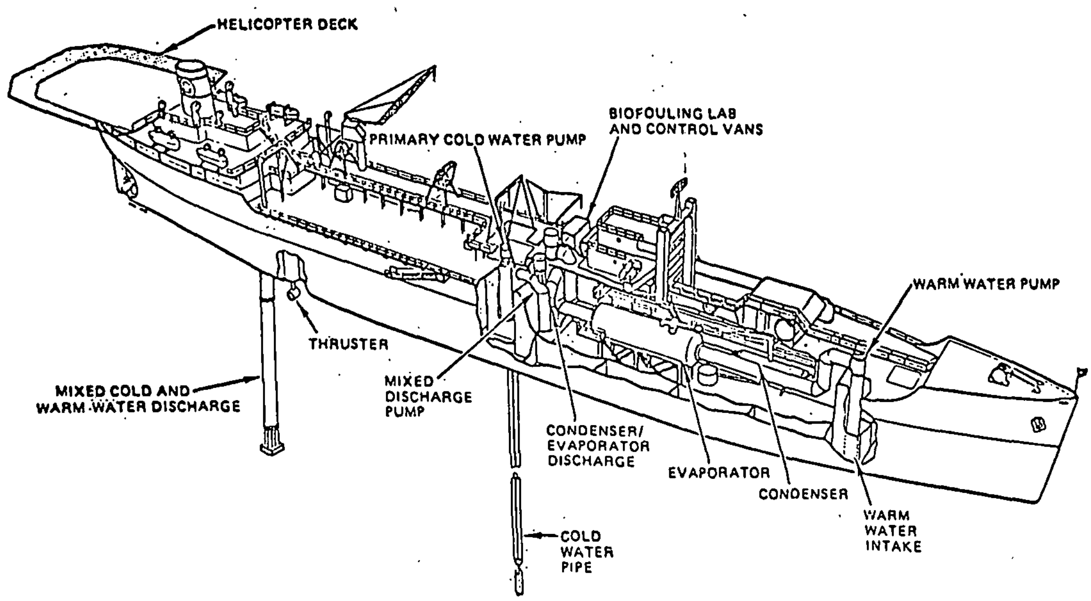
OTEC-1 is an ocean-going, engineering test facility designed to test major integrated systems with emphasis on heat exchanger performance and cleaning techniques. The proposed test program includes evaluation of two sets of shell-and-tube heat exchangers and one set of plate-type heat exchangers, assessment of the impact of biofouling and corrosion in the ocean environment and monitoring of the performance of the cold water pipe. The impact of the OTEC system on the surrounding environment must be evaluated for compliance with Environmental Protection Agency regulations. Baseline heat exchange data is obtained as soon as the system is operational, with the combined cleaning system operating at maximum rates. These tests are repeated at various stages of the program for evaluation of the growth in fouling film resistance. More accurate evaluation of the fouling film resistance is obtained from experiments performed on several modules equipped with Heat Transfer Monitor devices. Instrumented tubes within the tube bundles are used to evaluate effectiveness of Amertap and chlorination on tube performance and to assess the distribution of the Amertap balls across the tube field. ~~Tests are performed with single and multiplane spray feed and varying reflux rates to evaluate possible "dryout" in the bundle and determine optimum reflux ratios. The performance of the enhanced half bundle is compared to the performance of the plain tube half bundle in both sprayed feed and pool boiling modes of operation.~~ Vapor velocity tests are performed to obtain data for predicting performance of a larger scaled up heat exchanger.

Introduction

Part of the OTEC program goal of stimulating the development of commercially viable OTEC technology is the near-term objective of demonstrating the technical and economic feasibility of heat exchangers which can be credibly scaled up for application in commercial offshore and grazing OTEC power plants. The OTEC-1 platform is the vehicle selected for completion of an early test program with minimal technical risk. (Figure 1)

The selected site, Keahole Point, will provide a temperature difference of 40°F between surface water and water at a depth of 2200 feet.[1] The working fluid selected is ammonia, since its thermal characteristics at the predicted temperatures are ideal for the Rankine cycle concept. Selection of ammonia, however, dictates that no copper-bearing materials be used in the power system loop. Since marine installations normally use copper-bearing alloys and antifouling coatings to inhibit fouling, this restriction presents a potential problem for OTEC and tests must be performed to evaluate the impact of both microfouling and macrofouling. Most of the experience gained with the proposed countermeasures systems has been in coastal installations and is not considered directly applicable to deep ocean water systems such as OTEC plants. Therefore, a test program must be undertaken to evaluate the efficiency of the cleaning system and the ability to comply with EPA regulations.

Figure 1. OTEC-1 Ocean Test Facility



Instrumentation and data acquisition selection must be coordinated to assure the accuracy needed to obtain meaningful test data. Decisions must be made on the type of tests to be performed to meet the program objectives. The sequence of testing must take into account the predicted effects of fouling and efficiency of the proposed countermeasures as they affect the heat transfer experiments.

Several contractors and consultants are involved in the design and construction of OTEC-1 and, in order to meet the stated program objectives, their efforts must be integrated. As shown in figure 2, the responsibility for coordinating this effort has been delegated to the Test Director Contractor (TDC).

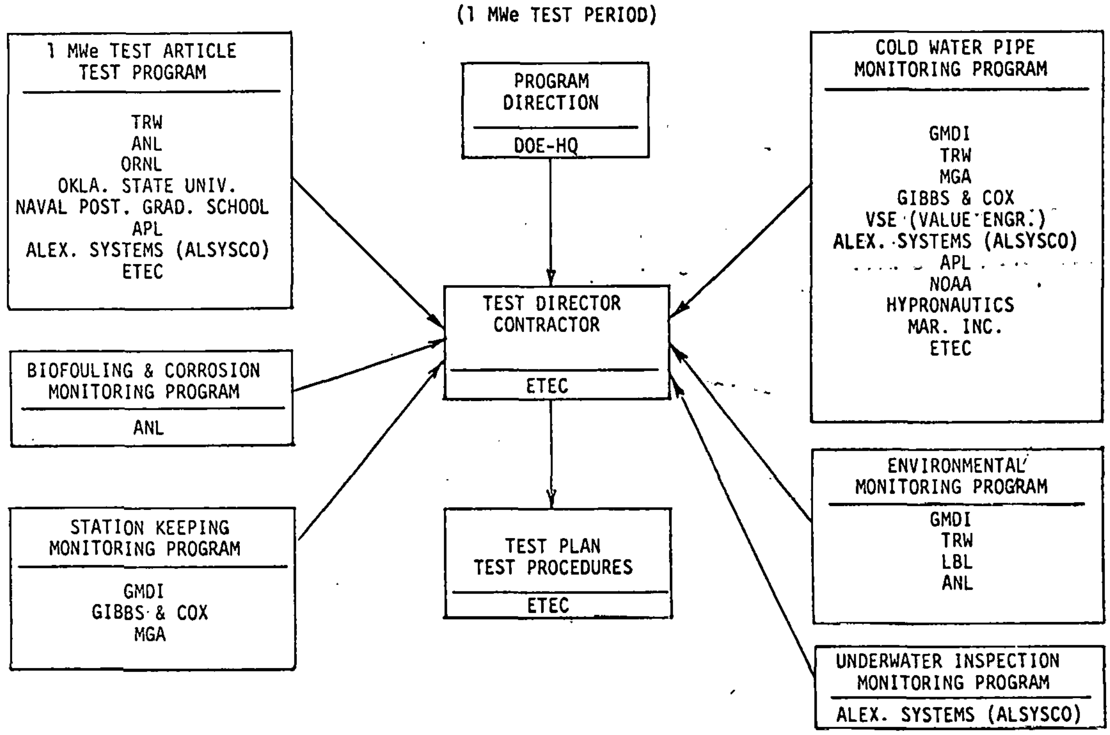
DOE selected the Energy Technology Engineering Center (ETEC) as the Test Director for the OTEC-1 program. ETEC is a government owned-contractor operated facility responsible for the management of the design and construction of facilities and of formalized programs for testing large heat exchangers, valves, pumps, mechanisms and instrumentation in support of the overall DOE energy program.

temperature of 72°F and discharge pressure of 133.4 psia. The heating fluid is 80°F seawater circulated through single-pass tubes at a nominal rate of 82,600 gpm. "Internal reflux" is available at rates from 256,000 to 1,280,000 lb/hr. The condenser is also a shell-and-tube unit with 40°F seawater circulated through single-pass plain titanium tubes at a nominal rate of 68,170 gpm. The condenser is designed to condense 265,000 lb/hr of 97.5% quality ammonia at a pressure of 86.4 psia and a temperature of 48°F.

Since the overall program schedule establishes an eight (8) month test program for the 1 MW_e test articles it is mandatory to make maximum use of system availability from the start of testing. Figure 3 presents a test schedule which is the result of several meetings with the Test Article Contractor and technical advisors during which priorities were established and time cycles developed for testing and configuration changes. In establishing the heat exchanger test sequence, the problem of biofouling had to be considered. An investigation by Fetkovich[3] et. al. at Keahole Point indicated that an induction period of about 4 to 8 weeks is experienced during which a "conditioning"

Figure 2. Organizational Input to OTEC-1 Test Plan

(1 MWe TEST PERIOD)

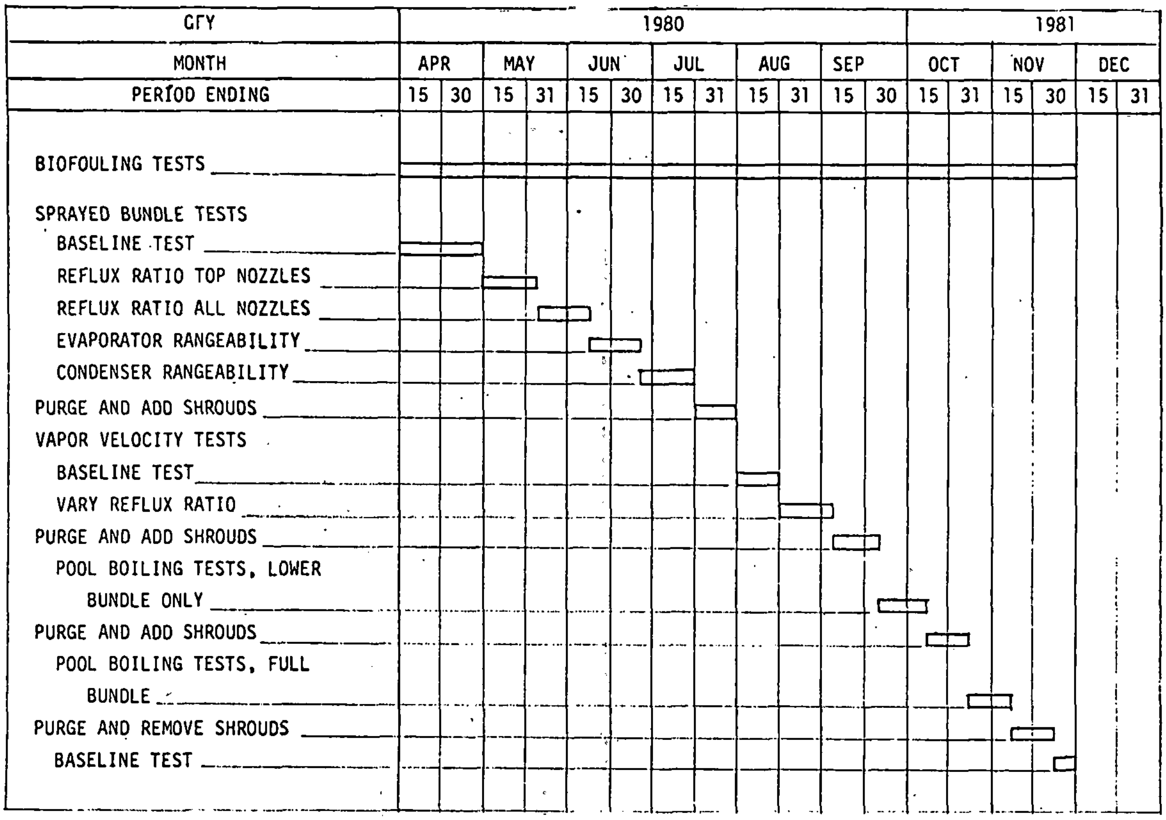


1 MW_e Heat Exchanger Test Plan

The first set of test articles comprises shell-and-tube heat exchangers designed for a nominal duty of 1 MW_e (40 MW_t).[2] The evaporator is a horizontal shell and tube unit with the tube bundle split horizontally into two sections. The upper section of the bundle consists of 3152 plain titanium tubes and the lower half consists of 3152 high flux tubes. The bundle is housed in a 13 ft. diameter shell 55 ft. long. The unit is designed to deliver 256,000 lb/hr of dry ammonia vapor at a

layer forms. After this layer is established the fouling curve becomes linear with time and the plan is to schedule the countermeasures systems to keep the fouling resistance (R_f) within an acceptable range in this linear portion of the curve. It was generally agreed by the technical advisors that baseline heat exchanger performance data should be obtained as soon as the system is operational. These tests will be completed while fouling is still in the induction phase and tests can then be performed to determine the optimum reflux ratios and to develop performance curves for the heat

Figure 3. OTEC-1 1 MWe Test Program Schedule



exchangers by doing rangeability tests with various water flowrates. The rate of growth in fouling resistance during this period will be slow enough to enable meaningful data to be obtained. If this series of tests is completed in accordance with the schedule, figure 3, the fouling film resistance coefficient, R_f, is expected to be stabilized, under control of the cleaning system, at some value less than 0.0005 hr.-ft^2-^{o}F/Btu. Both the Amertap system and chlorination will be operating from the start of the test program with Amertap operating at a rate of 1 ball per tube every five minutes and a chlorination dosage rate of 0.13 ppm.

Since the 1 MWe system does not include a turbine, the demonstration of the concept must rely on obtaining credible heat transfer data for comparison against the predicted performance of the heat exchangers. In order to achieve this credibility, it was agreed at a meeting of the technical advisors that it would be necessary to limit the uncertainty in the overall heat transfer coefficient "U" to 6 to 8%. Tests performed in the ammonia loop at ANL indicated the range of accuracies needed to meet this program goal.[4] A sensitivity study of influence coefficients showed that the value of U was less sensitive to flow than to temperature. Figure 4 shows that the temperature accuracy at the readout has to be $\pm 0.15^oF$, while a flow reading error of up to 5% can be tolerated and still maintain the uncertainty in U within the stated limits. Resistance Temperature Detectors (RTD's) were selected for the temperature measurements and multipath acoustic flowmeters for the seawater flows. The overall heat transfer coefficient U is calculated from:

$$U = \frac{Q}{A_o DT_e}$$

where Q is the heat duty, DT_e is the log mean temperature difference and A_o the outside surface area of the tubes.

The heat duty Q is determined from the temperature change of the water passing through the heat exchanger:

$$Q = m\, Cp\, (T_i - T_o)$$

where m is the water flowrate, Cp the specific heat and T_i and T_o the average inlet and outlet water temperatures respectively.

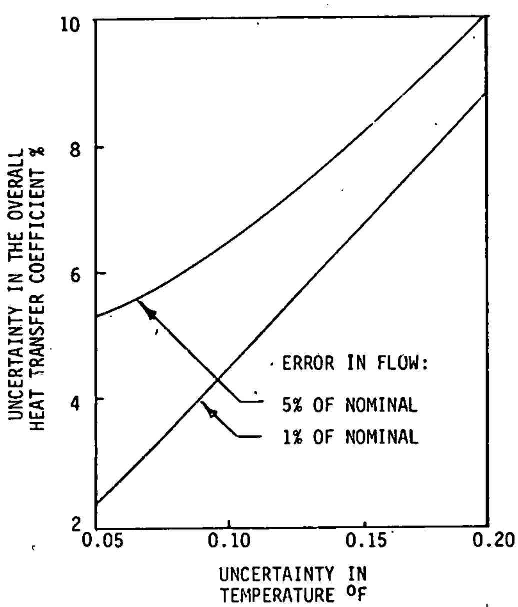
The log mean temperature difference is obtained from:

$$DT_e = \frac{T_i - T_o}{\ln\left[\frac{T_i - T^1}{T_o - T^1}\right]}$$

where T^1 is the ammonia temperature.

By substitution U can be determined from

Figure 4. Sensitivity of U to Temperature and Flow Variations



$$U = m\ Cp/A_o\ l_n \left[(T_i-T^1)/(T_o-T^1)\right]$$

which indicates that the overall heat transfer coefficient depends on the measured quantities m, T_i, T_o and T^1. These parameters will be recorded during initial operation of the 1 MWe heat exchangers at the following nominal flow conditions:

1	Cold water flowrate	68,170 gpm
2	Warm water flowrate	82,600 gpm
3	Cold water inlet temperature	40°F
4	Warm water inlet temperature	78-80°F
5	Ammonia feed to evaporator	768,000 lb/hr.

Starting with the evaporator in configuration 1 of figure 5 for baseline testing the following results are predicted:

	U Value Btu/hr.-ft^2-°F	
Condition	"Clean" Condition	R_f = .0002
Upper Half Bundle	465	425
Lower Half Bundle	760	699
Overall	612	562
Condenser	436	436

Once baseline data has been obtained the ammonia feed will be adjusted to provide the maximum rate of 1,280,000 lbs/hr. through the top spray nozzles only. While this feed rate will preclude "dryout" in the lower tubes, the heavy liquid loading on the upper tubes is expected to reduce the efficiency of the evaporator. During these reflux ratio tests the following measurements will be taken to enable a determination of ammonia quality leaving the evaporator:

1. Moisture separator inlet and outlet pressure
2. Moisture separator temperature
3. Liquid level in moisture separator drain
4. Liquid temperature and flowrate from the separator drain
5. Vapor flowrate leaving the separator.

Assuming that the vapor quality leaving the separator is "dry" and that the liquid level is maintained automatically so that there is no change in stored energy, then the mass flow from the evaporator can be determined using the mass balance:

$$M_e = M_l + M_v$$

where

M_e = mass flow from evaporator

M_l = liquid mass flow from separator

M_v = vapor mass flow from separator

The quality of the vapor leaving the evaporator will be determined from:

$$X_e = M_v/M_e$$

On completion of the test run with maximum feed to the top spray nozzles the reflux ratio will be reduced in increments of 25% of the nominal feed and data taken after the system has stabilized at each set of conditions. Feed will then be supplied at the maximum rate using both top nozzles and mid-plane nozzles. It is predicted that the mid-plane feed will improve performance by providing wetting of the lower tubes with less liquid loading on the upper tubes, resulting in more efficient vaporization in the upper half bundle. The ammonia feed rate will again be reduced in increments with data taken at each feed rate for determination of the optimum reflux ratio. The multi-plane feed is expected to improve the optimum ratio from an approximate value of 2:1 for top feed only, to the range of 1.3:1.

With nominal 1 MW_e flowrates re-established, seawater flowrates will be varied between 60% and 115% of nominal for each heat exchanger. Since the fouling film resistance, R_f, is expected to be controlled by the cleaning system, by this time, the seawater flowrate should be the only variable in the heat transfer equation and performance curves will be plotted from the resulting data for each exchanger. An R_f value for the evaporator will be determined from the data, using the Wilson plot technique, and compared to the value obtained from the ANL modules.

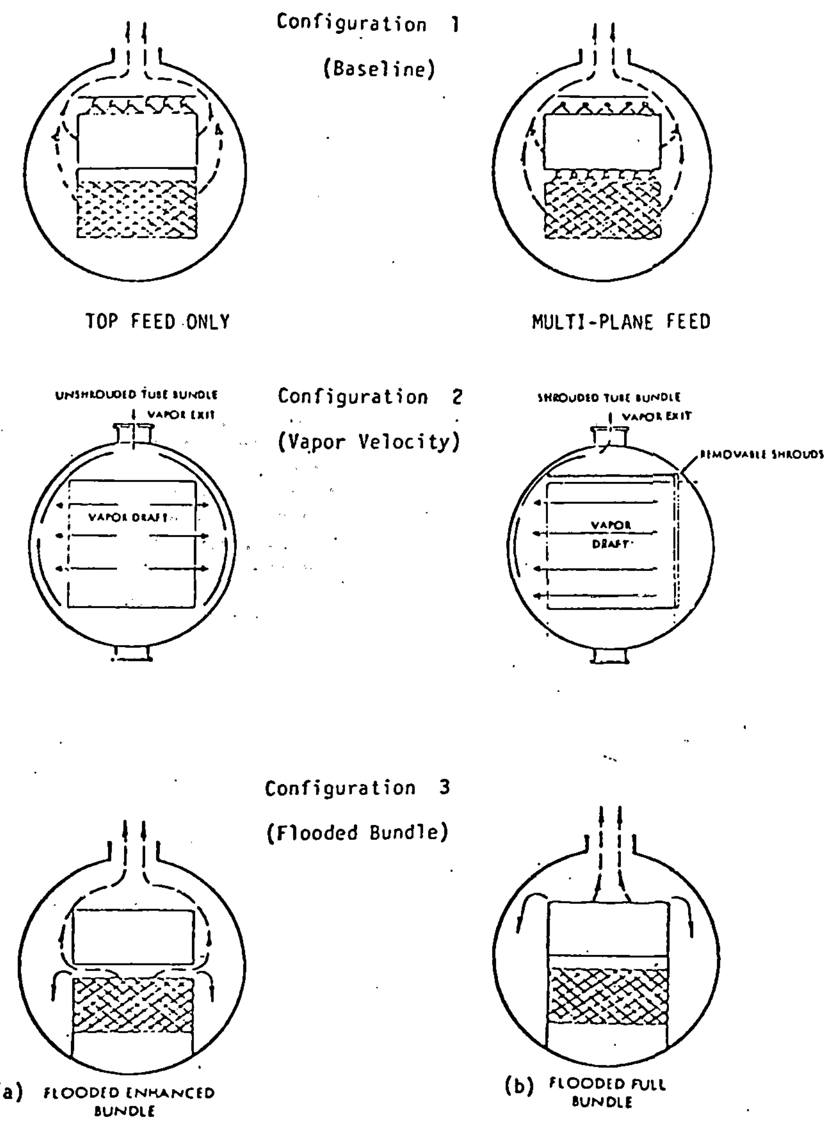
As soon as rangeability tests have been completed on the evaporator, the chlorine dosage will be reduced until a residual reading of .05 ppm continuous is obtained.

Seawater flow will be maintained at a velocity no less than 3 feet per second, to minimize the onset of macrofouling on the waterside. The ammonia side of the system will be drained and purged. Shrouds will be installed to provide configuration 2 of figure 5. In this configuration all of the vapor produced must flow in one direction and exit from one side of the tube bundle. This results in the maximum vapor velocity being doubled over the original configuration. The resulting vapor velocity distribution is identical to that in the corresponding half of a bundle four times as large as the 1 MW_e bundle. The data obtained from tests in this configuration will enable evaluation of an equivalent 4 MW_e unit and will provide an opportunity to investigate liquid ammonia entrainment in high velocity vapor streams. During this test sequence the Amertap rate will be changed from 1 ball per tube each five minutes to 1 ball per tube each ten minutes. The effect of this change on the R_f value will be continuously monitored.

Additional shrouds will be added to provide configuration 3b of figure 5. Pool boiling tests will be made using the full bundle and comparison made with performance in the sprayed mode. Performance in this configuration with the upper bundle being smooth tubes is expected to be less than baseline sprayed tube performance.

During the pool boiling experiments the chlorination system will be operated so that the residual rate is 0.2 ppm for 2 hours each 24 hours, unless the R_f value increases in which case the operation will revert to a residual rate of .05 ppm continuous.

Figure 5. Evaporator Configurations



The ammonia side will then be drained and purged and shrouds installed to provide configuration 3a of figure 5. The lower half bundle will be flooded with liquid ammonia to demonstrate pool boiling performance. Based on pool-boiling tests on an enhanced tube bundle at ANL it is anticipated that the performance in this mode will not be significantly less than in the sprayed bundle mode.[5]

Biofouling Tests

During the course of the 1 MWe test program various combinations of Amertap rate and chlorination dosage rates will be implemented as shown in figure 6. The effectiveness of these countermeasures will be assessed from data taken from a series of instrumented tubes in each heat exchanger and from experiments using the ANL modules.

Figure 6. Biofouling Countermeasures Schedule

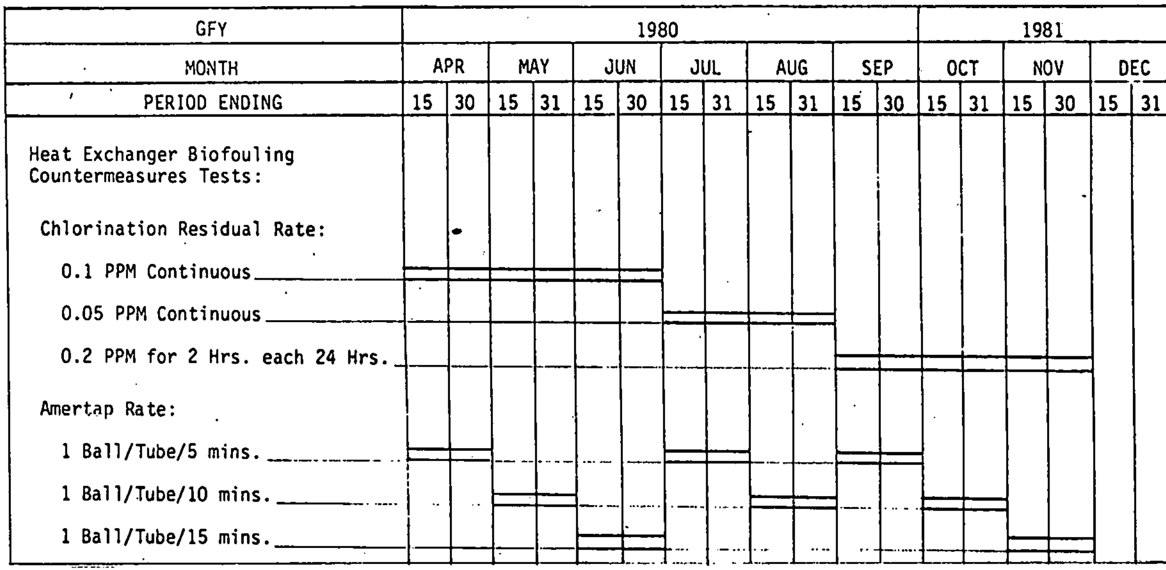
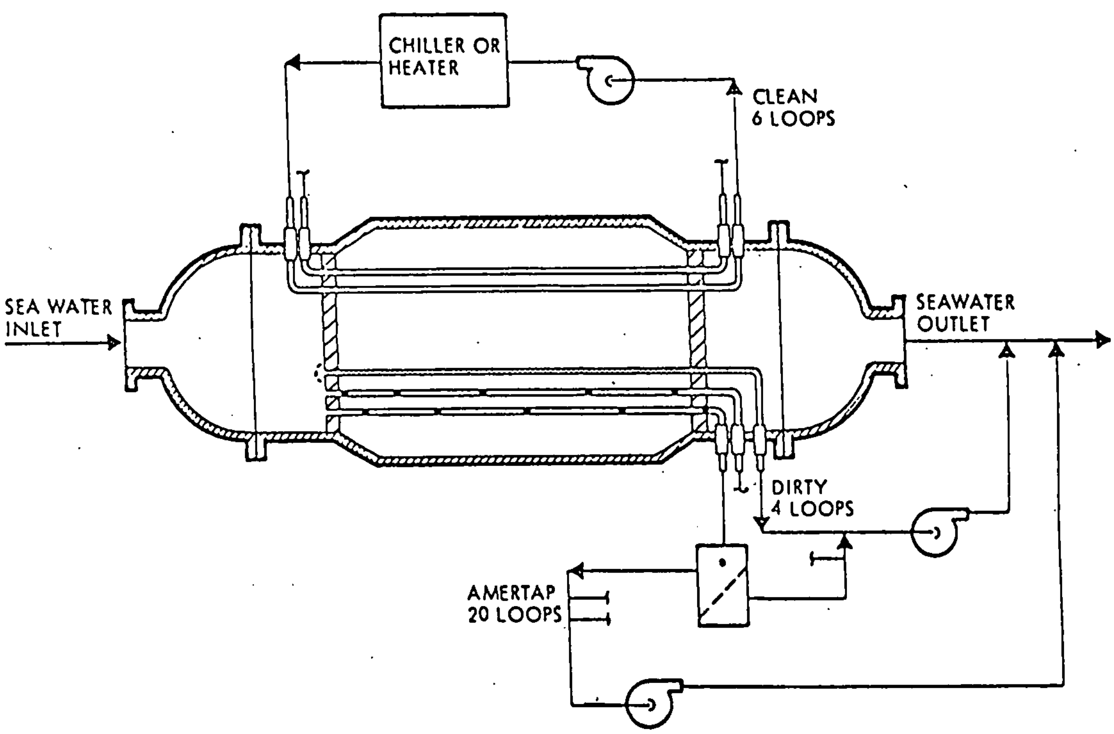


Figure 7. Instrumented Tube Loops



There are several sets of instrumented tubes, figure 7, located in a geometric pattern in each heat exchanger. The pattern is designed to provide an assessment of seawater flow distribution and Amertap ball distribution. Each set of tubes consists of one clean water closed loop, one tube cleaned with Amertap and chlorination, and one exposed to chlorination only. Flow and temperature measurements are made on each tube individually, and the Amertap balls are collected and counted. The inlet water temperature to the clean water, closed loop, tube is automatically controlled to duplicate the incoming sea water temperature. Data collected from each tube will be reduced for evaluation of the fouling buildup and comparison of the effectiveness of the countermeasures.

A set of six ANL biofouling modules will be installed in the OTEC compartment and connected to various water sources as shown in table 1. Flowrates through each module will vary from a nominal 6 ft/second for normal operation up to 12 ft/second during Wilson plot runs on each Heat Transfer Monitor (HTM) device. The flowrate will normally be maintained within 1% of the flow in the associated heat exchanger tubes. During Wilson plot runs over a total range of 2.5 to 12 ft/second the flow accuracy will be within 1% with temperature controlled within $\pm 0.1^{o}C$. Each module consists of a heat transfer monitor device to measure changes in heat transfer rates, a biofouling section and a corrosion section as shown in figure 8. Prior to each 1 MW_e test, the HTM devices will be used to obtain temperature and flow data from which heat transfer coefficients will be determined using Wilson plots. The flowrate will then track the flow through the heat exchanger tubes within 1% and data taken will be analyzed on a daily basis to monitor change in heat transfer coefficients caused by R_f buildup. Comparing data from the different modules will provide an assessment of the countermeasures effectiveness. Sample coupons will be removed from the biofouling and corrosion sections of tubing for characterization of biofouling, scale and corrosion film buildup. At each module takeoff from the main water system, samples will be taken for chemical analysis.

Environmental Monitoring

The environmental monitoring program includes physical, chemical, and biological measurements and meteorological observations.[6] Current meter-wave-rider buoys, as shown in figure 9, will be deployed to record speed, direction, depth, and temperature at discrete depths and to transmit wave rider buoy information via RF data link to a receiver installed either on-shore (initially) or on the platform (after deployment). The buoy system will be retrieved a minimum of every two months to collect the data. Detachable aufwachs coupons will be attached to each instrument of the array and each buoyancy sphere. These coupons will be recovered and replaced when the array is serviced. The plates will be preserved for analysis.

The environmental monitoring program will be carried out in six phases as shown in figure 10. Phase I will be a preoperational environmental survey with observations taken, before deployment of OTEC-1, from a current meter array. The meter array will be serviced at two-month intervals until the platform is deployed and the data reduced to provide a first-order assessment of time and space scales of motion and temperature variations in the vicinity of the OTEC site.

TABLE 1

BIOFOULING MODULE CONDITIONS

	Module #	Location of Water Source	Operating Condition	Amertap Mechanical Cleaning System	Chlorination
Evaporator	1	Warm Water Before Chlorinator	"Continuous" - No Shutdown Longer Than Two Hours	No	No
Evaporator	2	Warm Water After Chlorinator	Track Heat Exchanger Nominal Flow $\pm 1\%$	No	Yes
Evaporator	3	Warm Water After Chlorinator	Track Heat Exchanger Nominal Flow $\pm 1\%$	Yes	Yes
Evaporator	4	Warm Water At Exit From Evaporator	Track Heat Exchanger Nominal Flow $\pm 1\%$	Yes	Yes
Condenser	5	Cold Water Before Chlorinator	"Continuous" - No Shutdown Longer Than Two Hours	No	No
Condenser	6	Cold Water After Chlorinator	Track Heat Exchanger Nominal Flow $\pm 1\%$	No	Yes
Condenser	7	Cold Water After Chlorinator	Track Heat Exchanger Nominal Flow $\pm 1\%$	Yes	Yes
Condenser	8	Cold Water At Exit From Condenser	Track Heat Exchanger Nominal Flow $\pm 1\%$	Yes	Yes

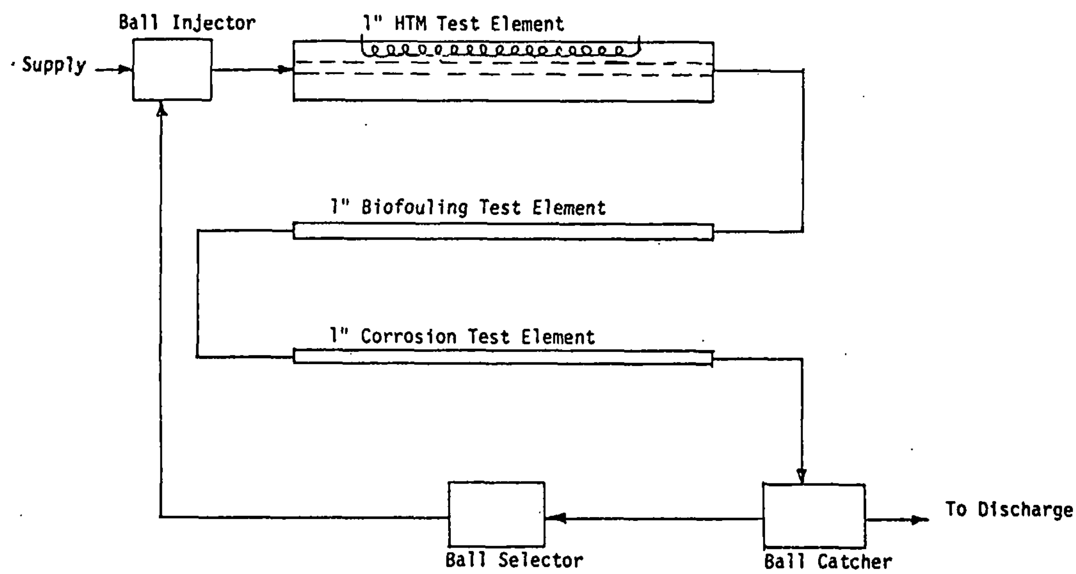
Phase II will provide data after deployment of the platform but prior to start of the test operations. Two additional instrument arrays will be deployed to form a triangle with the platform located at the center. Meteorological and biological observations will be made from the pltform itself. An instrumentation package located on the platform will measure:

1) Insolation
2) Wind velocity (speed and direction)
3) Air temperature (wet and dry bulb)
4) Barometric pressure.

4) Hydrocasts will be used to sample at standard oceanographic depths and at the chlorophyll maximum for biology and chemistry tests. The biology samples shall be taken for biological pigment, ATP and primary productivity. The chemistry measurements will include dissolved oxygen, pH, nitrate, nitrite, ammonia, silicate, dissolved phosphate, total phosphorus, alkalinity, POC, TOC, residual oxident, and trace metals of Cu, Al, Fe, Ti, and Cd.

In addition, net tows will be made at appropriate depths to determine populations and biomass. The tows will consist of surface and oblique plankton tows, both day and night, and tows for micronekton and organisms in the scattering layers.

Figure 8. ANL Biofouling & Corrosion Test Module (ANL)



WATER FLOW RATE: 6 FT/SEC (16 GPM) NOMINAL; 12 FT/SEC (32 GPM) MAXIMUM

Biological study observations will also be made from the platform for:

1) Hull scrapings
2) Setting racks
3) Acoustics for fish census and scattering layer
4) Mammalian and bird life.

Also, a workboat will be utilized to obtain additional information. During environmental cruises the following tasks will be accomplished:

1) Observations and data measurements will be taken to determine wind speed and direction, air temperature, barometric pressure, cloud cover, insolation, and relative humidity, no less than once an hour.

2) The instrument arrays will be serviced and data recovered.

3) A profiling system will be used to measure temperature, conductivity/salinity, light transmission, ocean currents, and irradiance.

After the OTEC-1 test program has started, the environmental monitoring program will continue to obtain data from, and maintain, the current meter-wave-rider buoy arrays, record the meteorological observations, and carry out the bimonthly cruises. In addition to these tasks, the effluent sea water plume will be monitored as to its extent. The plume will also be monitored for the chemical measurements made prior to the start of the test operation with an additional measurement made for the chlorine produced oxidants within the plume. Biological samples will be taken at the same points within the plume and a determination made if bio-accumulation of the chlorine produced oxidants is occurring. Tests will be made to determine if bio-stimulation is occurring.

When the platform leaves the site after testing the 1 MW_e test articles, one cruise will be made in the region to determine any significant effects caused by the presence and operation of OTEC-1. The current meter-wave-rider buoy arrays will be maintained and a complete set of physical, chemical and biological samples will be taken.

After deployment of the platform for the PSD-I and PSD-II test program, the complete environmental program described for the 1 MW_e program will be repeated.

Cold Water Pipe Monitoring

Although the Cold Water Pipe (CWP) is not prototypic, data can be taken and compared to the theoretical analysis to verify the design approach. The pipe must be towed from the fabrication site to the platform and will be subject to wave action during this period. Readings will be taken, from the strain gauges mounted on the pipe, during the towing operation, and analyzed for determination of the stresses experienced. As soon as the CWP is in position, baseline data will be taken and data will be monitored throughout the program for evaluation against predicted performance and for safety of the pipe itself. (Figure 11)

Assumptions made in predicting the performance of the pipe must be verified and instrumentation will be installed in locations selected to provide this verification. A series of LVDT's will be attached to the CWP in the location of the predicted maximum stress. A fourth pipe will be installed down the center of the main three-pipe cluster and will be used for the rest of the instrumentation. Forty (40) strain gages will be mounted on this inner pipe where the gages and connecting cables will be protected. These gages will also be mounted at locations selected to provide the best data for verification of the analytical program used in design of the CWP. They will also be used as a continuous monitoring function to provide information on pipe deformation for use in determining safe operation of the system.

A travelling package consisting of a digital compass and an accurate inclinometer will ride on a guide rail on the inside of the inner pipe and will be used to determine the angle of inclination and degree of twist at any level along the pipe string. The location of the end of the pipe will be monitored using the acoustic beacon mounted on the pipe. The data obtained will be used in analysis of the CWP performance and will be used to determine when to jetison the pipe in the event of adverse conditions building up.

Figure 9. Current Meter Buoy String

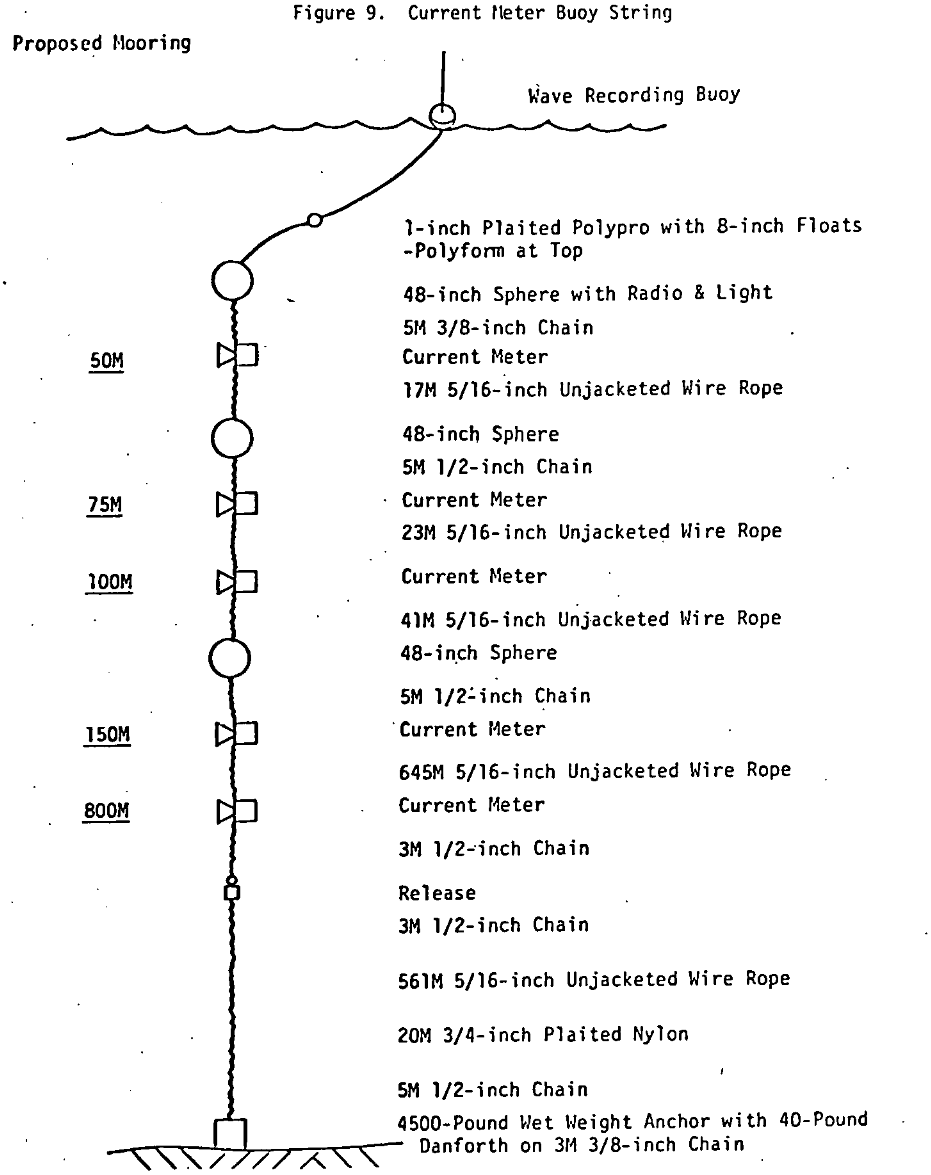
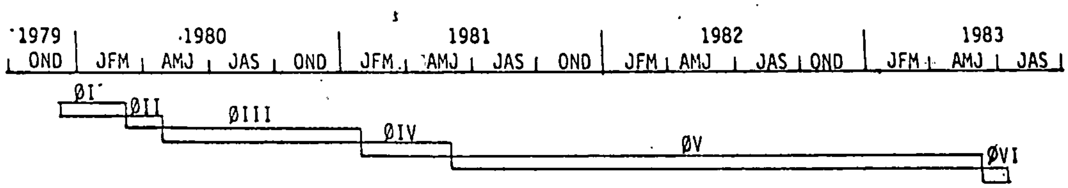


Figure 10. OTEC-1 Environmental Monitoring Phases



PHASE I (PRIOR TO ARRIVAL OF TEST PLATFORM AT SITE) - SINGLE CURRENT METER-WAVE RIDER BUOY

PHASE II (PRIOR TO START OF TESTING) - 2 ADDITIONAL CURRENT METER-WAVE RIDER BUOYS (3 TOTAL), PLATFORM METEOROLOGY, PLATFORM BIOLOGY, CRUISES

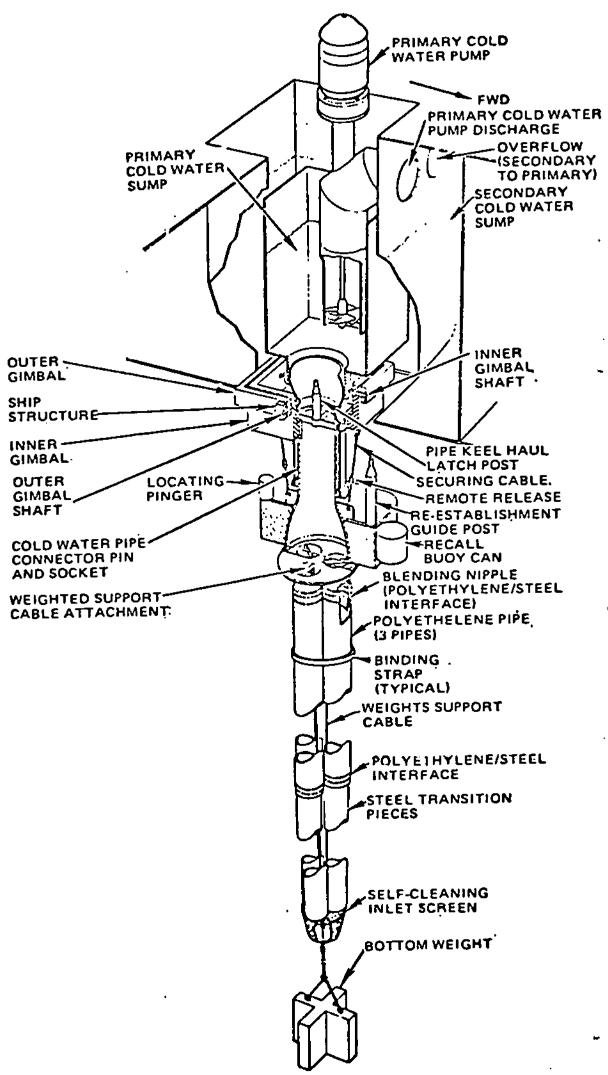
PHASE III (DURING TESTING OF 1 MWe TEST ARTICLES) - ALL DATA COLLECTION SYSTEMS TO BE USED TO MONITOR PHYSICAL, CHEMICAL, BIOLOGICAL, AND METEOROLOGICAL PHENOMENON BOTH AT SEA AND ON TEST PLATFORM

PHASE IV (AFTER TEST PLATFORM HAS LEFT SITE, AND FOR APPROX. 4 MONTHS) - MONITOR CURRENT-METER ARRAYS AND CONTINUE CRUISES OF WORK BOAT

PHASE V (AFTER PDS'S INSTALLED, TEST PLATFORM MOORED AND TESTING RESUMED) - CONTINUED MONITORING AS IN PHASE III

PHASE VI (AFTER TEST PLATFORM LEAVES SITE) - MONITOR AREA ENVIRONMENTAL, COLLECT EQUIPMENT, AND ISSUE FINAL REPORT

Figure 11. OTEC-1 Cold Water Pipe



Underwater inspections will be made periodically during the test program or, at the discretion of the Test Director, for possible damage or deterioration of the CWP and its attachments. During these inspections, the condition of the mooring system and the ship's hull will be examined.

On completion of the 1 MW_e test program, the platform will leave the site and transit to a shipyard, as yet undefined, where the 1 MW_e test articles will be removed. The PSD-I and PSD-II test articles will be installed and checked-out prior to return of the platform to the test site. This effort is scheduled to be accomplished in a three month period.

Advanced Heat Exchanger Tests

The ammonia power loop will be modified and set up to enable parallel testing of the PSD-I and PSD-II test articles. The PSD-I test articles are modular units sized to obtain data for scale-up to the 10/40 MW_e systems. The heat exchanger tubes are aluminum, the tube-sheets aluminum-clad steel, and the shells are steel. After completing heat transfer tests to verify the algorithms used in the design and establishing the countermeasures needed to maintain the R_f value at some constant level, the heat exchangers will be operated continuously for approximately two years.

The PSD-II test articles will be plate-type heat exchangers. Since these installations are still in the proposal stage, the test program and biofouling countermeasures systems are still to be developed. The test program will be designed to obtain data for evaluation of the comparative performance of these exchangers against the conventional shell-and-tube units.

PSD-I Heat Exchangers (Figure 12)

Heat transfer tests will be performed to obtain baseline data for the PSD-I test articles. Seawater flowrate, inlet and outlet pressures, inlet and outlet temperatures, ammonia vapor pressure and temperatures, and inlet and outlet delta temperatures will be recorded and the data reduced to provide an overall heat transfer coefficient U. This data will be obtained with no recirculation of vapor provided. The Amertap cleaning frequency will be based on the experience gained with the 1 MWe test articles. Distribution of the ammonia through the evaporator will be verified by measuring the vapor velocity from each of three zones. Pressure drops of seawater and ammonia across the heat exchangers will be measured. After determination that ammonia distribution is reasonably uniform, the blank will be removed from the vapor recirculation line and the system put into operation with nominal steady-state design conditions. From the data obtained the following heat exchanger calculations will be verified:

Log mean temperature difference (T_m):

$$T_m = \frac{(T_i - T^1) - (T_o - T^1)}{l_n \ (T_i - T^1)/(T_o - T^1)} \qquad (1)$$

where

T_i = seawater inlet temperature

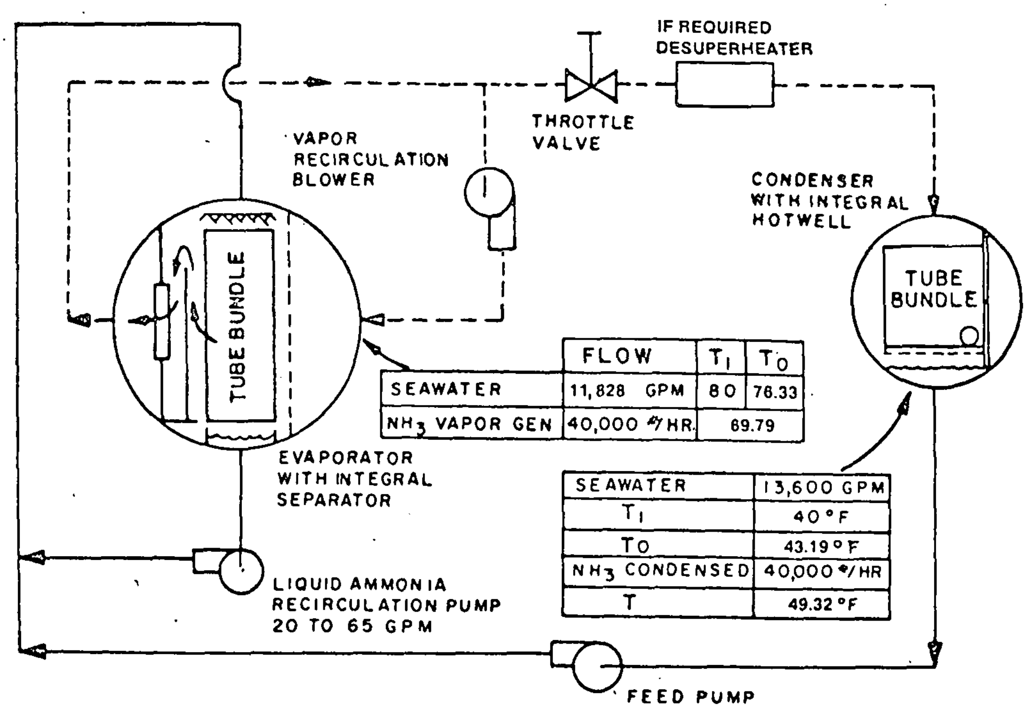
T_o = seawater outlet temperature

T^1 = ammonia vapor temperature.

Heat duty:

$$Q = m \ Cp \ (T_i - T_o) \qquad (2)$$

Figure 12. PSD-I Heat Exchanger Test Loop



where

m = seawater flow

Cp = specific heat - seawater

Overall heat transfer coefficient (U):

$$U - Q/A_o \, T_m \tag{3}$$

where A_o = outside area of tubes.

Seawater resistance (R_i):

$$\text{Nusselt number, } N_u = 0.023 \, Re^{0.8} \, P_r^{0.4} \tag{4}$$

Inside heat transfer coefficient (h_i):

$$h_i = nu \, K/D_i \tag{5}$$

where

K = conductivity

d_i = inside diameter tube, ft

$R_i = 1/h_i \, . \, A_o/A_i$

Tube metal resistance (R_m):

$$d_m = \frac{d_o - d_i}{n \, (d_o/d_i)} \tag{6}$$

Tube metal heat transfer coefficient (h_m):

$$h_m = t/12 \, K_m \, . \, d_o/d_m \tag{7}$$

where

t = tube thickness

$$K_m = 119.5 + 0.168 \, T_m \tag{8}$$

where

T_m = metal temperature

$$R_m = 1/L_m \tag{9}$$

Fouling resistance (R_f) assumed equal to zero for initial test runs.

Overall resistance:

$$R = 1/U \tag{10}$$

Outside resistance:

$$R_o = R - (R_i + R_f + R_m) \tag{11}$$

Outside heat transfer coefficient (h_o)

$$h_o = 1/R_o \tag{12}$$

The heat transfer coefficient is calculated as follows:

$$h_o = 5723 \, (DT_f)^{0.86} \text{ or } K(DT_f)^m \tag{13}$$

where

$DT_f = T_m \, (R_o/R)$ = temperature drop across film.

$$Q/A_o = 5723 \, (DT_f)^{1.86} \text{ or } K(DT_f)^{m+1} \tag{14}$$

If the two values of h_o from equations 12 and 13 are in agreement, the algorithm is validated. If they are not in agreement, additional data will be obtained by varying the seawater flowrate and, using the Wilson plot technique, better values of K and m will be determined and equations 13 and 14 revised accordingly.

Vapor velocity tests will be performed using the blower controls and throttle valve to control the flow. The tests will start with nominal steady-state design conditions and the vapor velocity will be varied into an overload region and observations made of film stripping and deflection. Flow from the separator section will be measured to determine the amount of entrainment resulting from exit velocity skimming.

Tests will be performed to determine the optimum ammonia reflux ratio. By manipulating the blower controls and throttle valve the ammonia feed to the evaporator will be increased until the level in the drain tank begins to rise. The ammonia recirculation pump will be operated until the drain tank level is constant. The ammonia feed will then be slowly increased until the evaporator vapor flow stops increasing. This test will be repeated several times to determine the ammonia reflux ratio.

After establishing nominal operating conditions, tests will be run with the seawater varied from 50% to 150% of nominal flow (11,700 gpm) at the evaporator, in 10% steps, and determining the heat exchange performance at each step. The seawater flow to the condenser will be varied in a similar manner around the nominal flowrate of 13,600 gpm and heat exchange performance calculated at each step.

The heat exchangers will then be operated at the nominal design conditions and periodic checks made on the heat exchange performance to determine that the fouling resistance is being controlled by the countermeasures system of chlorination and Amertap. The system will then be set up at nominal operating conditions and long-term testing initiated. Heat exchange performance will be monitored on a scheduled basis to enable detection of any change in performance caused by corrosion, erosion, or failure of the biofouling control system.

<u>Summary</u>

While it is generally accepted that the principle of the OTEC ammonia power plant is technically feasible, there are number of potential problems which could have significant impact on the economic viability of this source of power. Numerous individual research programs have been undertaken to answer such questions as:

1. What materials can be economically used in the construction of the plants and yet provide long term service in the hostile ocean environment.

2. What will be the nature and extent of the fouling experienced in tropical waters, particularly in the cold water system with water from the nutrient rich depths.

3. How will the cold water pipe perform over an extended period of time.

4. What will be the impact of an OTEC plant on the surrounding environment.

The answers to these and other questions are needed for development of the future OTEC programs. It is the objective of this test program to gather data, under actual OTEC operating conditions, which will provide practical answers to these questions.

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SYSTEM DESIGN CONSIDERATIONS FOR A FLOATING OTEC MODULAR EXPERIMENT PLATFORM

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Abstract

A baseline design of an OTEC modular experiment platform has been completed. This design addresses the grazing option of OTEC power utilization, whereby a self-propelled platform cruises a tropical ocean area and uses the OTEC power to synthesize an energy-intensive product (e.g., ammonia). ~~The paper describes~~ the system integration requirements and the resulting design of the hull structure, OTEC power system and cold water pipe are described. The cost for acquiring and deploying this platform outfitted with a 10 MW_e (net) OTEC power system is presented. These data are available to DOE to serve as baseline information to industry in subsequent design and construction phases of the OTEC program.

Introduction

The electrical power generated by an OTEC system can be used in several ways: (1) A deep-water moored platform can transmit baseload power via a submarine cable to a shore-based utility grid; (2) a lands-edge or shallow-water OTEC plant (with cold water inlet and discharge pipes extending out to sea) can also provide power to the local utility while eliminating mooring and cable costs; and (3) a self-propelled surface platform can cruise slowly in the tropical oceans, where ΔT is the greatest and storms are less severe, and use the OTEC power to produce a storable energy intensive product (e.g., ammonia). The latter option is favored by APL for it permits the harvesting of the OTEC resource over vast areas of tropical and subtropical waters, and in amounts that would have a significant impact on the world's energy shortage.[1] The product ammonia would be used for fertilizer or as a hydrogen carrier for fuel cells of electric utility companies.

As part of the Department of Energy (DOE) program to prepare for the acquisition of a 10 to 40-MW_e (net) modular experiment platform, APL has prepared a baseline engineering design, at the preliminary design level, of an OTEC platform that is configured to demonstrate the critical features of a self-propelled, cruising OTEC plantship.[2] This effort was funded jointly by DOE and the Maritime Administration, Department of Commerce. The principal features of the design are described below.

Baseline Design

The plantship design is dictated to a large extent by the type and arrangement of the power system components, principally the heat exchangers. In this design the APL type of folded tube, aluminum heat exchangers were selected because of (a) their predicted performance, (b) the test program being conducted to validate these predictions,[3] (c) concepts for cleaning the units *in situ* during OTEC cycle operation, and (d) their cost effectiveness. The selection of these heat exchangers, with the ammonia working fluid inside the tubes, led to a modular-partitioned barge-like structure which accommodates the gravity flow of sea water over the tubes and discharge through openings at the keel. The general arrangement is shown pictorially in Fig. 1.

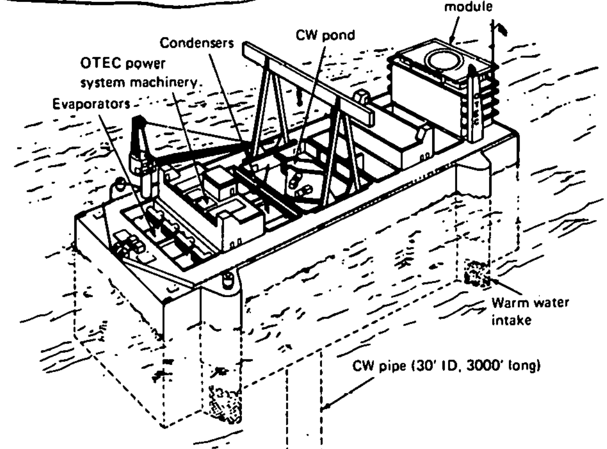

Fig. 1 OTEC cruising modular experiment platform.

The power system layout is symmetrical about the cold water pipe (CWP) which is located near midships. A 10 MW_e (net) power system is shown at the forward end of the platform. Four evaporator tube bundles are fed by 2 warm water pumps, located in sponsons near the corners, and four condenser tube bundles are fed by 2 cold water pumps located inside of the CWP. The heat exchangers are manifolded in pairs, and are connected to two 5 MW_e (net) power systems. The power system machinery, including turbo-generators, is housed between the evaporator and condenser banks. Space and weight allocations are provided at the aft end of the platform for an identical 10 MW_e (net) power system; alternatively, power systems of another design (possibly yielding up to 20 MW_e (net) at either end of the platform) could be installed.

The platform provides accommodations, equipment, repair facilities and stores to support the OTEC operation during essentially continuous at-sea testing.

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Hull Design

The platform has a length of 378 ft, a width of 121 ft (159 at WW pumps), an operating draft of 65 ft, an operating freeboard of 24 ft (plus 4 ft breakwater), and a gross weight of 67,900 long tons (including ballast).

Post-tensioned, reinforced concrete was selected as the best material for hull construction, due to its compatibility with the OTEC aluminum heat exchangers and marine environment, long-term durability with minimum inspection and maintenance, low cost, and the capability for construction in existing facilities in the U.S. The use of concrete for ship construction is not nearly as common as steel, but the material has performed well in the marine environment. The large ARCO LPG storage barge, built three years ago in Tacoma, Washington, is quite similar in size to the proposed OTEC platform. A ship of this size could be fabricated in a number of U.S. yards in the East, Gulf and West Coasts; in most cases construction would proceed in stages according to the depth of water available.

The platform has been designed to cruise in tropical waters of the Atlantic Ocean at site ATL-1 (5°-15°S, 20°-30°W), a prime site for commercial OTEC operations due to the high average ΔT and lack of hurricanes. The significant wave height in this area averages 5.1 ft, and it is estimated that wave action will force OTEC shutdown (significant wave height exceeding 18 ft) no more than one day per year.[4] The platform is ballasted with sea water to submerge the heat exchangers during operating periods; when a severe storm develops the seawater pumps are shut down and the ballast discharged to gain 10 ft of extra freeboard. In this configuration the platform can survive a 100-year return storm whose significant wave height is 29 ft, with the maximum expected wave height in 1000 encounters, H_{max} = 55 ft. (Note: Seakeeping predictions were made using the SCORES, CARGO and MIT computer codes, and utilize linear strip theory. Since the length-to-beam ratio of this platform is less than the accepted limit for the applicability of strip theory, it will be necessary to conduct model tests to validate the seakeeping predictions.)

Electric thrusters, each rated at 2500 HP are located at the corners of the platform beneath the keel to provide power to graze slowly during OTEC operation and to maintain heading during severe storms.

The platform has been designed for continuous, at-sea operation with hotel facilities for a crew of 38 plus 10 transients. A 60-ton traveling gantry crane will service pumps, motors, turbo-generators, etc. and is used to position a special framework for heat exchanger liftout and repair. A long-boom pedestal crane is used for sea-to-sea transfer and routine lifting. Ample space is provided onboard for storage, shops and repair facilities.

OTEC Power System Design

The APL concept for low-cost aluminum heat exchangers is illustrated in Fig. 2. The working fluid is in two-phase flow inside the multipass, 3-in O.D. tubes, and seawater flows over the tubes by gravity and out the bottom of the plantship. The 7 folded tubes are nested in parallel paths forming a vertical plane or "element" of the heat exchanger. Forty-one of these elements comprise the 2.5 MW_e (net) module shown. The evaporator and condenser tubing assemblies and supporting structures are identical, simplifying manufacturing and overall system design requirements. A 1.25-in horizontal clear space provided between the tube rows permit an ultrasonic or mechanical cleaner to be introduced to maintain a low waterside fouling coefficient. Heat exchanger leakage will require removal of the heat exchanger from the well to gain access to the liquid and vapor manifolds and tube ends. These manifolds and connections are sufficiently large to permit man and tool access for test and plugwelding of leaking tubes after ammonia pumpout, purge and ventilation.

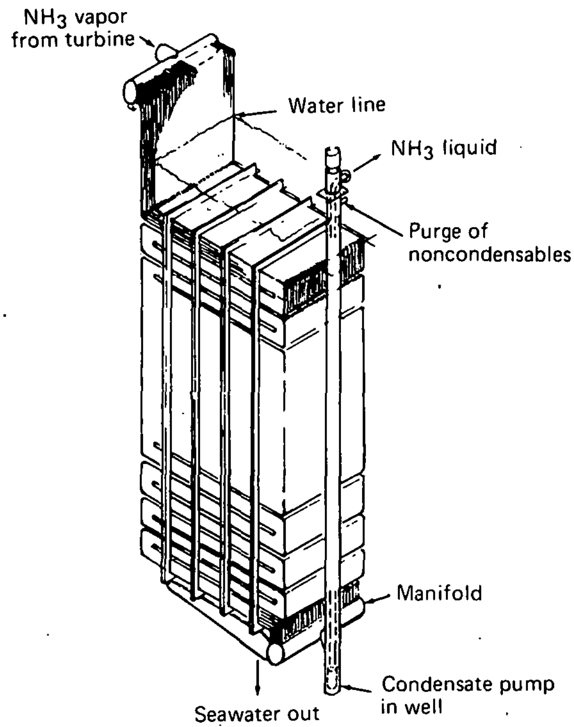

Fig. 2 APL 2.5 MW_e (net) condenser module.

In the basic power cycle warm water is pumped to head ponds and flows by gravity through the evaporators, and cold water is pumped up in the cold water pipe and flows by gravity through the condensers as indicated in Fig. 3. Liquid ammonia enters the evaporator inlet manifolds and is distributed to the multiple tubes comprising the heat exchangers. From computed heat transfer and ammonia pressures, evaporation begins in the first or second horizontal pass and with each subsequent pass the percentage vapor (quality) increases until at exit from the twenty-fifth pass it is approximately 70 percent. From the evaporator ammonia flows through a liquid separator, and dry saturated vapor is supplied to the turbine. The separated liquid ammonia is recycled to the system by a recycle pump and is mixed with the condensed ammonia in a pressure control tank.

The ammonia turbine is a synchronous machine with nozzle admission control of frequency, with the generator load controlled to match the available turbine power. The ammonia vapor is expanded in the turbine and exhausted to the condensers, and condensation occurs at relatively constant pressure and temperature. The condensate is pumped from the condenser sumps by well pumps to a pressure control tank where it mixes with the recycle liquid as noted earlier.

The OTEC electrical system has full controls and instrumentation at switchgear, switchboards and motor controllers to accomplish the normal operating and equipment protection functions required. All unit controls are connected to the central control-room console to permit one operator to monitor and control all system startup, shutdown and loading operations. OTEC startup is accomplished via the tie feeders from the platform thruster bus, with initially one cold water pump, one warm water pump, and the ammonia condensate and recycle pumps started in series. After ammonia vapor flow is established by the turbine bypass, the turbine generator can be started and when operating speed and synchronization are attained, the generator can be connected to the bus and can assume the loads via programmed computer control.

This design does not include a product demonstration plant or any useful power utilization from the OTEC generators other than platform and thruster load and the OTEC auxiliary (parasitic) loads. To attain the objective of demonstrating power generation and control capability, a water-cooled resistor-bank system is provided.

Sea Water Pumps

The seawater pumps are large low-speed, axial-flow pumps, whose overall efficiencies are extremely important. The pumps are vertically mounted with translation from vertical to horizontal flow at relatively low channel velocities. Pumps of larger capacities than these OTEC pumps are not uncommon, but the dynamic effects and oscillatory loads resulting from wave action and platform motion are unique. It has been assumed that OTEC operations could continue to some condition in sea state 6 where OTEC equipment would have to shut down, either due to excessive motions or to a degradation in the warm water resource. This is estimated to occur no more than one day per year.[4] Although the sea water pumps would not be operating under this shutdown condition, the driving loads exerted on the pumps will require vane feathering and shaft lock.

OTEC Performance

The gross power available from an OTEC plant is a strong function of the warm-to-cold water temperature difference, while the plant auxiliary or parasitic power, because of fixed equipment and sea water pumping power, is not affected proportionately. In the present design the net power available at the busbars varies approximately as $\Delta T^{2.5}$; thus in the grazing concept the plantship cruises to stay within the regions where the maximum ΔT is predicted to be available for maximum power generation. Figure 4 shows estimated power generation throughout the year for a selected site (ATL-1) where the annual OTEC plant outage from weather conditions is minimized.

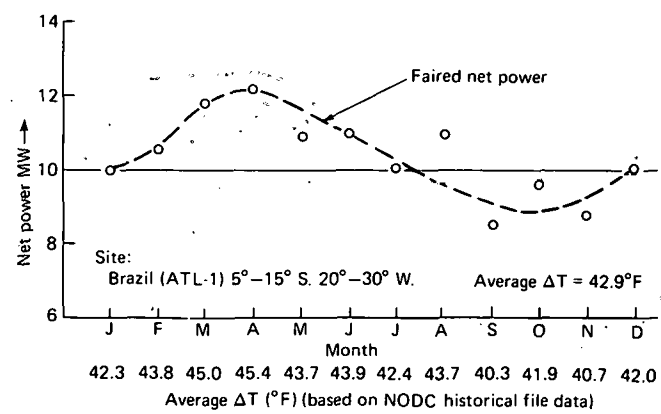

Fig. 4 **Predicted power output of cruising modular experiment platform with two-5 MW_e (net) APL power modules intalled.**

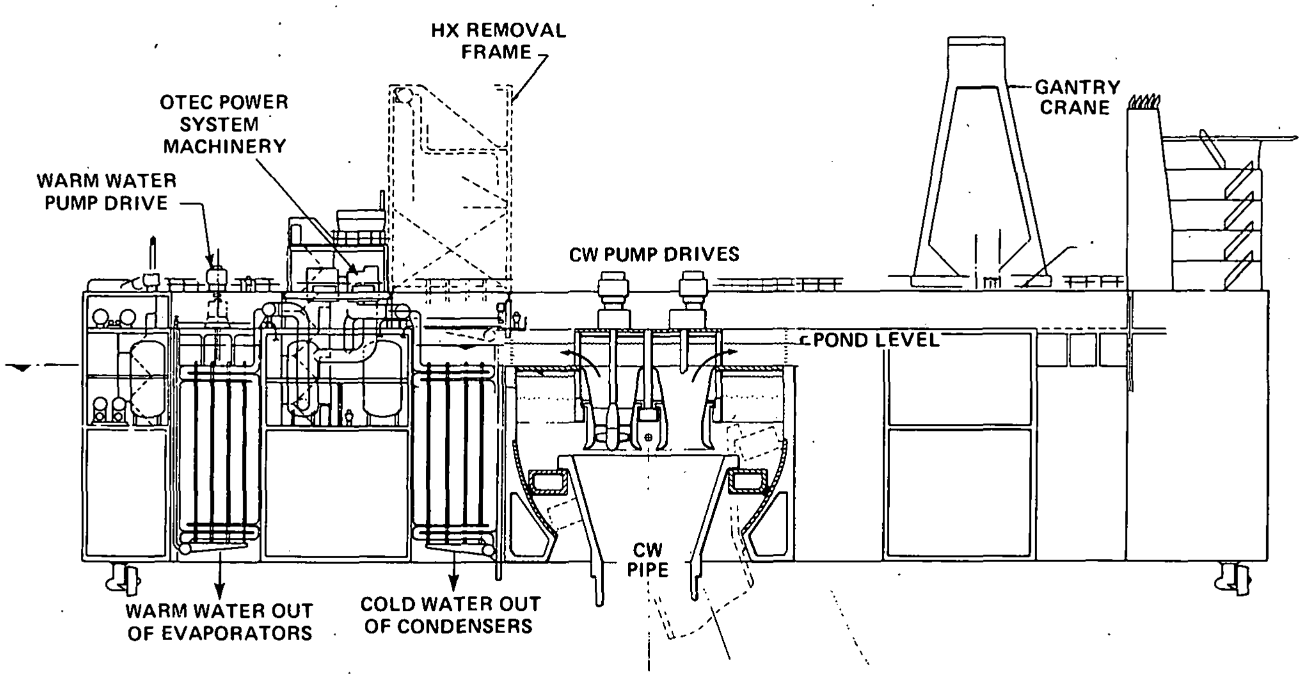

Fig. 3 **Installation of 10 MW_e APL power system in modular experiment platform.**

The Ships Service power plus the thruster power required for grazing for the present platform design is estimated at 2 MW_e. However, these requirements will not change with increased OTEC plant capacity that may be installed, and they will not increase significantly for commercial plant sizes, for which they may be equal to <1 percent of the net power. Power control and utilization are anticipated to be well within normal industrial practice and capabilities for plants producing energy intensive products. The resource is available essentially 24 hours a day continuously, and with generator loads controlled to match the power capability and control margins required, via matching the product load, the maximum utilization of the thermal resource will be made.[5]

Cold Water Pipe

A pipe diameter of 30 ft and a pipe length of 3000 ft have been selected to provide sufficient cold water for up to 40 MW_e of power demonstration. The analysis has been conducted using the special computer codes ATEC and ROTEC developed by Paulling.[6] Over 50 computer runs were made to study the behavior of the CWP, using a parametric design approach, and it was found that the elastic properties of the pipe and the introduction of hinges or compliant joints have a very significant effect on the induced bending moments, i.e., CWPs that are naturally compliant (plastic, rubber, etc.) or those that are made compliant by the introduction of hinges and joints tend to reduce material stresses.

Both the analysis and deployment studies favored the use of many, short segments of pipe, coupled with flexible joints. Such a concept suggests the use of post-tensioned concrete construction for reasons of material compatibility and resistance to the marine environment, as well as low construction costs that would result from simple, repetitive casting of segments. The diameter and length (50 ft) of the segments are similar to those being designed for the largest water irrigation and outfall projects. (It should be noted that other materials and CWP configurations are under active investigation in DOE-funded studies, and that the results of these and other CWP development programs could alter this judgement.)

A drawback to the use of conventional structural concrete, is its density, about 150 lbs/ft^3. For this reason, a specially formulated lightweight concrete weighing approximately 85 lbs/ft^3 has been developed for use in the CWP. The initial tests of this material indicate an ultimate compressive strength of over 4000 psi.[7] Additional tests and development is planned, including the fabrication and testing of a large (1/3 to 1/4 scale) CWP pipe and joint section. Using the lightweight concrete, each 50 ft segment will weigh about 135 LT in air and about 33 LT in sea water.

The segments are deployed vertically through a central well in the platform as shown in Fig. 5. A jacking rig, similar to those used by the offshore oil industry, will incrementally lower the pipe string as the segments are sequentially stacked and locked together with a rotating bayonet joint. This approach to CWP design, construction and deployment results in a structure that is intrinsically effective in shedding the hydrodynamic forces and utilizes proven concepts of concrete pipe fabrication and offshore construction technology.

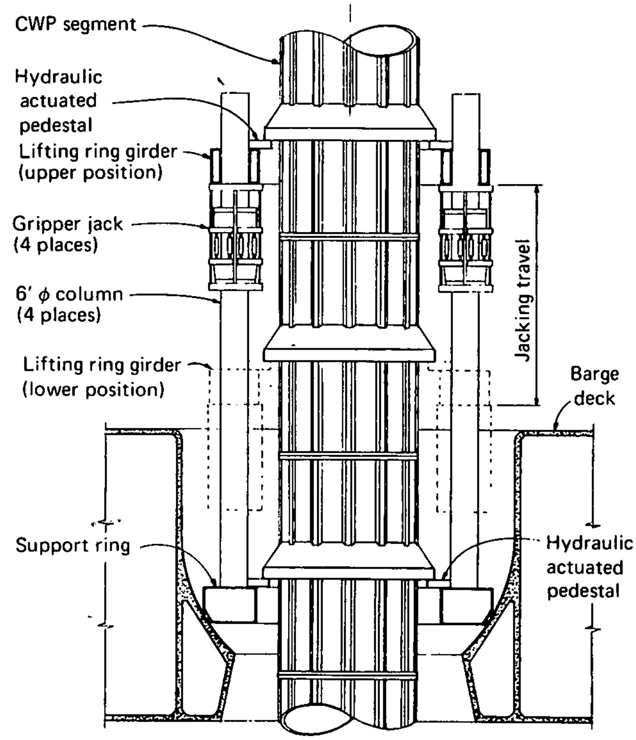

Fig. 5 CWP deployment rig.

Acquisition Cost

Acquisition cost estimates have been made in a consistent manner using DOE Work Breakdown Structure methodology. The cost of system acquisition, construction and deployment (WBS 3.0) is given in Table 1. The cost for constructing the concrete hull, the largest single item, was obtained from an independent professional estimator who was retained to conduct a step-by-step study of the construction sequence with an itemized list of all labor and material costs. Additionally, APL has been fortunate to receive informal assessments of construction and fabrication costs from firms that are not now a part of the design team. These independent assessments cover the concrete hull, the CWP (less deployment) and the fabrication of 8 heat exchanger modules, which together make up approximately half of the acquisition cost. In each case the second source estimates were equivalent or lower than costs reported in the table. Therefore, it is believed that the basic costs (not including inflation, contingency and profit) presented for fabrication, construction and deployment are realistic, and that the system acquisition cost of $71.0M (FY 80 dollars, no profit or contingency) is accurate for budget and planning purposes.

Areas of Uncertainty

Though the preliminary design has been completed in sufficient detail to serve as a baseline set of data for subsequent design and acquisition effort, there are some areas in which design details, analytics, performance, or material characteristics are uncertain at this time. These areas are discussed in the following sections, including the current and proposed efforts to eliminate the uncertainties.

Table 1 Cost Data for Acquisition, Construction and Deployment of OTEC Pilot Plantship

	Work Breakdown Structure	Cost(*)
3.1	Platform System	$33.4M
	3.1.1 Hull Structure	(15.8)
	3.1.2 Position Control	(6.7)
	3.1.3 Platform Support Systems	(1.7)
	3.1.4 Outfit, Furnishings	(5.6)
	3.1.5 Assembly Support	(0.4)
	3.1.6 Seawater Systems (10 MW)	(3.2)
3.3	Cold Water Pipe System	10.0
	3.3.1 Pipe	(7.7)
	3.3.2 Screen	(0.1)
	3.3.3 CWP/Hull Transition	(2.2)
3.5	Power System (10 MW)	16.2
	3.5.1 Evaporator System	(5.5)
	3.5.2 Condenser System	(5.3)
	3.5.3 Ammonia System	(1.0)
	3.5.4 Purge System (N_2)	(0.1)
	3.5.5 Generator System	(2.0)
	3.5.6 Auxiliary System	(1.8)
	3.5.7 Instrumentation, Control	(0.3)
	3.5.8 Biofouling, Cleaning	(0.2)
3.6	Energy Transfer System (Dissipation)	0.4
3.8	System T&E	0.3
3.9	Deployment Services	5.9
	3.9.1 Platform Deployment	(0.2)
	3.9.2 Power System Deployment	(0.1)
	3.9.3 CWP Deployment	(5.6)
3.10	Industrial Facilities	1.5
3.11	Engineering and Detail Design	3.3
	WBS 3.0 TOTAL	$71.0M

(*) FY 80 dollars; no allowance for contingencies or profit

CWP/Hull Connection

Analysis indicates that there will always be positive contact between the CWP support ring and the stainless steel socket of the hull, even during the 100-year storm at ATL-1. The concern lies with the ability to fabricate a steel socket with the required precision and to fit bearing pads that will support the pipe throughout its life time. This requires a more detailed study of tolerances and a materials evaluation effort, which are included in the FY 79 DOE program. A fall-back position is to replace this joint with a more costly gimbal or hydraulic support where the components could be serviced or replaced.

CWP Joints

The use of many joints is fundamental to the concrete CWP design. There have been questions raised as to the structural adequacy of the joint, its allowable rotation, and the service life of the bearing pads and seals. These issues need to be addressed with a more complete structural analysis, followed by large scale (1/3 to 1/4) model tests. This effort is funded by DOE for FY 79. A fall-back position is to select an alternate CWP design from the DOE-funded CWP studies underway.

Lightweight Concrete Material

Special lightweight concrete mixes, with densities of 75 to 85 lbs/ft^3, have exhibited compressive strengths in excess of 4000 psi. A more complete series of tests, including long-term moisture absorption, creep and fatigue, are required to verify the suitability of this concrete for use in the CWP. This effort is funded by DOE for FY 79. A fall-back position is the use of an alternate CWP, discussed above, or the use of regular density concrete with closer spaced joints or larger handling equipment.

Seawater Management

The dynamics of the flow of warm and cold seawater through the OTEC system, including the interactions of the platform motions, passage of surface waves and pump speed variations, have not been adequately addressed. A better understanding of this flow is essential in order to minimize pumping power, properly design and control the pumps, minimize pressure surges in the heat exchangers and upper CWP, and avoid reingestion of used water in the warm water intakes. DOE program plans for FY 79 include an RFP and contract award to investigate the seawater systems. The results of the investigations will be applied to the modular experiment design, and design modifications will be made as required.

Heat Exchanger Cleaning

Small-scale cleaning experiments with ultrasonic panels have been encouraging. More work is needed, however, to translate these test results and the concept design study of a cleaning system into a workable system that is capable of cleaning the pilot plantship heat exchangers on a regular basis. A test and development program is required to validate this approach. Funds are not currently earmarked for this activity. A fall-back position is a return to mechanical scrubbers, but much work needs to be done here, also.

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CONCEPTUAL DESIGNS AND COSTS OF OTEC 10/40 MW SPAR PLATFORMS

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ABSTRACT

~~This paper summarizes~~ studies of conceptual designs of 10 and 40 MW Modular Application Platforms (MAP) of the spar configuration are summarized. The feasibility studies, which preceded these designs, evaluated trends in spar platform size and configuration as a function of alternative power system components and materials. These studies, summarized in this document, led to selection of a steel spar supporting external vertical heat exchangers as the lowest cost configuration. The conceptual designs of the 10 and 40 MW spar concepts selected during the early feasibility studies are described, including platform arrangements and modularity characteristics. The efforts required to integrate the dynamic response of the platform, cold water pipe and mooring system are described in detail. Concepts for construction and deployment are described, and estimates are presented for total system cost and schedule.

INTRODUCTION

This Summary describes studies conducted by Gibbs & Cox, Inc. and its subcontractors for the U.S. Department of Energy, leading to conceptual designs of 10 and 40 MW OTEC Modular Application Platforms (MAP) of the spar configuration. The spar is one of several options being considered by DOE to follow OTEC-1, and to demonstrate the feasibility of generating electrical power using OTEC power components in the 10 and 40 MW size. Other options being considered are moored and grazing surface platforms and land-based plants. Design data on these three options will be made available as GFE with the Request for Proposal for OTEC 10/40 to be issued in FY 1980. Further studies of the spar are planned to expand the current Conceptual Design in areas of technical risk.

The current studies were conducted in two phases. Initial feasibility studies were performed to trade off variables having a major impact on platform size and configuration such as heat exchanger size and location, and hull materials. The results of these trade-offs led to Feasibility Study Baselines for both 10 and 40 MWe sizes. These baselines were then expanded and further defined in the Conceptual Design phase, along with tudies of construction, deployment, ost and schedule. The Conceptual esign Baseline resulting from this ffort will serve as the basis for follow-on studies.

DESIGN REQUIREMENTS

The Top Level Requirements established by DOE for these studies included the following key elements:

1. Mission - to demonstrate the technical and economic viability of the OTEC concept, and form a baseline for scaling up to commercial size.
2. System life of 30 years.
3. Ability to install 10 MW power system components while at sea.
4. Platform designed to commercial standards, and to suit ABS, USCG.
5. Power system as directed by DOE, based on PSD I studies.
6. Design for the Puerto Rico site.
7. Survive the 100 year storm, both with and without the CWP.
8. Construction in U.S. facilities (existing or modified).

FEASIBILITY STUDIES

The Feasibility Studies were conducted in two phases. The first considered the following variables:

- 2 Plant sizes (10, 40 MWe) assuming steel hull structure.
- Internal vs external heat exchangers.
- Horizontal vs vertical heat exchangers.
- Heat exchanger relative location within the platform.

Phase 2 considered the trade-off between steel and reinforced concrete structures.

Phase 1 Feasibility Studies

A family of ten point designs (5 each 10 and 40 MWe) were investigated during Phase 1. The heat exchangers had the following characteristics:

Internal, Horizontal	Length X Diam, Ft.	Operating Weight, Lbs.
Condenser	58 X 37	2,012,000
Evaporator	36 X 37	1,401,000

External, Vertical	Length X Diam, Ft.	Operating Weight, Lbs.
Condenser	30 X 15	924,000
Evaporator	30 X 15	972,000

The point designs were sized based on internal volume and buoyancy requirements, and weights and costs were estimated based on data extrapolated from earlier platform studies. The platform incorporating the larger internal, horizontal heat exchangers were found to displace about twice as much as those with vertical external heat exchangers. Much of this weight was the ballast required to submerge the large, relatively light density spar bodies to their operating drafts. Total platform costs for these designs were in the following ranges:

	10 MWe	40 MWe
Vertical H/X Options:	\$32-39M	\$57-61M
Horizontal H/X Options:	\$43-54M	\$94M

This led to selection of the options with external vertical heat exchangers as being optimum for the spar configuration, based on

- Lowest cost and lightest displacement.
- Better access to heat exchangers for modular changeout.
- More efficient seawater and ammonia system configurations.

Phase 2 Feasibility Studies

Designs of 10 and 40 MWe spars with external, vertical heat exchangers were developed using both steel and 160 pound per cubic foot reinforced concrete. The concrete hulls were larger, to provide the necessary buoyancy to offset their added weight. The weight of concrete hull structure was between 3 and 4 times that of the steel spars. Studies of construction and deployment indicated that such operations were feasible for either option, though the concrete options required more deepwater work due to their drafts.

Cost studies were based on unit steel costs for primary structure of \$2000 per ton, with corresponding concrete costs of \$1100 per cubic yard (\$573 per ton) including facility rental. These costs resulted in very similar structural costs for both 10 and 40 MWe spars. However, the greater size and weight of the concrete spars resulted in a significant increase in mooring system cost, resulting in total OTEC System costs which marginally favored steel:

Total Acquisition Cost	10 MWe	40 MWe
Steel	\$76.1M	\$155.8M
Concrete	\$77.6M	\$163.1M

Based on these studies, steel was selected for the following reasons:

- Slight acquisition cost advantage (less than 5 percent).
- Similar operational costs.
- Enhanced modularity (easier to modify structure, interface equipment).
- Less risk.

CONCEPTUAL DESIGN 40 MW SPAR

Platform Arrangemenmts

The arrangements and principal dimensions of the 40 MW spar MAP are shown in Figure 1. The central cylindrical hull is the primary source of static and reserve buoyancy for the spar, and contains the 4 turbo-generator sets, power conditioning equipment and portions of the platform auxiliary equipment. This hull is surrounded by 8 power modules, 4 each for evaporators and condensers, and each containing the required seawater pumps, ammonia sumps and pumps, and chlorination equipment. The modules are designed as mini-spars which can be towed to the site, upended, lowered into position and mated with the spar in a controlled ballasting sequence. This modular installation capability permits the modules to be added or removed throughout the life of the system, either to facilitate tube removal or major overhauls at shore facilities, or to install and test new power system concepts. Modularity is further enhanced by providing an airlock at the module-hull interface so that critical ammonia piping, electrical and instrumentation connections can be made in a shirtsleeve environment.

The surface-piercing column serves both to support the deckhouse and as a trunk for access, equipment removal and support service runs. The column has both an inner and outer ring to localize flooding in the event of minor collison.

The deckhouse contains all habitability and control functions, as well as the generators, shops, and helicopter facilities. Hatches are provided on centerline for removal of equipment via a 50 ton crane.

Structure

The structure of the spar hull is of high strength steel, ASTM A-572 Gr 50,

Figure 1

GENERAL ARRANGEMENTS - 40 MWe SPAR

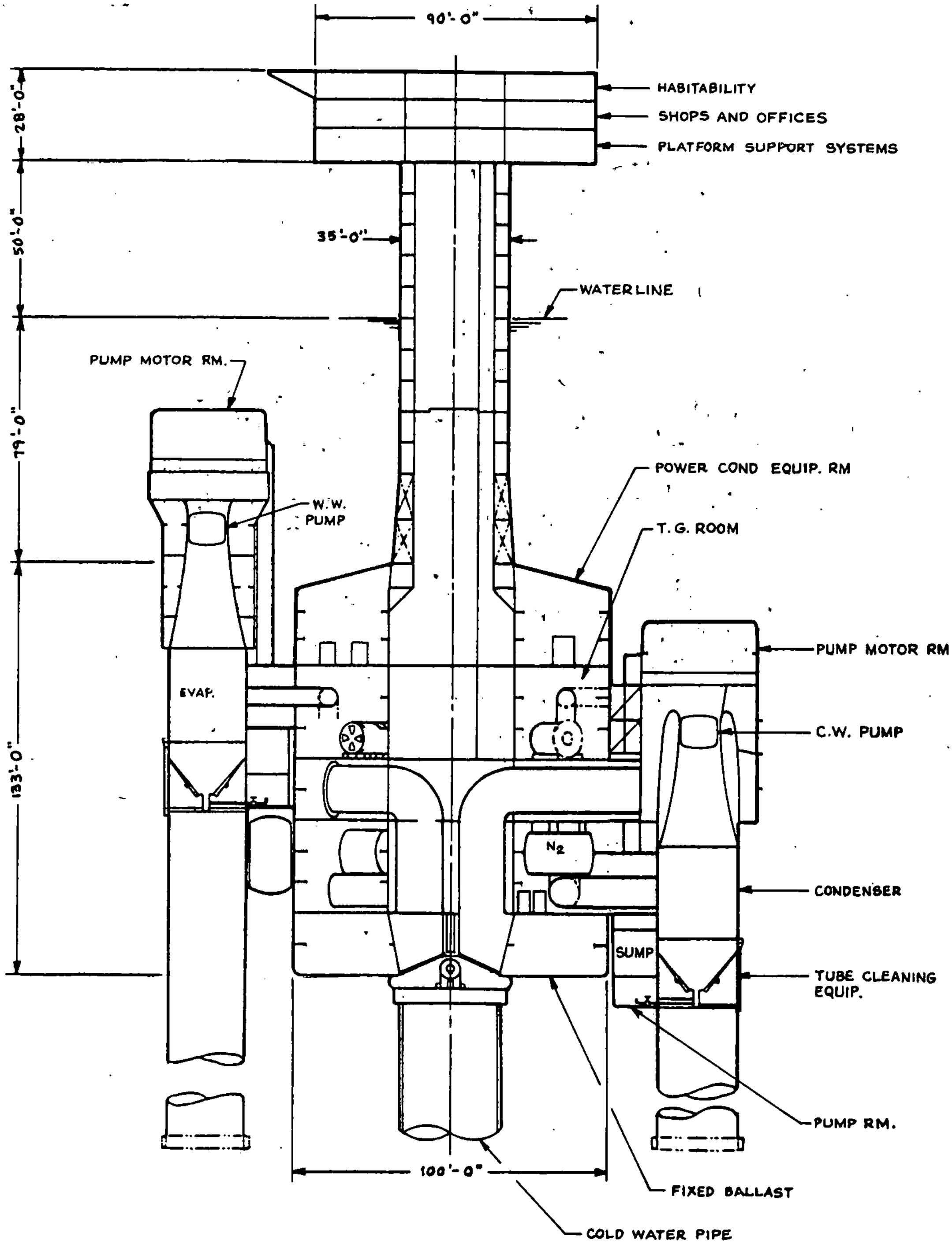
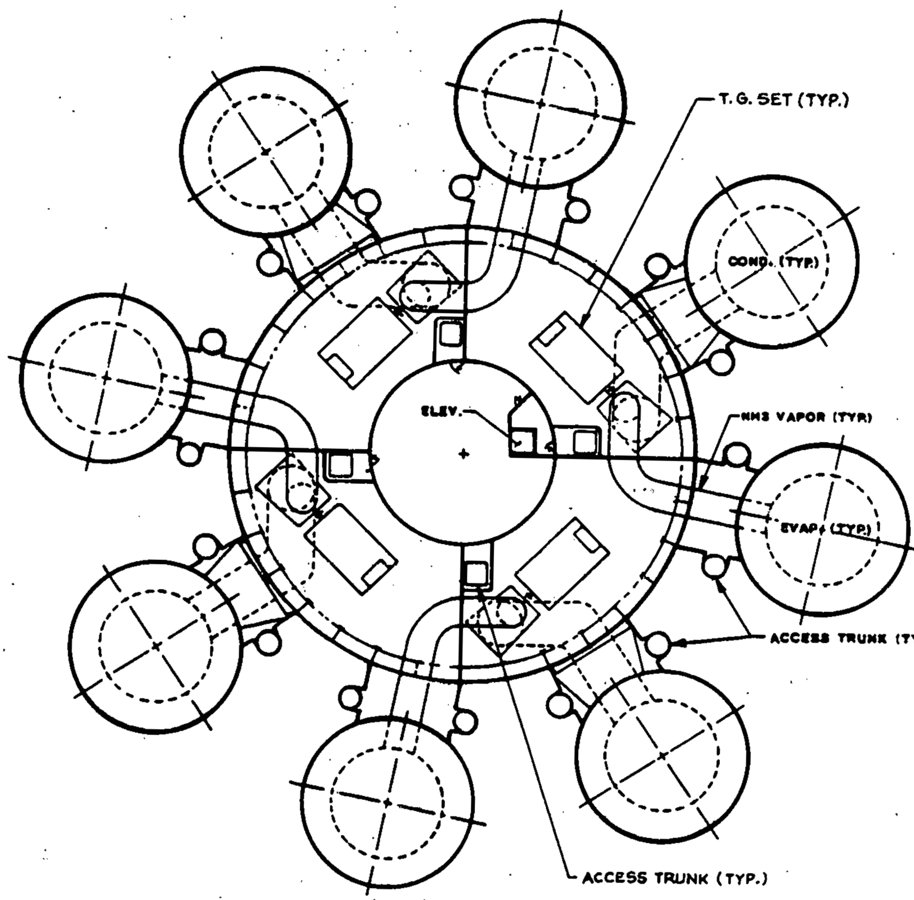


Figure 1 (Continued)

GENERAL ARRANGEMENTS - 40 MWe SPAR



3rd DECK (TYPICAL)

and consists of a grillage of vertical stiffeners supporting plate panels which in turn are supported by deep vertical webs, ring frames, quartering bulkheads and decks. High strength steel was selected both to reduce weight and cost for the highly-loaded structure. The column structure is carried down into the spar hull and through the deckhouse for strength and rigidity, and is stiffened by vertical frames supported by girth rings. Primary support for the deckhouse is provided by deep bulkheads tying into the column. All structure below the waterline and in the splash zone incorporates adequate corrosion allowances for a 30 year life in conjunction with active and passive corrosion control systems.

Mooring System

A number of concepts were evaluated for the mooring system including:

- Multi-point conventional catenary moor
- Tension leg moor attached to the platform
- Tension leg moor attached to the CWP

Preliminary studies indicate that the latter concept is less expensive, easier to deploy and contributes to reduced platform and CWP motions. This concept also offers the possibility of running the riser cable up the combined CWP/ mooring systems, and is discussed in greater detail later.

Weights and Stability

Table 1 summarizes the weights for both the 10 and 40 MWe spars, which indicates that the 40 MWe spar requires 2-1/2 times the weight for 4 times the power output. The spar is capable of surviving rupture of the column inner and outer bulkheads or any hull compartment, even if the CWP and mooring system are lost. Since the CWP/mooring system and the hull attachment are designed to survive the 100 year storm, the likelihood of this happening is very remote.

Preliminary platform motions for the spar platform were calculated using the Paulling computer simulation with a variety of mooring system pretensions, based upon the 100 year storm. For the tension leg hinged CWP concept described below, the significant (highest 1/3) motions were limited to 5 degrees of angular motion and 10 feet of heave. The accelerations corresponding to these motions are less than 0.02 "g", indicating that the tension leg moored spar is essentially free of motions even in the 100 year event. The steady-state angle of heel corresponding to the design watch circle of 10 percent in the 100 year storm is about 5 degrees.

Table 1

WEIGHT SUMMARY FOR 10/40 MWe MAP SPARS

DESCRIPTION	10MWe	40 MWe
Platform System	8057	22691
Cold Water Pipe System*	(4272)	(8435)
Power System	1241	4727
Energy Transfer System	187	352
Margin	619	1661
Light Ship Displacement	10104	29431
Load Items	562	1244
Fixed Ballast	4000	11500
Saltwater Ballast	894	4925
Mooring Tension	4800	7200
Total Operating Displacement	20360**	54300**

* Cold water neutrally buoyant-not included in light ship
**Excludes water in CWP and discharge tubes

Power System

The 40 MWe power system consists of 4 each condensers, evaporators, turbogenerators and associated sumps, pumps and controls, based on the TRW PSD I design. As noted earlier, the compact vertical shell-and-tube heat exchangers are incorporated in the external power modules, above the ammonia sumps and feed or recirculating pumps. The turbogenerators are located within the hull, arranged radially around the column, to minimize ammonia vapor runs. Power support subsystems include:

- Ammonia system capable of storing the entire on-board inventory.
- Nitrogen system to purge the gaseous ammonia loop.
- Mechanical tube cleaning system (Amertap or equal), with the retrieval mechanism located in a structural module attached to the bottom of the heat exchangers.
- Hypochlorite system for biofouling control using chlorine.

The modular approach permits tube plugging on site as well as removal for tube replacement at shore facilities.

Seawater System

The warm water modules incorporate a straight-through flow, starting at the inlet screen located about 40 feet below the waterline, through the pump, diffuser, heat exchanger, tube cleaning module and finally into discharge tubes. The discharge tubes consist of reinforced rubber panels wrapped around a 25 foot diameter "cage" of wire ropes suspended between the tube cleaning module and a ring-shaped concrete weight located 330 feet below the surface.

The cold water module draws water from the cold water plenum via four large pipes, and directs that water into an annular ring surrounding the pump and diffuser. The water is then drawn up, makes a 180 degree turn and goes down through the pump and other components similar to the warm water side. This more circuitous route is dictated by the physical constraints of the arrangement, particularly the desire to localize the piping intefaces in way of the airlock.

The head losses through these modules are 9.2 feet for the warm water side and 10.3 feet for the cold water side. Of this, 95 and 80 percent of the losses respectively are due to the heat exchangers.

The 10 MWe seawater pumps are of the low speed axial flow propeller type, direct driven by AC synchronous motors. Principal characteristics are as follows:

	Warm Water	Cold Water
Pump capacity, GPM	535,000	492,000
Pump speed, RPM	77	84
Motor HP	2100	2050
Total cost, $M	1.23	1.10

Alternates such as bulb type and high speed or gear driven pumps were considered, but the ease of maintenance and high reliability of the low speed, direct driven pumps were overriding considerations.

Auxiliary Systems

An electrical load analysis indicates a total ship service load of 706 KW for the platform. Two 700 KW ship service/ emergency generators are provided for use during deployment and other times when the OTEC plant is down. Total electrical load, including seawater pumps and OTEC system auxiliaries, is 15,126 KW corresponding to a parasitic load of 38 percent. Space and weight reservations are provided for a 5000 KW start-up generator, though shoreside power is proposed as the primary means of plant startup.

The heating, ventilation, air conditioning, fluid and gas systems are typical of offshore practice, with zero discharge of pollutants as a requirement.

Energy Transfer System

The electrical system for conditioning the 13.8 KV power generated by the plant to 138 KV and transmitting the power to shore consists of proven, off-the-shelf components. Four cables (3 active, one spare) are envisioned for transmission, though designs have not been developed as part of this study. Provision has been made in the design for a cable oil system, assuming an oil-filled design.

COLD WATER PIPE - 40 MWe

A number of options were considered for the 40 MWe CWP, based on a nominal 30 foot diameter and 3000 foot depth of intake. Although rigid pipes appear marginally feasible for a spar, the high stress induced by platform motions and vortex shedding strongly suggests a hinged CWP as optimum. A major concern is the feasibility of fabricating and deploying a rigid CWP, either from the platform itself or from a special construction platform.

Calculations indicate the likelihood of critical resonances at intermediate lengths during the deployment process, which makes the on-site construction of a rigid CWP from modules unacceptably risky. Likewise, the risks inherent in a float-and-flip operation of a 30 x 3000 foot pipe are very high.

Based upon these considerations, a concept has been developed incorporating a hinged, neutrally buoyant CWP constructed of steel, with segments nominally 460 feet long. Each consists of an inner tube 9-1/2 feet in diameter serving both as a buoyancy chamber and a tension-carrying member, and an outer shell 31-1/2 feet in diameter forming the CWP itself.

Figure 2 illustrates the concept. The outer shell could just as easily be constructed of fiberglass, rubber or other lightweight materials, which will be investigated at a later date. Cast steel, oil-enclosed universal joints are used as hinges.

The CWP is connected to a series of 200 foot hollow steel mooring links which extend down to a mud-filled cylindrical steel dead-weight anchor. This system is pretensioned to 7200 tons to prevent slackening as a result of platform motions.

10 MWe SPAR CONCEPTUAL DESIGN

Platform Arrangements

The arrangements and principal dimensions of the 10 MWe are indicated in Figure 3, and are generally similar to the 40 MWe spar concept except that only 2 power modules are required, and the reduced number of major internal components permits a reduction in the number of internal decks from 5 to 3. The total hull volume of the 10 MWe spar is about half that of the 40 MWe spar, with a corresponding reduction in displacement as shown in Table 1.

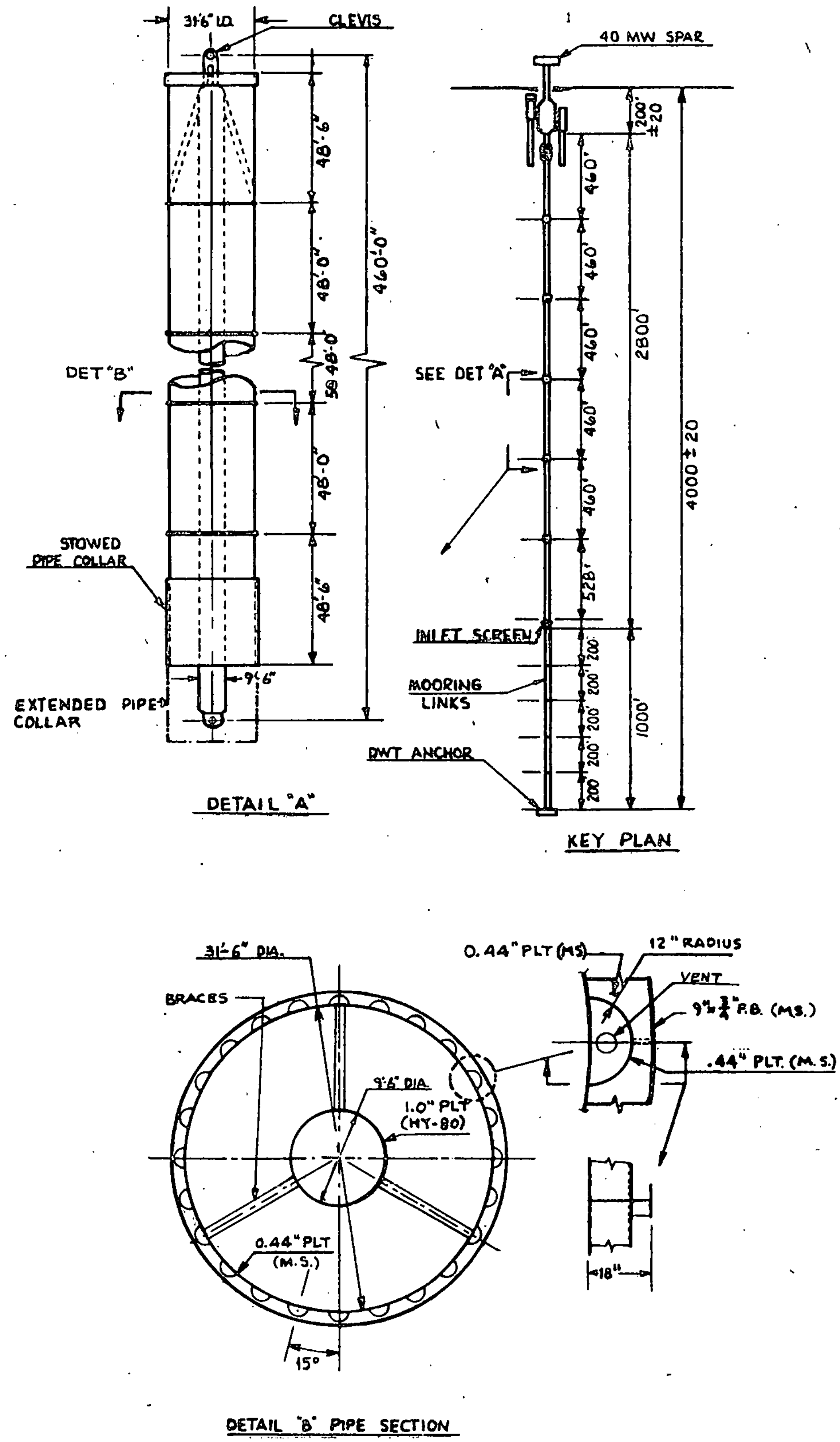


FIGURE 2
HINGED BUOYANT CWP

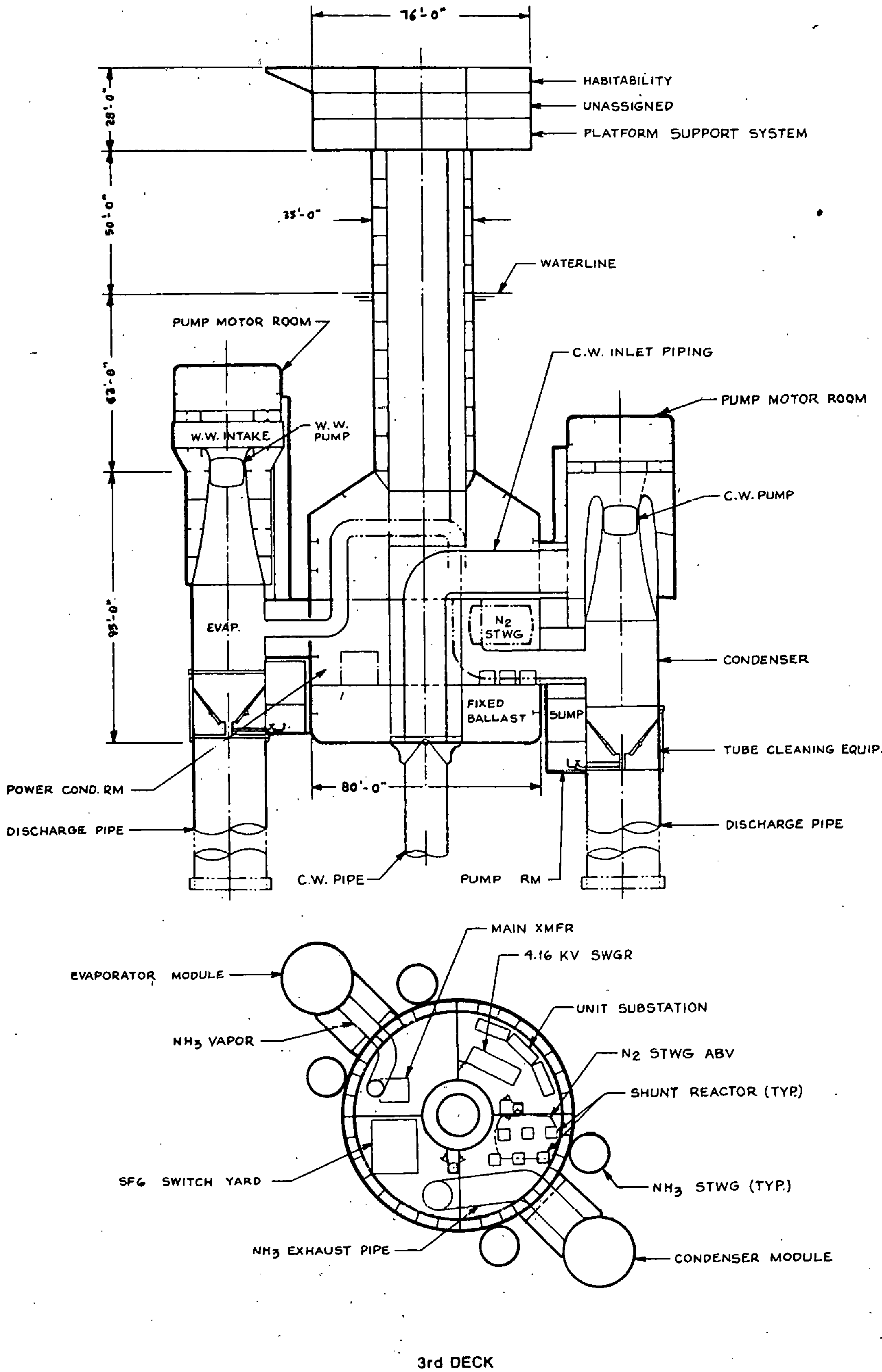


Figure 3
GENERAL ARRANGEMENTS - 10 MWe SPAR

The 10 MWe CWP is identical in concept to the hinged steel buoyant 40 MWe CWP, though the outer shell diameter is reduced to 16.75 feet, and the pretension is about 30 percent less than that for the 40 MWe system.

The Power and Seawater Systems for the 10 MWe spar are essentially one-quarter of those described for the 40 MWe spar, with cold and warm water modules located on opposite sides of the spar body. Due to slight differences in drag of these two modules, special consideration will have to be given to the torsional equilibrium of the 10 MWe spar in a wave and current environment.

The Energy Transfer System for the 10 MWe spar includes components similar to those on the 40 MWe plant, though reduced in capacity. The total weight of on-board equipment is about 240,000 pounds versus 650,000 pounds for the 40 MWe spar.

CONSTRUCTION AND DEPLOYMENT

Extensive investigations were conducted to evaluate the various methods of constructing and deploying the spar, CWP/mooring and power modules. These studies indicated that procedures can be developed which are within the state-of-the-art and at an acceptable level of risk. In the development of these concepts, primary consideration was given to minimizing offshore work and the inherent risks associated with such operations. Reduction of schedule for critical operations was of paramount importance.

Spar

The spar itself can be most easily constructed in a horizontal configuration, either in a graving dock or a typical Gulf Coast ground-level launchway as currently used for the construction of large jacket structures. In the former case, auxiliary buoyancy is required in most existing facilities to limit draft over the sill. After launch, the spar, minus deckhouse, would be towed to the site and upended or flipped by controlled ballasting, after which the deckhouse can be installed in a conventional manner. If a ground level facility is used, the spar would be slid onto a launch barge for transportation to the site and launching. The higher tow speed of this method offsets the rental cost of the barge, so that total cost is nearly identical to the graving dock method. In either case, the power modules would be installed after upending of the spar.

CWP/Mooring System

Figure 4 illustrates the proposed oncept, which involves towing the pre-assembled CWP/mooring link system on the surface to the site. After lowering the anchor and mooring links, each CWP section would be sequentially upended in a controlled operation, using auxiliary buoyancy as required. Due to the relatively large weight of the anchor and mooring links a large semi-submersible or equal is required to maintain positive control of vertical and horizontal position during this operation. After the final CWP link is upended, the assembly would be lowered into final position, filled with mud and ballasted to a slight positive buoyancy. The spar can then be moved into position and ballasted down to mate with the top section of the CWP. The spar can then be deballasted to develop the required pretension in the system, and a flexible seal can be installed at the interface.

Alternative concepts are to be investigated during subsequent studies to reduce the requirements for the dynamically positioned platform. These will include additional buoyancy or reduced weight of the system (e.g. fiberglass CWP outer pipe, buoyant mooring system links) and fewer links to reduce the number of upending operations.

Power Modules

The power modules will be towed, either afloat or on a barge, to the site for attachment to the spar. Each will be submerged by ballasting in a horizontal configuration, and linked up to a portable track guide system on the body of the spar. The module can then be rotated to a vertical orientation and lowered on the guide system until it aligns with the airlock foundation/ seal mechanism. Finally, hydraulic links will mate with the module, and draw it tight against the seats and gaskets. After latching, the airlock can be evacuated, creating a shirtsleeve environment for the make-up of ammonia, electrical and other connections. Figure 5 illustrates this operation

Schedule

The estimated schedule for constructing and deploying the 40 MWe spar is about 19 months as shown in Figure 6. The corresponding schedule for the 10 MWe spar is about 4 months shorter.

COST ESTIMATES

The assumptions used in estimating costs for management, preliminary design, and construction/deployment (including detail design) of the 10 and 40 MWe spars are summarized as follows:

GENERAL

- All costs FY 1980
- No cost margins

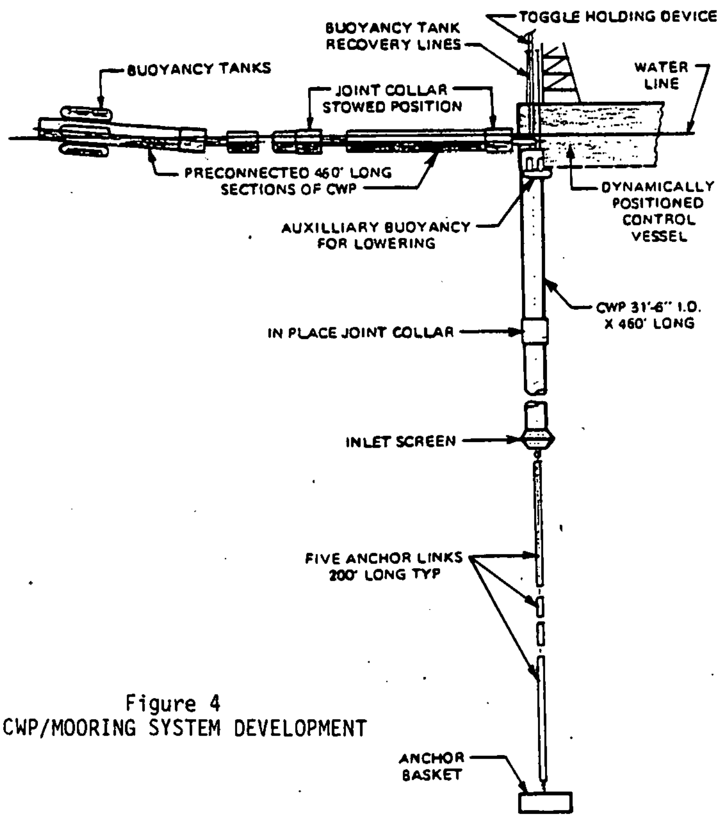


Figure 4
CWP/MOORING SYSTEM DEVELOPMENT

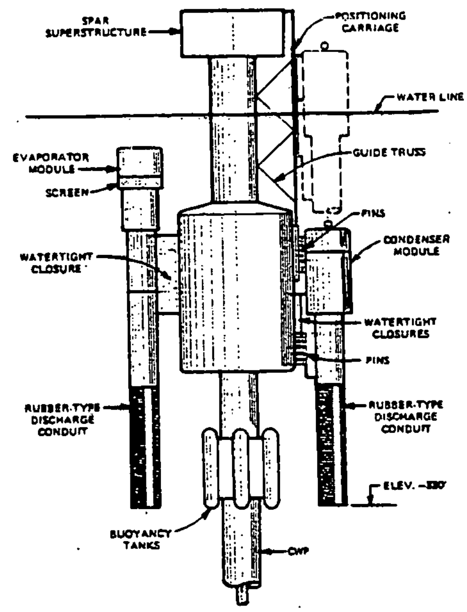
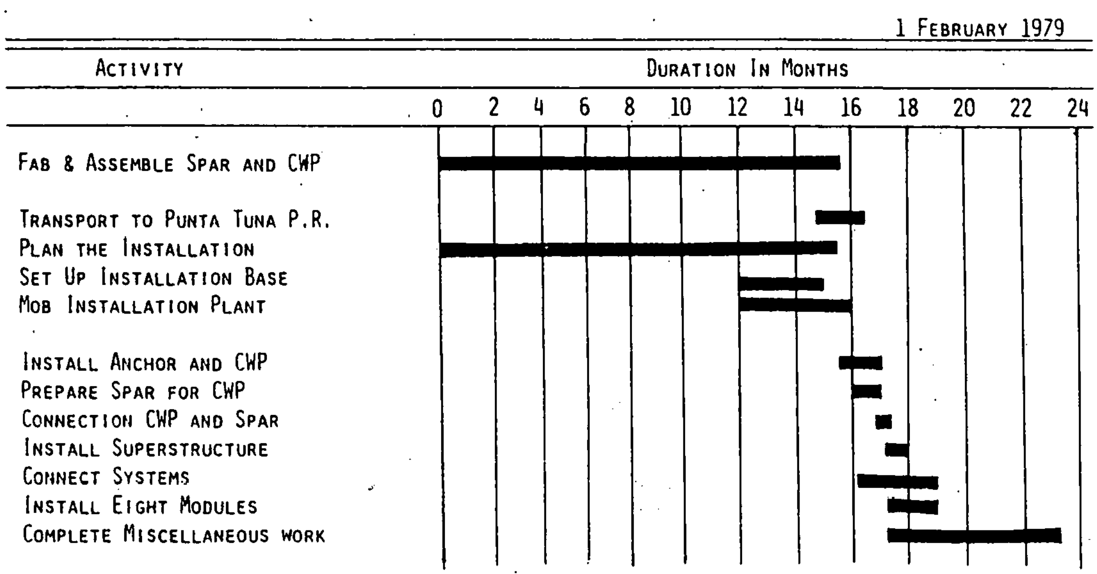


Figure 5
POWER MODULE INSTALLATION

Figure 6
40 MW STEEL SPAR
CONSTRUCTION AND DEPLOYMENT SCHEDULE



N.B. 1. No testing is included
2. No cable installation is included

- Procurement of single system to commercial standards
- Refit/modernization, major downstream overhauls not considered

GENERAL MANAGEMENT

- Includes only System Integration Contractor Program Management
- Staff = 8 during design/construction, 5 during operation

DESIGN

- Preliminary/contract design includes 3 competitive efforts @ $1.5M each.
- Detail design included in Construction/Deployment below

CONSTRUCTION/DEPLOYMENT

- Structure (HTS) = $2,416 per ton average
- Position control/CWP system = $3,580 per ton average
- Platform support services = $26,308 per ton average
- Outfit, Furnishings = $20,844 per ton average
- Major equipment costs based on vendor data
- Power system @ $22 M/10 MW module based on DOE input (plus installation)
- Energy transfer system @ $8.0 M (10 MW), $16.0 M (40 MW) based on DOE input (plus installation)
- Industrial Facilities includes allowance for special flotation; no yard improvements
- Deployment costs reflect the scenario described in Section 7

The average $/ton above result from dividing the total cost of a detailed, bottom-up cost estimate by total weight.

Table 2 summarizes the cost estimates based on the above criteria, which indicates a total cost through deployment of $109M for the 10 MWe spar ($10900/KW) and $244M for the 40 MWe spar ($6100/KW). Cost exclusive of the Power and Energy Transfer Systems are $79M and $140M ($7900/KW and $3500/KW) for 10 and 40 MWe platforms respectively. This indicates that the 40 MWe plant is more than twice as cost-effective as the 10 MWe plant, which verifies the anticipated economy of scale with OTEC plants.

The level of confidence in these cost estimates is difficult to assess, due to the varying level of design development for the various spar subsystems. Assuming a maximum range of uncertainty of + 50% and - 20% for the CWP and deployment cost elements and lesser ranges for other elements, a net raqnge of + 33% to - 12% can be postulated for the total costs in Table 2.

TABLE 2
CONSTRUCTION/DEPLOYMENT COSTS ($M)

	10 MWe	40 MWe
1.0 General Management	2	2
2.0 Preliminary Design	5	5
3.0 Construction Deployment		
• Platform System	31	71
• CWP Sys.	12	22
• Power System	22	88
• Energy Transfer	8	16
• Testing	2	3
• Deployment	22	30
• Detail Design	6	7
TOTAL 3.0	102	237
TOTAL 1.0-3.0	109	244

CONCLUSIONS

Platform

1. A spar is a technically viable platform for the OTEC 10/40 MW Modular Applications Platform mission. Although it presents a number of problems which must be addressed, no fatal flaws are identified at this time.

2. It is feasible to incorporate rapid refit concepts into a spar for modular changeout of heat exchangers and related Power System components.

3. The 40 MWe spar is nearly 2-1/2 times more efficient volumetrically than the 10 MWe spar. Both platforms require extensive ballast for stability and draft control.

4. The spar configuration is best suited to the use of vertical heat exchangers.

5. Steel is marginally superior to concrete for this application, since it permits the use of a smaller, lighter hull, with slightly lower total cost.

6. The stability of the spars can be maintained both with and without the CWP and mooring system intact. However, the spars are designed to survive the 100 year storm with the CWP and moorings attached.

7. Spar motions are minimal. With the proposed tension leg moor, vertical and angular motions are essentially non-existant, while horizontal motions and related accelerations are very small.

8. It is anticipated that a rather extensive active cathodic protection system will be required to protect the steel hull, CWP and mooring systems.

9. The optimum position control system appears to be a tension leg moor using a deadweight anchor and hinged rigid links attached to the CWP.

10. Platform support systems and outfit and furnishings can be based on state-of-the-art technology.

Cold Water Pipe

1. The cold water pipe remains the highest risk element of the OTEC Ocean Energy program, though current and near-term technology is considered adequate to the challenge of designing a 10 or 40 MWe pipe with acceptable risk by 1984.

2. The stresses introduced by vortex shedding create a significant stress profile which can have a severe effect on fatigue life, though suppression techniques are currently available.

3. A hinged or multi-section pipe offers significant advantages not only in reducing pipe bending stresses, but the weather window required for deployment. The latter is the primary factor leading to selection of a CWP with the following characteristics (subject to further study):

- Hinged at the platform and at approximately 460 foot intervals.
- Hinges designed to allow 90 degreee rotation of each segment as required for deployment in a controlled manner.
- CWP segments are of steel construction, with an inner tube serving both as a tension member and buoyancy chamber to make each link neutrally buoyant.
- Outer shell of steel to serve as a cold water duct. Fiberglass, rubber or concrete could also be used for this shell since it is structurally independent of the inner tube.
- Hinged to the upper end of the mooring system, to provide a continuous tensile load path from the deadweight anchor to the spar itself.
- Universal joints at each segment interface, immersed in an oil bath for long bearing life.

4. Further study is required to optimize such variables as segment length, outer shell materials, joint design and inter-segment sealing.

Power System

1. The external vertical heat exchangers developed by TRW for the Power System Development I (PSD I) program are considered optimum for this specific application relative to other PSD I candidates.

2. Modular removal and installation of the Power System can be achieved by incorporating the heat exchangers and related equipment into 10 MWe Power Modules which can be floated out, upended and submerged to align with matching structure on the spar.

3. The spar as presently configured can accommodate both mechanical tube cleaners (Amertap or equal) and chlorine injection.

4. All major Power System components can be removed either with the power modules or via the column for service ashore.

Energy Transfer System

1. The on-board components of the Energy Transfer System can be procured off-the-shelf.

2. The spar with integral CWP/mooring system in a tension leg configuration offers an ideal solution to the riser cable problem, by providing an essentially rigid connection between the seafloor and a platform with minimum motions.

Deployment Services

1. A steel spar of either 10 or 40 MWe size can be built in a number of existing U.S. facilities with no required facility improvement.

2. Both launch barges or float-out with auxiliary flotation are possible for the hull, and appear to be about equal in cost.

3. CWP deployment is simplified, and the risks are greatly reduced, by the use of the hinged concept proposed. Offshore work is greatly reduced, both in terms of manhours of effort and overall schedule. This in turn reduces the risk of running into bad weather during the deployment operation.

RECOMMENDATIONS

In developing recommendations for further studies, the following risk areas have been identified:

1. Fatal Flaws
   o None are identified

2. <u>High Risk Areas</u>

- User acceptance of spar concept
- Cold water pipe dynamic response, fatigue life
- Platform safety (flooding, fire, toxic gases)
- Riser cable and interface

3. <u>Medium Risk Areas</u>

- Platform design criteria acceptable to regulatory agencies
- Non-redundant mooring system
- Cathodic protection of hull, CWP
- Platform, CWP deployment
- Module/platform interface
- Environmental considerations

Studies recommended to reduce these risks to an acceptable level are as follows:

<u>High Risk Areas</u>

- Develop 40 MWe surface platform designs as an alternative to the spar.

- Continue with proposed CWP validation program (analysis, model tests, at-sea tests) directed toward recommended CWP concepts.

- Conduct in-depth studies of stability, fire protection, deluge and other safety-related systems in conjunction with USCG.

- Conduct platform/riser cable integration studies to investigate solutions to the cable problem.

<u>Medium Risk Areas</u>

- Initiate dialogue with ABS, USCG and other regulatory agencies for input to design.

- Evaluate alternative tension leg concepts to reduce the risk of gross failure.

- Investigate the effectiveness of impressed current systems, and hull/heat exchanger compatibility. At-sea testing is also recommended.

- Conduct in-depth studies of platform and CWP dynamics and stability during the launch/deployment sequence, backed by model tests.

- Conduct similar studies of module deployment.

- Begin initial environmental assessment studies including plume analysis.

DESIGN CONSIDERATIONS ON 100-MW$_e$ COMMERCIAL SCALE OTEC POWER PLANT AND A 1-MW$_e$ CLASS ENGINEERING TEST PLANT

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Ocean Thermal Energy Conversion

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Abstract

The current status of the feasibility study conducted by the OTEC Committee in Japan is presented. Based on the advanced design of 100-MWe OTEC power plant carried out in FY 1976 - FY 1977, the results of the power system optimization examined in relation to the plant site conditions are discussed. An analysis of the external forces and dynamic properties are given for installation of a cold water pipe on a surface-ship-type and a submerged-cylindrical-type platform under the design criteria at two representative sites (Osumi and Toyama). The conceptual design of 1-MWe engineering test facility, which is the next step in Japan, is introduced.

Introduction

Energy consumption in Japan has increased year by year, and the import ratio of Japanese energy sources reached about 88% recently. Therefore, it is important for our country to develop alternative energy sources such as solar, geothermal and ocean.

Ocean thermal energy is supplied to the ocean surrounding Japan by "Kuroshio", which is one of the largest currents in the world. Hence, Japan potentially has a large amount of ocean thermal energy resource. Feasibility studies on Ocean Thermal Energy Conversion (OTEC) have been conducted in Japan under the "Sunshine Project", which has been promoted by the Agency of Industrial Science and Technology, MITI. In FY 1974, the OTEC Committee was established to carry out a feasibility study on OTEC in various aspects such as engineering, economics, environmental assessment, and energy resources. Since 1974, the OTEC Committee has integrated Japanese OTEC research. The results presented in this paper are mainly based on studies which the OTEC Committee conducted in FY 1978.

Design Considerations for a 100-MWe OTEC plant
Site Selection

Based on oceanographic and atomospheric data, the site must be carefully selected taking into account the mutual interacion between the environment and OTEC power plant as well as socio-economic aspects. Five representative sites for OTEC power plants in Japan were evaluated by the OTEC Committee ; (1) Iriomote island, (2) Okinawa island, (3) Toyama bay, (4) Osumi islands, and (5) Izu peninsula. These sites were evaluated with regard to power generation condition, ocean engineering, and socio-economics.

The monthly changes of the temperature difference (ΔT) between the surface and deep water (600, 800, 1000m deep) at these sites were obtained from the data accumulated at the Japan Oceanographic Data Center, Maritime Safety Agency, Ministry of Transportation. In every month the capacity factor was determined by the equation $\varphi = 0.1(\Delta T - 11°C)$, that is, at $\Delta T = 21°C$, φ equals 1.0, and at $\Delta T = 11°C$, φ is zero. The mean capacity factors were obtained from the average of each monthly capacity factor under three cold water intake depth as shown in Table 1.

Oceanographic and atomospheric conditions were investigated from the viewpoint of installation of a marine structure. The following factors were considered; (1) maximum wind velocity, (2) the number of times of appearance of higher waves and longer period waves, (3) significant wave height, and (4) the number of times of typhoon.

The social aspects to be considered; (1) effect on the fishing grounds, (2) effect on ocean route and ocean field for fleet exercises, and (3) the need for open seas near the mainland in connection with restrictions on the installation of OTEC power plants.

The following factors were evaluated from the viewpoint of the maintenance and future power demand; (1) the shortest distance to land, (2) the distance to a major port, and (3) the distance to the nearest airport. The evaluation from these aspects is summarized in Table 1 for each representative OTEC site. As a results, the sites offshore of Osumi islands (Osumi) and Toyama bay (Toyama) were selected as the representative Japanese sites. The five site locations are shown in Fig. 1.

Power System Optimization

Based on the advanced design of a 100-MWe(gross) OTEC power plant presented at the 5th Conference[1], the power system was evaluated relative to site conditions, because the design conditions and the cost of the OTEC power plant depend strongly on the site. Therefore, it is necessary to optimize the power system for each site.

Plate-type heat exchangers were adopted in expectation of higher heat transfer coefficients under the appropriate pressure drop, smaller size, and lower cost than the conventional shell-and-tube

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Table 1 Characteristics at five representative OTEC sites in Japan

Condition	Parameter	Iriomote	Okinawa	Osumi	Toyama
Generating	Capacity factor				
	0 - 600	0.66	0.49	0.70	0.62
	0 - 800	0.93	0.75	0.86	0.62
	0 -1000	1.00	0.90	0.94	
Oceanographic	Maximum wind velocity (m/s)	60.9	73.6	78.6	39.6
	Annual time of Occurring the wave height more than 7.75 m (%)	0.22	0.22	7.67	
	Significant wave height estimated (m)	14.0	15.0	13.0	8.5
	Annual times of typhoon (No/year)	41-60	41-60	41-60	21-40
Social	Effect on ocean route or oceanic fleet exercise (yes or no)	no	no	yes	no
	Effect on fishing (yes or no)	no	no	no	no
	Open sea near the mainland or not	no	yes	no	yes
Geographic	Shortest distance to a land (km)	10-15	20	20	3-4
	Distance to the major port (km)	450	100	200	20-25
	Distance to the nearest airport (km)	60	30	70	20

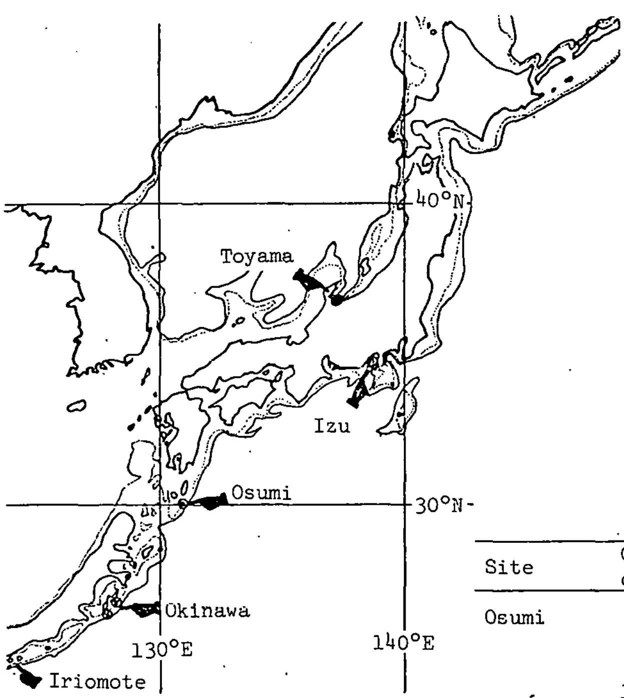

Fig. 1 The representative OTEC site locations in Japan

type. The evaporator consists of vertical plates which are sealed with gaskets to each other. The warm sea water flows horizontally through the passages between adjacent plates from the outer shell. The working fluid is sprayed from the passages over the vertical plate, the other side of which is heated by the warm sea water[2]. The condenser is a vertical-plate-type similar to the evaporator. The working fluid is vertically condensed on the plate and drained by the drainaged gutters. Basic experimental data of the heat transfer performance have been obtained with Fron 114. Based on these data, the overall heat transfer coefficients, including the fouling factor at the sea water side were estimated 5000 $kcal/m^2h^oC$ for the evaporator and 4500 $kcal/m^2h^oC$ for the condenser under the water flow rate of 2 m/s using ammonia as a working fluid. From the optimization process, the overall heat transfer coefficients for the evaporator and the condenser were selected to be 3476 $kcal/m^2h^oC$ and 3015 $kcal/m^2h^oC$ respectively, and the corresponding water flow rates were 0.79 m/s and 0.87 m/s, respectively.

For the advanced platform design, a submerged cylindrical type was selected for the Osumi site to avoid unacceptably large forces on the platform in the stormy season. A surface-ship type, which has advantages from the viewpoints of both engineering and cost, because of the application of baseline shipbuilding technology, was chosen for the Toyama site.

Each of the four 25-MWe (gross output) power modules consists of an axial flow turbine, two evaporators, two condensers, two pumps for the working fluid, two pumps for warm water intake, two pumps for cold water intake, and auxiliary equipments. Optimization analyses were conducted at Electrotechnical Laboratory in cooperation with industrial firms to select the optimal ratio of the construction cost to the net power output (i.e., the gross power output 100-MWe minus total pumping power). Cost functions were developed for an evaporator, a condenser, a turbine generator, pumps and a cold water pipe. Calculations were carried out to minimize total cost, and the allocations of temperature to each component and flow velocities of the warm and cold water through the heat exchangers were determined.

The design temperatures of warm and cold water must be carefully determined by consideration of the annual temperature variation of the plant site. Under the several assumptions, the optimum design inlet temperatures of warm water at Osumi and Toyama were determined as shown in Table 2. In the Table the optimum design temperature, annual total power output normalized by the base power output at overall temperature difference T = 21°C, and

Table 2 Optimum design temperature at Osumi and Toyama

Site	Cold water depth	Optimum temperature	Annual.total power output ratio	Available factor ratio
Osumi	600 m	28.2°C	0.452	69.4 %
	800	28.2	0.811	100
	1,000	28.1	0.989	100
Toyama	300	26.4	0.518	52.8
	400	26.1	0.548	55.6
	500	25.9	0.560	59.4

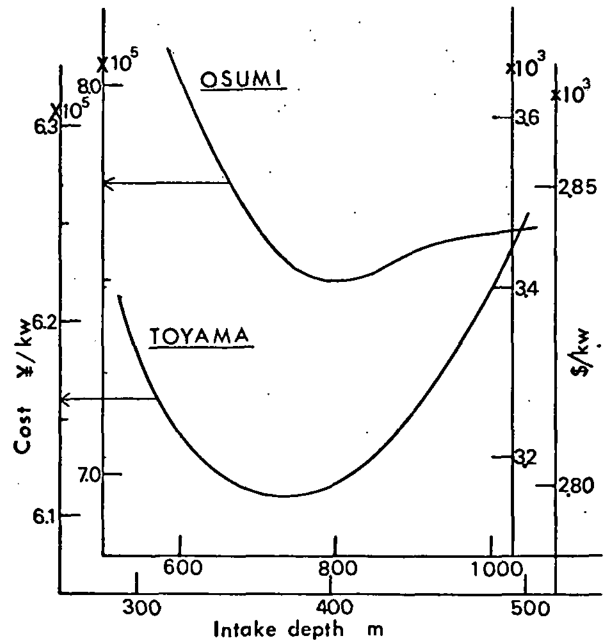

Fig. 2 Cost change with cold water intake depth at Osumi and Toyama

available factor are shown for three cold water intake depths. The design temperatures of warm water are 28°C and 26°C at Osumi and Toyama, respectively.

The temperature profile at Osumi is quite different from that at Toyama. For example, at 500-m depth the temperature at Osumi is 9.18°C; at Toyama, 0.35°C. The optimum intake depth was determined from the trade off between T (i.e., power output) and cold water pipe cost and friction loss (see Fig. 2). The optimum intake depths are 790 m and 370 m at Osumi and Toyama, respectively.

The optimization analysis showed that the OTEC power plant could deliver net powers at busbar of 78.8-MWe at Osumi and 83.1-MWe at Toyama as shown in Table 3. The major specifications and costs are summarized based on the 1975 and 1976 design results. In the 1978 design, unit power cost at busbar includes the available factor and capacity factor of the plant, so that the costs are slightly higher than the costs of previous designs. The estimated OTEC power costs for baseload power are comparable to projected costs for fossil-fuel fired power plant.

Corrosion and Biofouling for OTEC Heat Exchangers

Corrosion control is a very important problem in the design of OTEC heat exchangers. The materials being tested at the Government Industrial Research Institute, Chugoku, include titanium, aluminum-brass, hastelloy, copper-nickel alloys and stainless steels. The test was begun by measuring the polarization behavior and the rate of corrosion of the materials in artificial seawater. It is assumed to extend the test from the general corrosion characteristics to the localized one, such as pitting, crevice, galvanic, dealloying corrosion, stress corrosion cracking, both in the artificial sea water and ocean environment. In addition to the corrosion problem, the performance of the heat exchangers is affected not only by corrosion in the working fluid, namely ammonia, but also by fouling

Table 3 Major specifications and costs of 100-MW OTEC power plant

Items	1975 design	1976 design	1978 design Osumi	1978 design Toyama
Gross power output ; MW	100	100	100	100
Net power output , MW	73.9	77.2	78.8	83.1
W. F. flow rate , 10^7 kg/h	1.18	1.114	0.99	0.908
Warm water temp. , °C	28	28	28	26
Intake warm water , 10^8 kg/h	9.88	9.74	7.82	6.93
Cold water temp. , °C	7	7	4.56	0.747
Intake cold water , 10^8 kg/h	1.01	8.09	6.16	5.69
Evap. heat transfer area, 10^5 m^2 unit	3.2 16	3.11 8	2.62 8	2.14 8
Cond. heat transfer area, 10^5 m^2 unit	3.3 16	3.51 8	2.94 8	2.37 8
T/G output, unit	25 4	25 4	25 4	25 4
Type of platform	Rect. barge	Submerged cyl.	Submerged cyl.	Surface ship
Unit construction cost, 10^6 yen/kw 10^3 $/kw	78.0 3.55	64.5 2.93	78.1 3.55	59.3 2.69
Unit power cost at the busbar , yen/kwh mills/kwh	11.75 53.4	9.56 43.5	12.21 55.5	12.36 56.2
Power transmission cost, yen/kwh mills/kwh	- -	1.5(D.C) 6.8	0.66(A.C) 3.0	0.66(A.C) 3.0

from inorganic deposits and organic slimy films. Ammonia may cause a corrosion problem when it is contaminated with oxygen or water, and in certain cases leads to a fracture of materials.
Biofouling may also promote corrosion of materials and degrade the heat transfer coefficient. It is, however, generally incompatible to invest the materials with the suppressive character to fouling and the protective against corrosion, a decrease of corrosion due to protective coatings, formation of protective layers, application of electric current, and others, all lead to possible fouling even of copper and copper base alloys. It is, therefore, assumed at present that the prevention and removal of biofouling is done by chemical and mechanical treatments, but depending on the chemicals and means used in the treatment, new corrosion problem will arise. Although plate-type heat exchangers are small size and have low initial costs, they are difficult to keep clean at the present stage. Therefore, this type must interrupt its operation for maintenance. On the other hand, the shell-and-tube type, which is widely used for fossil-fuel and nuclear power plant condensers can be kept clean enough to run with some established mechanical means such as rubber balls and brushes. If the plate-type heat exchanger needs scheduled maintenance periods, they will increase its operating cost and decrease its capacity factor. As a results, if the plate-type needs maintenance for one month a year, and by this time the heat transfer coefficient decreases to 90 % of the initial value, the cost performance is almost the same for both type of heat exchangers. In advance of R&D on enhancement of plate-type, cleaning technology for plate-type heat exchangers should be developed and tested under real sea conditions.

Platform Structure and Cold Water Pipe

Several types of ocean platform structures have been evaluated. A rectangular barge type was selected for the baseline design by the OTEC Committee for a plant site in the tropical ocean. For the advanced design, different platform configurations were selected corresponding to plant sites around Japan, because of the large forces that might attack the ocean structure in the stormy season. As mentioned earlier, the surface ship is attractive from the viewpoints of reliability and economics. Both types have been investigated by Shimizu Construction Co., Mitusbishi Heavy Industry, Ltd., and Ishikawajima-Harima Heavy Industries Co., under the leadership of Tokai University.

For the design study, the characteristics of sea states were selected as shown in Table 4 as the representative design criteria in Japan. The external loads and dynamic properties were calculated on two types platform configurations installed 100-MWe OTEC power system at both sites (Offshore of Osumi islands and Toyama bay). The size and displacements for both types including the cold water and discharged water pipes were specified as shown in Table 5. The resultant horizontal force was computed for each type of platform with a cold water pipe (CWP) in both ordinary and extraordinary sea states. In the case of a surface ship type, the following calculation procedure was used:

1) The external forces were analyzed for various directions of wind, wave and current individually.
2) Each external force was divided into the longitudinal and lateral components in terms of ship direction.

Table 4 Design criteria of platform at both offshore of Osumi islands and Toyama bay

		Osumi	Toyama
Maximum wind velocity (mean) , m/s		60	60
Current velocity , m/s			
Surface		1.74	1.00
Depth of	50 m	1.08	-
	100 m	1.15	0.30
	140 m	1.15	-
	500 m	0.25	0
Wave			
Significant wave height, m		13	8.4
Significant wave period, s		13	13

Table 5

Dimensions and displacement of surface ship type and submerged cylindrical type design

Ship type		Submerged disc type	
Length , m	230	Diameter , m	110
Beam , m	60		
Height(depth) , m	27		40
Draft , m	13		-55
Displacement , 10^3 t	145		409
One cylindrical pipt for cold water extraction		Two cylindrical pipes for water discharge	
Diameter , m	11.2		8
Length , m	500		100

Table 6 Estimation of external force and dynamic properties at Osumi and Toyama

Site	Sea state	Force	External forces: Surface ship	External forces: Submerged cylinder		Dynamic properties: Surface ship	Dynamic properties: Submerged cylinder
Osumi	Extraordinary sea state	Wind force (t)	440	163	Surge (m)	2.4	2.5
		Current force (t)	400	703	Sway (m)	2.3	-
		Wave drifting force(t)	350	405	Heave (m)	5.0	2.6
					Roll (°)	20.9	-
		Total (t)	1190	1271	Pitch (°)	4.3	0.1
	Ordinary sea state	Wind force (t)	30	10	Surge (m)	0.6	0.5
		Current force (t)	400	703	Sway (m)	0.7	-
		Wave drifting force(t)	130	82	Heave (m)	1.2	0.6
					Roll (°)	2.8	-
		Total (t)	560	795	Pitch (°)	1.5	0
Toyama	Extraordinary sea state	Wind force (t)	445	163	Surge (m)	1.6	1.6
		Current force (t)	95	143	Sway (m)	1.9	-
		Wave drifting force(t)	180	169	Heave (m)	3.0	1.7
					Roll (°)	12.7	-
		Total (t)	720	474	Pitch (°)	2.9	0.1
	Ordinary sea state	Wind force (t)	30	10	Surge (m)	0.6	0.5
		Current force (t)	90	143	Sway (m)	0.6	-
		Wave drifting force(t)	140	82	Heave (m)	1.1	0.6
					Roll (°)	2.6	-
		Total (t)	260	235	Pitch (°)	1.5	0

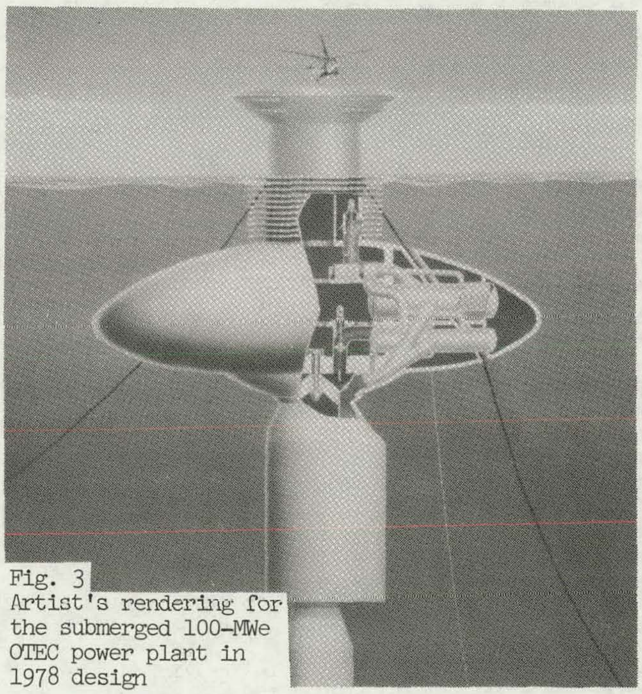

Fig. 3
Artist's rendering for the submerged 100-MWe OTEC power plant in 1978 design

3) The calculations were continued to obtain the condition that the total longitudinal force became the maximum with the total lateral force being zero.

The resultant longitudinal forces are shown in Table 6. The wind forces are large in the stormy season. In the case of submerged-cylindrical type the total external force was deduced from the scalar summation of each force, because of symmetrical shape. The dynamic characteristics were calculated from the computer model. The current force dominates in the ordinary at Osumi. The larger resultant force comes from the larger volume of the submerged-cylindrical type as compared to the ship type because of necessity for additional ballast space. In order to reduce the total external loads, there exists further work on the arrangement of power system configuration and outer shape of the platform. For example, an elliptical outer space was proposed for a submerged-cylindrical type as shown in Fig. 3 to reduce the external force due to current.

The dynamic analysis on the motion of the platform connected with the CWP was achieved in the

Table 7 Anchoring system specification

External force	Chain	Total weight for 1650-m chain	Sinker weight	Sinker concrete volume
1200 t	16	11,502 t	800 t	8650 m^3
800	8	5,751	840	4540
400	8	2,800	440	2380

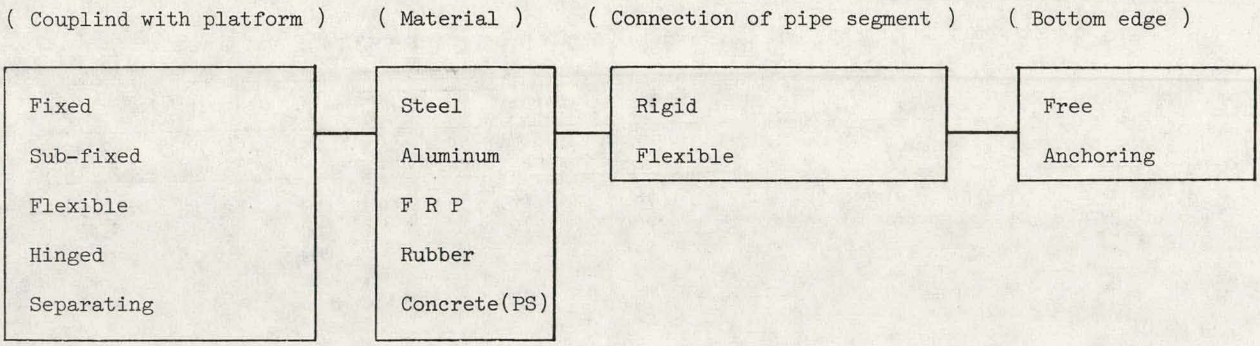

Fig. 4 Cold water pipe approach

preliminery stage in order to optimize the OTEC plant configuration and its mooring system and to clarify the design of the power transmission cable. To evaluate the precise dynamic characteristics, it is proposed as future work to investigate that coupling the platform and CWP, CWP material, CWP connection method and the state of bottom part of CWP as shown in Fig. 4.

The design of the anchoring systems based on the above resultant external loads and baseline anchoring technology are presented in Table 7. In the calculations the holding power coefficient of a sinker made of concrete is 0.65 and a safety factor is 1.3.

Environmental Aspects on OTEC Power Plant

Environmental assessment has been conducted by an oceanophysicist group in collaboration with a marine biologist group belonging to the OTEC Committee under the leadership to Tokyo University. Some disturbance of sea water temperature profile due to large scale operation of OTEC Plant might affect meteorological, oceanic, and biological environments.

A preliminary study as for the behavior of cold water discharged plume and biological effect has been carried out. In addition to the evaluation of the plume behavior at the near field, the effect of the Coriolis force at the far field was investigated with a basic experiment using a small-scale rotation water bath at Tokyo University. The results indicated peculiar characteristics of the dilution of water by the rotation effect qualitatively. The water plume did not sink and was kept near the surface by the influence of the rotation. The entrainment of nekton by the CWP was estimated to be about 5000 tons per month for a 100-MWe plant. And the entrainment by the warm water pipe was estimated to be 70000 tons (wet weight) of nekton per month based on the data on the vertical distribution of nekton[3]. The entrainment of micronekton and spawn is an important problem, even though ecological compensation might be expected.

Conceptual Design of 1-MWe Engineering Test Facility

In order to demonstrate proof of the OTEC concept, a 1-MWe test facility is planned to be operated at sea to generate electrical power using ammonia as a working fluid under the real sea states. The test objectives are:

1. Power system
   a) Proof of OTEC power generation concept using real sea water
   b) Evaluation on the characteristics of the OTEC power system in comparison with the results of small scale OTEC test facility on land and also theoretical simulation model

Table 8 Major specification of 1-MWe Engineering Test Facility

Items		Specification
Warm water temperature		28°C
Cold water temperature		7°C
Overall Δ T		21°C
Evaporating temperature		22.9°C
Condensation temperature		12.5°C
Turbine efficiency		0.6
Generator efficiency		0.96
Gross power output		1000 Kwe
Rankine efficiency (net)		1.98 %
Evaporator,	Heat duty	4.34×10^7 kcal/h
	Area	$4.13 \times 10^3 m^2$
	Pressure	9.58 kg/cm^2abs
	Size	4.9 m x 9.15 mL
	K_E	3476 $kcal/m^2h°C$
Condenser,	Heat duty	4.25×10^7kcal/h
	Area	$4.70 \times 10^3 m^2$
	Pressure	6.83 kg/cm^2abs
	Size	5.9 m x 7.7 mL
	K_C	3015 $kcal/m^2h°C$
Working fluid ammonia flow rate		1.48×10^5 kg/h
Warm water flow rate		12.640 m^3/h
Cold water flow rate		10.526 m^3/h
Intake & discharge pipe,	Warm water	1.20 m
	Cold water	1.30 m x 500 mL
	Discharge	1.80 m x 100 mL
Pumping power,	Warm water	123 kw
	Working fluid	32 kw
	Cold water	221 kw
Net power output		624 kw

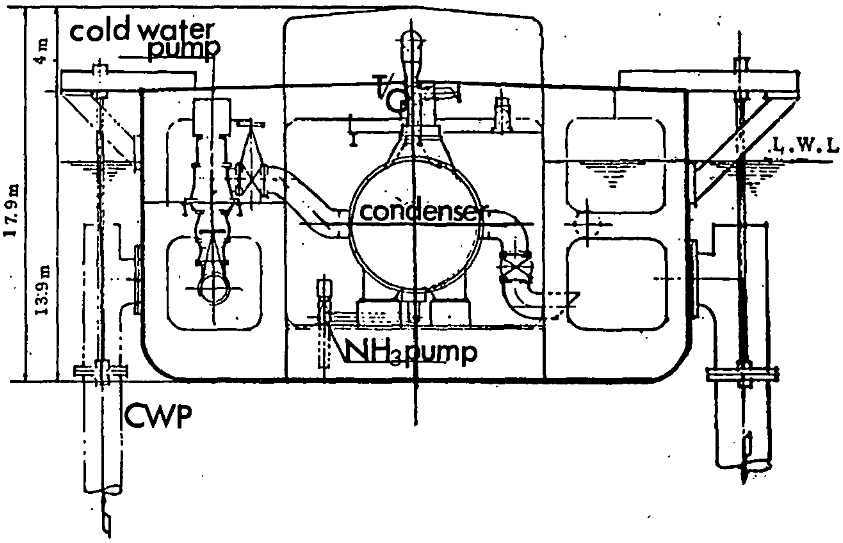

Fig. 5 Sectional view of 1-MWe Engineering Test Facility

   c) Evaluation on the performance of evaporator and condenser, in paticular, biofouling and corrosion effect
2. Ocean engineering
   a) Measurement of external force and dynamic response to the barge in order to get basic data for large scale structure
   b) Measurement of external force and stress on the cold water pipe
   c) Mooring test by the usage of anchoring system and thruster alternatively
3. Environmental assessment
   a) Evaluation on the behavior of discharged plume at the near field of outlet of discharged pipe
   b) Evaluation on the biological effect due to the discharged cold water including the entrainment effect

The principal object is the performance test of heat exchangers. Test barge is planned to be applied with a used 32,000 freight tonnage oil tanker, which is 202 m long, 26.5 m wide, 13.9 m deep and 10.4 draft. The power system is installed in the part of 4 central oil baths. The specifications are shown in Table 8. The temperature allocation comes from the results of optimization of 100-MWe OTEC power plant. The geometric arrangement is shown in Fig. 5. Various types of the heat exchangers will be tested to obtain design data for large-scale OTEC plants. The test barge will be located, in sequence, at several representative sites as described in the previous section.

Conclusions

The conclusions of this study are:

1. OTEC power cost is highly dependent on the oceanic and atomospheric conditions, which paticularly determine the design temperature difference and capacity factor.
   For the two representative sites (Osumi and Toyama) the costs were estimated by optimization analysis. The resultant costs at the two sites were almost the same 12.2 - 12.4 yen/kwh, even though the CWP lengths and capacity factors were quite different.
2. A cost-effectiveness comparison of shell-and-tube type and plate-type heat exchangers including the maintenance for cleaning yielded nearly equal costs for the two type. It is suggested that the development of the cleaning method should be the first consideration for plate-type heat exchangers.
3. The external forces and dynamic properties were evaluated for a surface ship and a submerged cylindrical type platform coupled with a cold water pipe. Based on these results the anchoring system was estimated.
4. In the course of the investigation on the behavior of plumes, the effect of the Coriolis' force was qualitatively clarified by small-scale rotation water bath experiments.
5. The main emphasis in the next step in OTEC R&D activity in Japan is on the construction and operation of 1-MWe OTEC test facility built on a barge which is a converted oil tanker. Various types of the heat exchangers will be tested at several representative sites.

Acknowledgement

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STATION KEEPING SUBSYSTEM DESIGNS FOR MODULAR EXPERIMENT OTEC PLANTS

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Abstract

Various alternative concepts for mooring of the SPAR and BARGE type platform hulls were considered, evaluated, and the best two concepts for each hull were established. The screening and evaluation efforts included compilation of site environmental data and design criteria, and the development of a reliability/performance/optimization methodology.

At the end of the conceptual design phase of the study, multi-point spread catenary moorings using composite wire rope/chain lines were found to be suitable for positioning both platforms. Also applicable to the SPAR platform was a vertical tension leg type mooring, and to the BARGE platform a single point mooring with tension buoy.

The multi-point catenary mooring concepts were selected as baseline for the next phase, development of preliminary designs, since they use state-of-the-art components.

TABLE 1

Principal Characteristics of the SPAR type 40 MWe Platform

Dimensions, ft.	
Main body depth	133
Main body dia. (min.)	100
Main body dia. (over discharge pipes)	196
Column dia.	35
Weights, Long Tons	
Platform System	22691
Cold Water Pipe System*	(8435)
Power System	4727
Energy Transfer System	352
Light Ship Without Margin	27770
Margin 10% (equipment only)	1661
Light Ship Displacement	29431
Load Items	1244
Fixed Ballast	11500
Saltwater Ballast	4925
Mooring Tension	7200
Total Operating Displacement	54300**

*Neutrally buoyant cold water pipe not included in light ship

**Excludes water in CWP and discharge tubes

Introduction

The first large scale power generating Ocean Thermal Energy Conversion (OTEC) plant will be the 10/40 MWe Modular Experiment Platform. Current DoE plans are to construct and deploy such a plant by 1984.[1] This paper presents the results of a conceptual design study on Station Keeping Subsystems (SKSS) for two candidate platforms for this application. The study was performed under contract to the U.S. Department of Commerce within the Department of Energy's OTEC Ocean Engineering Program.

The platform candidates are a submerged SPAR type hull with outboard heat exchangers and a surface BARGE type hull. Preliminary designs on both of these platforms have been prepared by Gibbs & Cox, Inc.[2] and Applied Physics Laboratory (JHU),[3] respectively. Tables 1 and 2 list the principal characteristics of these platforms to be used as the baseline hulls for SKSS studies. Other subsystems of the Modular OTEC platform are presently being studied by various DoE contractors, e.g. cold water pipe by TRW[4] and SAI,[5] riser cable by Simplex,[6] etc.

Prior investigations of SKSS for OTEC platforms at the deployment site have been conducted by Westinghouse[7] and others, and a compilation of state-of-the-art was prepared by Frederick R. Harris, Inc.[8] The present study is limited to a specific site, Punta Tuna at the south east coast of Puerto Rico, and to a primarily moored SKSS configuration with thrusters, if necessary, to be used as secondary systems.

TABLE 2

Principal Characteristics of the 10/20 MWe Pilot Plantship Hull

Dimensions, ft.	
LOA (overall length)	381.5
LWL (length of waterline)	378
Beam (Hull)	121
Beam (Over WW Pumps)	159
Depth to Main Deck	77
Depth to Upper Deck	89
Draft (DLWL, Hull Only)	65

Conditions Defining Displacement:
- 65-ft draft
- Assumes APL HE's at all wells
- Vessel displacement includes displacement of HE's
- CWP in place
- Hull fully outfitted for 20 MWe operation

Displacement	67,901 LT
Ballast (To 65' WL, 20 MW Oper.)	6,757 LT

*Project Manager

†Head, Naval Architecture

**Senior Naval Architect

In this study, MR&S has been assisted by the following subcontractors:

- o Oceanics, Inc.
- o John Gadbois, Consultant
- o Bryant Engineering
- o Linnenbank International
- o Marine Supply Co.
- o A. H. Glenn & Associates
- o McClelland Engineers, Inc.

The basic approach consisted of the following steps:

- o Develop matrix of all candidate concepts
- o Adopt and exercise the quasi-static approach to the mooring system stress analyses
- o Adopt and exercise a framework reliability, performance, optimization (RPO) analysis scheme.

A flow chart detailing the sequence of analyses is given in Figure 1.

SKSS Concepts

Table 3 shows the final matrix of SKSS concepts considered. It will be noted that the matrix consists of tension leg and catenary systems. Rotary mooring concepts (turret mooring) were also considered but eliminated from the matrix due to the difficult problems associated with them. One of the tension leg concepts has the CWP integrated with the mooring system. This is the vertical tension mooring with the tension leg inside the CWP. This concept has been considered in the matrix, and a cost estimate has been developed, but essentially, the decision was made to consider in detail only those concepts which are independent of the CWP. It was foreseen that when the platform, CWP, SKSS, and riser cable preliminary designs are all completed, much more will be known about these subsystems and an integration effort to combine the effects and requirements into a complete system will be more meaningful at that time.

Deployment Sites

The specified region for this study is the southeast coast of Puerto Rico. Of the three possible OTEC MEP mooring sites in this region, as shown in Figure 2, two are close to the Puerto Rico island shelf and oneis farther offshore in

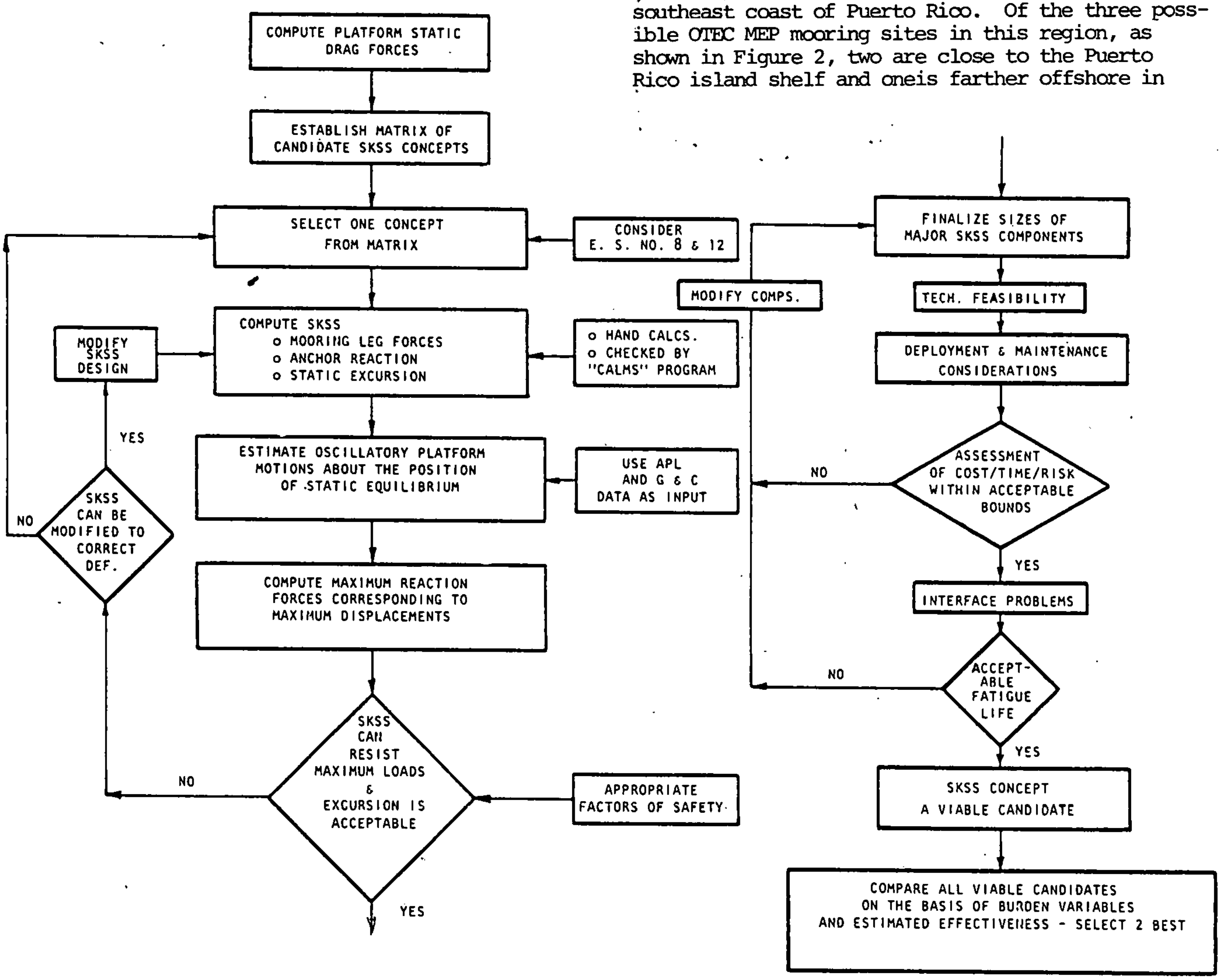


FIGURE 1: Conceptual Design Approach

TABLE 3 Final Matrix of SKSS Concepts

TYPE OF MOORING	BARGE	SPAR
SPREAD CATENARY MOORINGS		
4 LEG HCL*	x	x
4 LEG SBL+	x	x
4 LEG WIRE ROPE/HCL/CHAIN		x
8 LEG HCL/CHAIN	x	x
8 LEG KEVLAR/CHAIN		x
8 LEG KEVLAR		x
8 LEG WIRE ROPE/CHAIN		x
8 LEG WIRE ROPE/CHAIN/NYLON (3 LINES PER LEG)	x	
16 LEG WIRE ROPE/CHAIN (2 LINES PER LEG)	x	
TENSION LEG MOORINGS		
VERTICAL (TENSION RODS) 3 LEG		x
INCLINED (TENSION RODS) 3 LEG		x
VERTICAL, SINGLE LEG INSIDE CWP		x
SINGLE BUOY WITH 3 LEGS	x	

* HOLLOW CYLINDRICAL LINK
\+ SOLID BAR LINK

the Virgin Island Basin. The former would place the platform approximately 3 miles offshore at 4000-5500 ft water depth; the latter, 8 miles offshore at about 6000 ft depth.

Figure 3 shows the bottom profile at the No. 2 mooring site. The environmental conditions at the site, both surface wind, wave, current, and the bottom data, were investigated and the environmental states shown in Table 4 were established for use in the conceptual design analyses.

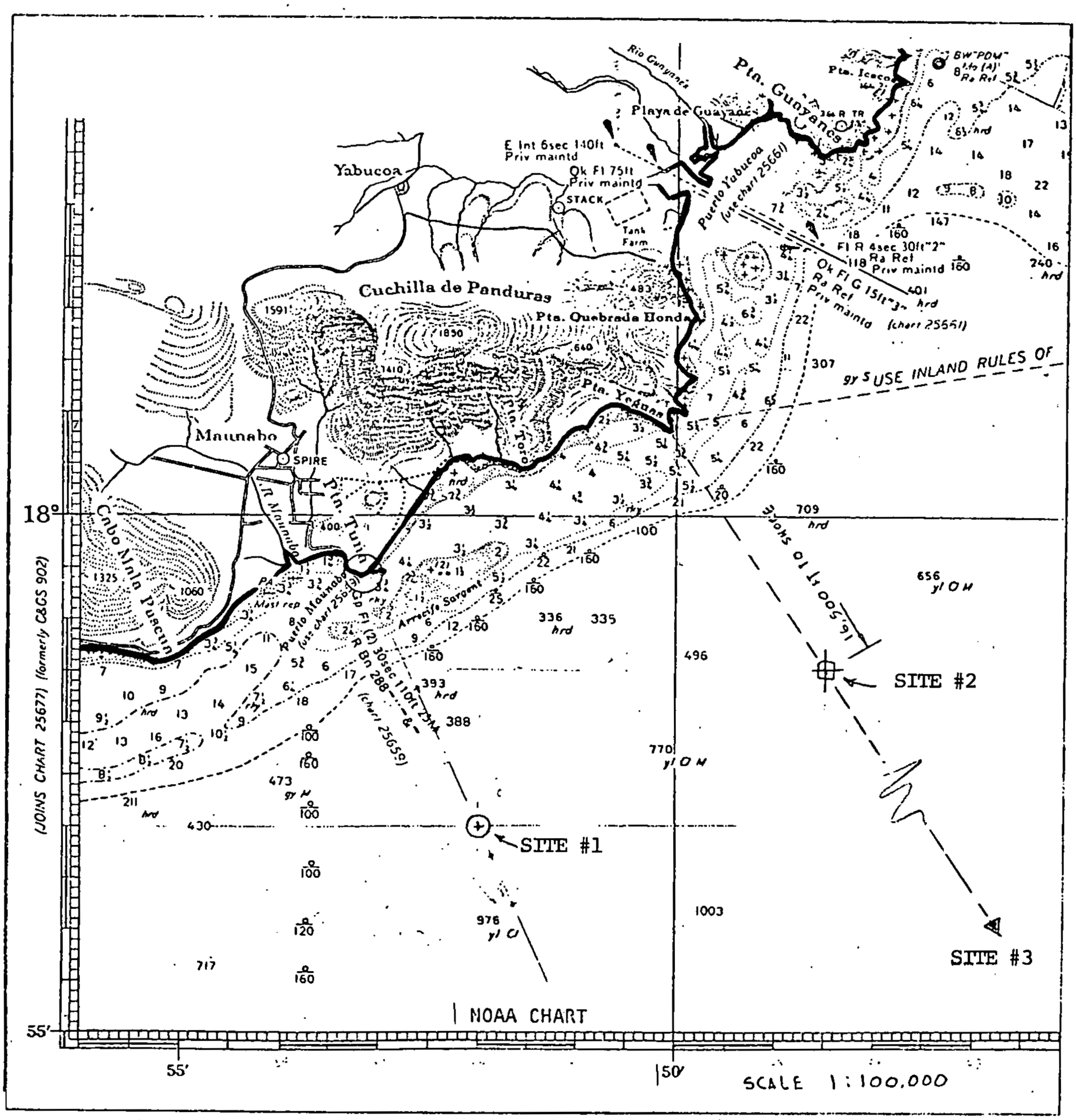


Figure 2: OTEC MEP SITES AT PUERTO RICO

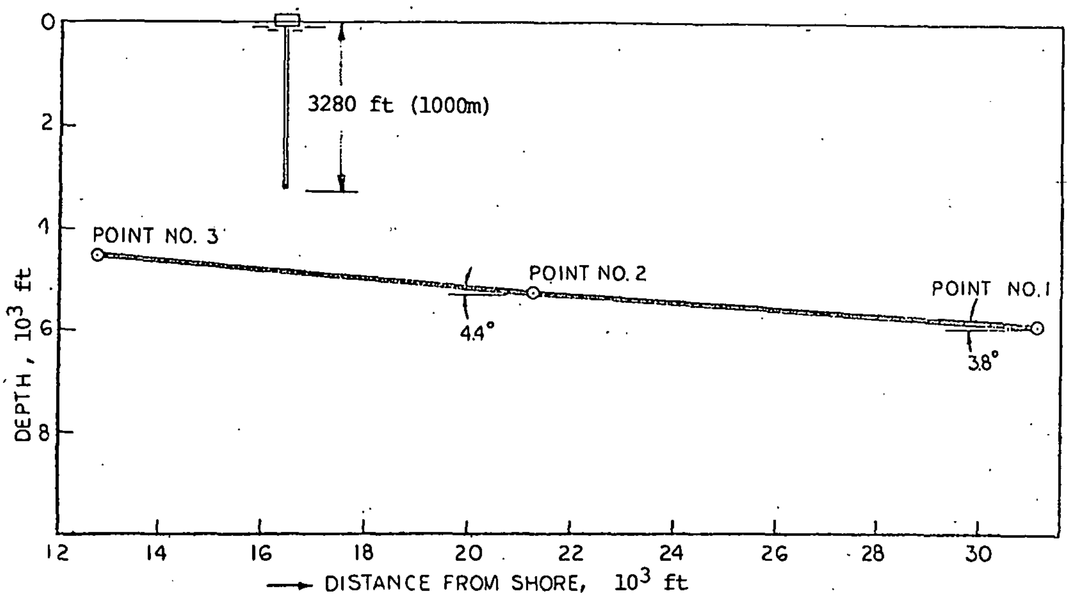


FIGURE 3: Bottom Profile at Site #2

TABLE 4: Environmental states for the Punta Tuna site used in conceptual design calculations

ENV. STATE	$h_{1/3}$ ft	$t_{1/3}$ ft	WIND SPEED KNOTS	SURFACE CURRENT KNOTS	Pe %
1	2	4.0	10	1.5	38.0
2	4	4.7	14	1.6	21.9
3	6	5.4	17	1.6	12.4
5	10	6.8	22	1.7	3.4
8*	25	10.4	40	2.1	1.8×10^{-2}
10	35	11.7	80	2.6	1.6×10^{-5}
12**	45	13.0	94	2.7	1.4×10^{-6}

$h_{1/3}$ Significant wave height

$t_{1/3}$ Significant wave period

Pe Probability of exceedance

* Corresponding to "Design Operational Sea State"

** Corresponding to "Design Extreme (or Survival) Sea State"

Stress Analyses

Although a complete dynamic analysis of the mooring system will be performed during the next phase of this study, the great number of variables and concepts for the conceptual design phase dictated a more simplistic procedure. Consequently the more common quasi-static analysis procedure was adopted. Basically this consists of the following steps:

- Compute the extreme wind, current and wave forces for the operational and survival environmental states. Consider these as static.
- Compute the first-order seakeeping motions of the vessel.
- Apply the static loads to a model of the mooring system and compute the tensions.
- Add the additional tension due to first order motions.
- Compute line reactions and component stresses.

Included in the adopted quasi-static procedure were the consideration of environmental conditions, factors of safety, wave drift forces, and extreme loads.

Table 5 summarizes the static loads on the SPAR and BARGE platforms. The computer results for excursion radii, line tensions, and horizontal anchor forces for the most promising SKSS concepts are listed in Table 6. Figures 4 and 5 depict the most promising concepts for the SPAR; Figures 6 and 7 for the BARGE.

TABLE 5: Static loads on the platforms for various environmental states

a) 40 MWe SPAR

ENVIRONMENTAL STATE	H_s (ft.)	F_{wind} (LT)	$F_{current}$ (LT)	$F_{avg.\ drift}$ (LT)	TOTAL (LT)	(KIPS)
1	2	0.83	209.35	0.57	210.75	472.08
3	6	2.36	217.28	4.08	223.72	501.13
4	8	3.24	220.52	6.09	229.85	514.86
6	15	5.93	238.13	11.49	255.55	572.4
8	25	12.99	268.90	18.75	300.64	573.4
10	35	52.0	298.95	27.57	478.52	1071.8
12	45	71.78	492.43	32.97	597.18	1337.6

b) 10/20 MWe BARGE

1	2	1.47	193.59	4.08	199.14	446.07
3	6	4.25	202.85	54.6	261.7	586.21
4	8	5.88	206.77	102.3	314.95	705.49
6	15	10.72	226.41	354.6	591.74	1325.5
8	25	23.51	253.04	802.17	1078.72	2416.2
10	35	93.93	379.51	1259.1	1732.54	3880.8
12	45	129.91	467.11	2256.9	2853.92	6392.7

Reliability, Performance, Optimization (RPO)

At the conceptual level, the RPO analysis must necessarily be partly qualitative and partly quantitative. The following variables were considered:

- o Performance
- o Reliability
- o Cost
- o Deployment
- o Maintenance, repair and replacement
- o Time schedules
- o Interface with other subsystems

Performance is considered in the maximum operating and survival conditions only. In the operating condition, the platform must not exceed an excursion radius equal to 10 percent of the depth while in the survival condition the allowable leg tension must not be exceeded. Reliability assessment is qualitative and takes into account system redundancy, past experience with similar designs, state-of-the-art of component development, number of possible modes of failure, and handling during deployment and maintenance operations. Costs have been estimated for the SKSS concepts having an operating condition excursion just meeting or slightly less than the maximum allowable. Deployment, maintenance, repair, replacement, time schedules and interface considerations have been included in the cost estimates as well as qualitative burden variables affecting each concept. The average annual costs and present values for each concept were calculated. Table 7 shows a sample cost calculation for the SPAR 8-leg wire/chain catenary mooring concept. The criteria used in measuring the variables are presented in Table 8.

Optimization was inherently performed by proceeding through the design spiral a number of times.

The results of the evaluation to determine the best concepts are summarized in Table 9 for both platforms.

TABLE 6: Computed SKSS results for the two platforms

a) SPAR (total load at survival condition: 1338 kips)

SKSS CONCEPT	PRETENSION KIPS	WATER DEPTH FT	HORIZONTAL DISTANCE TO ANCHOR, FT	HORIZONTAL PULL ON ANCHOR, KIPS	EXCURSION FT	LINE TENSION AT SURVIVAL CONDITION, KIPS
4 point cat HCL/CHN legs	700	5,470	7,880	-	370(0)	1,935
4 point cat SBL legs	3000	5,470	3,382	2,004	164(0)	5,100
8 point cat WR/CHN (nylon)	400	5,470	7,919	669	455(0)	1,065
8 point cat WR/CHN	1081	5,470	7,056	-	142(0)	1,513
8 point cat WR/CHN	290 to 480	4,000	6,600 to 14,350	440	255(S)	650
3 point inclined tension leg mooring	276	4,650	508	2,355	270(S)	13,174
3 point vertical TL mooring	431	4,650	0	1,087	450(S)	4,308
single tension leg (CWP) mooring	12,085	4,650	0	1,801	460(S)	18,460

b) BARGE (total load at survival condition: 6393 kips)

SKSS CONCEPT	PRETENSION KIPS	WATER DEPTH FT	HORIZONTAL DISTANCE TO ANCHOR, FT	HORIZONTAL PULL ON ANCHOR, KIPS	EXCURSION FT	LINE TENSION AT SURVIVAL CONDITION, KIPS
4 point cat HCL legs	2606	5,470	4,510	7,000	626	12,000
4 point cat SBL legs	4000	5,470	4,510		608	12,100
8 point cat SBL legs	6400	5,470	4,557	3,817	274	9,400
8 point cat 3 WR/CHN/NYL lines per leg	555 to 921	4,000	4,055 to 10,000	1,450	Below limit (540 @ survival condition)	1,820
16 point cat 2 WR/CHN lines per leg	1000	6,000	15,362	1,000	338	1,740
single buoy mooring with 3 SBL legs and buoy	8333	4,600	0	4,042	Below limit (585 @ survival condition)	12,117

Note: Allowable excursion radius @ operating condition: 10% of water depth

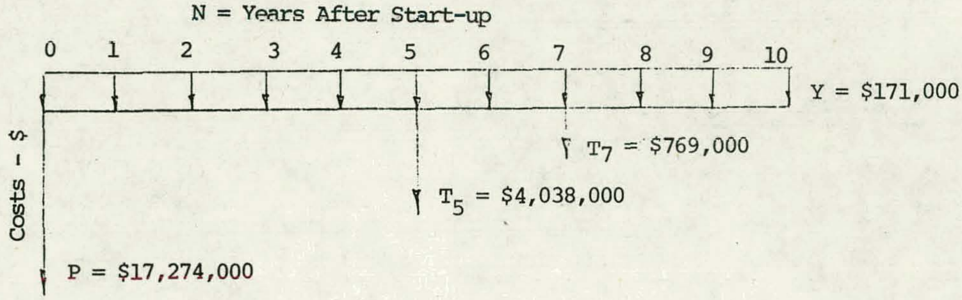
CAT: Catenary   HCL: Hollow cylindrical link

SBL: Solid bar link   WR: Wire rope   CHN: Chain

(O): Operating condition   (S): Survival condition

TABLE 7: Cost calculations for SPAR with 8-leg wire/chain catenary

CASH FLOW CHART



AVERAGE ANNUAL COST

P = $17,274,000	i = 0.09 (9%)	P = Initial Cost
Y = $ 171,000	$(CRF)_{N=10}$ = 0.1558	Y = Annual Operating Cost
		T = Major Maintenance Cost at year N
$T_{N=5}$ = $4,038,000	$(PWF)_{N=5}$ = 0.6499	i = Interest Rate
		CRF = Capital Recovery Factor
$T_{N=7}$ = $769,000	$(PWF)_{N=7}$ = 0.547	PWF = Present Worth Factor

$$AAC = (171 + 0.1558 \times 17,274 + 0.1558 \times 0.6499 \times 4,038 + 0.1558 \times 0.547 \times 769) \times 10^3 = \$3,337,000$$

PRESENT VALUE : PV = 3,337,000/0.1558 = $21,418,000

Figure 4

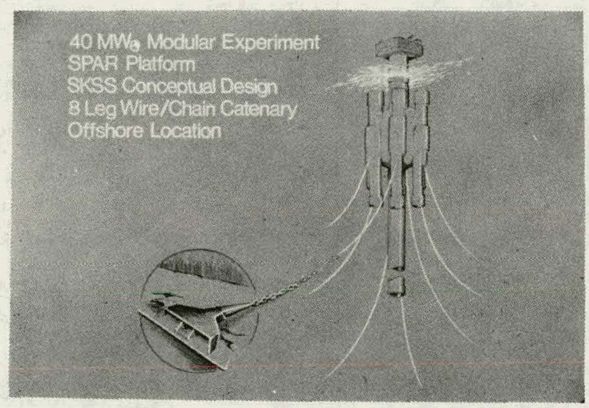
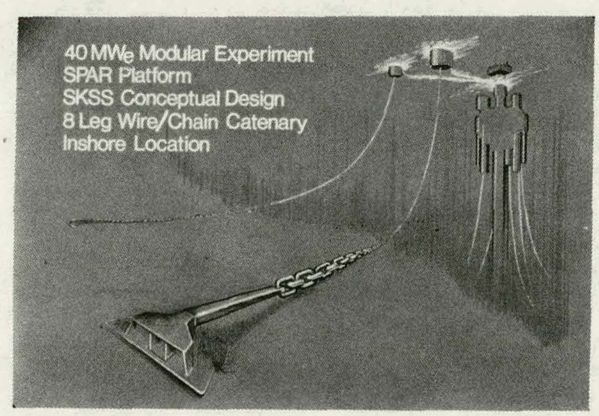


Figure 5



Conclusions

On the basis of this evaluation, the two best SKSS concepts for the MW_e Modular Experiment SPAR Platform are identified as:

- o A multi-leg spread catenary using wire rope and chain for mooring lines
- o A vertical tension leg type mooring.

The two best SKSS concepts for the BARGE platform are:

- o A multi-leg spread catenary using wire rope and chain for mooring lines
- o A single point mooring with buoy and rigid arm, (Figure 7).

Due to the fact that the tension leg type mooring for the SPAR and the single point mooring for the BARGE both require the use of solid bar link type line components which are not yet state-of-the-art, the government has recommended the multi-point spread catenary moorings for both platforms to be used as baseline in the forthcoming preliminary design studies.

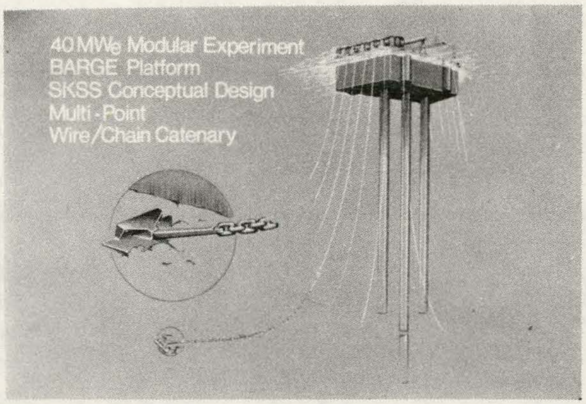


Figure 6

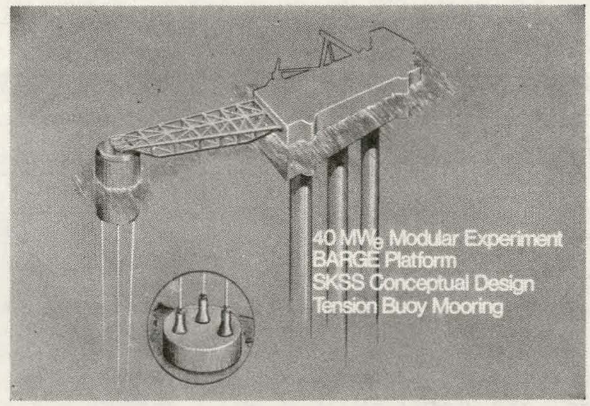


Figure 7

TABLE 8: Criteria for Evaluation of SKSS Concepts

	EXCELLENT	GOOD	FAIR	POOR
EXCURSION	STAYS WITHIN 10% LIMIT EVEN AT SURVIVAL CONDITION	WITHIN 10% LIMIT AT OPERATING CONDITION	HIGHER INITIAL PRETENSION REQUIRED TO LIMIT EXCURSION	LIMIT EXCEEDED
RELIABILITY	• ONE LEG FAILURE CAN BE TOLERATED • SOA COMPONENTS • EXTENSIVE FIELD EXPERIENCE	• SOME REDUNDANCY EXISTS • MORE LEGS CAN BE ADDED TO ASSURE REDUNDANCY	• ONE LEG FAILURE MAY LEAD TO TOTAL FAILURE • STRINGENT QUALITY CONTROL NECESSARY	• NO REDUNDANCY • MAJOR COMPONENT PROBLEMS
DEPLOYMENT	• ALL METHODS AND EQUIPMENT SOA • PREVIOUS EXPERIENCE EXISTS	• SIMILAR METHODS HAVE BEEN USED IN THE PAST • CAN PROCEED AT A FAST PACE	• TIME-CONSUMING • REQUIRES TOO MANY CHARTERED VESSELS	• VERY DIFFICULT AND TIME-CONSUMING • REQUIRES HANDLING OF VERY HEAVY LEGS
M & R AND REPLACEMENT	• WELL PROVEN PROCEDURES • NO REPLACEMENT WITHIN LIFETIME • MINIMUM NUMBER OF CHARTERED VESSELS	• ONE COMPLETE REPLACEMENT IN FIVE YEARS • REPAIR AND REPLACEMENT CAN BE MADE EASILY	• REQUIRES FREQUENT INSPECTION • MANY CHARTERED VESSELS • TIME-CONSUMING	MAINTENANCE AND REPAIR OPERATIONS EXTREMELY DIFFICULT
TIME SCHEDULES	ALL SKSS EQUIPMENT AND MATERIALS AVAILABLE OFF-THE-SHELF	• ALL COMPONENTS SOA • SOME AVAILABLE OFF-THE-SHELF	• SOME COMPONENTS NOT SOA • REQUIRES EXTENSIVE DEVELOPMENT • 1984 DATE POSSIBLE	• COMPONENTS NOT SOA • REQUIRES VERY EXTENSIVE R&D • 1984 DEPLOYMENT DOUBTFUL
INTERFACE	• HULL INTERFACE MINIMIZES MOTIONS • NO PROBLEMS WITH CABLE INTERFACE	• MOORING ASSISTS DESIGN OF CWP • HULL MOTIONS MAY BE REDUCED • RISER CABLE NOT AFFECTED BY EXCURSION	• INTERFERENCE WITH CWP OR RISER CABLE	• VERY SIGNIFICANT INTERFERENCE WITH CWP • POWER TRANSMISSION VIA RISER CABLE DIFFICULT

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8. Frederic R. Harris, Inc., "State-of-the-Art of Mooring and Dynamic Positioning", August, 1977

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TABLE 9: Evaluation and comparisons of OTEC SKSS concepts

a) SPAR Platform

CANDIDATE SKSS CONCEPT	EXCURSION	RELIABILITY	DEPLOYMENT	M&R REPLACEMENT	TIME SCHEDULES	INTERFACE	(PV) COST $M
4 point catenary (SBL)*	G	G	F	F	G	G	49.3
4 point catenary (HLC & CHN)	G	F	F	F	F	G	62.6
8 point catenary (wire rope & chain)	G	E	G	G	E	E	21.4
slant leg tension (3 tension rods)	E	F	F-G	F-G	G	G	54.4
vertical tension (3 tension rods)	E	F	G	G	G	F	28.5
vertical tension (1 leg inside CWP)	E	P	P	P	F	G	34.2

b) BARGE Platform

CANDIDATE SKSS CONCEPT	EXCURSION	RELIABILITY	DEPLOYMENT	M&R REPLACEMENT	TIME SCHEDULES	INTERFACE	(PV) COST $M
4 point catenary (HCL)	F	F	P-F	P-F	F	F (cable) - P (CWP)	210
4 point catenary (SBL)	F	G	P	P-F	G	F (cable) - P (CWP)	107
8 point catenary (SBL)	G	E	P	F	G	G	158
single buoy mooring (3 SBL tension legs)	E	G	F-G	G	G	E (hull) - F (cable)	66
8 point catenary (3 wire rope/ chain lines per leg)	G	G	P	F	G	G	96
16 point catenary (2 wire rope/ chain lines per leg)	F	G	P-F	G	G	F	152

P = poor, F = fair, G = good, E = excellent

* See footnotes to Table 6 for mooring abbreviations.

A FEASIBILITY STUDY OF AN OTEC GUYED TOWER CONCEPT

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Abstract

A bottom-supported, guyed-tower structure may provide a good solution for stationary Ocean Thermal Energy Conversion (OTEC) plants supplying electric power directly to the U.S. mainland. This approach would avoid some of the problems with cold water pipes and electric riser cables seen with floating OTEC platform designs. A novel solution for efficient OTEC fabrication is suggested, based on our experience in designing, fabricating, and installing various structures in oceans all over the world, one of which is designed to withstand hurricane loading in over 1000 ft of water. The OTEC plant would be supported above water by four double-walled steel columns (one at each corner of the plant, with appropriate cross-bracing) which would also serve as the cold-water pipes. Results of this limited study indicate that a guyed-tower OTEC plant would have superior life and stationkeeping ability, and that the optimum plant size, from a structural support standpoint, would exceed 1000-MW_e (net). Cost optimization studies, however, remain to be done.

Introduction

Most people who have shown interest in OTEC know that Georges Claude built and operated a 22-kW_e (gross) OTEC plant in Cuba fifty years ago. Some may not know that offshore oil production started a few years before that. Until recently, OTEC was virtually forgotten, but offshore oil production has undergone massive growth and technical development. Because this paper depends on the offshore oil technology, a brief review is included here.

Offshore oil production had the relative luxury of stepping gradually over fifty years from shallow water to deeper water. In 1909 the first platform structure for drilling was built at Ferry Lake in Cado Parrish, Louisiana. In the 1920s the first of the over 5000 platforms now in Lake Maricaibo, Venezuela were built. In 1955 the deepest water in which a fixed structure has been built for oil production was 100 ft; in 1965, 285 ft; in 1975, 474 ft; and 1978 we completed a structure for 1000 ft. The latter four structures (Fig. 1) were built by McDermott. The first and fourth, in the U.S. Gulf of Mexico, are designed to withstand severe hurricane loading. The second is in British North Sea waters. The third, in the Santa Barbara Channel off California, is designed to withstand severe earthquake loadings as well as the 100-year storm.

This progression is remarkable for the advances in the design technology and construction art that have been required; however, the progression has been limited primarily by economics. Every one of the thousands of offshore structures built had to appear to be justified in terms of rate of return on capital investment for risk ventures. Virtually every such structure has been essentially a one-of-a-kind prototype.

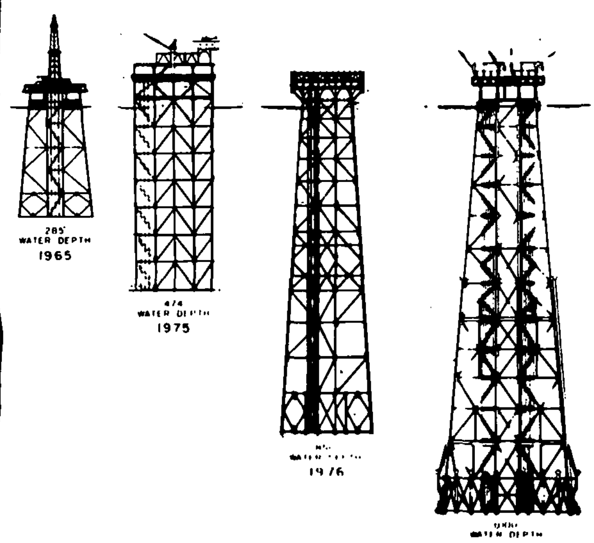

Fig. 1 Size progression of offshore oil drilling structures.

OTEC plant development will not have the luxury of stepping out gradually. Water depths in excess of 3000 ft could be required to operate the first commercially viable OTEC plant. For purposes of this study, we have assumed that the OTEC technology will have been adequately demonstrated prior to construction of the first major commercial OTEC plant. Then the repetitive work of constructing further OTEC plants in the Gulf of Mexico for delivery of power by undersea cable to the U.S. mainland will give the industry a basis for justifying and financing new or expanded facilities. The industries needed are available on the Gulf Coast where the bulk of the world's demand for offshore oil production facilities has historically been.

The OTEC development technology can be logically combined with the offshore oil development experience to yield very reliable guyed-tower OTEC plants.

Oil Production Offshore

The oil industry has often been symbolized by the drilling derrick. Most exploration drilling has been done from barges, ships, and the modern semi-submersibles. But drilling is not oil production. Most production has been done from bottom-supported structures. One very good reason for this is that floating facilities can not continuously keep station year after year. In bad weather they must cease operations and reballast for survival. When the worst of weather is forecast, they usually slip their moorings and often move away.

When a single application must bear the entire capital cost of facilities and continuous operation is desirable, bottom-supported structures have been used almost without exception. The design techniques are well developed arts, soundly based in theory and proven in application. Conductor pipes through which both drilling and production occur are analogous to the cold water pipe required for OTEC. For years high-voltage/high-power submarine electric transmission lines have been terminated on bottom-supported structures and have operated without noteworthy difficulty. No such parallel history of success exists for floating facilities.

Structures 3 and 4 in Fig. 1 are brute force solutions directly extrapolated from a vast amount of prior art. They offer a surety and reliability superior to modern high-risk buildings or bridges on which many people stake their lives nearly every day.

The Guyed Tower Concept

The most promising concept for deeper water is the guyed-tower, compliant structure developed primarily by Exxon Production Research (EPR) in Houston, Texas. Considerable information about the concept and tests of a model in 300 ft of water has been published by EPR in appropriate trade journals. More complete data have been made available by EPR for nominal fees. In the industry, both R & D and specific application studies continue. Construction of full-scale production facilities using the guyed tower in water depths of 1000 ft to 3000 ft seems imminent.

For an OTEC plant, reliability and dependability should be such that power could be sent ashore during a hurricane, provided that storm-driven turbulence does not eliminate the needed seawater temperature difference (ΔT). Cities below sea level, like New Orleans, dependent as they are on electric pumps to control flooding, could benefit from such reliability.

Elimination of cold-water-pipe (CWP) and riser-cable problems with a guyed-tower platform was proposed by Dr. W.L. Green of our Research and Development Division. The author further developed that concept and drew heavily on the experience and suggestions of other McDermott employees.

The fixed concept also has obvious merit for offshore OTEC plants because:

1) There are no CWP structural dynamic problems;

2) There are no difficulties with the transmission line or its termination;

3) The station can operate continuously even during the most severe weather;

4) Programmed maintenance of all major system components, including the CWP(s) and pumps, is practical while operating at a substantial fraction of full capacity;

5) The design life of the entire system may be as long as desired without significant outlay for structural systems beyond that required for a nominal ten or twenty year life;

6) No new technology with respect to structural systems design or construction is required. Extrapolations of the art are required using scale factors not significantly different from those already encountered and successfully handled by the offshore oil industry.

The concept (Fig. 2) developed for this study proceeded from the following considerations:

1) The vast majority of the world's offshore oil structures are made of steel, a highly advantageous structural material, which is used for this concept.

2) For commercial OTEC application to feed electric power to the U.S. mainland the station is assumed to require: a) operation at a fixed station in 3500 ft of water, b) survival of hurricane loadings with safety factors appropriate for a manned structure, and c) support of a 300,000 ton payload of OTEC and power generation equipment and machinery.

3) The structure is designed for the maximum single wave in the design storm; this is substantially more severe than design for the significant wave because fixed structures can not run from

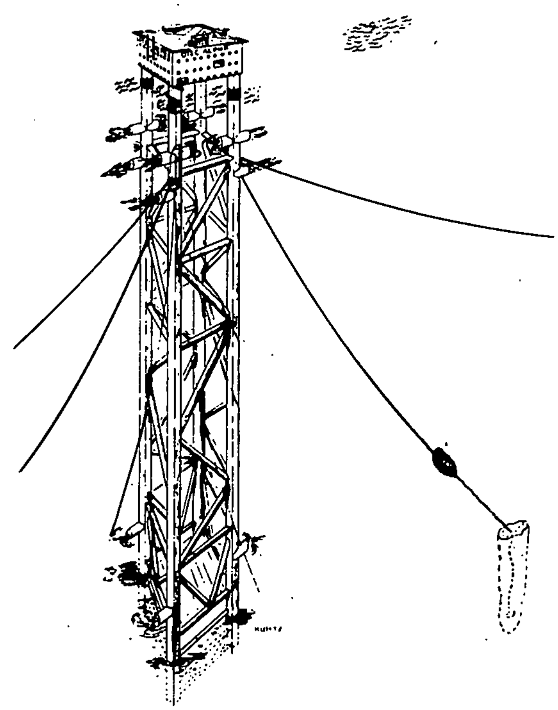

Fig. 2 The OTEC guyed tower concept.

storms, and the single wave of unusual steepness can do much damage. For this reason we have elected to conform with American Petroleum Institute (API) Recommended Practice 2A for the design of offshore structures.

4) A severe distribution of current has been assumed for survival conditions. The load-governing design is current-dominated if it is assumed that:

a) No portion of the system including CWPs, riser cables, or mooring lines is expendable under survival conditions.

b) A smoothly diminished current to extreme depth is present under design storm conditions when currents are driven by tidal or barometric conditions even though wind-driven surface currents do not generally extend to depth. The design storm for the areas where an OTEC plant could be located to supply electric power to the U.S. mainland is the hurricane. Hurricanes are very large barometric disturbances which can give rise to deep flowing currents even though the local sea state and surface currents over open deep water may be less severe than produced by other storms of a non-cyclonic nature. Tidal currents also extend to considerable depth in major portions of the Gulf of Mexico, Florida straits, and portions of the Southern Atlantic Seaboard. These tidal currents would be additive to barometric driven currents depending on the position of the storm. A slow moving storm will likely produce the worst condition of barometric and tidal current in coincident direction.

Without rigorous analysis (which should be reserved until there is cause to advance the design) it can be predicted that structural motion of the guyed tower probably would be imperceptible (except by sophisticated instrumentation) under normal operating conditions.

Study Procedure

The major columns of the tower must have a reasonable axial stress level while leaving margin for additional flexural stresses. Preliminary axial stress levels were held to about one-fourth of yield stress. Common structural-grade carbon steel was used for greatest latitude in fabrication and to avoid joint design difficulties. Plate thickness was limited to 3 in. For adequate stability of tubular column sections having diameter/wall-thickness ratios ≥ 60, double-wall construction was assumed. The axial capacity of stiffener elements was neglected. Employing a column steel such as ASTM A-36 at 9 ksi, 1852 linear ft of 3 in. plate develops the column cross-sections. Combinations satisfying this set of requirements would include one 295-ft O.D. column, three 98-ft-O.D. columns, or four 74-ft-O.D. columns. By similar development, if column axial stress is permitted to be as high as 15 ksi, these outside diameters would be reduced to 177, 60, and 44 ft, respectively.

The four-column structure (Fig. 2) would make the most efficient use of the structural material in resisting flexure. Allowing for 5-ft-double-wall construction and computing the internal conduit area required to provide appropriate cold-water flows through these legs at a water velocity of 5 ft/sec, I computed the required leg O.D. as a function of plant size, and the consequent structural system weight (which closely relates to cost) increases by less than direct proportion to plant size as shown in Table 1. The weight/plant-power ratio decreases from 2.4 tons/kW$_e$ for a 100-MW$_e$ plant to 0.6 tons/kW$_e$ for a 1200-MW$_e$ plant. Furthermore, the larger sizes self float for tow-out in a stable condition with a manageable draft, whereas the smaller sizes would be penalized with worse than indicated structural efficiencies because it would be necessary either to change to a less efficient tripod configuration or provide supplementary buoyancy for tow-out stability.

Table 1. Cold-Water flow rates (at 5-ft/s velocity), column O.D.s, supporting structure weights, and tow-out drafts for guyed-tower OTEC structures

Plant size MW$_e$	CW flow, 10^3 ft^3/s	Leg O.D., ft	Structure weight, 10^3 tons	Tow-out draft, ft$^{(a)}$
50	7.5	32	182	32 U
100	15.1	41	242	31 M
200	30.2	54	330	33 M
400	60.3	72	450	36 S
600	90.5	86	545	37 S
1200	181.0	117	754	41 S

a) U = unstable, M = marginal, S = stable

The maximum stress level during fabrication was found by considering the structure in various stages of completion using a reasonable erection sequence and considering simultaneous application of dead weight, construction loads, and hurricane force wind load. The maximum stress during fabrication was calculated in a full sag condition with the structure assumed to be bridging 3500 ft. The survival stress condition was calculated using a conservative 6-ft/sec full depth current. Approximate calculations for these three critical conditions are shown in Table 2.

Table 2. Appropriate critical stress levels (ksi)

Plant size, MW$_e$	Maximum fabrication stress	Maximum installation stress	Maximum 100-yr-storm stress
600	6.7	20.8	22.1
1200	6.2	20.8	21.7

Actual conditions for survival or installation would be less severe than assumed here, hence a reduction in structural weight could be realized on final design. The fact that preliminary stress levels for various critical conditions are manageable indicates that:

1) The concept including fabrication and installation techniques is entirely within the experience of the established offshore construction industry and is technically feasible.

2) No significant reason exists to employ expensive falsework in fabrication or to take on the high risk offshore assembly of components.

The study concept was modeled using the McDermott marine structures analysis system. Wave force and current loading data were generated using arbitrary design storm data appropriate to a severe Gulf-of-Mexico hurricane. Precision static and dynamic analyses were not performed since three factors made it apparent that the expense was unjustified: 1) preliminary calculation by hand indicated that substantial reductions in the structural weight are in order before rigorous analysis; 2) the natural period of the structure in its fundamental rocking mode is so long that dynamic and wave force interaction effects are small; and 3) the steady-state static solution for current alone is a good approximation for worst conditions of structural behavior. The tower response is primarily a function of the guy system.

Guy systems were analyzed from the base of experience gained in designs of the guyed-tower concept for offshore oil production applications and the actual installation of a 300-ft-water-depth, guyed tower. Current-induced drag loadings in three dimensions and variable submerged weight of the guy lines were considered in rigorous static analysis. It was verified that current-induced drag forces on the guy system itself are not significant and that satisfactory guy systems would require wire rope and chain sizes slightly larger than available today. An appropriate type of wire rope of 10-in. diameter is commercially available at the present time. The larger OTEC plants would require wire rope of 16 to 20 inch size for the guy systems to retain reasonable configurations. The wire rope division of U.S. Steel advised us that the commercial production of wire rope in such sizes is feasible if an adequate demand develops.

An intentionally non-linear, clumped guy system was assumed. The extreme non-linearity contributes to favorable structural behavior. Unlike taut systems, this allows anchor system to avoid being subjected to uplift forces.

Fabrication Concepts

Since commercial OTEC plants will be of vast size, it is doubtful that investment in sophisticated permanent facilities for the production of one OTEC after another is in order. Elaborate dry dock or graving dock facilities are not justified if they produce a structure only once every two or three years. Conversely, the character of the work is more like that of a vast onshore process plant: there are vast numbers of identical components in a single plant, for which the highest practicable degree of automation and sophistication is justified, as well as mobilization of considerable construction capability to an appropriate job site.

Two fabrication concepts have been considered. The selection between them is probably site dependent. For either concept, a site adjacent to an adequate channel and downstream of all bridges is required. For sites with stable underlying geology such as the Limerock Base in South Florida it would be practical to strip, dewater, and excavate an area about 700 ft by 4000 ft to a depth of 50 ft below sea level. The structure could be fabricated in the pit with necessary shops on the adjacent high ground. When fabrication is complete, the pit would be flooded to float the structure and the high ground between the pit and channel excavated away for egress. Such facilities are rarely reused for their original purpose. More often they are retrofitted as minor harbors or major marinas.

An alternative exists which is more appropriate when site conditions would make dewatering to 50 ft either impractical or unsafe. The structure is designed to span its own length under its own weight. This strength can be exploited to advantage to permit the structure to be fabricated on high dry ground without waiting for excavation. Launchways are not required. They could prove to be prohibitively expensive even if the adjacent channel is deep enough for launching. When the structure is substantially complete (even though as much as a year's fit out and detail work remains) deep flooded excavations are made near each end of the structure to about 70 ft. Piles (if required) are driven into the bottom of the excavation and a tremie concrete pile cap/foundation mat is cast in the bottom of the excavation. Two sets of cribbing falsework are built on each foundation, and the structure is brought up to clear the ground by alternatively jacking on the sets of cribbing. Once the structure is clear of the ground, the ground may be excavated or dredged away. After the excavation and fabrication are complete, the falsework is removed. This technique can reduce the construction critical path to a minimum time as excavation and outfitting are concurrent activities.

Installation Concepts

The fabrication objective is to construct, at a convenient and sheltered location, the entire structure and all major mechanical systems as a single seaworthy unit. When fabrication is complete, the structure would be floated from the fabrication site and towed to the installation site. The required channel depth and width would be approximately 50 ft by 1000 ft. The primary tugs would tow the structure from one end using hawsers of adequate length to provide maneuverability and effective decoupling of the small boats' pitching and heaving motions. While in confined waters, a substantial fleet of steering tugs would also be required. Preliminary calculations indicate that three 13,500 HP class sea tugs could make headway with the tow into a full gale.

Prudence and experience dictate that the tow would not be initiated until a good weather window is forecast. Also, the tow should be rigged to survive hurricane force conditions as forecasts are not always reliable.

The guylines and anchorages would have been installed and layed down at the installation site previously. Submarine buoys with sonar transponders would be attached to the head end of each guy line by a messenger line adequate to raise the guy. A section of high strength wire net at the head end of each guy makes graple recovery practical in the event a buoy is lost.

The tower structure will be designed to withstand the stresses of tow in severe sea states and of being tilted up to a vertical position. Once the structure is on location and favorable weather is forecast, the tilt-up is accomplished by controlled flooding of the members. The stable vertical position is reached with the base of the structure several hundred feet above the sea floor. The tower is positioned by the tugs and gently lowered to the sea floor by continued controlled flooding. The structure is stable by its buoyancy derived uprighting moment. To be stable under design storm conditions the base of the structure must penetrate the sea floor sufficiently for lateral load capacity development. To accomplish this and to insure adequate safety factor with respect to vertical loads, the structure would be preloaded. To accomplish preloading the guy system is first attached to the tower and pretensioned.

The vertical components of the guy line tensions (though large) are not adequate to make a significant contribution to preloading. Large amounts of sea water must be taken on in segments of the structure which would be dry for normal operations. Flooding of the structure is resumed and the system sacrifices the stability derived from buoyancy, becoming entirely dependent on the guy system. After sufficient preload ballast has been taken on and the base of the structure achieves the required sea floor penetration the ballast is dumped and selected portions of the structure are blown and returned to atmospheric pressure. Structural installation is completed by readjusting guy line tensions and removal of protective falsework from portions of the super structure. Final instrument installation and electrical hookup can then be made prior to commissioning and startup.

Conclusions and Recommendations

With respect to structural design, construction, and installation, the guyed tower structure is feasible as a reliable platform to support OTEC operations. No new technology is required for the structural design, construction, and installation of such a tower.

For its size alone, an OTEC guyed tower requires reasonable extrapolation of present day construction arts. This extrapolation is within the demonstrated capacity of the offshore construction industry.

Further work is recommended to refine the estimated structural steel requirement and to integrate such data with OTEC process equipment data is required. Once this data is available, cost estimates and meaningful comparisons with other energy systems can be made.

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LAND-BASED OTEC PLANTS - COLD WATER PIPE CONCEPTS

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Abstract

A feasibility design study of land-based OTEC plants has been completed. It concludes that OTEC power plants can be sited on land that is adjacent to deep ocean water, with sufficient thermal gradient, and generate electric power in the 10 MWe to 40 MWe range. Both the Keahole Point, Hawaii, and Punta Tuna, Puerto Rico sites are acceptable candidates for these plants.

The deep water portion of the cold water intake pipe presents the most significant engineering aspect of a land-based OTEC plant. ~~This paper emphasizes~~ the conceptual solutions to the problems of pipe materials, pipe design, and installation methods, are emphasized.

Presented is the rationale used for selecting two different pipe materials and two different installation methods for two different sets of specific ocean floor conditions. The selected pipe design and installation method are then presented. Finally potential risk is discussed and a test program to evaluate or reduce risk is outlined.

Introduction

The vast majority of OTEC investigations have been concerned with power plants sited on various types of floating structures. The power thus developed could be used either in the form of electric power suppplied to a land grid through a subsea cable, or for a "plant ship" which would deliver an energy intensive product such as ammonia or aluminum. An OTEC power plant sited on land offers an attractive alternative to floating plants for certain applications in specific locations. Deep Oil Technology has completed a feasibility study for two plant sizes - 10 MWe and 40 MWe, to be located at two locations - Keahole Point, Hawaii and Punta Tuna, Puerto Rico. This paper emphasizes the conceputal solutions developed in this study to the problems of cold water pipe (CWP) materials selection, design, and installation method. Additional study of the land-based cold water pipe and preliminary design of the 10 MWe to 40 MWe plant will probably be conducted in FY 1980.

The design and installation of the CWP is the only unique construction aspect of a land-based OTEC plant construction. The large diameter requirements (nominally 30 feet for 40 MWe and 15 feet for 10 MWe) and the depth of water for installation (3,000 feet) both exceed the demonstrated capabilities of the offshore pipe laying industry. However, the design and installation methods presented here are deemed to be workable, involving the minimum extension of existing technologies. These are largely dictated by the specific seafloor conditions at the proposed sites.

Pipe Materials Selection

The selection of pipe design and material must be made in conjunction with considering the installation method and the resulting pipe stresses. The most fundamental categorization of possible installation methods depends on whether the pipe should be light (buoyant) or heavy (resting on the seafloor). Thus the candidate pipe materials are grouped in these two categories. Table 1 presents a short list of candidate pipe materials and the estimated cost for manufacturing a 5,800-foot length of pipe for the two plant sizes.

Table 1 Comparative Costs for Cold Water Pipe Materials

			MANUFACTURED COST ($1000)			
			10 MW Plant		40 MW Plant	
			22 to 25-Ft. Diameter		30 to 33-Ft. Diameter	
Candiate Materials		Data Source	$/Lin Ft	Total $	$/Lin Ft	Total $
Heavy	Concrete	Ameron	1.667	9,670	2.583	14,981
	Steel	Kaiser/Fluor	3.505	20,330	5.874	34,070
Buoyant	Fiberglass Reinforced Plastic	Fiberglass Structural Engineering	2.640	15,320	4.004	23,230
	Rubber Coated Nylon	Goodyear	6.56	38,050	8.941	51,860

This comparison is for the deep water, steeply sloped, portion of the cold water intake pipe only, because in the shallow water zones the pipe will be trenched and buried, eliminating any advantages of buoyant pipes. Further the selection of pipe materials for the shallow water zones can be based primarily on cost comparisons since industry has closely related construction and installation experience for nuclear plant cooling water and large sewer outfalls.

In addition to the four materials examined (concrete, steel, fiberglass reinforced plastic, and rubber coated nylon) several other materials have been suggested for floating OTEC plant cold water pipes. These include: polyvinyl chloride (PVC), corrugated metal culvert, and composite steel and concrete pipe.

Of the four materials examined, concrete was found superior for most applications. It not only has the least cost, but also has excellent durability in sea water. Industry has experience in both the manufacture and shallow water installation of pipe in large diameters. Both 18-foot and 21-foot diameter prestressed concrete pipes are currently being constructed. The 18-foot diameter pipe is being installed offshore for the cooling water systems of the San Onofre Nuclear Plant.

The primary disadvantage of conventional concrete pipe is its large weight. This necessitates more costly handling equipment during installation and it imposes large loads on anchor or foundation systems when the pipe is deployed on a steeply sloping seafloor. Lighter weight concrete mixes are being developed and have been proposed for use with floating OTEC plants. Perhaps this technology could be applied to land-based OTEC plants.

Amongst the candidate light or buoyant pipe materials, the choice is less obvious. Fiberglass Reinforced Plastic (FRP) pipe was selected primarily because it has been used to construct cylinders of the required OTEC diameters. A 1,215-foot long section of 24-foot diameter FRP pipe was fabricated for use as a chimney liner. A 34-foot diameter FRP cylinder 80-feet long was fabricated for use as a "cross-flow scrubber." FRP also has the advantages of excellent durability in sea water and ease of transporting raw materials to a field fabrication facility. FRP disadvantages include relatively high cost (compared to concrete), necessity to maintain rigid quality control to assure a uniform product, and lack of familiarity with the material by construction workers.

Another light-weight (though not buoyant) pipe material being examined is corrugated metal culvert. This material is readily available in the required diameters, is relatively inexpensive, and has a history of use for water transport, even in marine environments. It is shipped in many plate segments that are bolted together on the job site. Its disadvantages include a high friction factor and susceptibility to corrosion. Although both these problems might be overcome with suitable coatings and liners, further work is needed to determine the structural limits and modes of failure of this material.

Installation Methods

In selecting the best methods to install the cold water pipe in the deep water, steep slope zone, several methods were defined in categories and partially developed conceptually. These methods may be categorized with respect to whether the pipe is installed in a single length (see Figure 1) or multiple sections (see Figure 2). These are next separated by whether the pipe is transported to its final position along the seafloor or on the surface. The final division relates to whether the pipe deployment proceeds toward the deeper water or toward the shallow water. Defining the installation options in this manner provides assurance that all possibilities are being considered.

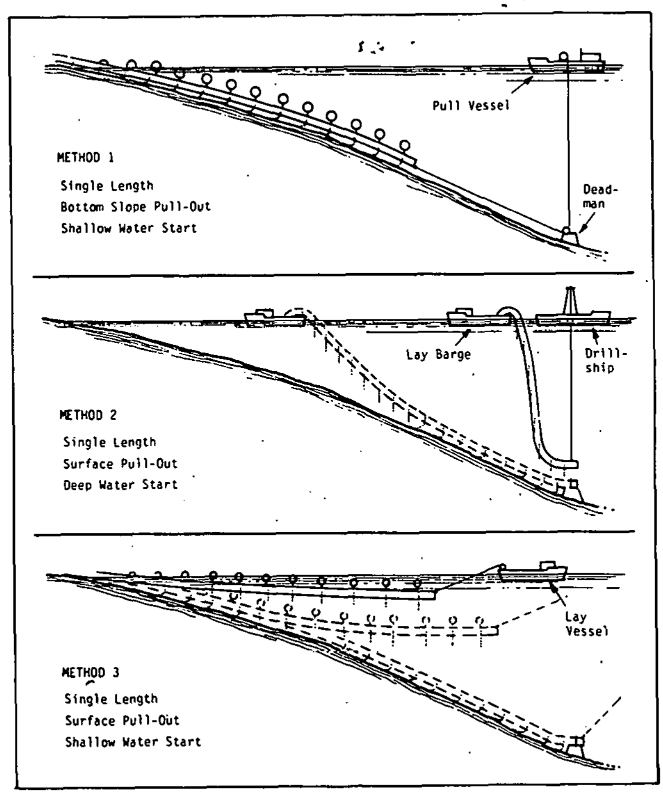

Figure 1 Single Length Pipe Installation Methods

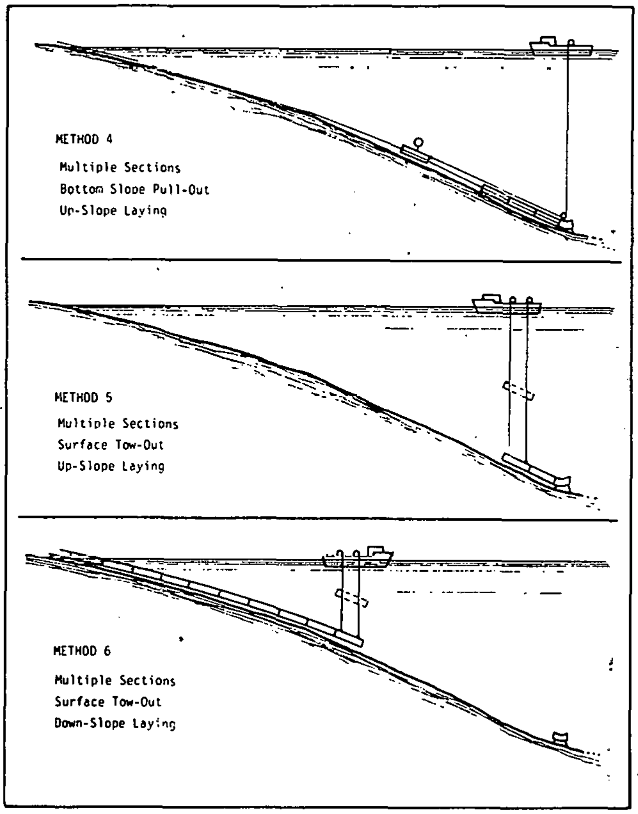

Figure 2 Multiple Pipe Sections Installation Methods

These installation methods were evaluated together with pipe material and site preparation considerations using a subjective Cost/Technical Risk Factor. On this basis, Method 4 (multiple sections, bottom slope, pull-out, up-slope laying) was selected for the concrete pipe sections for the Hawaiian Site, while Method 3 (single length, surface pull-out, shallow water start) was selected for the FRP pipe for the Puerto Rican Site.

The seafloor conditions along a CWP line route greatly influence pipe design and the workability of an installation method. For the deep water portions of the pipeline route any extensive bed preparation was considered to be impractical, or at best, too expensive. Therefore, the pipe design must accommodate essentially existing seafloor conditions. These are quite different for the two locations studied.

The Hawaiian site presents a comparatively smooth uniform lava flow (dense, hard basalt) with an average slope of 26 degrees in the deep water portion. This steep slope begins about 2,500 feet from shore (see Figure 3).

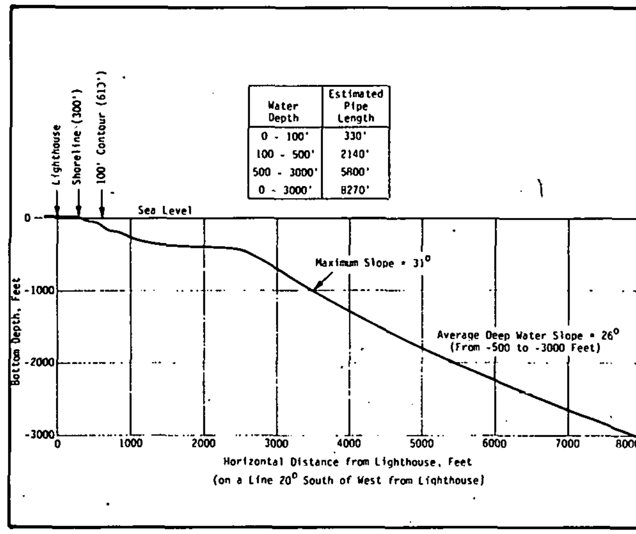

Figure 3 Seafloor Profile, Keahole Point, Hawaii

The Puerto Rican site appears to be extremely irregular and very steep in the deep water portion, though no detailed surveying has been done in the proposed locations. The average slope beyond the 300-foot water depth is about 45 degrees, with some portions exceeding 65 degrees. This steep irregular slope begins over 5,000 feet from shore (see Figure 4).

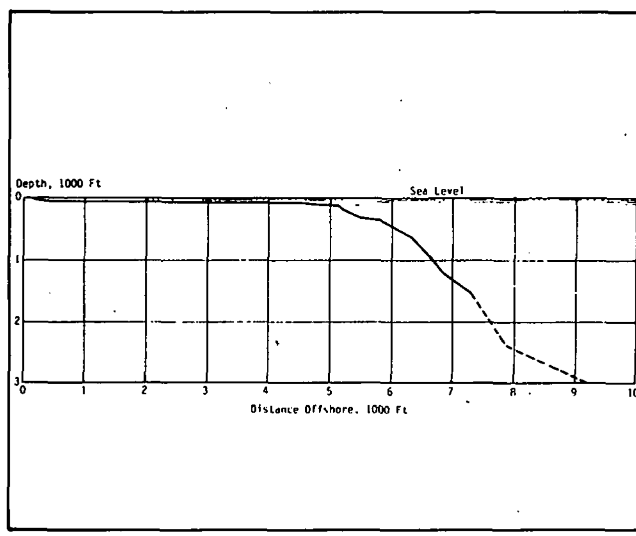

Figure 4 Seafloor Profile, Punta Tuna, Puerto Rico

The relatively uniform and structurally competent seafloor in Hawaii indicated that a pipeline could be installed to rest on the seafloor. Since concrete was the least costly of the candidate pipe materials, a method for its installation was devised. This method consists of first installing a deep water termination block anchored to the steep seafloor with drilled and cemented pilings (see Figure 5). The termination block would anchor two large wire rope guidelines, routed back to shore along the pipeline route. Onshore segments of prestressed concrete pipe would be post-tensioned together into 100-foot long joints (see Figure 6) and launched. A removable transport buoyancy module would be secured to the pipe section. The pipe section would proceed down the pipeline route guided by sliding along the guidelines and partially supported by a surface transport vessel (see Figure 7). These pipe sections would be successively joined together by positioning their spherical mating surfaces (see Figure 8). A pull-in line would also be rigged to the termination block to provide an axial force, controlled from the shore front, to assist joint make-up. The near shore sections of pipe would be layed in an excavated trench and backfilled for protection from surf zone forces.

The steep seafloor irregularities and rock outcrops present in the deep waters off Puerto Rico necessitate a different CWP design and installation method. Here it would be advantageous if the pipe were buoyant and prevented from coming in contact with the seafloor. Amongst the candidate materials studied, the Fiberglass Reinforced Plastic (FRP) pipe appeared best suited because of its somewhat lower cost, related fabrication experience in the required large diameters, and its durability in the sea.

The installation method calls for a detailed bathymetric survey to select a preferred pipeline route and locations (small structurally competent shelfs) for approximately six deadweight anchor clumps. The pipe would be filament wound on 60-foot mandrels in a bay front facility. Multiple helix angles would be used for both the inner and outer walls of a sandwich pipe wall construction. The core between these walls would contain a light weight filler material to increase the pipes buoyancy (see Figure 9). These sections would be welded together and launched in approximately 800-foot long segments. These segments would be joined, with bolted flanges, into one continuous piece approximately 5,000 feet long, while floating in an existing sheltered bay.

Previously the six deadweight anchors would have been landed using a two part wire line. Both ends of this lowering line would be secured to a buoy (see Figure 10). The single length of FRP would be towed a short distance and secured to one end of each of the buoy supported ropes. The entire buoyant pipe would then be pulled down to within 100 feet of the steep seafloor (see Figure 11). Here the pull down lines would be secured, permanently tethering the pipe in its operating position (see Figure 12).

Both of these installation methods are further described in Reference 1.

Potential Risk Evaluation/Reduction

Since laying the CWP is the only principle unproven or high risk construction aspect of a land-based OTEC plant, emphasis should be placed on evaluating and reducing its risk elements. Since pipe design and installation method are quite site specific, the design characteristics of the most probable type of site need to be defined. Alternatively, a specific design and installation method could be developed; its limitations, calculated, and then a site with acceptable conditions located. However, early establishment of a probable plant site, or sites where the power generated could be beneficially introduced into the local power grid, should lead to the earliest commercialization of a land-based OTEC.

To evaluate or reduce risk prior to committing to full plant construction a test program should be conducted. This should be initiated with some small scale qualitative underwater tests. However, the most instructive testing of the actual installation procedures should be done in full scale. The testing could be done on dry land to reduce costs compared to the expense of offshore operation. A land terrain typical of the selected offshore terrain could be located. The selected installation method could be tested and perfected using an appropriate shorter length of pipe or a few joints of pipe. The effect of buoyancy may be simulated on land with a lighter pipe material or perhaps an overland crane. An appropriately designed test could substantially reduce risk in a land-based OTEC plant as evaluated by the DOE, potential installation contractors, and potential utility companies.

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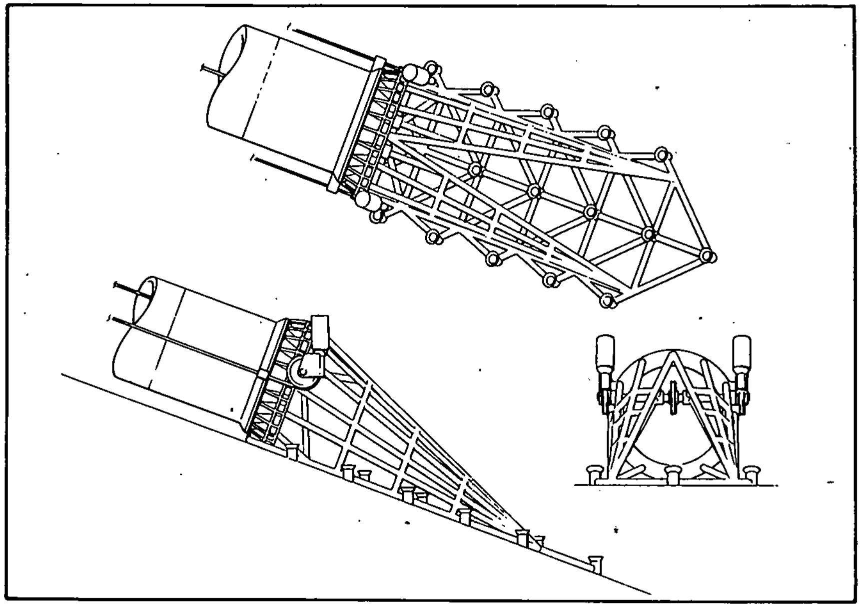

Figure 5. Deadman Template, Cold Water Intake

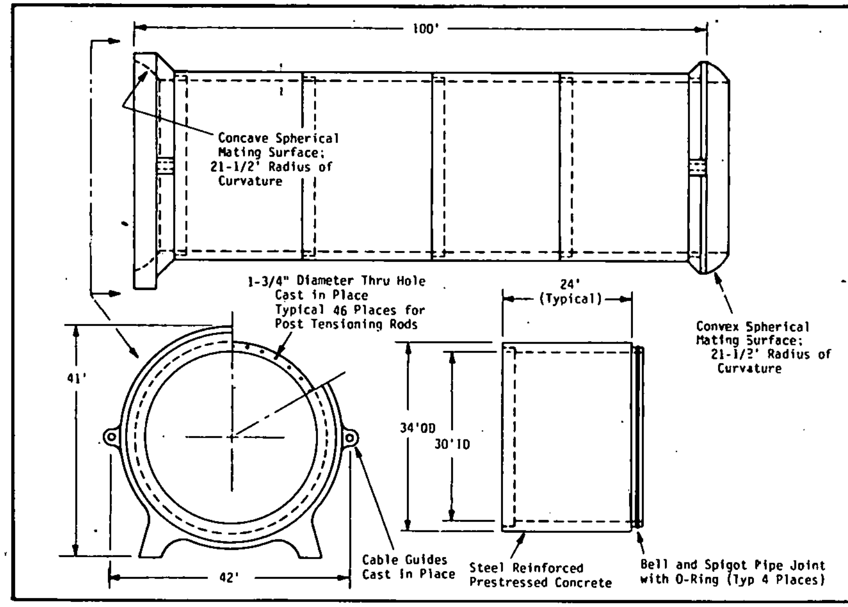

Figure 6. Typical Concrete Pipe Section

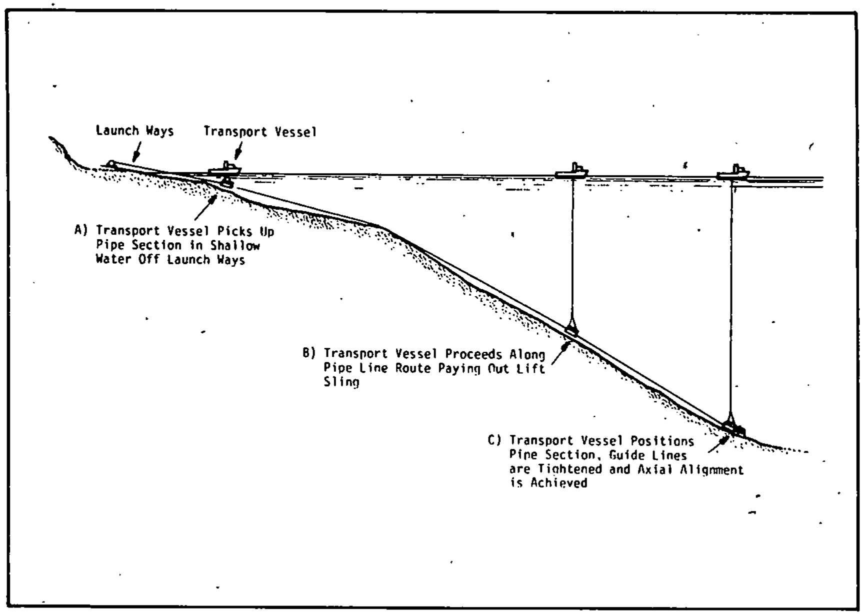

Figure 7. Transporting First Pipe Section

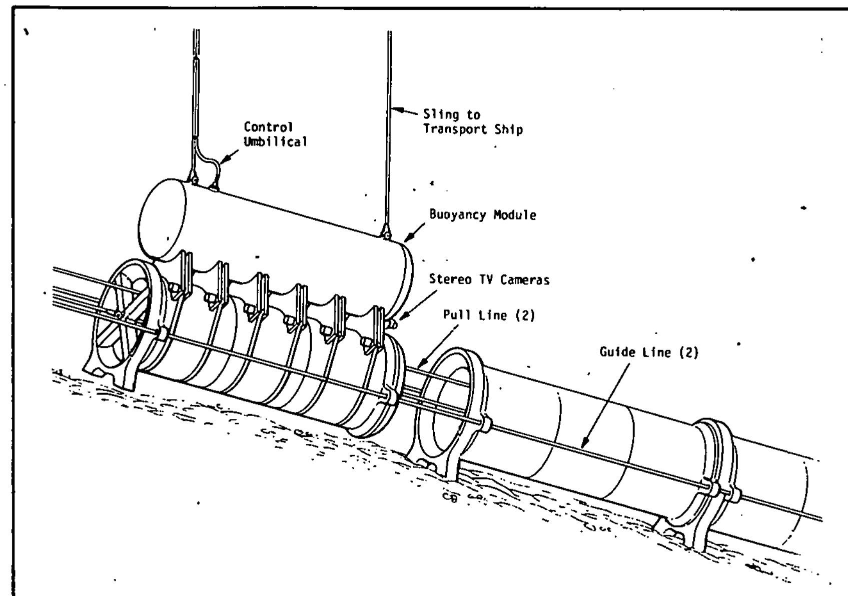

Figure 8. Pipe Section Remote Installation Technique

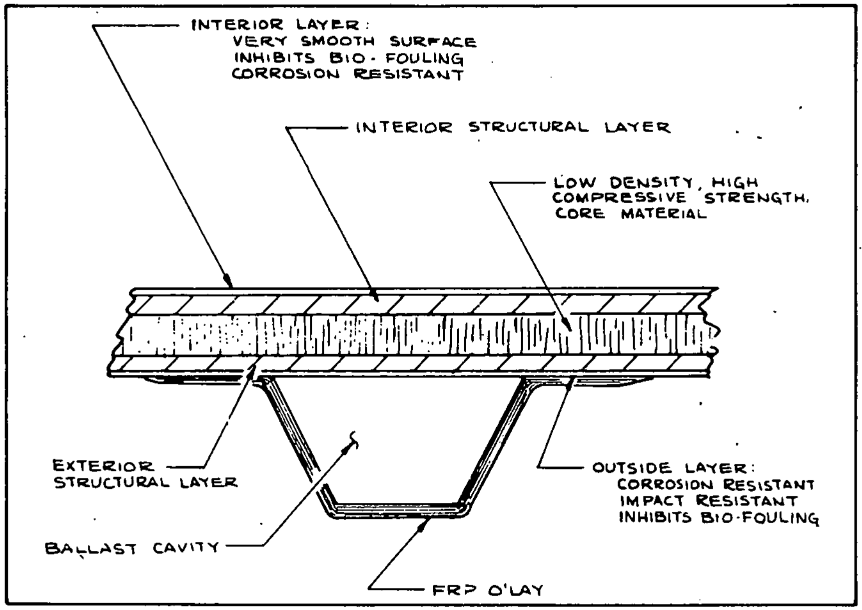

Figure 9. FRP Pipe Wall Construction

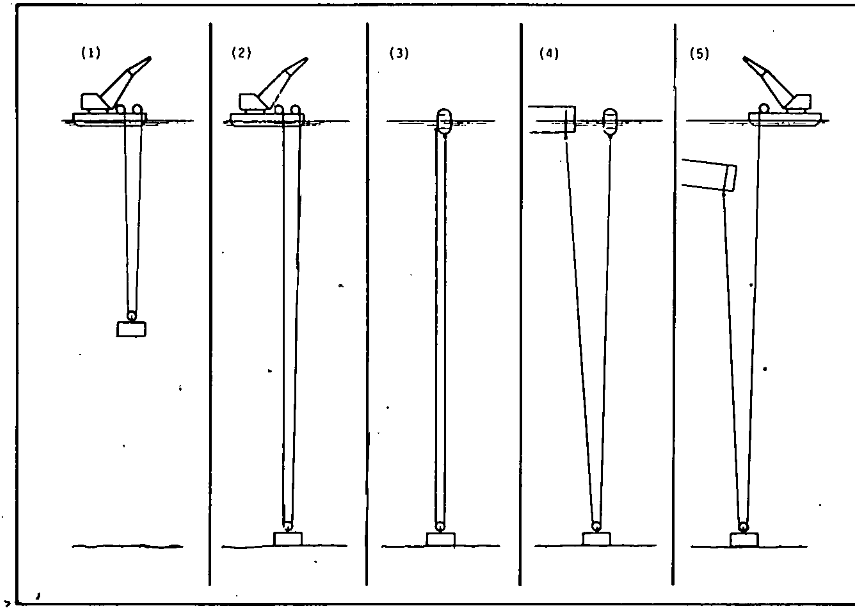

Figure 10. Deadweight Anchor Deployment Method

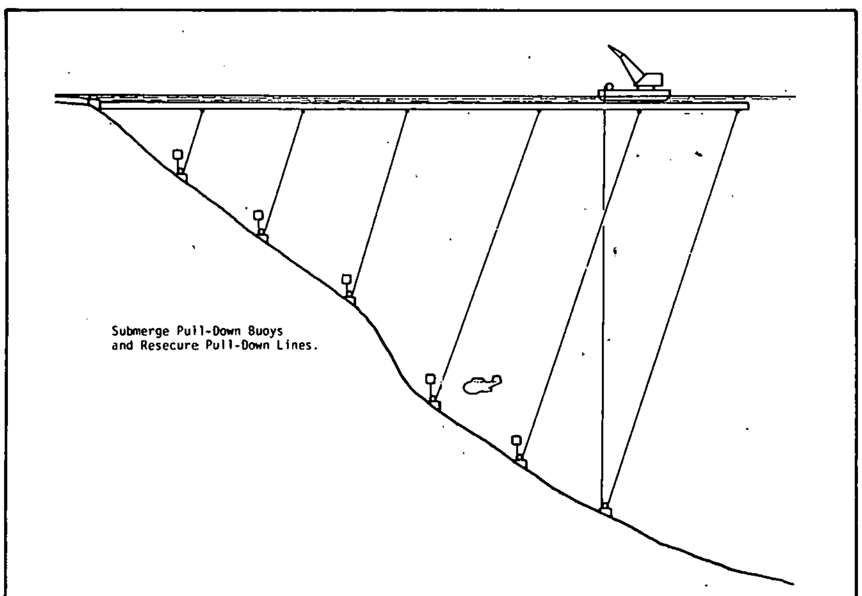

Figure 11. Buoyant FRP Pipe Installation, Step 2

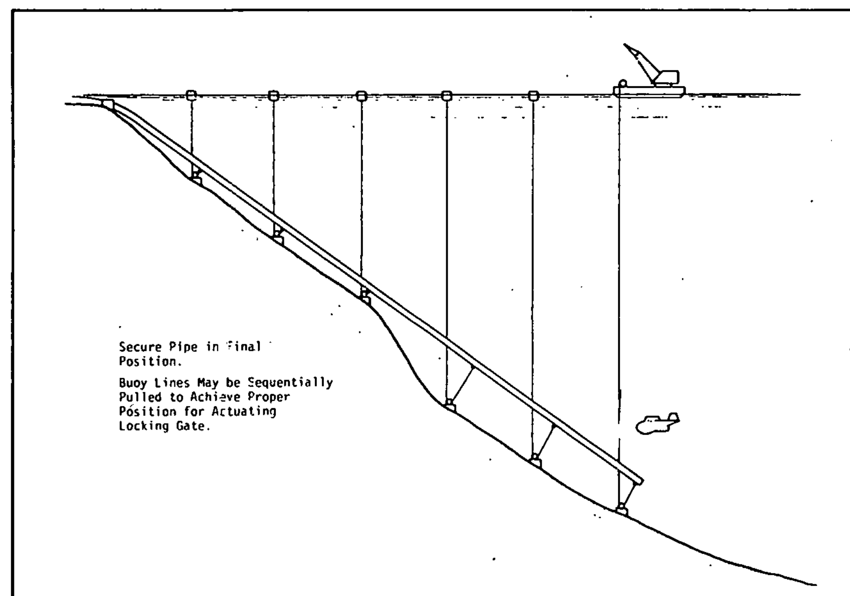

Figure 12. Buoyant FRP Pipe Installation, Step 4

6. COLD-WATER PIPES

PRELIMINARY DESIGNS OF COLD-WATER PIPES FOR BARGE- AND SPAR-TYPE OTEC PLANTS

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Abstract

In September 1978 the National Oceanic and Atmospheric Administration (NOAA) awarded parallel 1-yr contracts to two industry teams, ~~(1) TRW, Inc. (prime) and Global Marine Development, Inc. and (2) Science Applications, Inc. (prime) and Brown and Root Development, Inc.~~ to develop and compare design concepts for cold water pipes (CWP's) for ocean thermal energy conversion (OTEC) plants. The investigations have covered 4 generic CWP types (rigid with flexible joints, compliant, stockade, and bottom-mounted buoyant), 5 materials (steel, reinforced concrete, fiberglass reinforced plastic (FRP), elastomer/fabric, and polyethylene), and 3 possible 10/40-MW_e pilot plant applications (a barge-type plant-ship cruising in site Atlantic-1 east of Brazil, and both barge- and spar-type plants moored off Punta Tuna, Puerto Rico with electric cables to shore). The pipes were specified to be 30-ft (9-m) I.D. x 3280 ft (1000 m) long with 30-yr life. Each team devised a cost/risk evaluation methodology and qualitatively screened many concepts. Team 1 then studied further and rated 16 concepts; Team 2, 10 concepts. In March 1979, NOAA directed the teams to continue work on 3 concepts each: Team 1--FRP-sandwich for the cruising barge, elastomer for the moored barge, and multiple polyethylene pipes for the moored spar; Team 2--ring-and-stringer reinforced steel for the cruising barge, FRP-sandwich for the moored barge, and bottom-mounted, ring-and-stringer-reinforced steel for the moored spar, respectively. Descriptions of these 6 CWP systems and preliminary cost estimates including deployment--26, 36, 77, 62, 26, and 74 million dollars, respectively--are presented.

Introduction

Prior OTEC CWP Investigations

The history of OTEC cold-water-pipe (CWP) construction and deployment schemes began with Claude's successful deployment of a 6-ft-diam., 1-mi.-long, corrugated steel CWP off Cuba in 1929-30. His CWP did its job, in that he was able to demonstrate power generation by an open-cycle OTEC system, but then it was wiped out by a hurricane.[1] In 1955 a French government company, "Energie de Mers," successfully laid and recovered a 500-ft length of a 6.6-ft-diam. pipe off Abidjan on the Ivory Coast of Africa. The pipe comprised 20-ft-long steel sections connected by 8-ft-long flexible sections made of steel-spring-reinforced rubber. It was designed to serve a 3.5-MW_e (net), onshore, open-cycle plant, which was never completed because it proved uneconomical when compared to hydroelectric and oil-fired power plants.[2] In the 1960's the Andersons of York, Pennsylvania proposed a closed-cycle OTEC plant with a steel CWP.[3] In 1974, the University of Massachusetts proposed an aluminum CWP for a submerged, closed-cycle OTEC plant off Miami, Fla.[4] A bottom-mounted, reinforced-concrete pipe for a submerged spar platform was suggested by Carnegie-Mellon University.[5] In 1975, industrial teams investigating closed-cycle OTEC plants, headed by Lockheed and TRW, proposed a segmented, reinforced-concrete CWP for a spar-buoy platform[6a] and a fiberglass-reinforced plastic (FRP) CWP for a cylindrical surface platform,[6b] respectively. The Andersons proposed a stockade concept (a CWP comprising a circle of smaller pipes connected by welded gussets).[7]

The foregoing investigations (plus others) considered static loads only. Meanwhile, dynamic analyses of CWP loads and stresses were being formulated.[8,9] Initial results indicated that concrete or steel pipes rigidly connected to the platforms would have to be excessively thick, heavy, and expensive. Analyses of the benefits of hinging the pipe at the platform, and of adding compliant joints at intervals down the pipe, showed large reductions in dynamic bending stresses and hence in required wall thickness and weight.

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In mid-1978 The Johns Hopkins University's Applied Physics Laboratory and its subcontractors, ABAM Engineers and Tokala Offshore, began an investigation to determine the most advantageous reinforced-concrete CWP design for cruising, barge-type OTEC plants. In addition to employing multiple joints, advantages were seen in development of a lightweight concrete to reduce loads (including deployment loads) and stresses further.[10a,b]

Present Investigations

In September 1978 the National Oceanic and Atmospheric Administration (NOAA) awarded parallel 1-yr contracts to two industry teams, (1) TRW, Inc. (prime) and Global Marine Development, Inc. (GMDI) and (2) Science Applications, Inc. (SAI, prime) and Brown and Root Development, Inc. (B&RDI) to develop and compare design concepts for CWP's. The investigations have covered 4 generic types (A through D in Fig. 1), 5 materials (steel, reinforced concrete, fiberglass-reinforced plastic (FRP), elastomer/fabric, and polyethylene) and 3 possible 10/40-MW_e pilot plant applications (a barge-type plant-ship cruising in site Atlantic-1 east of Brazil, and both barge- and spar-type plants moored off Punta Tuna, Puerto Rico with electric cables to shore). The pipes were specified to be 30-ft (9-m) I.D. x 2500-3280 ft (760-1000 m) long with 30-yr life. Data on material properties, environmental design conditions, cost assumptions, work breakdown structure, and platform motions and configurations were standardized for the two teams.

Each team devised a cost/risk evaluation methodology and qualitatively screened many concepts. Team 1 then studied further and rated 16 concepts; Team 2, 10 concepts. In March 1979, NOAA directed the teams to continue work on 3 concepts each: Team 1--FRP-sandwich for the cruising barge, elastomer for the moored barge, and multiple polyethylene pipes for the moored spar; Team 2--ring-and stringer reinforced steel for the cruising barge, FRP-sandwich for the moored barge, and bottom-mounted, ring-and-stringer-reinforced steel for the moored spar, respectively (Fig. 2). NOAA selected these 6 concepts with the objective of supplying a range of baseline design data for the pilot plant(s). The following general instructions were given to TRW and SAI for all concepts:

1) Make all pipes 1000 m long to facilitate comparisons.
2) Provide fail-safe devices and procedures for recovery of pipe.
3) Detach CWP from platform for routine maintenance in benign seas.
4) Maintain pipe integral to platform during hurricanes.
5) Assess the impact on pipe design of detaching and maintaining the CWP near station for 20-ft-significant-wave conditions.
6) Define effects of biofouling on the outside of the CWP over its length and life and resolve, if necessary, for each CWP concept.
7) Formulate a development plan for each concept including details of the pipe's fabrication, transportation, deployment, and operation.
8) Conduct a life-cycle-cost analysis from design through development and operations over a 30-yr life.

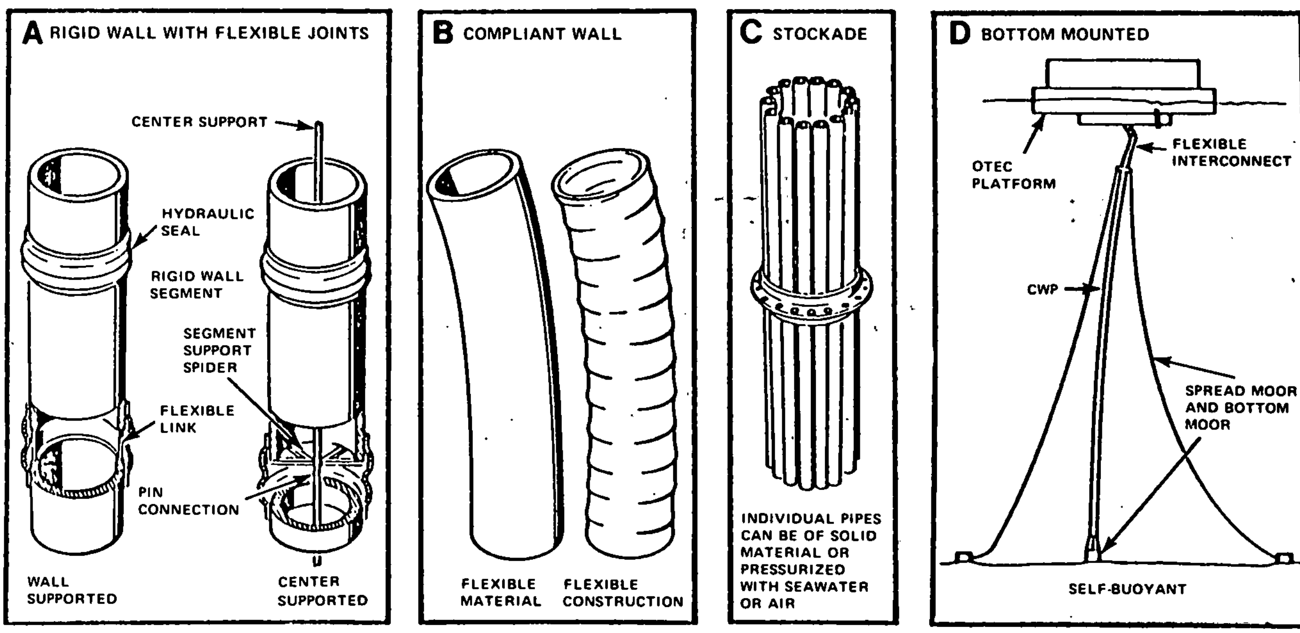
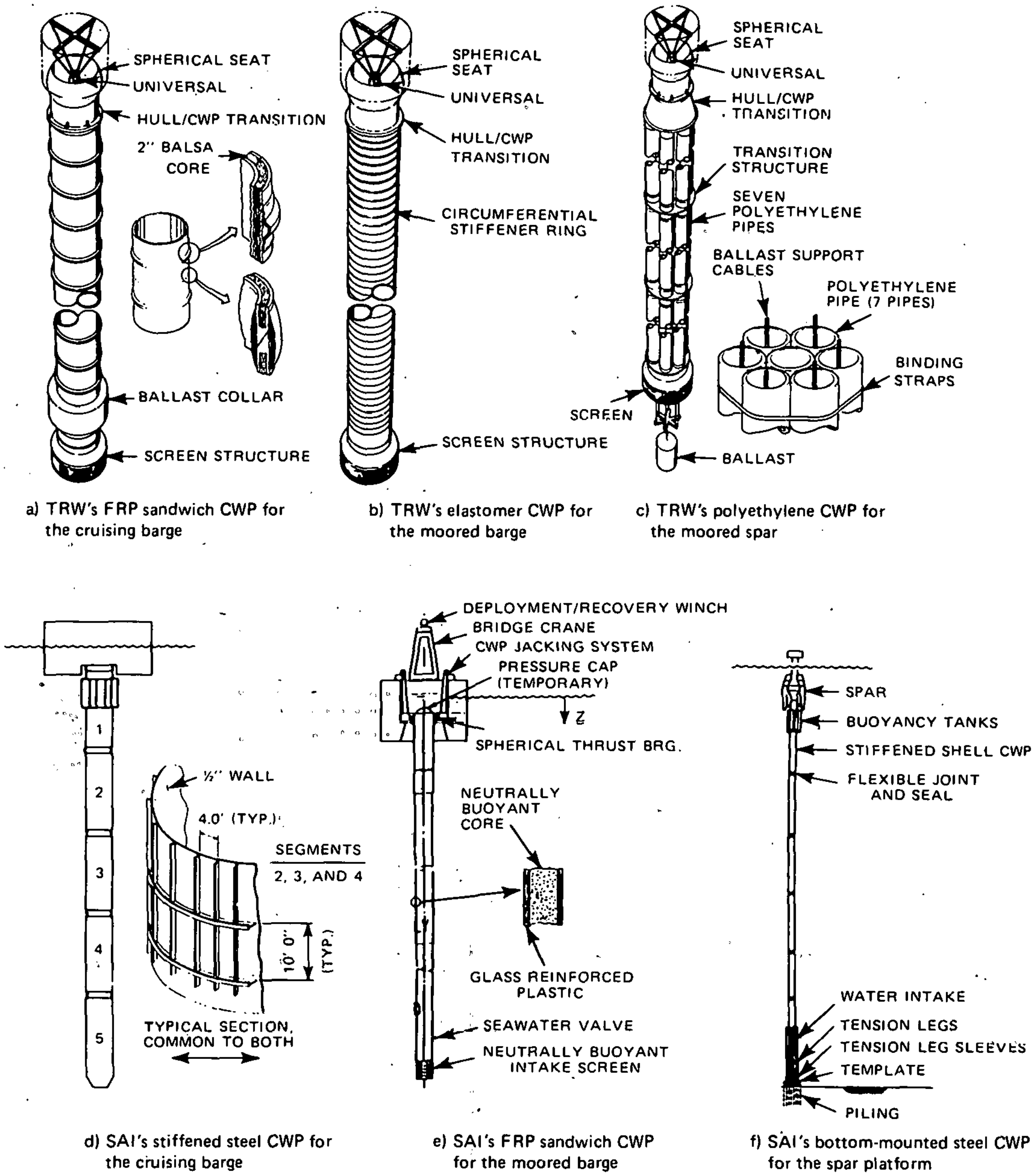


Fig. 1 Generic types of cold water pipe (CWP) configurations shown in the request for proposals (RFP) issued by NOAA.

The methodologies, results and conclusions of the TRW/GMDI team are presented first, followed by those of the SAI/B&RDI team. It will be shown that two types of CWP's investigated in some depth by these teams--FRP sandwich and elastomer--are estimated to have deployed costs in the $26-36



a) TRW's FRP sandwich CWP for the cruising barge

b) TRW's elastomer CWP for the moored barge

c) TRW's polyethylene CWP for the moored spar

d) SAI's stiffened steel CWP for the cruising barge

e) SAI's FRP sandwich CWP for the moored barge

f) SAI's bottom-mounted steel CWP for the spar platform

Fig. 2 The six CWP configurations selected by NOAA for the final investigations by the contractors.

M range. Steel pipes cost more, but a bottom-mounted configuration at $74M could prove attractive because it would also serve the mooring function (worth about $25 M for Puerto Rico or Gulf of Mexico sites) and would alleviate problems with the riser cable of the cable-to-shore transmission system.

Work by the TRW/GMDI Team

Preliminary Screening of Concepts[11]

During the proposal effort this team developed 67 CWP concepts from the list of materials and wall types stated in the request for proposal (RFP) plus built-up wall types (sandwich, honeycomb, etc.), -ossible use of articulating joints, and in some ases use of a central member to support the weight f the pipe. These concepts were screened by a oard of senior technical personnel, who eliminated many of them either by finding a major technical problem or a design/ material incompatibility or by comparing two similar concepts and deciding on one.

The 16 concepts selected for further analysis are shown in Fig. 3, arranged by the wall types suggested in the RFP. Because of the promise and/ or interest engendered by the FRP, concrete, and steel concepts, much of TRW's overall effort was spent in analyzing these concepts. Analysis of the elastomer CWP's was more difficult because of the proprietary nature of the material and fabrication techniques used by the companies associated with them, but progress was still possible.

Ranking of 16 Concepts

Figure 4 illustrates the methodology employed to

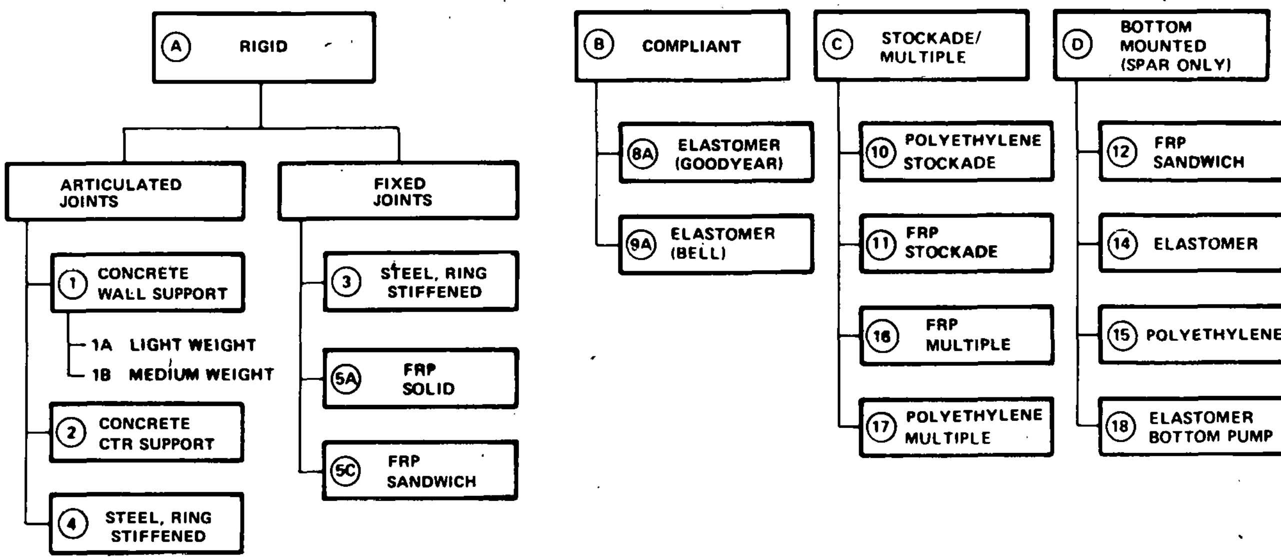


Fig. 3 The 16 pipe concepts selected by the TRW/GMDI team for further investigation.

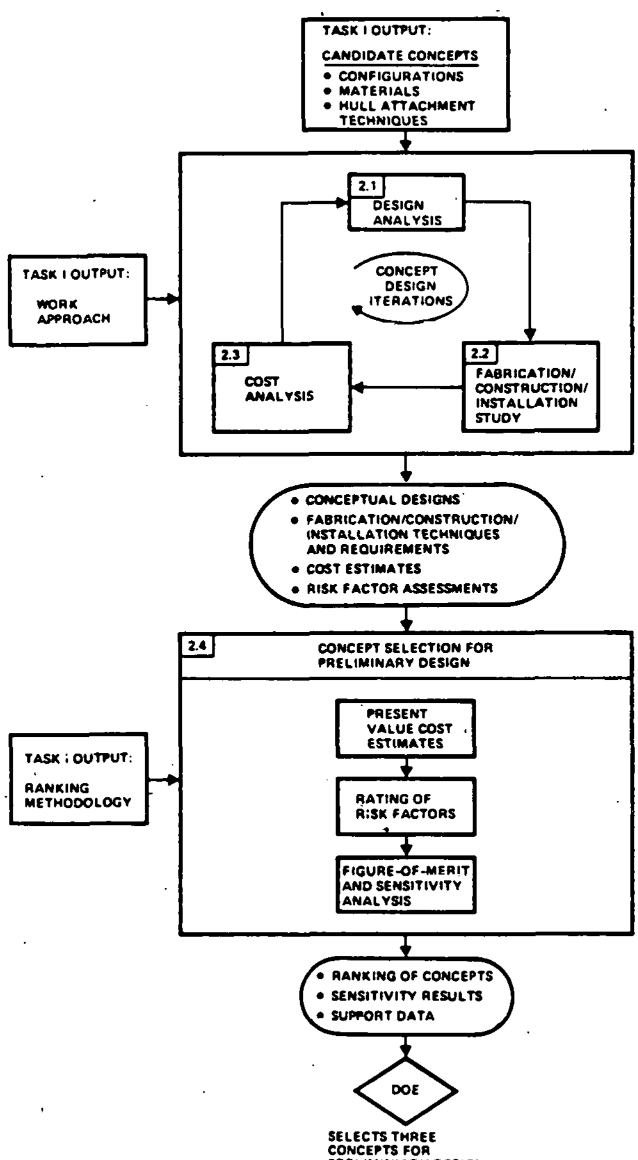
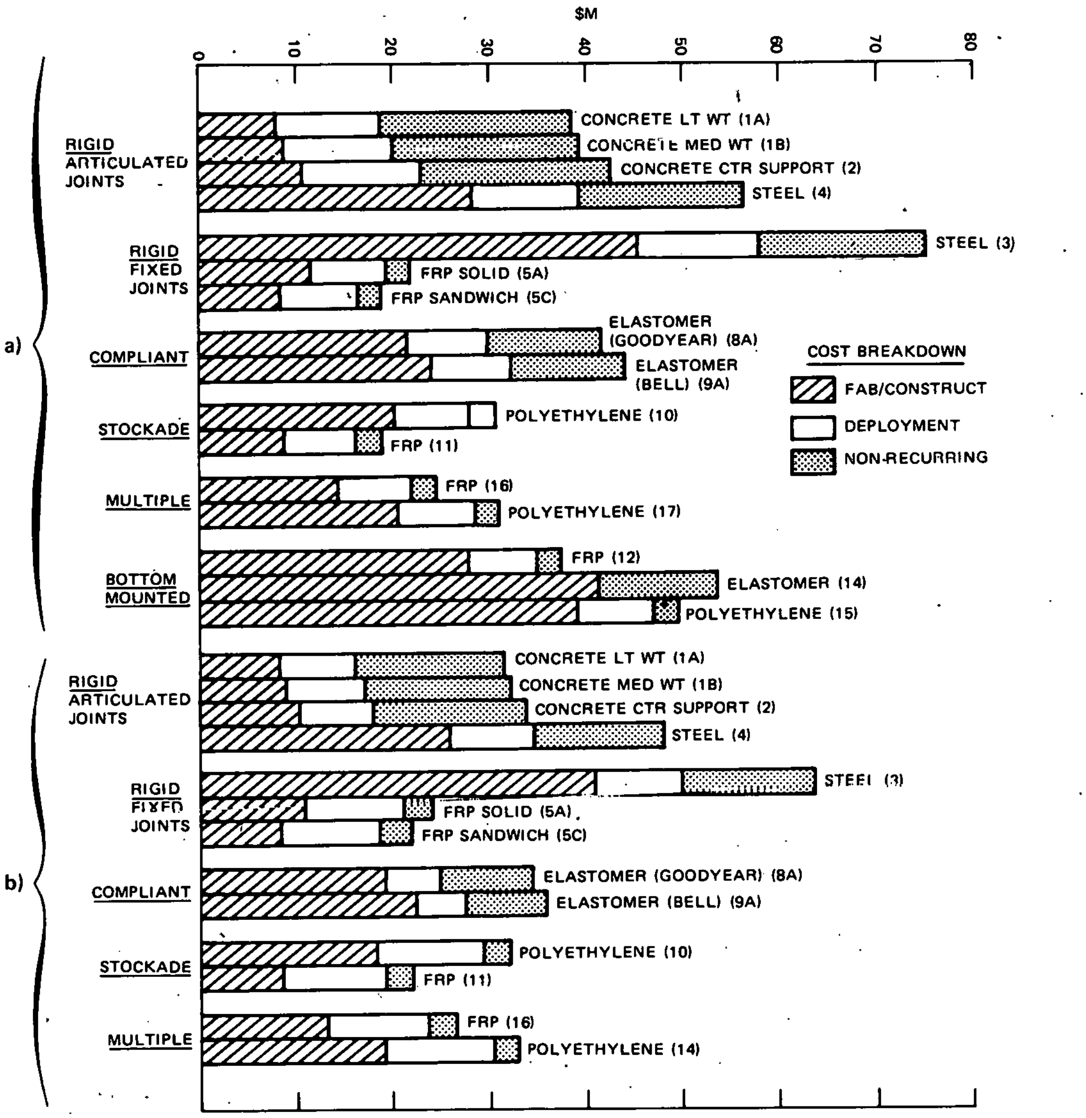


Fig. 4 TRW's methodology for concept selection.

rank the 16 concepts. Following a technology review, formulation of system requirements, and generation of an evaluation methodology, the analysis began. First, hydrodynamic, dynamic-response, and structural analyses were performed.[11] Then the CWP-hull joint was investigated, and a fabrication/construction/installation study and cost/risk analyses were conducted. Where necessary, iterations were conducted until a self-consistent set of data had been developed for each design. Preliminary estimates of costs for the designs analyzed for the Puerto Rico spar and South Atlantic barge applications are presented in Fig. 5 in terms of recurring fabrication/construction costs, deployment costs, and the non-recurring facility and/or tooling costs which would be reduced if more than one pipe were to be built.

The recurring fabrication/construction and deployment costs for the first two concrete and the first three FRP types are lowest, in the $16-22 million range. The non-recurring costs estimated for the concrete types are considerably higher than those for FRP types. The material costs are higher for the other types. The ring-stiffened steel pipes, with or without articulating joints, are the most expensive because of the enormous amount of steel required to withstand the loads imposed on the pipe by the environment and because of the high cost of corrosion protection. The costs for bottom-mounted pipes are also high because of the expenses of the compliant, flexible, CWP-hull coupling required.

Figure 6 illustrates the risk rating system which was applied to each of five areas: wall design, joint design, hull attachment, fabrication/construction, and deployment. Figure 7 shows the average risk ratings. On the 1 to 10 (extreme risk to no risk) scale, none of these concepts is rated below 4 because the initial screening had eliminated the highest risk concepts, and none is rated above 8 at present because there is no 30-ft-diam, 3280-ft-long CWP that can be built in the near future without some development effort. (Some of the required effort is already under way, and more will be done in 1980.[10c]) The



a) MOORED SPAR PLATFORM AT PUERTO RICO   b) CRUISING BARGE PLATFORM AT SOUTH ATLANTIC SITE -1.

Fig. 5 TRW's initial comparisons of estimated costs through deployment for two platform-site combinations.

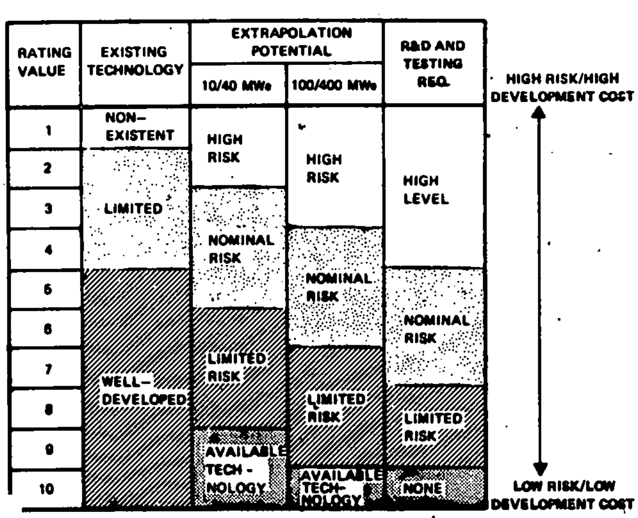


Fig. 6 The technical risk assessment approach.

FRP pipe with a sandwich wall with balsa wood (or a syntactic foam) for the core material is judged to incur the lowest overall risk, followed by the steel pipe without joints and a cable-reinforced, ring-stiffened elastomer pipe. Because of the 30-yr-life requirement, all concepts requiring articulating joints are judged to be of relatively high risk. However, use of the joints does substantially reduce stresses and hence pipe weight. The polyethylene pipes are considered to represent higher risk than the FRP pipes because less is known about the mechanical properties of polyethylene.

In the evaluation process, cost and risk rankings were considered together using varying degrees of emphasis on one or the other in a vector addition scheme (resultant vector length when the vector of a risk parameter is measured on an abscissa scale and the vector of a deployed cost parameter is on an ordinate scale. The best system is the one with the shortest resultant vector.)

Table 1 lists the ten highest ranked concepts. The top five are FRP/balsa sandwich, solid FRP, cable-reinforced, ring-stiffened elastomer (Goodyear Aerospace Corporation type) and two more FRP

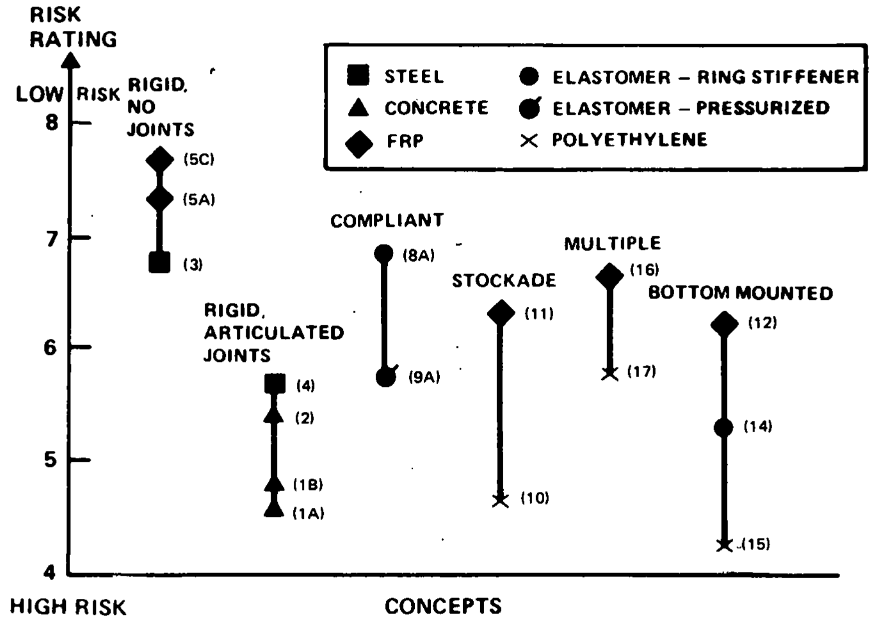


Fig. 7 Results of TRW's evaluation of averaged risks.

concepts. The medium-weight concrete (100 lb/ft^3 dry density compared to APL's 85 lb/ft^3) and steel pipe concepts were ranked eighth and ninth, respectively.

Preliminary designs of the three concepts shown in Figs. 2a-c were completed in October 1979. Brief descriptions of each concept and its deployment scenario follow, then estimated costs are compared.

TRW/GMDI's FRP/Balsa-Core Pipe (Fig. 2a)

Fabrication. The fiberglass comprises E-glass rovings and a resin matrix, 1:1 by volume or 60/40 glass/resin ratio by weight. A 1:1 ratio of axial

Table 1 TRW's ranking of CWP concepts based on first unit cost and risk

Rank	Category	Material	Design features
1	Rigid	FRP/(balsa)	Sandwich wall
2	Rigid	FRP	Solid wall
3	Compliant	Elastomer	Cable-reinforced, ring-stiffened (Goodyear)
4	Stockade	FRP	Pultrusions
5	Multiple[a]	FRP	Six 12-ft-diam. pipes
6	Compliant	Elastomer	Pressurized double wall (Bell)
7	Multiple	Polyethylene	Seven 12-ft-diam. pipes
8	Rigid	Concrete	Articulated joints
9	Rigid	Steel	Welded joints
10	Bottom-mounted	FRP	Sandwich wall

[a]Special category considered by TRW/GMDI - not in RFP.

(0°) to $\pm$ 60° layers is used. The ultimate tensile strength after 30 years (10^7 loading cycles) in seawater is 20% of the initial value. The compressive strength degrades very little. The balsa (or syntactic foam) core provides the buckling strength as well as providing slightly positive buoyancy to the pipe. Sections of 60-ft length are made by a) wet-winding a 0.79-in-thick inner FRP wall on a horizontal, rotating mandrel, b) laying up a 2-in.-thick, end-grain balsa core in panels that are backed with fiberglass mats having a finish compatible with the impregnating resin of the FRP walls, c) wet-winding a 0.79-in.-thick FRP outer wall over the core and d) wet-winding an FRP stiffening ring, essentially semicircular in cross-section and 8 in. high, at the midpoint of the 60-ft-long section. Upon completion of the section the mandrel is collapsed inward to release the section (an existing technique).

The sections are then placed in cradles on rail cars at the seaside assembly site and are joined by adding inner and outer wraps over a 32-in. length, with a maximum thickness of 1.0 in. each at the joint location.

Deployment. Mobilization of equipment and vessels is done at New Orleans because of the greater certainty of obtaining the required vessels in sufficient numbers there. The tugs, barges, and work boats are sent to the seaside assembly site (Salvador, Brazil), where the upper end of the CWP is placed on the fantail of the tow/work boat and made fast. The lower end of the CWP is placed on a barge which has a lowering winch. Weather prediction service is contacted, and at the appropriate "window" time, the tow begins.

At the deployment site the tow/work boat pulls alongside the cruising-barge-type platform. The keelhaul lines from the platform are attached to the upper end of the CWP, which is placed in the water with the load supported by a towing winch on the work boat. A cementing hose is attached to the ballast collar on the lower end of the CWP, which is on the barge with the lowering winch. The lower end of the CWP is placed in the water and is allowed to fill with water, and the bottom of the CWP is lowered at a rate of 10 ft/min. When the CWP is vertical, the lowering cable is released.

The work boat lowers the CWP to a predetermined depth. As strain is taken on the keelhauling cable from the platform, the work boat pays out the support cable. When the CWP is directly under the platform, the work boat cable is released. Underwater television cameras and divers observe and direct the procedure by which the CWP's upper end is hauled up through the moonpool, clear of the water. The crane on the platform picks up the CWP/hull transition piece and holds it over the upper end of the CWP so that the flange can be bolted to the transition piece. The CWP is lowered into brackets in the hull which accept the upper spider assembly, and divers attach the seal structure and seal shoes. Then the ballast collar is filled with cement, and the cementing hose is detached. The support vessels then return to New Orleans.

Costs. The costs of the FRP pipe are summarized in the first column of Table 2. The estimated cost through deployment is $25.6M. The additional costs during 30 years of operation (shown below the dashed line in Table 2) are small, because TRW considers this FRP pipe (and the elastomer and polyethylene pipes) to be essentially maintenance-free.

TRW/GMDI's Elastomer CWP Based on the Goodyear Concept (Fig. 2b)

Fabrication. The compliant wall membrane for the pipe will be made from 6-ft wide standard conveyor belting modified by the use of a 1/4-in.-thick neoprene matrix on both sides of the longitudinal steel reinforcing cables. The present design uses 1/2-in.-diameter cables spaced on 3-in. centers. These belts are obtained in continuous lengths and will be joined longitudinally in 75-ft-long tubular sections by vulcanizing at the manufacturer's so the sections will have reinforcing cables in the pipe's axial direction (Fig. 2b). The cable ends will terminate in circumferential steel rings by means of bolted swaged eyes, or shackles, or captured swaged balls. To the outer pipe wall membrane, circumferential stiffener tubes will be attached with through-bolts attached to internal steel backing plates. The circumferential stiffener tubes will be placed at 6-ft intervals along the length of the pipe. The stiffener tubes have a 7.25-in. O.D. with a 0.25-in. wall at the top of the CWP and will vary at cross-section intervals down the length of the pipe to a 5.25-in. O.D., 0.188-in. wall tube at the bottom of the pipe.

Deployment. As in the scenario for the FRP pipe deployment, the necessary vessels are mobilized at New Orleans and moved to the shore assembly site, where the first half (22) of the 75-ft CWP sections are loaded onto the transit barge. This barge is towed to the deployment site and tied up to the barge platform, whose crane off-loads a few CWP sections. The transportation barge and tug stand by as the barge is used to stow sections during deployment.

The first section of CWP is placed into the deployment structure by the platform crane. The slipjacks are in the up position, and the hydraulic rams on the slipjack ring are extended to support the CWP at the upper circumferential ring. The slipjacks are lowered 75-ft until the hydraulic jacks attached to the platform hull take the weight of the deployed section(s) from the slipjack ring. The slipjacks are raised 75-ft to accept the next section of CWP, which is picked up by the platform crane.

Table 2 Estimated Costs (Millions of 1978 Dollars) for the Final Six CWP Concepts

Team:	TRW/GMDI						SAI/B&RDI					
CWP type:	FRP,		Elastomer,		Polyethylene,		Steel,		FRP,		B-M steel,	
Platform:	Cruising barge		Moored barge		Moored spar		Cruising barge		Moored barge		Moored spar	
Site:	S. Atl.		P.R.		P.R.		S. Atl.		P.R.		P.R.	
Cost Item / Figure:	2a		2b		2c		2d		2e		2f	
Acquisition and Deployment												
Management		1.2		1.8		3.5		1.9		1.9		1.9
Prelim./contract design		2.3		3.3		6.7		3.4		3.4		4.0
Eng'g des./constr'n/deploy't		22.1		31.2		67.0		56.2		21.0		68.2
Engineering design	0.4		0.6		1.3		3.0		3.0		3.0	
Construction facilities	5.1		10.5		10.9		3.1		4.8		3.5	
Construction of:												
Pipe system	10.8		12.6		44.9		24.3		7.9		28.6	
Screen	0.1		0.1		1.2		3.8		0.6		1.8	
CWP/hull transition	1.9		2.0		2.7		1.2		2.5		1.2	
Systems for biofouling & corrosion control	0.0		2.8		0.7		3.0		0.5		3.5	
Mooring, flotation	0.2		0.0		0.2		3.0		N/A		4.5	
Acceptance testing	0.2		0.3		0.7		1.1		0.4		1.2	
Deployment	3.4		2.3		4.4		13.7		1.3		20.9	
Subtotal, Acquisition & Deployment		25.6		36.3		77.2		61.5		26.3		74.1
System Operation & Support		1.2		1.8		3.5		46.0		18.9		60.1
General management	0.1		0.2		0.3		1.1		1.1		1.1	
Offshore support	1.1		1.6		3.2		28.4		8.0		37.3	
Tests	--		--		--		2.5		2.5		2.5	
Refit	--		--		--		8.2		6.8		9.2	
Disposal	--		--		--		5.8		0.5		10.0	
Life Cycle Cost		26.8		38.1		80.7		107.5		45.2		134.2

The inside doubler plate ring segments are attached by a few bolts to hold them in place to the bottom of the section to be attached. The crane places the section in the deployment structure, and the slipjack rams are extended again to support the upper-circumferential ring. The slipjacks are moved to align the position of the lower end of the section to be attached with the deployed section(s) of the CWP. The outer doubler plates are added, and the two sections of the CWP are joined. After all of the bolts are torqued, the air gap formed within the doubler plates is potted with sealant from three locations 120° apart. The slipjacks are raised to take the weight of the deployed CWP. The lower rams are retracted and the slipjacks lower the assembled CWP sections 75 ft until, again, the lower rams can take the deployed load.

The foregoing sequence is repeated until the total CWP is assembled. The tug then tows the barge back for the second load of CWP sub-assemblies. When the total length of the CWP is deployed, the CWP/hull transition section is positioned above the CWP and is attached to it by the same method that was used to attach sections of the CWP. The transition with the attached CWP is lowered into brackets in the hull which accept the upper spider assembly and support the deployed CWP. The spider assemblies are attached to the hull, and divers attach the seal structures and the seal shoes. The deployment vessels return to the shore assembly site and thence to New Orleans for demobilization.

Costs. The costs of the elastomer CWP are shown in the second column of Table 2.

TRW/GMDI's Multiple Polyethylene Pipe (Fig. 2c)

Fabrication and deployment. The polyethylene CWP assembly comprises 7 pipes of 12-ft-diameter, each of which is assembled from three section of 1068-ft length including a 14-ft-long transition/joint section at each end. The wall thicknesses are 5.88, 5.5, and 4.88 in., respectively, for the top, middle, and bottom sections. These long sections are made by fusion-welding together 50-ft-long sections which are made by helically winding high-strength, extruded polyethylene onto a heated, rotating, collapsible mandrel. The substrate also is heated externally as the winding is applied. The assembly of the pipe is done in a partially flooded graving dock.

The deployment procedure is similar to that for the FRP pipe, except that when the tow/work boat pulls alongside the spar platform, off the Guayama, Puerto Rico shore assembly site, the keel-haul lines are attached to the transition piece. When the CWP has been keel-hauled under the spar, it is raised up to mate, under water, with the transition attached to the platform, with the underwater TV and divers watching.

Costs. The costs of the polyethylene pipe are shown in the third column of Table 2.

Work by the SAI/B&RDI Team

Screening and Concept Selections[12]

The flow charts employed for concept selection and definition by SAI and B&RDI are shown in Figs. 8 and 9. In the conceptual design phase, government-furnished information was reviewed and candidate baseline concepts were developed in the RFP categories. Design studies were initiated to define subsystems and perform selected sizing analyses. Vendors were contacted to establish material properties and costs. Fabrication/construction/installation plans were developed. A qualitative screening eliminated unworkable systems and procedures. Remaining uncertainties and design issues were addressed through design studies until the system concept was judged to meet the system requirements on a best estimate basis.

The ten concepts remaining after the qualitative screening are identified in terms of RFP category and subsystem in Table 3. Some subsystems are common to each design (e.g., intake screens, not shown in the table), while others require variations, depending on configuration (mooring the platform through the CWP) or pipe weight (bearings). (Stockade pipes were not considered because of the relative difficulty of construction and low fatigue life.) Finally, detailed information was developed for successful candidates, and a quantitative evaluation was performed.

Risks were evaluated by applying statistical decision theory to key parameter uncertainties. Concept ranking was then done on the basis of the probability of meeting technical and cost goals.

An example of the process is illustrated in Fig. 10. Nominal cost estimates for each WBS element are accompanied by low and high estimates, along with a measure of the confidence that these values are not

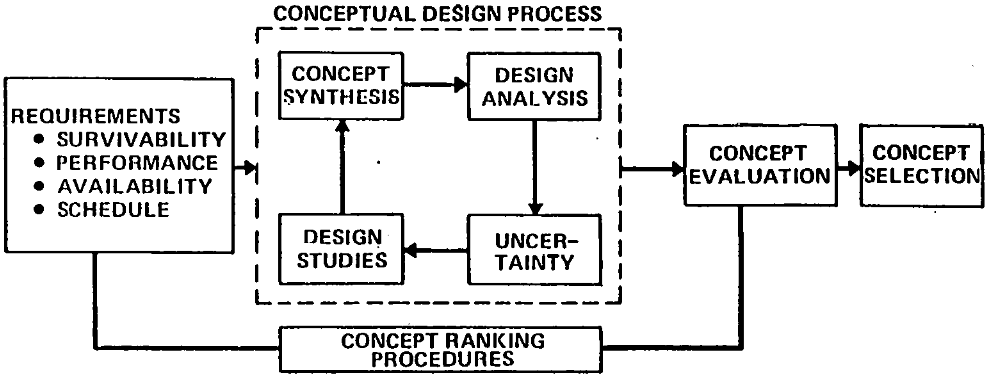


Fig. 8 SAI's concept selection methodology.

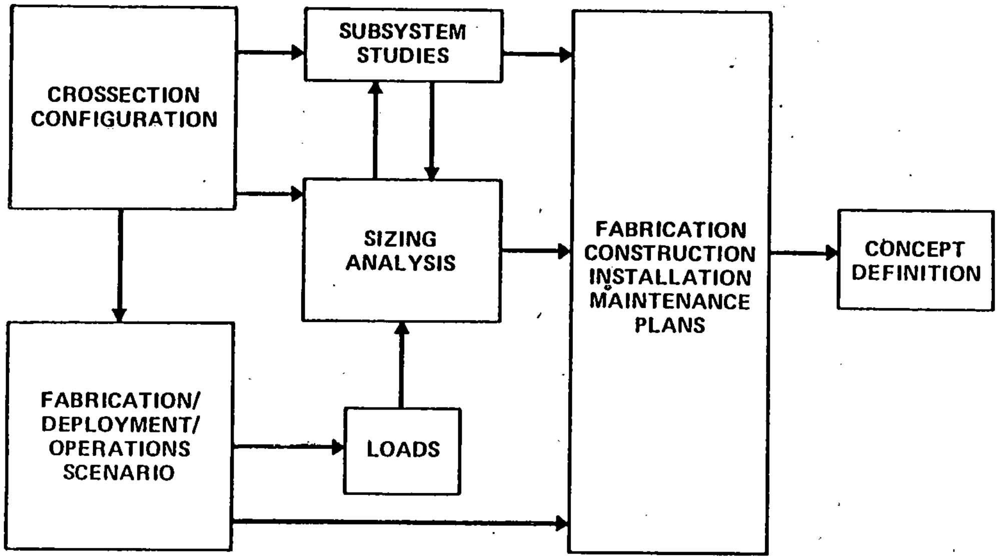


Fig. 9 SAI's concept definition methodology.

Table 3 Baseline concept alternatives evaluated by SAI/B&RDI

	Rigid Category					Compliant Category			Bottom-mounted buoyant	
Material	Steel	Steel	Concrete	FRP	FRP	Elastomer	Fabric	Plastic	Steel	FRP
Deployment[a]	V	V	V	H	H	V	V	H	V	H
Mooring[b]	--	--	--	--	--	--	--	--	1	2
Bearings[b]	1	1	2	3	3	--	--	3	4	3

[a]H = horizontal tow and keel-hauled; V = vertical, through platform.
[b]Numbers indicate different approaches.

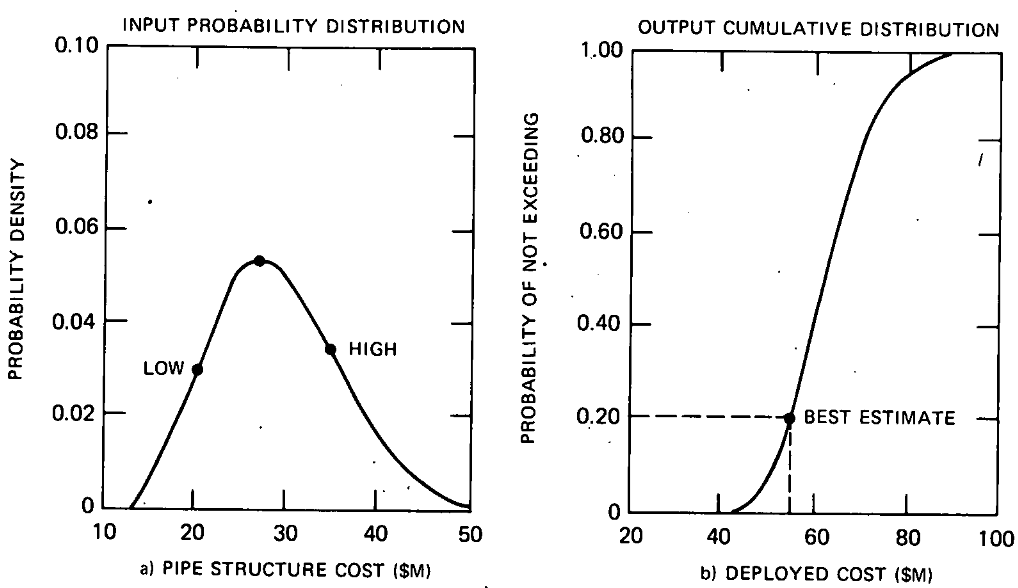


Fig. 10 Example analysis for cost through deployment.

exceeded. A probability density function (here, Weibull) is fitted to these input data, and a Monte Carlo routine is used to sum the uncertainties. The result is a cumulative probability distribution which describes the uncertainty in the cost data for a particular CWP concept. Note that the distribution is skewed with a high probability (0.8) that the nominal cost estimate will be exceeded.

Evaluation Results

The 10 candidate concepts were first evaluated within their respective categories and then the top two concepts in each category were evaluated together. Top ranked candidates in the rigid wall category were a ring-and-stringer (tee) reinforced steel (hereinafter designated R&SR steel) design and a corrugated-wall fiberglass design. In the compliant wall category, the favored concepts were a polyethylene design and an elastomer design. Favored candidates in the bottom-mounted buoyant category were a slack-moored FRP design and a taut-moored R&SR steel design.

Concepts were ranked in terms of cost through deployment (Fig. 11), schedule, and technical feasibility (Fig. 12) criteria using the scheme previously indicated in Fig. 6. The ranking procedure was to order the concepts in terms of probability of a deployed cost less than $50M, probability of having a technical feasibility factor greater than 5 (i.e., low risk), and a combined probability of meeting these technical and cost goals. Results are summarized in Table 4.

Based on these evaluations, the SAI/B&RDI team was directed by NOAA to prepare preliminary designs of the following three designs depicted in Figs. 2d-f, briefly described as follows.

SAI/B&RDI's Stiffened Steel CWP (Fig. 2d)

The stiffened steel CWP is designed to be attached to a cruising barge to operate off the coast of Brazil. This pipe comprises five vertical segments with flexible joints between them. The top segment is 700 ft long and the other four are 645 ft long. The basic cross section of each segment has a diameter of 32.3 feet and an equivalent wall thickness--taking into account the longitudinal and ring stiffeners--of 1 in. in the top and bottom segments and 0.8 in. in the three middle segments. The overall weight (in air) of this design is 14.25 million pounds. The pipe will be assembled into approx. 100 ft sections on shore and transported to the site on barges. To deploy it, a derrick will lift the sec-

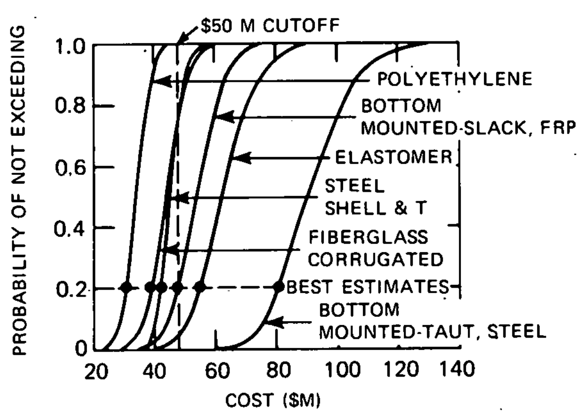


Fig. 11 SAI's initial estimates of cost through deployment.

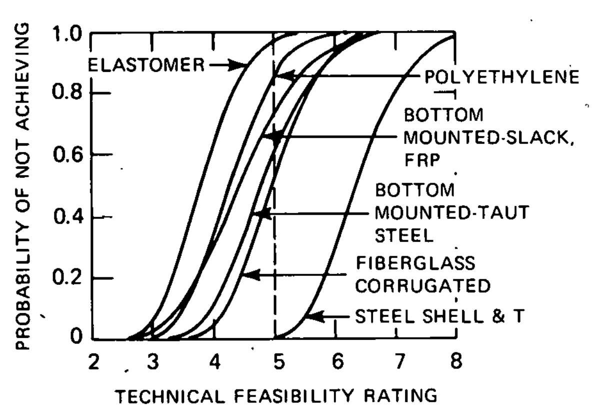


Fig. 12 SAI's evaluation of technical feasibility.

Table 4 Concept ranking by SAI

Rank	Ranking Criteria[a]			
	P_S(Cost < \$50 M)	P_S(T < 48 months)	P_S(TF > 5)	P_S(Cost < \$50 M)(TF > 5)
1	Polyethylene (30)[b]	R&SR steel	R&SR steel	R&SR steel
2	R&SR steel (42)	Polyethylene	FRP-sandwich	FRP-sandwich
3	FRP-sandwich (38)	Elastomer	BM-R&SR steel, taut	Polyethylene
4	BM-FRP-slack (47)	BM-R&SR steel, taut	BM-FRP-slack	BM-FRP-slack
5	Elastomer (55)	FRP-sandwich	Polyethylene	Elastomer
6	BM-R&RS steel, taut (80)	BM-FRP-slack	Elastomer	BM-R&SR steel, taut

[a] P_S = probability of success; T = time required; TF = technical feasibility (see Fig. 6); R&SR = ring-and-stringer reinforced; BM = bottom mounted

[b] Numbers in parentheses are initial estimated costs through deployment (Fig. 11)

tions off the transporter barges and align each vertically for welding to a previous section supported by a buoyancy tank. This procedure is repeated until the five segments are assembled. Costs are shown in the fourth column of Table 2. The estimated cost through deployment, \$61.5M, is somewhat greater than the \$50M target cost mentioned earlier. The estimated additional costs for system operation and support over a 30-yr life include substantial allowances for corrosion control (cathodic protection), inspections and tests, repairs and refitting.

SAI/B&RDI's FRP Sandwich (FRP) Pipe (Fig. 2e)

The SAI FRP pipe is designed for attachement to a moored barge at the Puerto Rico site. The pipe, 30 ft in inside diameter, is monolithic. The inner and outer filament-reinforced skins, each 1/2-in. thick, are separated by a low-density polyester core with a thickness of 6.5-in. near the top, tapering in steps down to 4.5-in. at the bottom. The structure has slightly negative buoyancy. However, if filled with a suitable mixture of fresh and salt water, the pipe becomes neutrally bouyant for towing to the operational site. No bottom weight is required. The overall weight of the CWP system (in air) is 12.2 million pounds. Deployment of this pipe is initiated by holding the upper pipe at the platform and allowing seawater to enter the pipe bottom slowly. The increasing weight causes the pipe to rotate into the vertical position. Costs are shown in the fifth column of Table 2.

SAI/B&RDI's Bottom-Mounted, Stiffened-Steel Pipe (Fig. 2f)

The SAI bottom-mounted CWP system was designed for attachment to a spar buoy at the Puerto Rico site. This design combines into one structural element the CWP function and the mooring function, hence it has an additional function when compared to the other five CWP designs. The 14.2-million-pound, R&SR steel pipe and its deployment are the same as previously described for the cruising barge. In addition, however, before deployment of the pipe, the bottom mount must be installed. Seventeen 2-ft-diameter pipe piles, drilled through a template to a depth of 200 ft and grouted, must be emplaced.

Costs for this CWP/mooring system are shown in the sixth column of Table 2. Since the mooring function could cost \$25 M or more if done separately, this approach may be a cost-competitive contender for use on spar platforms.

Concluding Remarks

These investigations by the TRW/GMDI and SAI/B&RDI teams have led to estimated costs through deployment for six types of CWP's as summarized in Table 5. The costs for FRP types for installation on barge-type OTEC platforms by the two teams are the lowest and are in good agreement at approximately \$26M. The elastomer pipe installed on a barge is next lowest in cost (\$37M). The other pipes cost more, but none should be ruled out at this stage. Costs of maintenance over the lifetime (herein specified as 30 years) of the pipes' use are being estimated and will also have to be taken into account by contractors proposing OTEC pilot plants or pilot plantships. While first estimates of these costs have also been indicated earlier in the lower part of Table 2, they may be subject to larger revisions than the acquisition and deployment costs upon more thorough investigation. It should be noted, too, that the useful life of any 10- to 40-MW_e pilot plant or plantship may be only 10 years or so, because larger and more cost-effective commercial plants would be expected to follow successful pilot demonstrations. Such commercial plants will use the relevant cost data developed by the pilot operations but may

Table 5 Summary of features of CWP concepts chosen by NOAA for preliminary design

Material/ cross-section	Category	Plat- form	Site	Deploy- ment[a]	Cost estimate,[b] \$M	Risk	Team's ranking
Team 1, TRW/GMDI							
FRP-sandwich	Rigid	Barge	S. Atl. cruise	H	26	Low	1
Elastomer	Compliant	Barge	Puerto Rico	V	36	High	3
Polyethylene- 7 pipes	Compliant	Spar	Puerto Rico	H	77	Med.	7
Team 2, SAI/B&RDI							
Steel-R&SR	Rigid with joints	Barge	S. Atl. cruise	V	62	Low	1
FRP-sandwich	Rigid	Barge	Puerto Rico	H	26	Low	2
Steel-R&SR	Bottom-mounted taut	Spar	Puerto Rico	V	74	Med.	6

[a]H = horizontal, float and flip; V = vertical, through platform by sections.
[b]Costs through deployment; see Table 2.

also benefit from other technology improvements in the course of a decade or so. Thus, less efficient, smaller pilot plants would surely be phased out in less than 30 years.

It may be noted that neither of the two contractor teams chose a reinforced-concrete pipe among its top candidates. The primary reasons were perceived risks with respect to a) any multiple-joint approach with concrete and b) use of the lightweight concrete proposed by APL and ABAM Engineers. These aspects and others are being investigated by material and model tests in the ongoing program.[10a-c]

The Department of Energy, through NOAA, is considering the fabrication and deployment of a 1/3-scale FRP pipe during FY 1980. In addition, small scale tests of pipe/platform interactions and loads are planned. Thus, the ongoing CWP work should provide more information for industrial use in concept definitions of pilot plants in the near future.

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COLD WATER PIPE VERIFICATION TEST

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Abstract

Tests of a 5-ft-diam steel pipe, made up in 20 ft sections to total lengths up to 500 ft, were conducted off Santa Catalina Island, California during December 1978 and January 1979. The pipe was supported from Deep Oil Technology's X-1 semi-submersible platform by a gimbal joint. The platform was moored with spring buoys in 1000 ft of water. The objectives were to evaluate the at-sea performance of various configurations and to use the test results to verify or improve the existing time domain and frequency domain analyses of cold-water pipes for Ocean Thermal Energy Conversion (OTEC) plants. The configurations tested were: platform alone; platform with 120-ft pipe, with 300-ft pipe, with 500-ft pipe, and with 384-ft pipe including a U-joint at 162 ft. The results available by mid-July indicated that in general, the frequency locations of the spectral peaks can be predicted well, but the magnitudes of the peak values are in poorer agreement. The analyses are continuing and final results will be given in a comprehensive report.

Introduction

An at-sea test of a large diameter pipe was conducted by Deep Oil Technology, Inc. under contract to the Applied Physics Laboratory. The main objectives of the test were to evaluate the performance of a large pipe in the open sea and to verify computer programs used in the design and analysis of cold water pipes for OTEC platforms.

The test was conducted off the coast of Santa Catalina Island, California during December 1978 and January 1979 using the Deep Oil X-1 platform (see Fig. 1). The platform, which is a triangular semi-submersible, 120-ft on a side, was moored with spring buoys in 1000 feet of water. There was no attempt to pump water through the pipe during the test.

Test Description

Platform

The Deep Oil X-1 platform is a fully instrumented experimental vessel originally designed as a Tension Leg Platform (TLP). It was selected to carry out

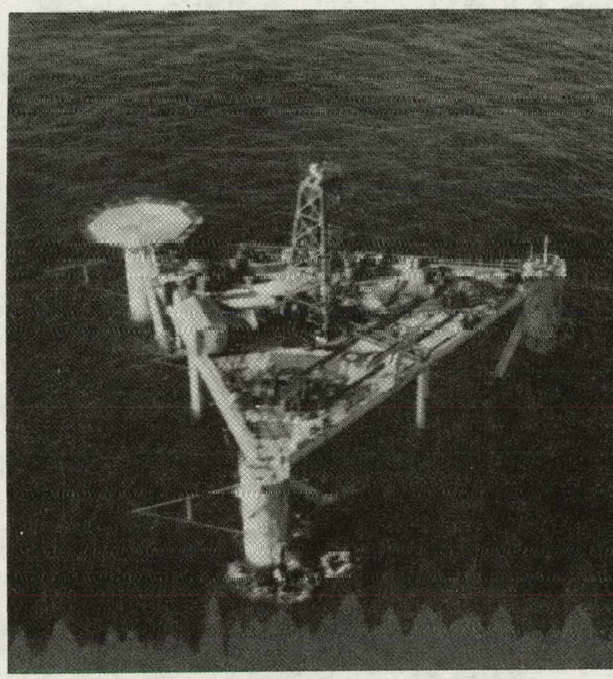

Figure 1 Test Platform "Deep Oil X-1" Used in Cold Water Pipe Sea Test

these tests because its size is adequate and it is thoroughly instrumented to measure directional waves as well as its own motions. For these tests, the Deep Oil X-1 was anchored in a three point catenary moor, similar to a conventional semi-submersible. There were three reasons for this mooring: 1) the water required for the tests was much deeper than the design depth for the vessel as a TLP, 2) the weight of the pipe was higher than what the vessel can carry as a TLP, 3) the platform moored as a TLP is so stable that very small pipe motions and stresses were expected if it is attached to a TLP, contrary to the purpose of the tests. The stiffness of the mooring system was made very low in order to assure unrestricted movement of the platform.

The Deep Oil X-1 as configured for these tests had the following characteristics. Note that these

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characteristics apply to the platform alone without the pipe.

Draft:	44 ft
Displacement:	662.5 long tons
Height overall:	66 ft
Length on side: (Center-to-Center)	120 ft
Vertical CG:	33 ft
Heave Natural Period:	11.5 sec
Roll and Pitch Natural Periods:	18 sec

During all tests, the draft was maintained at 44 ft by deballasting the platform in amounts equal to the weight of the pipe taken onboard.

Cold Water Pipe

The cold water pipe (CWP) had a 5-ft outside diameter and a 3/16-in wall thickness. These dimensions were chosen so that, for the lengths of pipe deployed, several of the pipe natural frequencies would be in the range of the expected wave frequencies. The total length of pipe was limited by the amount of weight the platform could support. Each pipe section, 20-ft in length, was fabricated with mating flanges so that the sections could be bolted together during deployment (see Fig. 2). A single universal joint was available for use during the test. Two special sections were prepared which were identical to the other sections but had two strain gage stations 5 ft from each end configured to measure bending strains. The universal joint and the instrumented sections could be placed anywhere along the pipe string. Enough sections were fabricated to achieve any length of CWP up to 800 ft.

The CWP was connected to the platform through a gimbal arrangement. There was provision to lock out the gimbal by a set of three wire-rope lines that effectively restrained the lateral motions of the pipe at the lower level of the platform. In this way a nearly "built-in" connection was simulated.

Four guide tubes were precisely located and welded inside each pipe section to accommodate the accelerometer pods described in the following section.

Instrumentation

Wave heights were measured with six resistance wave staffs that were located around the periphery of the platform to provide a reliable means of determining the directional sea spectrum at the time of recordings. In addition, a string of self-contained current meters was placed in the vicinity of the platform to record current velocity and direction at various depths to 800 ft during the tests.

Platform motion instrumentation consisted of a redundant set of ten accelerometers to record the six-degree-of-freedom accelerations of the platform.

The Cold Water Pipe was instrumented with accelerometers and strain gages. The accelerometers were mounted in pairs which measured pipe accelerations in the horizontal plane. They were placed in watertight housings (see Fig. 3) with five such housings spaced about 70 feet apart on the armored

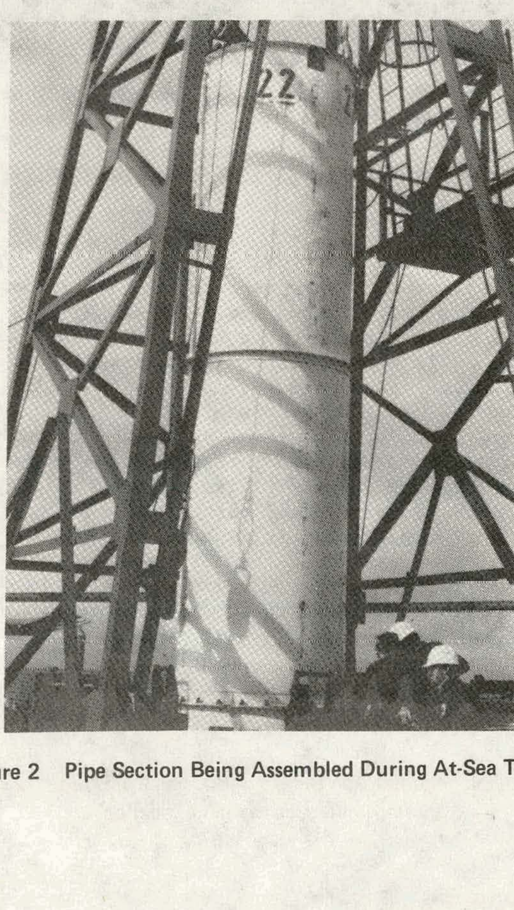

Figure 2 Pipe Section Being Assembled During At-Sea Tests

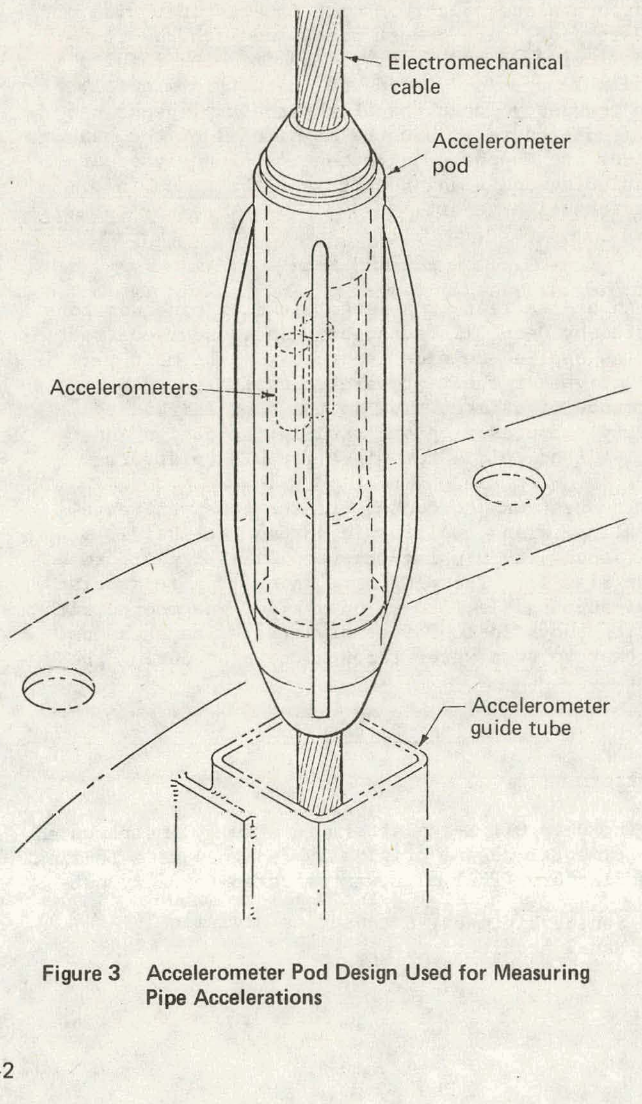

Figure 3 Accelerometer Pod Design Used for Measuring Pipe Accelerations

cable. After the pipe was deployed a string of accelerometers could be lowered into position through a guide tube rigidly attached to the inside of the pipe. The advantages of this procedure were that (1) the accelerometers could be deployed after the pipe; (2) the locations of the accelerometers along the pipe could be changed easily; and (3) as a check on the output, two or more accelerometer strings could be positioned at the same levels so their readings could be checked for agreement. Strain gages were installed on two of the 20-ft pipe sections and these could be inserted at any location along the length of the pipe.

Data Acquired

Initially data were collected from the platform alone with no pipe deployed. Then 300 ft of pipe was deployed and additional data were taken from the platform and the pipe. For the next phase the pipe length was increased to 500 ft and more data were collected. During a storm on the night of January 5, 1979 there was a failure in the pipe flanges and 400 ft of pipe was lost along with one string of accelerometers and one of the strain gauged sections of pipe. Available data indicate that the failure, which was due to inadequate flange design, was progressive, involving the loss of three shorter pipe lengths (200 ft, 100 ft, and 100 ft) over a period of six hours during the storm. Each break was near a point of maximum stress for the length involved as predicted by the computer analysis. After recovery of the remaining 100 ft of pipe the test was continued with 400 ft of pipe and two strings of accelerometers. Subsequently data were collected for a 384 ft pipe with the U-joint at 162 ft and for 300 ft of pipe without a U-joint. The test was concluded on January 24, 1979.

A total of 39 files of data were taken during the test period, each file consisting of 17 minutes 40 seconds worth of data taken with a sample rate of 2 per second, giving 2120 data points for each channel of data on each file. The amount of data files collected for each configuration along with the range of significant wave heights is summarized in Table I.

Table 1

Number of data files collected during CWP verification test

Sig. wave ht. (ft)	1.5 – 3.0	3.0 – 4.5	4.5 – 6.0	6.0 – 7.5	Total
Platform alone	3				3
120 ft pipe		1	2		3
300 ft pipe	1	2	8		11
300 ft w/restraint			3		3
500 ft pipe	2		1	4	7
384 ft w/u-joint			4	6	10
384 ft w/u-joint and restraint			2		2
					39

Data Reduction and Analysis

The initial processing for a file of data involved the generation of twelve "derived" channels of data, i.e. the six-degree-of-freedom accelerations at the center of the platform and the platform heave accelerations at the locations of the six wave staffs. The recorded data from the ten platform accelerometers provided a redundant set of measurements which was used to define the accelerations at the center of the platform. A computer program was developed which transformed the measured platform accelerations at the sensor locations into surge, sway, heave, roll, pitch, and yaw accelerations at the center of the platform and produced six added channels of data. Similarly, the platform heave accelerations at the six wave staff locations were calculated and stored in an additional six channels of data. These six channels were used to remove platform heave motion from the measured wave heights. The twelve derived channels of platform data were in the same format as the actual test data so that they could be processed as part of the test data. Consequently, throughout the data analysis process the original data and the twelve added channels were all treated as though they were recorded during the test.

An initial evaluation of each data channel in a given file was made by computing the basic statistics (e.g. standard deviation, maxima, minima, mean) and testing for normality and stationarity. The data were also checked for consistency. For example, one consistency check was made with the five platform heave accelerometers. The data from these five sensors were compared to ensure that there were no contradictions in the readings.

Periodically during the test, two sets of accelerometers were positioned at the same vertical station on the pipe, thus providing a redundant set of pipe acceleration data. A comparison of the measurements of two such accelerometers is given in Figure 4. This is a log-log plot of the spectral output of two accelerometers 59.6 ft above the U-joint during the test of a 384 ft pipe. As can be seen in the figure, the agreement between the output of the two accelerometers is quite good; this enhances confidence in the reliability of the accelerometer data.

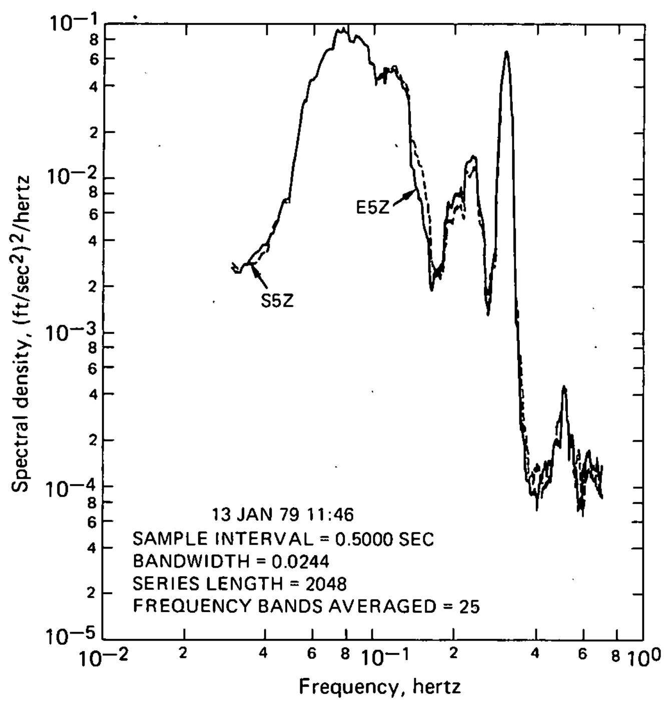

Fig. 4 Comparison of output spectra from two accelerometers at the same location.

Since the wave staffs were mounted to the X-1 platform the measured wave data included a component due to the heaving of the platform. In order to obtain the absolute wave height history it was necessary to remove the platform heave component from the measured data. This was done using the six channels of data giving the platform heave accelerations over the wave staff locations. These six acceleration time series were numerically transformed into Fourier series by a Fast Fourier Transform subroutine. The acceleration Fourier series were then integrated to give heave displacement Fourier series. (When the data are in the form of a Fourier series the integration amounts to a division of the acceleration series by the square of the frequency to produce the displacement series.) Then the wave staff data are transformed into Fourier series and the platform heave displacement is removed. The result is six Fourier series for the absolute wave heights at the six wave staff locations. These Fourier series are then used to generate the sea spectrum. Deep Oil Technology used the data from the six wave staffs to produce directional sea spectra.

Most of the data analysis was done in the frequency domain. For this the power spectra were generated using a time series length of 2048 points with 25 frequency bands taken in a moving average. The resolution bandwidth was 0.0244.

Computer Program Comparisons

Comparisons between measured data and computer predictions have been made with Prof. R. Paulling's ROTEC program[1] which performs a frequency domain analysis, and Deep Oil's MRDAP[2] program, a time domain program originally developed for the dynamic analysis of marine risers.

Comparisons with ROTEC

It was necessary to modify the ROTEC program so that a direct comparison could be made between the test results and the program results. One modification allowed the input sea spectrum to be specified pointwise as a function of frequency rather than as a Bretschneider spectrum, which is the usual input format for ROTEC. A second modification was made to include the gravity component in the predicted cold water pipe accelerations. During the test, whenever the pipe would bend, the accelerometers would tilt out of the horizontal plane; this would cause the instruments to measure a component of the acceleration due to gravity in addition to the acceleration of the pipe. ROTEC was modified so that it included this gravity component. Similarly, the surge and sway accelerometers were affected by the pitch and roll of the platform, so ROTEC was modified to include these effects in its predictions.

One final modification was made so that ROTEC would print the acceleration spectra for the platform and pipe in addition to the RMS values.

All comparisons between test results and ROTEC predictions were made on the basis of power spectra, hence the results presented here will show measured and predicted spectra plotted on log-log graphs. Comparisons of three different configurations will be given: (a) the X-1 platform with no pipe; (b) 500-ft pipe with no U-joint and no restraint; and (c) 384-ft pipe with a U-joint at 162 ft and no restraint.

Platform Alone. A contour plot showing sea spectrum power densities as a function of frequency and direction for this file of data is given in Figure 5. The corresponding unidirectional spectrum is shown in Fig. 6; this was used as input to the ROTEC program with all of the sea energy specified from the 110° direction, even though there was a

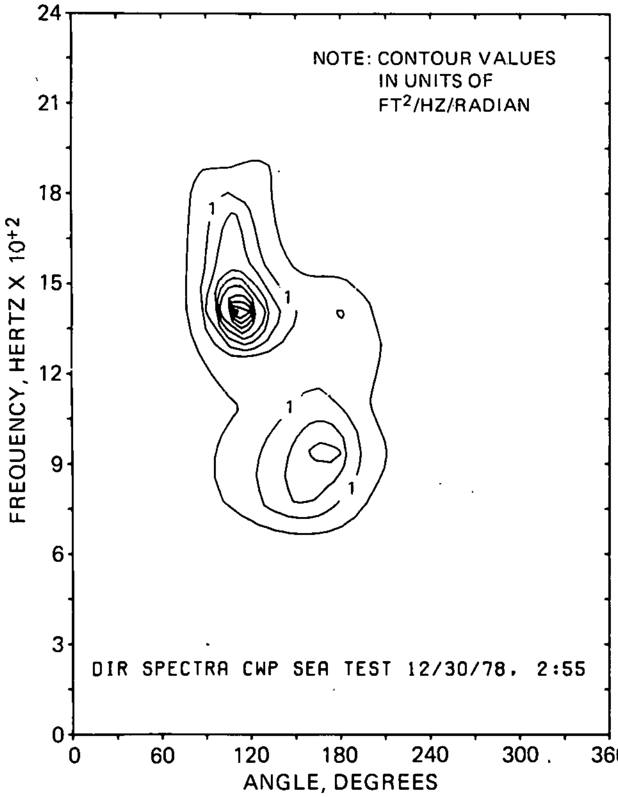

Fig. 5 Directional sea spectrum for platform-alone case.

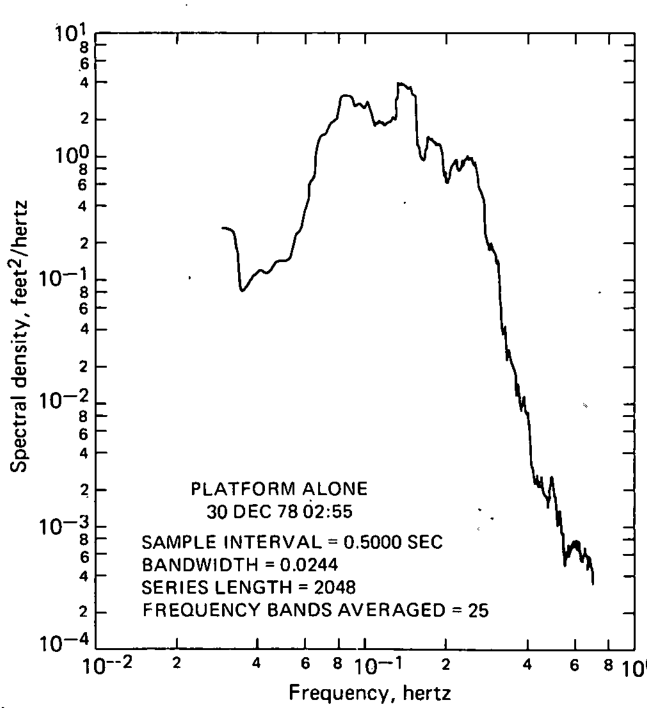

Fig. 6 Unidirectional sea spectrum for platform-alone case.

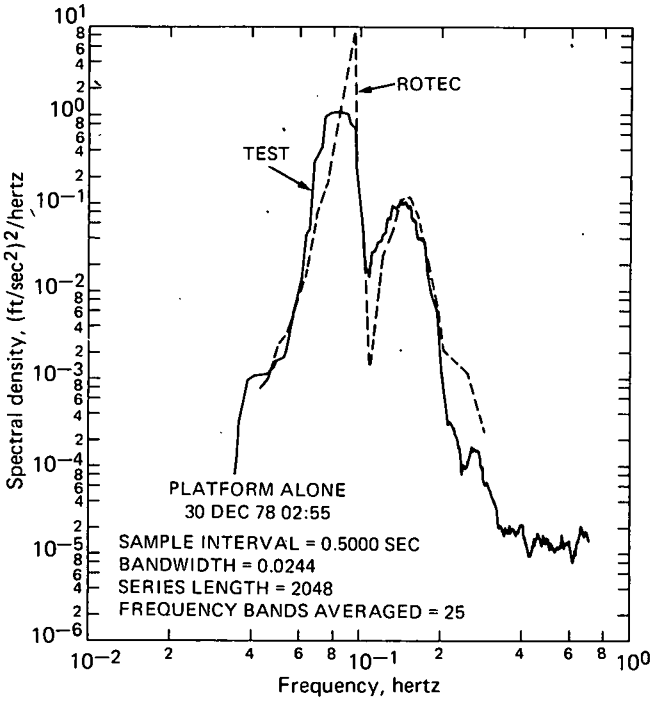

Fig. 7 Heave acceleration spectra for the platform-alone from the test and from ROTEC.

lower energy peak at 160°. Figure 7 shows the platform heave acceleration spectrum as measured in the test and as predicted by the ROTEC program. The ROTEC results follow the shape of the measured spectrum well, but the magnitude of the peak value is overpredicted. A similar comparison is shown in Figure 8 for platform pitch response. The ROTEC prediction shows a large spike at about 0.055 Hz which indicates a resonant response. However, a large response at that frequency does not appear

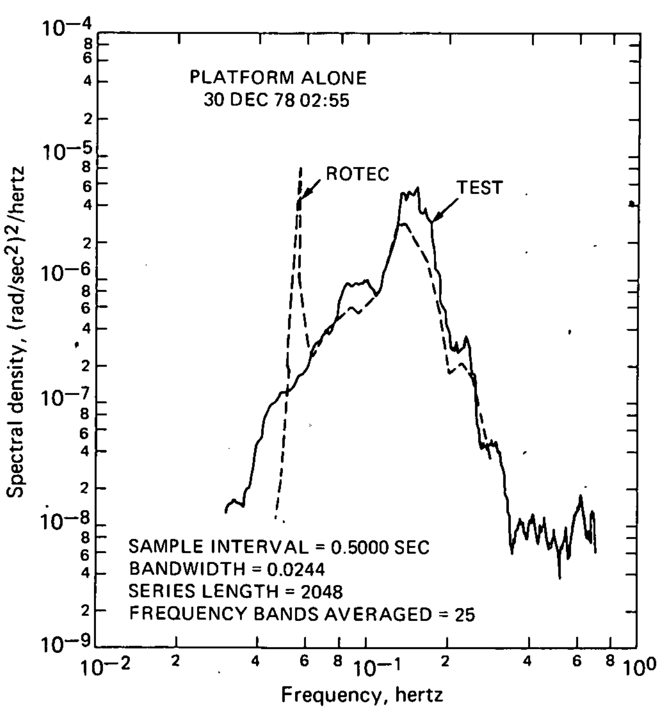

Fig. 8 Pitch acceleration spectra for the platform-alone from the test and from ROTEC.

in the test data. As will be seen, this anomaly in the ROTEC predictions occurred for other configurations also. The ROTEC program predicts high spectral power levels at about 0.055 Hz, but the test data do not show such a response.

<u>500 Ft Pipe With No U-Joint and No Restraint.</u> The directional sea spectrum for this case is shown in the contour plot in Figure 9. This was a multidirectional spectrum with the low frequency (0.075 Hz) swells coming from about 180° (bow-on) and the high frequency (0.14 Hz) waves coming from 75° and 110°. Since the ROTEC program is not designed to handle a multidirectional sea spectrum it was decided to model this case with two ROTEC runs. One had an input spectrum from the 180° direction which matched the low frequency portion of the measured spectrum and the other case had an input spectrum from 110° which approximated the high-frequency part of the measured spectrum. These two spectra are shown in comparison to the measured sea spectrum in Figure 10.

The test results for accelerometer station #2 (measuring in the surge direction 72 feet from the bottom of the pipe) are compared with the ROTEC results in Figure 11. Since the input spectrum for the 180° case matched the test spectrum best at the lower frequencies, comparisons with ROTEC results for this case are made at the lower frequencies; similarly the 110° ROTEC results are compared with test results at the higher frequencies. As shown in Figure 11, the ROTEC results generally follow the shape of the test results but they disagree in magnitude. As before, the ROTEC program predicts a large spectral peak near 0.055 Hz, but the test data do not have such a peak. Figure 12

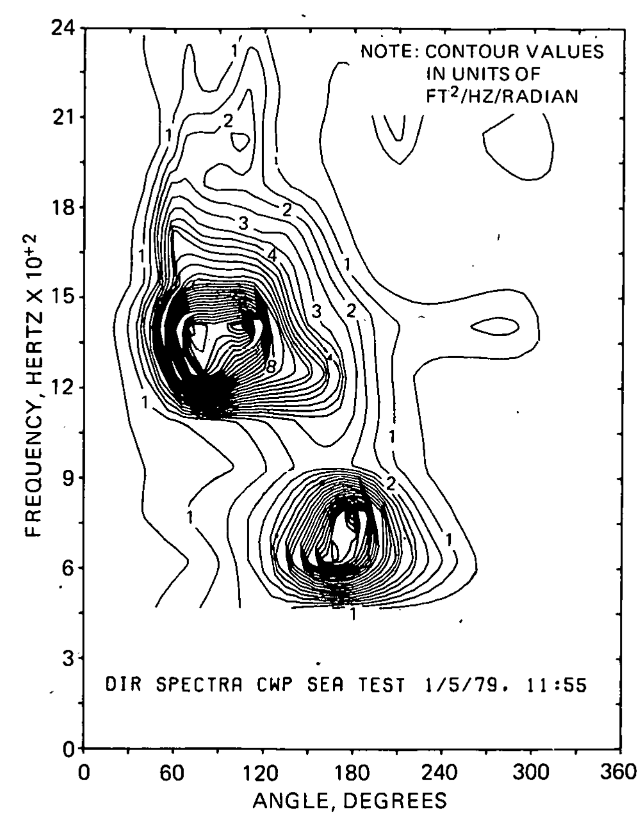

Fig. 9 Directional sea spectrum for 500-ft pipe case.

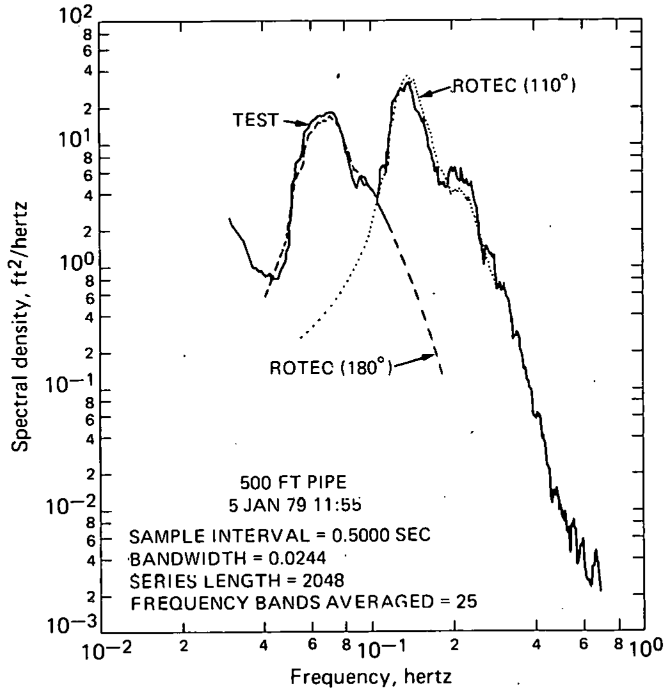

Fig. 10 Unidirectional sea spectrum and ROTEC input spectra for 500-ft pipe case.

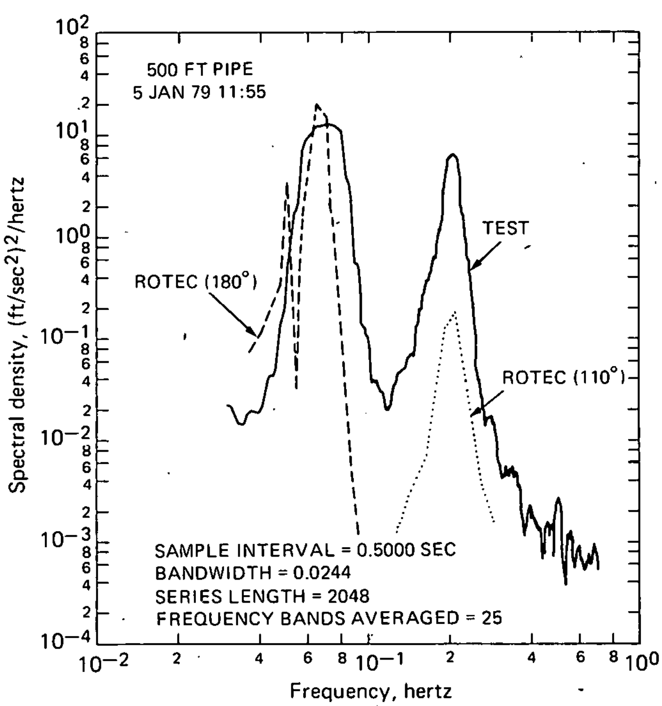

Fig. 11 Pipe station No. 2, acceleration spectra from test and ROTEC.

shows the results for pipe accelerometer station #5 in the sway direction. This accelerometer is located about 278 ft from the bottom of the pipe. The ROTEC program with a sea spectrum from 180° predicts no acceleration in the sway direction, so only the results for 110° are shown. The program results for frequencies above 0.10 Hz have the same general shape as the test results but the magnitudes do not agree well. However, this could be due to the fact that we are modeling the response

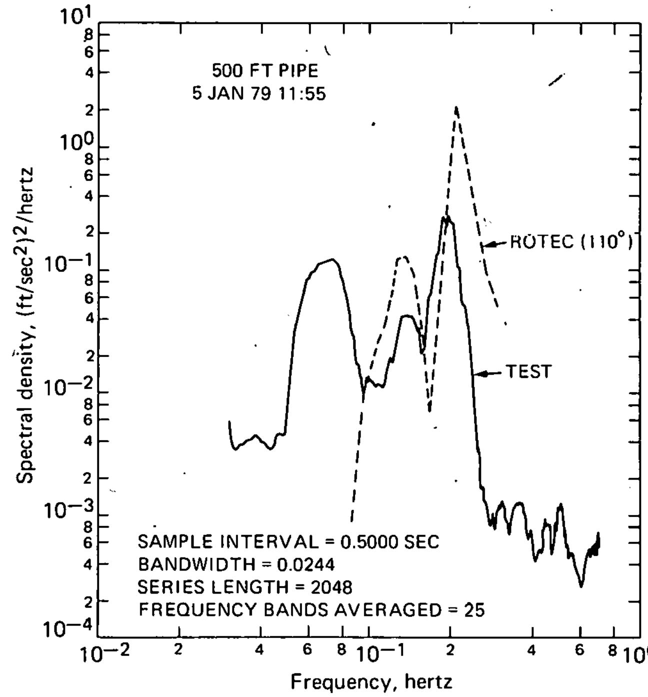

Fig. 12 Pipe station No. 5, acceleration spectra in the sway direction from test and ROTEC.

to a multidirectional sea with a unidirectional program.

<u>384 Ft Pipe with U-Joint at 162 Ft From The Top of the Pipe and No Restraint</u>. For this case the sea was rather unidirectional, coming from 180° (bow-on) as shown by the contour plot in Figure 13.

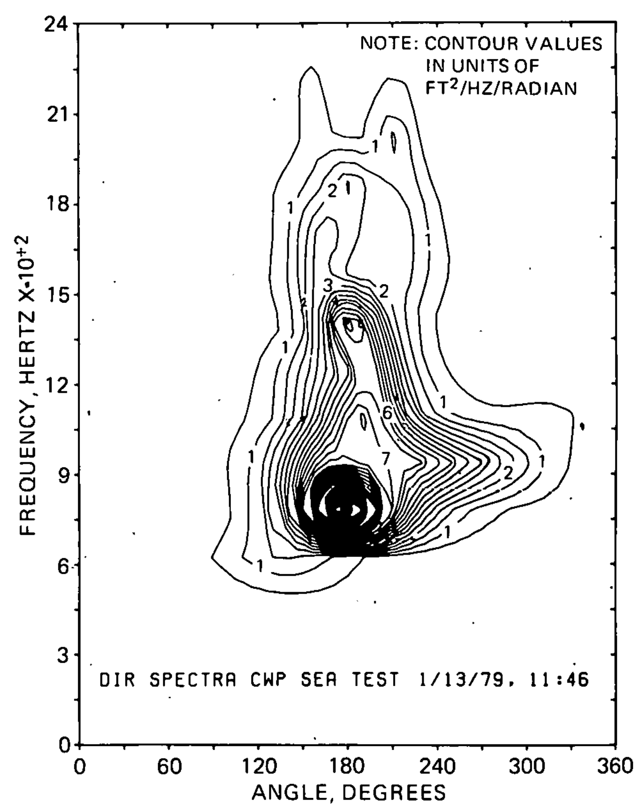

Fig. 13 Directional sea spectrum for 384 ft pipe case.

The corresponding unidirectional spectrum is shown in Figure 14. This spectrum was used as input to the ROTEC program and the resulting platform and pipe accelerations were determined. The predicted and measured spectra for platform pitch are compared in Figure 15. The agreement is good except at the 0.055 Hz frequency where the ROTEC program shows a large spectral amplitude which is not in the test data. A similar comparison is shown in Figure 16 for accelerometer station #1 which is below the U-joint, 218 ft from the U-joint and 4 ft from the bottom of the pipe. The large response

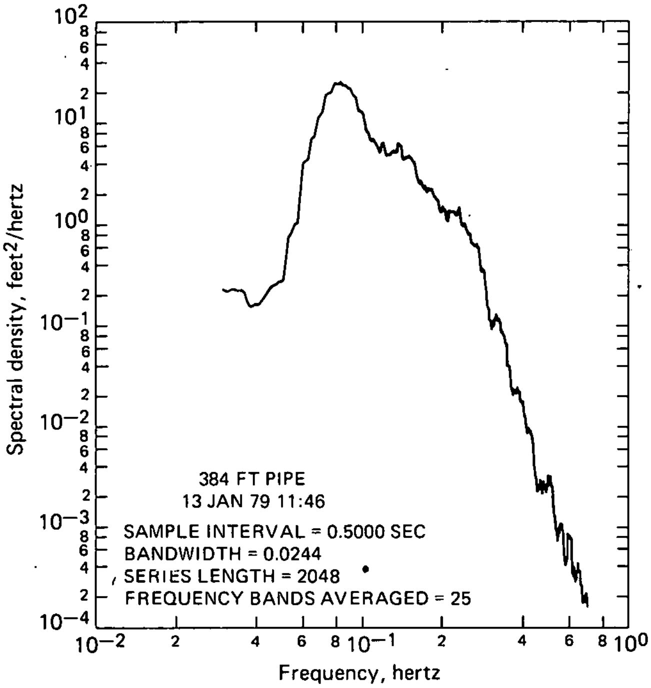

Fig. 14 **Unidirectional sea spectrum for 384 ft pipe case.**

spike at 0.055 Hz is still present in the ROTEC results but not in the test data. In the region away from that frequency the ROTEC program predicts a peak spectral amplitude which is larger than the experimental response. The agreement between experimental and predicted data is much better for accelerometer station #5 as shown in Figure 17. This is located above the U-joint, 277 ft from the bottom of the pipe. Although the ROTEC prediction still has the anomaly in the response near 0.055 Hz, the agreement is quite good at other frequencies.

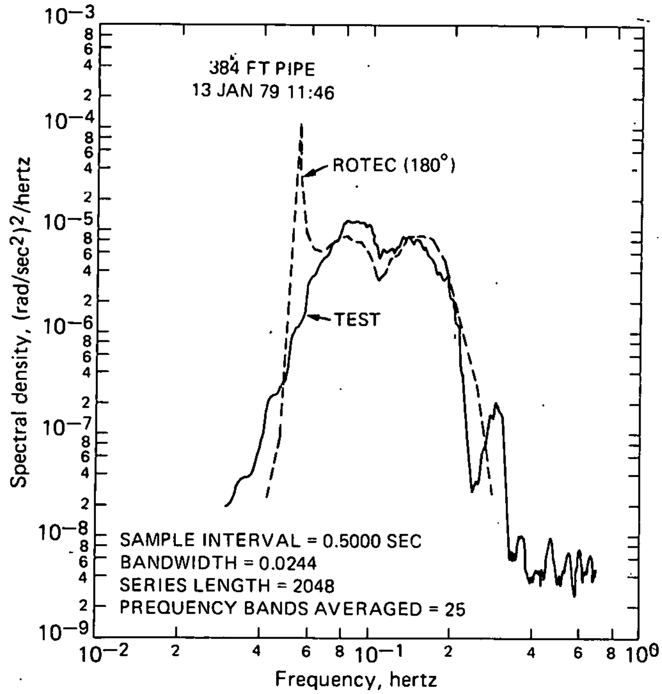

Fig. 15 Platform pitch acceleration spectra from test and ROTEC.

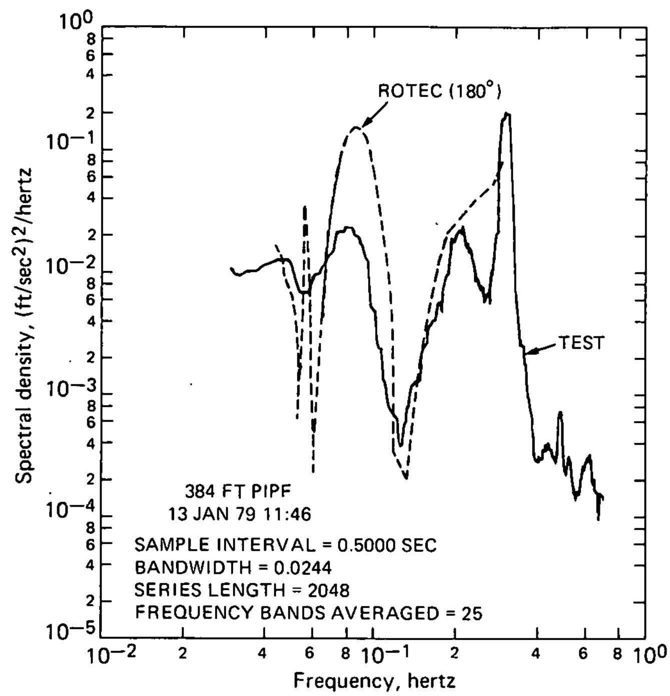

Fig. 16 Pipe station No. 1, acceleration spectra from test and ROTEC.

Comparison with MRDAP

MRDAP is a Deep Oil Technology proprietary computer program for the totally non-linear time-domain analysis of petroleum risers in directional seas. Because, unlike cold water pipes, petroleum risers are normally much smaller than the platforms to which they are connected, it is usually assumed in their analysis that the platform affects the riser motions but not vice-versa. This assumption is poorly realized in a CWP-platform system and should probably not be used in the analysis of cold water pipe behavior until its effects can be quantified. Although a revision to MRDAP that includes the effects of the CWP on the platform response is underway, it has not been completed to date. Using the present version of MRDAP, comparisons were made for two files: the same 500-ft-pipe file used in the ROTEC comparison and a file very similar to the one used in the 384-ft ROTEC comparison. In general agreement is not very good, possibly because of the above mentioned assumption. As an example, Figure 18 shows

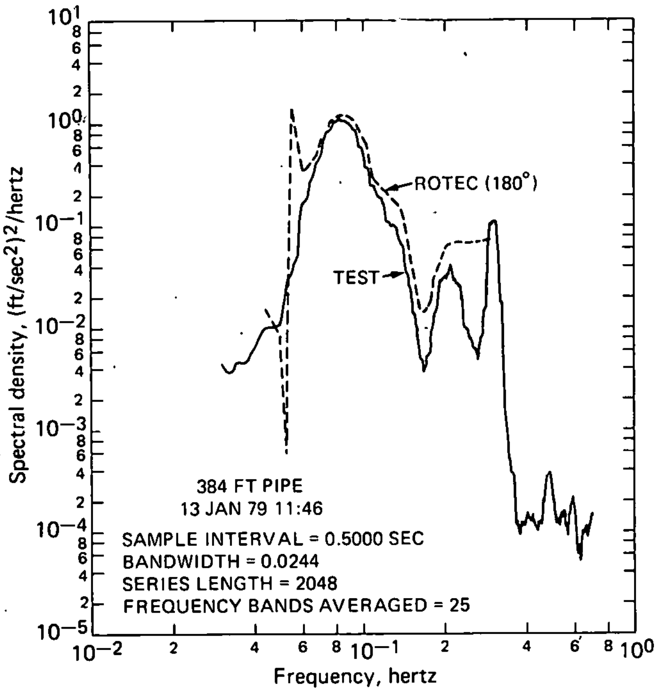

Fig. 17 Pipe station No. 5, acceleration spectra from test and ROTEC.

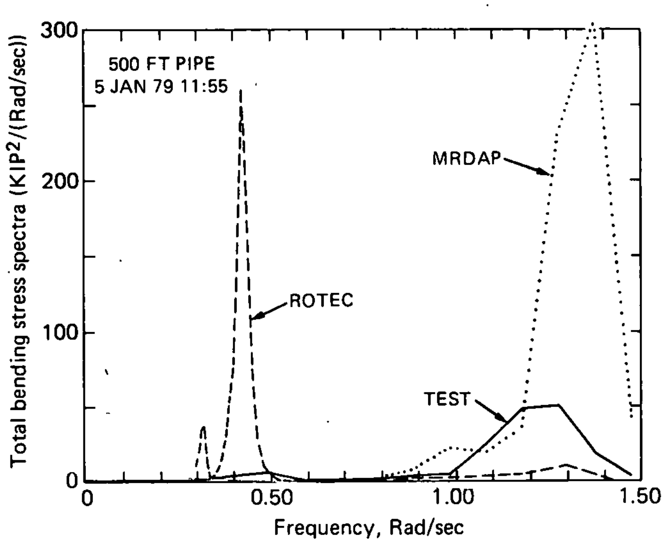

Fig. 18 Total bending stress spectra from test, MRDAP, and ROTEC.

the total bending stress spectra for a location 65 ft from the top of the pipe for the 500-ft case. (Note that this figure is plotted on a linear scale.) At low frequencies (below 0.9 rad/sec), MRDAP results are very close to the experiment, whereas ROTEC still shows the peculiarity at the pitch natural frequency. At higher frequencies, MRDAP overpredicts the bending stress considerably but ROTEC underpredicts it.

Conclusions

The at-sea test has provided a reliable set of data for use in verifying computer programs. These data are now being compared with results generated by the ROTEC frequency-domain program and the MRDAP time-domain program.

The comparisons between the test data and the ROTEC results indicate that, in general, the program does well in predicting the frequency locations of the spectral peaks but is less reliable in predicting the magnitude of the peak values. The ROTEC results show a large spike in the response spectra at the platform pitch frequency but this is not present in the test results. The influence of platform hydrodynamic damping on predicted pipe response will be investigated. Some of the differences between ROTEC results and test data are undoubtedly due to the fact that ROTEC is based on unidirectional sea spectrum whereas the test conditions were not unidirectional.

The MRDAP program does not give good agreement with the test results. This program does not account for the effects of the cold water pipe on the platform, but Deep Oil is modifying the code to include these effects.

The data analysis is continuing in three main areas:

i) more data files are being analyzed and compared with the ROTEC program results;

ii) an effort is underway to determine the cause of the anomaly in the ROTEC platform response spectra; and

iii) the effect of using a unidirectional sea spectrum in making ROTEC predictions is being evaluated.

References

1. J. R. Paulling, "Frequency-Domain Analysis of OTEC Cold Water Pipe and Platform Dynamics", OTC Paper 3543, Offshore Technology Conference, Houston, 1979.
2. "Engineering Tasks Related to the 'Verification Test for OTEC Cold Water Pipe'" Appendix F; Prepared by Deep Oil Technology, Inc. for the Applied Physics Laboratory, June 21, 1978.

ANALYTICAL AND EXPERIMENTAL METHODS FOR DETERMINING OTEC PLANT DYNAMICS AND CWP LOADS

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Abstract

Ocean Thermal Energy Conversion (OTEC) plant dynamics and CWP loads and structural design are among the most important OTEC technology areas. Significant effort has been devoted to development of theoretical methods for predicting CWP/platform dynamic motions, loads and stresses. A wide variety of computational techniques have been used, but to date few methods have been well validated. Several sets of laboratory tests, using small-scale models, and one at-sea test with a large-scale model, have been carried out to measure dynamic motions and loads. Such experiments are complicated by the difficulties in correctly scaling all important phenomena. The paper discusses, in some detail, three important theoretical methods, discussing their capabilities and shortcomings and concludes that no method currently incorporates all desired features. It is concluded that with the use of specially constructed CWP models, it is possible to scale most important factors even when small-scale (1/50 to 1/100) models are used.

Introduction

The difficulties in designing and building practical cold water pipes (hereafter called CWP) for OTEC plants have been recognized for several years, and significant effort has been devoted during the past two years to developing CWP technology and design concepts. This effort has identified the response of the OTEC CWP/platform configuration to ocean wave and current induced dynamic loads as the crux of the CWP design problem. Various theoretical methods have been developed for analyzing CWP dynamic response and laboratory model tests have been used to predict CWP behavior and to provide data for validating theoretical methods.

At-sea tests of large scale models of OTEC plants appear to be an attractive means for evaluating CWP dynamic response and stresses, but such test present many problems. The foremost problems are cost, time and the lack of control of environment, particularly waves. Such tests, when included in an OTEC plant design, will probably be conducted near the end of the design cycle to verify a final CWP design. The OTEC plant and CWP designer must have other, less costly, means for use during the complete design cycle. To date theoretical methods, sometimes supplemented by small-scale model tests, have been used by designers. It seems likely that theoretical methods and improved laboratory model test methods will remain the primary bases for design.

This paper describes and evaluates available theoretical and experimental methods for analyzing CWP dynamic responses. Some shortcomings of theoretical and small-scale laboratory methods are discussed, with emphasis on experimental methods which have, to date, received far less attention than theoretical methods. It is intended that the paper provide the OTEC designer with a better basis for evaluating available methods and for selecting the most appropriate method(s).

A Brief History of CWP Theoretical Analysis

Rigorous theoretical CWP analysis appear to date from about 1976, when a method for predicting the coupled response of CWP and platform to ocean waves was published[1]. This method, which treated the CWP as a rigid or inelastic body, was replaced, by early 1977, by various methods[2,4], which treated the CWP as an elastic beam.

Two of these methods, developed by HYDRONAUTICS and J.R. Paulling,[2,3], treated the linear, coupled response of the platform and CWP in the frequency-domain. The other two methods, developed by F.R. Harris and Southwest Research[4], treated the non-linear, uncoupled response of the platform and CWP in the time-domain. The method developed by Harris was subsequently modified to include CWP/platform coupling[5]. The features of these four methods were compared in some detail, but with some errors, in Reference 4, and also in Reference 6. Reference 4 provided detailed comparisons of calculated results for the OTEC-1. These comparisons indicated significant differences in calculated results for some sea states and wave heading angles. Only the HYDRONAUTICS results were compared with the OTEC-1 model test data[7]. The methods of References 2 and 3 have been used for a number of design studies.

Today a number of theoretical methods exist for analysis of coupled CWP and platform response to wave and current induced loads. These methods all treat the CWP as an elastic beam. Only three of these methods are now publicly documented. These include frequency-domain analysis methods developed by Paulling[9], and by HYDRONAUTICS[10]; and a time-domain method developed by TRW[11].

The method of Paulling[9], which analyzes the CWP using finite element methods, is a much extended version of the earlier method of Reference 3. The latest version can treat multi-jointed pipes, and includes an equivalent linearized treatment of CWP hydrodynamic damping and interaction of waves and steady current. It can treat directional or short crested waves as well as uni-directional waves. Computer programs for this method are not generally available.

The latest HYDRONAUTICS Methods[10], analyze the CWP using the transfer matrix method of Ref. 12. This method can treat multi-jointed pipes, includes an equivalent linearized treatment of CWP damping and can treat arbitrary distributions of static and dynamic loads and mooring attachments along the CWP. The method does not presently treat directional waves or wave-current interactions. Computer programs and documentation are available to qualified users through NOAA.

The recently developed TRW method[11], uses a finite difference method for CWP analysis. It can

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treat multi-jointed pipes, directional waves and wave-current interactions, as well as arbitrary distributions of unsteady loads. Because this is a nonlinear analysis no approximations are required in the treatment of CWP damping or wave-current interactions. Some approximation is required for treatment of platform hydrodynamics, however. Computer programs are not available.

A Brief History of CWP Model Tests

The history of CWP model tests is even briefer and sparser than the history of CWP theoretical analyses. Tests to date have been designed to provide both quantitative and qualitative CWP response data and CWP loads due to waves and current. These tests which have, with one exception, been carried out in laboratories, have involved both complete CWP/platform configurations and sections of a CWP.

The limited use of laboratory model tests is probably due both to cost and to limitations on available test facilities. With available U.S. test facilities such as those at HYDRONAUTICS, Incorporated, Offshore Technology Corporation (OTC) and DWTNSRDC limited to water depths of 42 to 30 feet, small-scale models (typically 1/75 to 1/100 scale) of complete CWP's must be used. The limitations imposed by the use of such small-scale models are discussed in later sections.

Complete CWP/Platform Configurations

The earliest known tests of a complete CWP/platform configuration were Froude scaled tests of a 1/50 scale model of a proposed OTEC-1 configuration using the Hughes Mining Barge at HYDRONAUTICS[7]. Tests were conducted in regular and random waves using two CWP models, one very stiff and one quite flexible. Measurements included platform motions and CWP bending moments at two points along the CWP. These tests were intended primarily to provide validation data.

Tests of a Kelp farm design, which uses a CWP, and of a model of the Deep Oil Technology (DOT) X-1 platform with the proposed test CWP[13], have been carried out by OTC. These tests involved selected measurements and determination of motions from photographic records. In both cases, however, the CWP model was not structurally scaled and was not instrumented to measure bending moments.

Recently, Froude scaled tests of a 1/110 scale model of a proposed 400 MW spar platform and CWP were conducted at HYDRONAUTICS to provide validation data[14]. Tests were conducted with one relatively stiff CWP with rigid and pin connections to the spar. Platform motions and CWP bending moments at four locations along the CWP were measured for three representative sea-states.

In January of this year, a five foot diameter, thin (3/16 inch) wall steel CWP of three lengths was tested at-sea using the DOT X-1 platform[15]. In these tests, which were directed by APL, measurements were made of platform motions, CWP accelerations and bending moments and wave and current conditions. Analysis of the large body of test data is now in progress; early results appear promising. The early CWP failures, which occurred under severe sea conditions, serve to emphasize the difficulty of the CWP design problem.

Tests of CWP Sections and Circular Cylinders

High Reynolds number data for CWP loads and hydrodynamics can be most easily obtained from towing tank or wind tunnel tests of circular cylinders representing a section of the CWP. While a rich body of circular cylinder test data exist, as described in Ref. 15, these data provide only part of the needed information on CWP loading. As a result, several sets of cylinder tests have been carried out to provide needed CWP loads data.

Tests of a small cylinder at Reynolds numbers up to 1.3×10^6 were carried out by SAI[16], as a preparation for tests of a larger model, at Reynolds numbers up to 10^7. While these latter tests were never conducted, the small model data are of some interest. Ref. 15 contains an analysis of these and other available data and presents some controversial conclusion about correlation lengths of unsteady, vortex shedding induced loads due to steady flow.

Wind tunnel tests of cylinders in steady uniform and sheared flows at Reynolds numbers between 10^5 and 10^6 have been recently conducted at VPI, although to date only limited results for tests in uniform flow have been presented[17]. Towing tank tests of a section of a multi-pipe CWP design have been conducted recently at OTC, but results are not known.

The tests of Sarpkaya[18], while not related to OTEC, should be noted as they provide unique data for cylinder and CWP hydrodynamic forces for oscillatory flow and motions.

Status of Theoretical Analysis Methods

As noted in an earlier section, there now exist a number of theoretical methods for analyzing the dynamic response of OTEC CWP/platform configurations. Three of these methods[9,10,11], are sufficiently well documented to assess and compare their capabilities and potential accuracy. These methods and their capabilities are probably representative of other available frequency- and time-domain methods. The general agreement of response to waves from the methods of Ref. 9, 10 and 11 are illustrated by Figures 1 and 2.

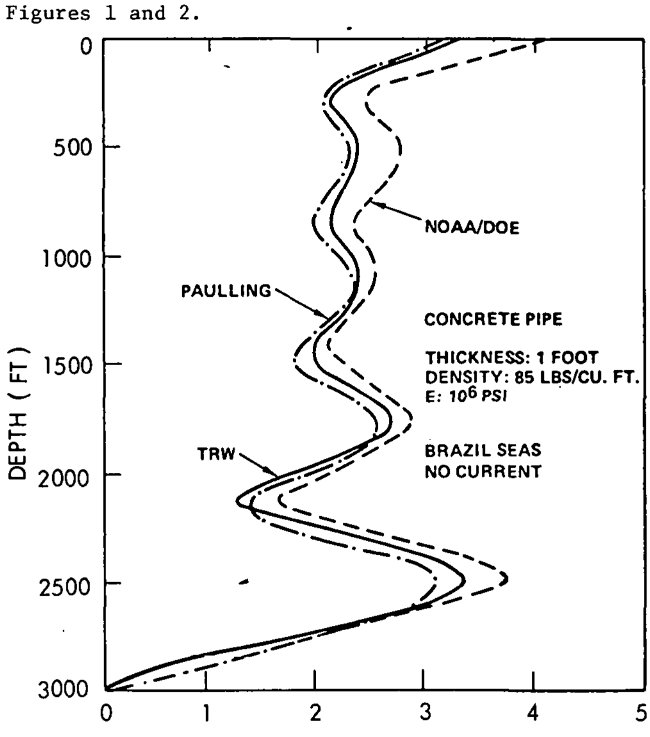


FIGURE 1 - COMPARISON OF RESULTS FROM VARIOUS METHODS FOR UNJOINTED CONCRETE CWP - From Reference 11

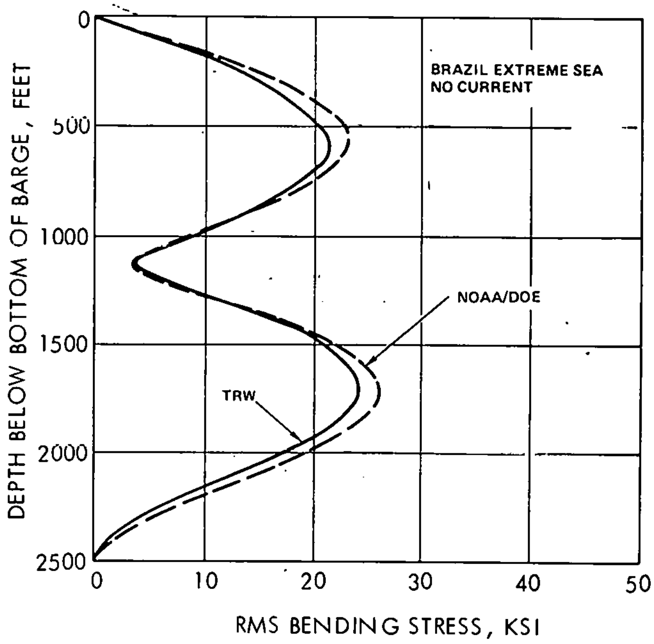


FIGURE 2 - COMPARISON OF RESULTS FROM TRW AND NOAA/ DOE METHOD FOR UNJOINTED STEEL CWP - From Reference 11

NOAA/DOE Methods

These methods, developed by HYDRONAUTICS, are described in detail[10]. These are the only methods for which computer source decks and detailed User's manuals are generally available (through NOAA).

These methods treat the coupled, linear, frequency-domain response of a wide variety of CWP and platform designs to uni-directional, regular and random waves and the linear, frequency-domain response of the CWP alone to steady and unsteady loads due to current, vortex shedding, etc.

The CWP is treated as an elastic beam, and the CWP/platform attachment is characterized by arbitrary stiffnesses. The CWP can have attachments to moorings or to bodies other than the platform. Non-linear CWP damping is represented through equivalent linearized damping and variation of damping coefficient along the CWP length. Flow velocity within the CWP is considered. A separate program treats CWP longitudinal (heave) and torsional dynamics.

The platform is treated as a rigid body with frequency dependent added mass, damping and wave exciting forces. The programs can compute these platform hydrodynamic forces for a variety of ship or barge semi-submersible and submarine platforms using the method of Ref. 19. Hydrodynamic forces for axi-symmetric platforms can be accepted as input. An empirical, equivalent linearized platform roll damping can also be accepted as input.

Paulling Methods

Two distinct methods have been developed by Paulling[9]. These methods are incorporated in computer programs ROTEC and SEGPIP for continuous and jointed CWPs, respectively. These methods treat the coupled, linear, frequency-domain response of a wide variety of CWP and platform designs to uni-direction regular waves. The methods are currently being modified to include treatment of directional waves and interactions between waves and steady current velocities.

The CWP is treated as an elastic beam and the CWP/platform attachment is characterized by a stiffness and damping in all degrees of freedom. Non-linear CWP damping is represented using equivalent linearized damping and an automatic iteration scheme for damping can be used. Flow velocity inside the CWP is not considered.

The treatment of the platform is very similar to the NOAA/DOE methods, although the method of Ref. 20 is used for semi-submersibles. Paulling has also modified the platform program of Garrison[21], to provide direct input to these programs for complex platform geometries.

TRW Method

This method, which is briefly described[11], is based on marine riser analysis methods. It treats the coupled, non-linear, time-domain response of CWP and platform designs to uni-directional, regular and irregular waves, to current and to unsteady CWP forces due to vortex shedding, etc. It treats interactions between wave and current velocities.

The CWP is treated as an elastic beam and the CWP/platform attachment is characterized by stiffness. Quadratic damping with a constant drag coefficient is used. Flow velocity within the CWP is considered.

The platform is assumed to have constant added mass and damping forces, which are accepted as input and must be generated by independent methods. The assumption of constant hydrodynamic forces might cause significant errors for some platforms.

Comparison of Capabilities

It is clear from the preceeding descriptions that each of the three methods share a number of basic features and capabilities, while each method also has certain unique features and capabilities. Table 1 presents a comparison of capabilities of these methods in a number of areas considered to be most important for typical analyses of CWP/platform dynamic and static responses.

Table 1 Comparison of Capabilities of Three Theoretical Analysis Methods[a]

Feature/Capability	NOAA/DOE	Paulling	TRW
Loadings			
Uni-directional Waves	Yes	Yes	Yes
Directional Waves	No	IP	No
Steady Current	Yes	IP	Yes
Vortex Shedding	Yes	No	Yes
Arbitrary Loads	Yes	No	-
Internal Flow	Yes	No	Yes
CWP Geometry			
Joints	Yes	Yes	-
Non-linear Damping	AP	AP	Yes
Wave/Current Interaction	No	IP	Yes
Moorings, etc.	Yes	No	Yes
Platform			
Frequency Dependent Coef.	Yes	Yes	No
Full CWP Coupling	Yes	Yes	Yes

a IP denotes work in progress; AP, approximate treatment

The TRW method enjoys certain actual or potential advantages in treatment of non-linearities, wave current interactions and directional waves. It is disadvantaged in the treatment of platform hydrodynamics. It is likely that the TRW method has a greater computational costs, although no suitable basis exists for comparison of costs with the three methods.

Validation of Methods

Considerable differences exist in the state of validation of these methods. Based on the limited comparisons of References 10 and 11 and Figures 1 and 2 it is concluded that the three methods are in good agreement for cases of relatively stiff CWP's attached to a barge-like hull (the APL barge[14]), operating in uni-directional head waves, while the NOAA/DOE and Paulling methods are also in good agreement for beam seas.

A rather extensive validation effort has been carried out for the NOAA/DOE methods[10,14]. Comparison of theoretical prediction with model test data[7,4], have shown generally good agreement. This is illustrated by sample comparisons shown for the OTEC-1 in Figure 3 and for a 400 MW spar design in Figures 4 and 5. Predicted and measured platform motions are in generally very good agreement. Predicted and measured CWP bending moments are in generally good agreement, although some significant differences in peak bending moments do exist (see for example Figure 5). Various capabilities of these methods have also been extensively validated through comparison with analytical and independent theoretical methods[10].

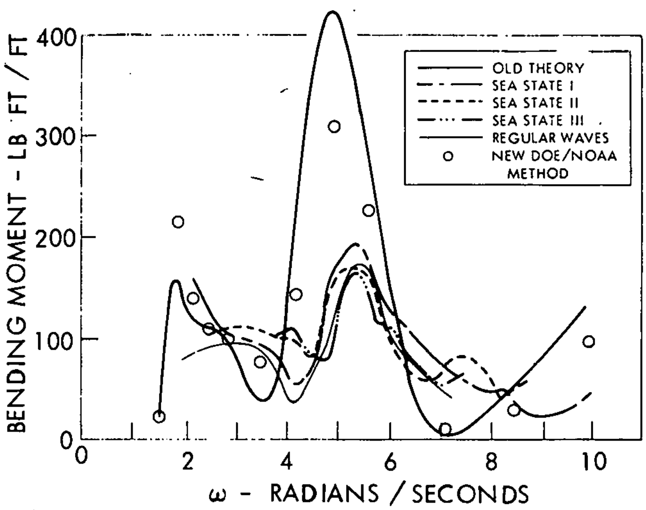


FIGURE 3 - COMPARISON OF MEASURED AND PREDICTED CWP BENDING MOMENTS FOR STIFF OTEC-1 CWP WITH RIGID ATTACHMENT.- Reference 10

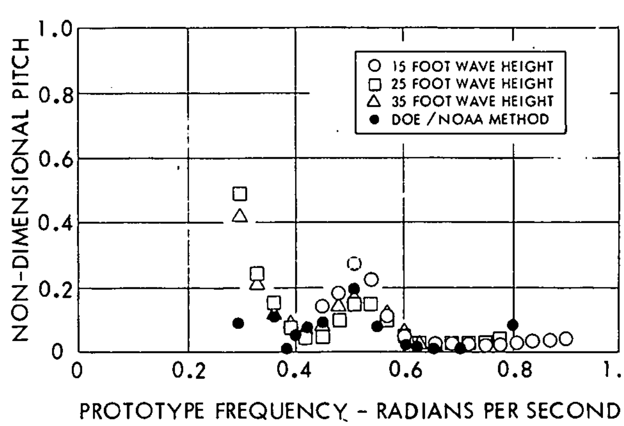
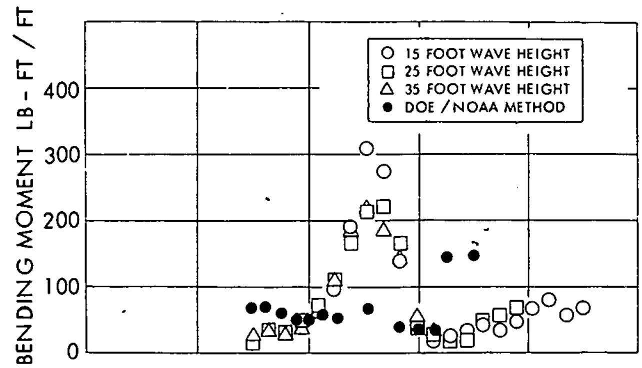
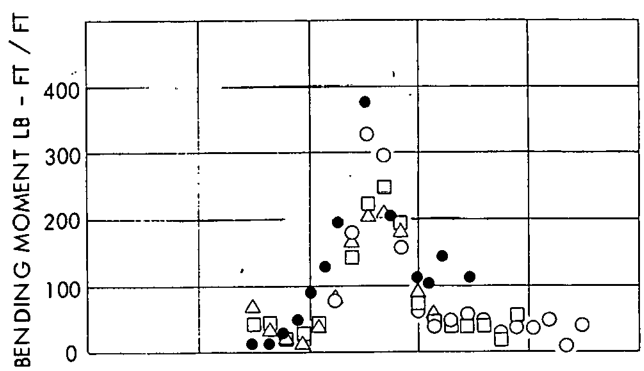


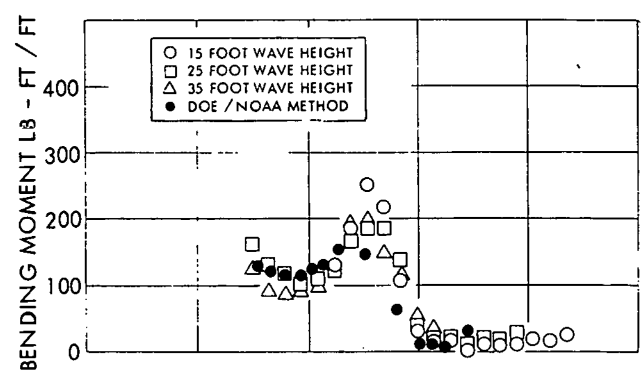
FIGURE 4 - COMPARISON OF MEASURED AND PREDICTED PITCH MOTIONS FOR 400 MW SPAR WITH RIGIDLY ATTACHED CWP - Reference 15



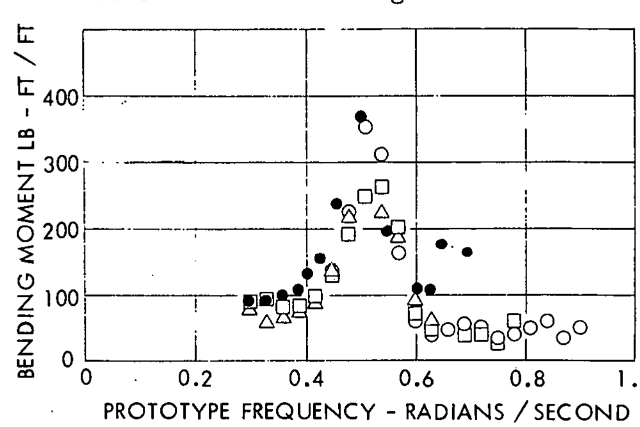
a. Top of CWP



b. 25 Percent of Length Down CWP



c. 50 Percent of Length Down CWP



d. 75 Percent of Length Down CWP

FIGURE 5 - COMPARISON OF MEASURED AND PREDICTED BENDING MOMENTS FOR CWP RIGIDLY ATTACHED TO 400 MW SPAR - Reference 15

The TRW methods for response to waves were validated by comparison with results from the other programs for two non-jointed CWPs attached to the APL barge[22]. Some other capabilities, such as static response to current, have also apparently been validated, although details are unknown.

The Paulling methods have been validated, to date, only by the comparisons shown in Figures 1 and 2. It is intended that data from the at-sea tests of the DOT X-1 platform be used as the primary basis for validating this method.

Status of Model Test Techniques

To date two basic types of CWP model tests have been carried out. Froude scaled models of complete CWP/platform designs have been tested in waves and/or current, and circular cylinders, representing a section of a CWP, have been tested, usually in a steady flow representing a current. These tests are quite different but have complementary purposes.

Tests of Complete CWP/Platform Configurations

These tests are designed primarily to model the dynamic response of a CWP/platform configuration to ocean waves and to measure platform and CWP motions and CWP loads. In laboratory tests to date, platform motions, joint loads and CWP bending moments have been measured. In the DOT X-1 at-sea tests CWP lateral accelerations were also measured. The primary difficulty in such tests is the proper scaling of CWP stiffness and weight or tension.

HYDRONAUTICS has developed methods for building CWP models of large scale ratio with correct structural stiffness[14]. Figure 6 shows a section of an 8.75 inch diameter, 0.013 inch wall thickness model CWP. This model is built of a single layer of glass fibre to minimize stiffness and uses spirally wound glass fibre roving to provide adequate buckling stiffness; a serious problem with such models is bending induced buckling. This model has a low density and models an essentially naturally buoyant prototype CWP. Special techniques, as described in the next section, are required to scale tension as well as stiffness. These techniques have not been used to date.

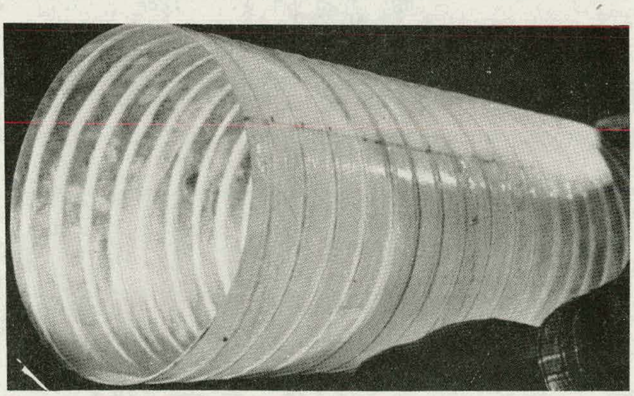


FIGURE 6 - SECTION OF GRP CWP MODEL

Because of the severe limitation imposed by Froude scaling, it is desirable to conduct supplementary, forced oscillation tests of complete CWP models, with or without steady velocity simulating current. Such tests, and the required model scaling, are discussed in the following section. While no tests of this type have yet been conducted, capabilities and equipment for such tests exist at HYDRONAUTICS, and several proposals have been made for such tests.

Tests of Circular Cylinders

As noted earlier, there exist a large body of data for rigid circular cylinders. While most tests have been in steady, uniform flow, data do exist for tests with velocity shear[15] and for tests with oscillatory flow[18]. In addition some data exist for tests of flexible cylinders and elastically supported cylinders. Reference 15 provides a good introduction to these data. While it is not possible to discuss these data in any detail, several aspects of the data seem worthy of note.

Cylinder oscillatory added mass and damping forces are dependent on the amplitude of fluid and/or cylinder motion. The influence of combined steady and oscillatory flow on these forces is unknown, as test data do not exist. Data are badly needed for small amplitude oscillatory motions and for combined steady flow and oscillatory flow or motion.

Reference 15 concludes, from subcritical Reynolds number data, that with expected ocean current velocity shears, the correlation length for vortex shedding will be approximately equal to the CWP length, and hence that vortex shedding induced loads can be a significant source of CWP loading unless effective vortex supressors are used. This conclusion is disputed in Reference 11, where a maximum correlation length of 10 diameters is proposed. While a definate conclusion about correlation length cannot be reached with available data, the conclusion of Reference 11 seems much more realistic.

Required Scaling For Modelling CWP Hydroelastic Responses

Hydroelastic Responses

The types of hydroelastic response considered most important for an OTEC-CWP are lateral beam response, local shell response and longitudinal beam response. Although local shell response (ovalling and buckling) and the heave or longitudinal beam response of the CWP are important and subject to modelling, the following analysis is directed toward understanding the requirements for modelling the lateral beam response. Similar analyses of the other modes should be carried out.

Modelling Lateral Beam Response

In order to achieve complete similarity between a model and prototype, it is necessary that all forces have the same proportion in model and prototype. Since the CWP lateral response is most often considered to be harmonic for a harmonic input, it is necessary to examine only the response to a single harmonic input to determine similarity requirements.

Mode Shape and Natural Frequencies. A vertical beam oscillating sinusoidally with amplitude, X_o, and frequency, ω, will have discrete natural frequencies corresponding to different mode shapes. The mode shapes and natural frequencies are determined by the end conditions, the non-dimensional static deflection response of the beam (the stiffness resistance of the beam to static lateral displacements) and the mass distribution along the beam.

The CWP stiffness distribution is determined by the force required to obtain unit CWP displacement from the vertical as resisted by (1) beam rigidity, (2) tension due to pipe weight and (3) changing the direction of momentum flux in the pipe. These stiffnesses at a prescribed location along the pipe (z/L) will be proportional to (1) EI/L^3, (2) W/L, and (3) $\rho AU^2/L$, respectively. The natural frequencies are determined by the ratio of these stiffnesses to the mass. Thus the first requirements for achieving similarity between a model and its prototype, whose mass distributions are similar along the CWP, are:

$$\left[\frac{EI}{L^3}\left(\frac{z}{L}\right)\right]_{model} = K\left[\frac{EI}{L^3}\left(\frac{z}{L}\right)\right]_{prototype} \qquad (1)$$

$$\left[\frac{W}{L}\left(\frac{z}{L}\right)\right]_{model} = K\left[\frac{W}{L}\left(\frac{z}{L}\right)\right]_{prototype} \qquad (2)$$

$$\left[\frac{\rho AU^2}{L}\left(\frac{z}{L}\right)\right]_{model} = K\left[\frac{\rho AU^2}{L}\left(\frac{z}{L}\right)\right]_{prototype} \qquad (3)$$

Clearly the constant K must be identical in all three equations so that the distribution of total lateral stiffness is similar in model and prototype. The requirement that the mass distribution be similar may be expressed as:

$$\bar{m}\left(\frac{z}{L}\right)_{model} = \bar{m}\left(\frac{z}{L}\right)_{prototype} \qquad (4)$$

where $\bar{m}$ is the sum of pipe mass, internal water mass and water added mass per unit length.

The square of a CWP natural frequency with arbitrary end conditions can be expressed as:

$$\omega_n^{\ 2} = \frac{A_n}{B \cdot m_o}\left[C_1\frac{EI}{L^3} + C_2\frac{W}{L} + C_3\frac{\rho AU^2}{L}\right] \qquad (5)$$

where,

A_n is a coefficient of the n^{th} mode and depends only on the end conditions.

C_1, C_2, and C_3 are coefficients which depend only on the distribution of EI, W and ρAU^2 along the length of the CWP and the mode number, n. Equation (3) may be rewritten as:

$$\omega_n^{\ 2} = K_1\frac{EI}{m_o L^3}\left[1 + K_2\frac{WL^2}{EI} + K_3\frac{\rho AU^2L^2}{EI}\right] \qquad (6)$$

For example, for a simply supported beam with uniform EI and upward pipe flow and with uniform tension W (all weight concentrated at the bottom); K_1 is $(n\pi)^4$ and K_2 and K_3 are $(n\pi)^{-2}$.
Thus, if the stiffness distributions are similar, the following conditions must be met to maintain identical proportions between stiffnesses:

$$WL^2/EI = \text{const}; \quad \rho AU^2L^2/EI = \text{const} \qquad (7)$$

It is important to understand the effect of the requirement that the parameter WL^2/EI be equal in model and prototype. If the volume enclosed by the pipe is assumed to have an average density $\bar{\rho}$ then the requirement of constant WL^2/EI may be written as:

$$\frac{\left(\frac{\bar{\rho}}{\rho w} - 1\right)_m}{\left(\frac{\bar{\rho}}{\rho w} - 1\right)_p}\left(\frac{D_m}{D_p}\right)^2\left(\frac{L_m}{L_p}\right)^3\frac{(EI)_p}{(EI)_m} = 1 \qquad (8)$$

If EI/m_oL^3 is constant in model and prototype and $\rho AU^2L^2/EI$ is negligible, the ratio of model to prototype natural frequency may be obtained from Equation (5) as $(W/m_oL)_m/(W/m_oL)_p$. If the model is geometrically similar and Froude scaled or if $(\omega_{n,m})^2/(\omega_{n,p})^2 = L_p/L_m$, then $(\bar{\rho}/\rho_w - 1)_m = (\bar{\rho}/\rho_w - 1)_p$ and the added mass coefficients $C_{m,m}$ and $C_{m,p}$ must be equal. These equations show that if the frequency is to be Froude scaled, the model diameter cannot be distorted from geometric similarity unless the model EI and total weight are correspondingly distorted. Such a weight distortion cannot be permitted in a combined platform-CWP model, so that a CWP model tested with a platform cannot have diameter distortion.

Forces. It is instructive to examine the displacement and kinematic and dynamic forces acting on the CWP by treating a unit length of the CWP as a simple damped oscillator with natural frequency ω_n. Let us assume that this simple oscillator is initially displaced by a constant force (due to steady current) and is excited by the oscillatory displacement, $\chi_o e^{i\omega t}$, where $\chi_o = KX_o$ and where X_o is the displacement at the top of the pipe, Figure 7. The constant K is defined by the CWP mode shape.

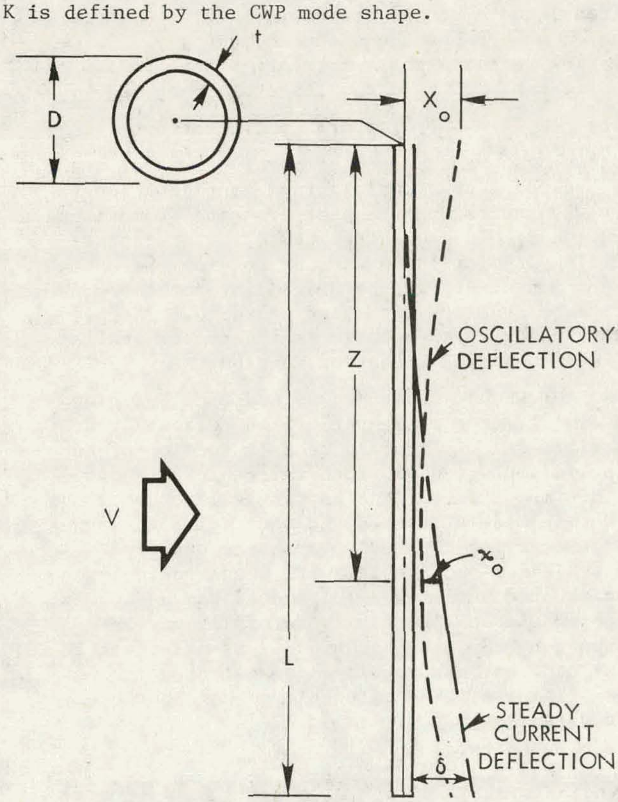


FIGURE 7 - DEFINITION SKETCH

The steady force per unit length due to a steady current is:

$$F_d' = k\left(\frac{X_o}{L}\right) = \omega_n^{\ 2} \cdot \bar{m} \cdot \chi_o \qquad (9)$$

where k is the beam stiffness $(m_o\omega_n^{\ 2})$ and δ is the steady deflection. The amplitudes of the displacement force due to oscillatory excitation, damping force and inertia force per unit length of CWP are, respectively:

$$F_k' = C_D\tfrac{1}{2}\rho_w\, D(\omega\chi_o)^2 = C_D\tfrac{1}{2}\rho_w\, D\left(\frac{\omega}{\omega_n}\right)^2\omega_n^{\ 2}\chi_o^{\ 2} \qquad (10)$$

$$F_i' = \bar{m}\,\omega^2\chi_o = \bar{m}\left(\frac{\omega}{\omega_n}\right)^2\omega_n^{\ 2}\chi_o \qquad (11)$$

Since the motion (is harmonic) about the steady deflection, δ, the total force per unit length is:

$$F'_{total} = C_{D_o} D\tfrac{1}{2}\rho w V^2$$

$$+ \left[1 + \frac{C_D \, \rho w D \, \frac{\omega}{\omega_n}^2 \, X_o}{\overline{m}} + \left(\frac{\omega}{\omega_n}\right)^2\right] \overline{m} X_o \omega_n^{\;2} \qquad (12)$$

Complete dynamic similarity will be achieved only if each of the terms in brackets is identical for model and prototype, or if:

$$C_D \, D \, X_o \rho_w \, / \overline{m} = \text{const}; \; \omega/\omega_n = \text{const} \qquad (13)$$

For oscillatory motions the ratio of model force to prototype force is:

$$\frac{F_{model}}{F_{prototype}} = \frac{\left(\overline{m} \, x_o \omega_n^{\;2}\right)_m}{\left(\overline{m} \, x_o \, \omega_n^{2}\right)_p} \cdot \left(\frac{L_m}{L_p}\right) = \frac{\left(m_o \; \omega_n^{\;2} \; x_o\right)_m}{\left(m_o \; \omega_n \; x_o\right)_p} \qquad (14)$$

It is not necessary that the ratio of model and prototype steady force satisfy equation (13); it is only necessary that the non-dimensional deflection curve be identical in model and prototype. This requirement for similar deflection curves will be satisified if:

$$C_{D_o} \, D\rho_w V^2 / \, \overline{m} \, L \, \omega_n^{\;2} = \text{const} \qquad (15)$$

The mass terms $\overline{m}$ and $m_o = \overline{m}L$ depend on the pipe geometry and added mass. Thus for a pipe of external diameter D and thickness t

$$\overline{m} = C_o \frac{\Pi}{4} \rho_w D^2 = \frac{\Pi}{4} \rho_w D^2 \left[\frac{\rho_m}{\rho_w} + C_m - 1 + (1 - 2\tfrac{t}{D})^2 \, \frac{\rho_i}{\rho_w} - \frac{\rho_m}{\rho_w}\right] \qquad (16)$$

where

$$C_o = \frac{\rho_m}{\rho_w} + C_m - 1 + (1 - 2\,\frac{t}{D})^2 \, \frac{\rho_i}{\rho_w} - \frac{\rho_m}{\rho_w}$$

If the average density of the CWP and its internal water is $\overline{\rho}$, the coefficient C_o is:

$$C_o = \left(\frac{\overline{\rho}}{\rho_w} - 1 + C_m\right).$$

where,

- ρ_w = water density
- ρ_m = pipe material density
- ρ_i = internal density ($\rho_i = \rho_w$ in the prototype)
- C_m = cylinder added mass coefficient
- t = pipe wall thickness
- D = pipe outside diameter

Substitution of Equation (16) into Equations (13) and (15) gives the following conditions for similarity:

$$C_{D_o} \, D \, V^2 / C_o \, L \, D^2 \omega_n^{\;2} = \text{const} \qquad (17)$$

$$C_D / C_o \times X_o / D = \text{const}$$

with this similarity, Equation (14) becomes:

$$\frac{F_m}{F_p} = \frac{\left[C_o \; L \; D^2 X_o \; \omega_n^{\;2}\right]_m}{\left[C_o \; L \; D^2 X_o \; \omega_n^{\;2}\right]_p} \qquad (18)$$

Hydrodynamic Effects. It is important to understand the factors that influence the hydrodynamic coefficients $C_{D,o}$, C_D, and C_m (C_m affects C_o). Furthermore, steady flow past a cylinder causes periodic vortex shedding which imposes oscillatory forces that must be taken into account in modelling a CWP.

The steady drag coefficient $C_{D,o}$ is dependent on the ratio of roughness height, k, to diameter, k/D, and the Reynolds number, VD/ν, where V is the current velocity and ν is the kinematic viscosity. For small scale models, C_{D_o} is approximately 1.25 (for any roughness) while full scale C_{D_o} varies from 0.6 to 0.9 for relative roughnesses of 10^{-5} to 10^{-3}, respectively[15,18].

The oscillatory drag coefficient C_D is dependent on the oscillatory Reynolds number, $\omega x_o D/\nu$, where x_o is the amplitude of motion and ω is the frequency; the relative roughness, k/D; and the relative amplitude of oscillation (x_o/D). For typical values of (k/D) and (x_o/D), Ref. 18 shows that the value of C_D becomes constant for Reynolds numbers greater than about 3×10^5 and that it is sometimes possible to match model and prototype values of C_D by properly selecting the model roughness.

The virtual mass coefficient C_m has the same type of dependence on $\omega x_o D/\nu$, k/D, and x_o/D as does C_D. All remarks concerning C_D therefore apply also to C_m.

The frequency of vortex shedding, f_s, is defined by the Strouhal number, $S = f_s D/V = \omega_s D/2\pi V$. Reference 15 shows that the Strouhal number is approximately 0.2 for roughness Reynolds numbers less than 50. Hydroelastic similarity between model and prototype can thus be achieved, when vortex shedding is an important factor, only if the ratio of shedding frequency to natural frequency is identical, or ω_s/ω_n equals a constant. Vortex shedding produces an alternating side or lift force F_s whose root mean square value is $\overline{F}_s = \overline{C}_L \, \rho_w A V^2/2$ where $\overline{C}_L$ is the rms lift coefficient. The existence of this side force imposes the requirement that

$$\overline{C}_L \; D \; V^2 / C_o \; X_o \; D^2 \omega_n^{\;2} = \text{constant} \qquad (19)$$

Reference 18 gives the value of $\overline{C}_L$ as approximately 0.1 for kV/ν < 600 and 0.28 for kV/ν > 600. Obviously if the values of $C_{D,o}$, C_D, C_m, S, and $\overline{C}_L$ were identical in model and prototype, the modelling problem would be greatly simplified. However, this is not the case, but it is possible, as noted below, to devise models which account for all or at least most of these hydrodynamic effects.

Summary of Model Laws

The requirements for similarity in the lateral hydroelastic response of CWP models may be summarized as follows: (1) Similar non-dimensional distribution of (EI/L^3), (W/L) and $(\rho A U^2/L)$ (see Equation 10; (2) Similar non-dimensional distribution of $\overline{m}$ (see Equation (2)); (3) Identical values of the following non-dimensional parameters in model and prototype:

$$(1/C_o)(E/\rho_w V^2)(I/L^4)(V/\omega_n D)^2, \; WL^2/EI,$$

$$(\rho_w A U^2 L^2/EI, \; \omega/\omega_n, \; (C_{D_o}/C_o)(D/L)(V/\omega_n D)^2,$$

$$(C_D/C_o)(X_o/D), \; \frac{\omega_s}{\omega_n} = (2\pi S)(V/D\,\omega_n) \text{ or } S\,(V/\omega_n D), \text{and} \qquad (20)$$

$$(\overline{C}_L/C_o)(D/X_o)(V/\omega_n D)^2;$$

(4) The ratio of model to prototype oscillatory forces will then be:

$$\frac{F_m}{F_p} = \frac{\left[C_o \; L \; D^2 X_o \; \omega_n^{\;2}\right]_m}{\left[C_o \; L \; D^2 X_o \; \omega_n^{\;2}\right]_p} \qquad (21)$$

Geometrically Similar Froude Model

For cases where the CWP and platform models are to be tested in waves, it is necessary to model according to the Froude criterion (that is, identical model and prototype accelerations). If the model is also geometrically similar, the Froude criterion will be satisfied if $E_m = \lambda \; E_p$, $C_{D_{o,m}} = C_{D_{o,p}}$, $C_{D,m} = C_{D,p}$,

$$\overline{C}_{L,m} = \overline{C}_{L,p}, \text{ and } S_m = S_p. \qquad (22)$$

With the proper choice of model relative roughness, $C_{o,m}$ can be made equal to $C_{o,p}$. For small models of thin walled CWP's with low values of E_p it can become impractical to construct a model with $E_m = \lambda E_p$.

Reference 15 summarizes available date for $C_{D,o}$. These data indicate that for Reynolds numbers greater than 2×10^5, C_{Do} depends only on the relative pipe roughness and is approximately 0.75 and 1.0 for $k/D = 10^{-5}$ and 10^{-3}, respectively. For Reynolds numbers less than 2×10^5, C_{D_o} is approximately 1.25 and not greatly influenced by the relative roughness. Therefore it is possible to achieve identical values of $C_{Do,m}$ and $C_{Do,p}$ only if the model Reynolds number exceeds 2×10^5.

Figure 8 presents the typical effects of model scale ratio and roughness on the hydrodynamic coefficients $C_{D,o}$, C_D, C_m, S, and $\overline{C}_L$ based on prototype diameter, current velocity and oscillatory velocity amplitude of 50 feet, 1 fps and 28 fps, respectively. Figure 8 illustrates that the value of $C_{D,o}$ cannot be simulated in a Froude model unless the model scale ratio is greater than about 1/10. For a 3000 feet long prototype CWP, such a large scale will require a facility of 300 feet depth.

Figure 8 does show that by proper choice of model roughness it is possible to select model scale ratios much less than 1/10-th that will reproduce the full scale values of all coefficients except C_{Do}. With a large value of $k/D = 10^{-2}$, the smallest scale model that will properly scale $\overline{C}_L$ in the model is 1/30th. However, it is not necessary to exactly simulate the value of $\overline{C}_L$ in the model in order to study the coupling of the CWP with shed vortices as long as the value of S is simulated and the actual value of $\overline{C}_L$ is known or can be estimated.

The deepest existing U.S. facility that includes wave making capability is the HYDRONAUTICS Ship Model Basin which has an 11 feet diameter, 42 feet deep section. For a 3000 feet long CWP, the resulting model length scale ratio is 1/75. This scale ratio of $\lambda = 1/75$ is indicated in Figure 8. Except for simulating the value of $\overline{C}_L$, Figure 8 shows that the scaling problems are essentially identical for scale ratios between 1/75 and 1/10. Thus, a facility approximately 3 times as large as the HSMB will add the capability of simulating the side force coefficient $\overline{C}_L$, but will still not simulate the value of C_{Do}.

For a 1/75 model scale the problems associated with a geometrically similar Froude model may be summarized as follows:

- o For a full scale material with a low elastic modulus it will be necessary to distort thickness/

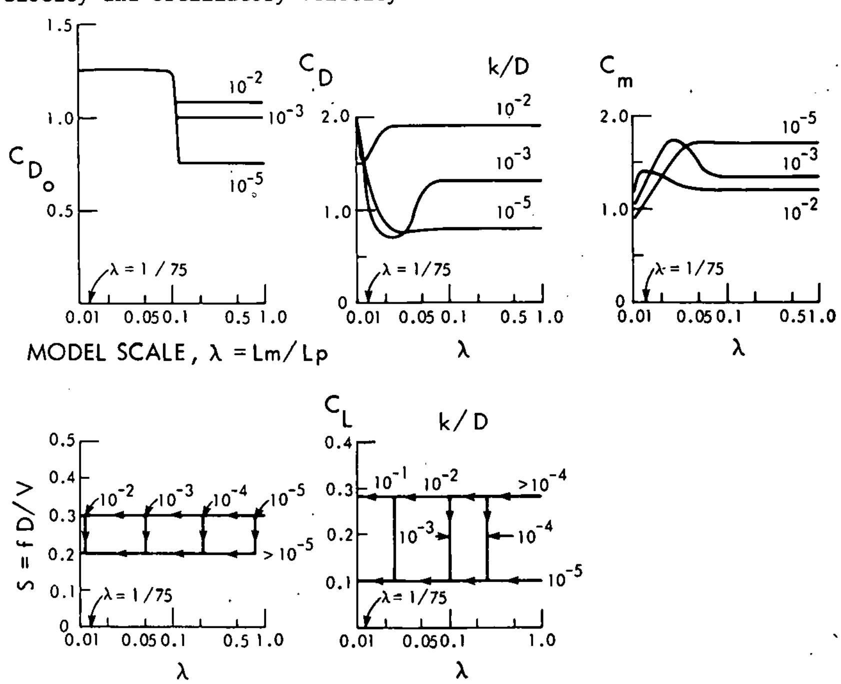


FIGURE 8 - INFLUENCE OF MODEL SCALE RATIO ON THE HYDRODYNAMIC COEFFICIENTS FOR FROUDE MODEL DEFINED IN TEXT

diameter ratio (or use a built-up model) while keeping constant the basic parameters

$$(EI/\rho_w\, V^2\, L^4) \text{ and } (WL^2/EI)$$

- o It is not possible to maintain C_{D_o} constant in model and prototype in an undistorted Froude model, and hence the model static deflection will be greater than the prototype deflection.
- o It may be possible to select a model roughness such that C_D is constant in model and prototype.
- o Eddy shedding caused by oscillatory motion alone will probably not generally exist and thus (ω_s/ω_n) scaling may not be important in the absence of a current.
- o With a current, eddy shedding will be important. The ability to achieve identical values of S in the model and prototype by using appropriate roughness is marginal, and will depend on the particular case.
- o Thus the magnitude of $\overline{C}_L$ and of the oscillatory side forces will be about 65 percent less in the model than in full scale.

It is therefore not possible to achieve complete dynamic similarity in a geometrically similar (1:75) Froude model if a current is present. In the absence of a current the response can generally be made similar if the effect of eddy shedding caused by the oscillatory motion is negligible and if the model pipe relative roughness is properly selected. If the model Strouhal number is the same as full scale, the effect of vortex shedding can be studied in a Froude model if the approximately 50 percent increase in static deflection is acceptable.

Distorted Models

The fact that C_{D_o}, S, and $\overline{C}_L$ are generally different in model and prototype makes it impossible to exactly simulate the lateral response of the CWP when a current is present. However, if the diameter of the model and its excitation amplitude are distorted then similarity can be achieved in the steady deflection and vortex shedding frequency, although the magnitude of side force will not be correctly simulated. To achieve similarity in the shedding frequency with current and oscillatory excitation, it is necessary that:

$$(V/\omega_n\, D)\; s = \text{constant} \tag{23}$$

$$(X_o/D)\; S = \text{constant (if } \omega/\omega_n \text{ is constant)} \tag{24}$$

Substituting Equations (23) and (24) into Equation (20) leads to the following requirements for the distorted model:

$$\frac{(D/L)_m}{(D/L)_p} = \frac{(C_{D_o}/C_o)_p}{(C_{D_o}/C_o)_m}\left(\frac{S_m}{S_p}\right) \tag{25}$$

$$\frac{(X_o/D)_m}{(X_o/D)_p} = \left(\frac{S_p}{S_m}\right) \tag{26}$$

$$\frac{{}_{,m}}{{}_{,p}} = \left(\frac{V_m}{V_p}\right)\left(\frac{S_p}{S_m}\right)\frac{(C_{D_o}/C_o)_m}{(C_{D_o}/C_o)_p}\left(\frac{L_p}{L_m}\right) \tag{27}$$

It has been previously noted that if the parameter $(WL^2)/EI$ is important, it is not possible to consider a diameter distortion if the model is to be attached to a platform model. However, if the CWP alone is excited by a mechanical oscillator, distorted models can and should be considered.

Oscillator Models

Much important information about CWP lateral hydroelastic response can be obtained without testing a platform of a CWP model in waves. A CWP model may be attached to a mechanical oscillator, with properly simulated attachment end conditions, and oscillated over a range of frequencies and amplitudes in water and with current and its response recorded. Oscillatory forces imposed by wave orbital velocities are not simulated, but all other factors influencing pipe response can be studied. Mechanical oscillators can directly measure in- and out-of-phase forces at the top of the CWP making possible the study of CWP response and the deduction of effective value of C_D and C_m.

When Froude scaling is not imposed, higher model frequencies may be used and thus higher values of the oscillatory Reynolds number can be achieved. The oscillator model technique for such a case is outlined below for a model whose length scale is 1/75.

Reference 18 shows that C_D is dependent only on relative roughness and, to a slight extent amplitude of motion, when the Reynolds numbers is greater than about 3×10^5. It will therefore be instructive to devise model tests that reach a Reynolds number of at least 3×10^5 for amplitude ratios typical of full scale CWP's and hence not covered by Reference 18. For most CWP configurations of interest, model Reynolds number will exceed 3×10^5 for a geometrically similar model if $\omega_{n,m}/\omega_{n,p}$ is set equal to L_p/L_m or $(1/\lambda)$ rather than $(1/\lambda)^{\frac{1}{2}}$ as in a Froude scaled model. Since the resulting model and full scale velocity are identical, such a model is called a constant velocity model.

Various relative roughnesses could be used on the constant velocity model and the values of $\overline{C}_D$ and $\overline{C}_m$ measured by the oscillator mechanism for various amplitudes and frequencies. Such data obtained on a CWP oscillating in realistic modes are needed to augment or verify available data such as that of Ref. 18.

Model Examples

In order to illustrate these modelling techniques three examples are presented. In the first two examples a 30 foot diameter prototype CWP with 0.2 foot wall thickness and 2×10^6 psi modulus of elasticity and a model length scale of 1:75 are assumed. In the third, the 1:5 scale ratio CWP tested at-sea on the DOT X-1 platform is considered.

Froude Model. It is impractical to model the prototype with a geometrically similar model having $E_m = 1/75\ E_p$. A built-up model with a central structural spine is therefore assumed. Although the spine may be solid, a hollow spine is assumed to be more desirable. The model O.D. cannot be distorted and is thus 0.4 feet or 4.8 inches. If a model spine of glass reinforced plastic ($E = 2 \times 10^6$ psi) and a 0.75 inch diameter is assumed and if a light polyurethane foam is used to attach the mass segments so as not to affect the effective model EI (some experience is required to determine the actual effect of the method of mass attachement), the spine thickness is given by

$$\frac{t}{d} = \lambda \; (E_p/E_m)(D_m/d)^4 \; \frac{t_p}{D_p} = 0.15$$

and hence the spine wall thickness is 0.11 inches.

The average density of the complete 4.8 inch diameter model must be identical to the full scale, and hence

$$(\bar{\rho}/\rho_w)_m = 1.0266$$

If the model CWP is assumed to have hollow, nonstructural mass segments with outside diameter D_m and inside diameter 0.25 D_m, and the specified spine, it can be shown that mass segment density ratio ρ_m/ρ_w must be 1.046. The resulting mass segments might be selected as hard wood with lead ballast inserts.

Constant Velocity Model. To better understand the effects of oscillation amplitude and relative roughness on CWP coefficients C_D and C_m, it is desirable to test a constant velocity model, in which $(EI)_m = \lambda^4 (EI)_p$.

For the specified prototype, the model EI can be achieved in an undistorted diameter model by making $E_m = E_p$, and $(t/D)_m = (t/D)_p$. A 1/75 scale GRP model would have a practical thickness equal to 0.036 inches. A thicker wall of lower modulus or a central spine model could also be considered. For this constant velocity model the average specific gravity could be achieved either by using a central structural spine with surrounding mass elements or by using a weight chain, within the thin wall GRP CWP. This weight chain must be designed to have a negligible effect on the bending stiffness of the pipe shell.

Scaling of X-1 Steel CWP. The unjointed steel CWP tested on the DOT X-1 platform was selected to be a 1/6 scale model of a 30 foot diameter, 3000 foot long modular plant CWP. It is interesting to examine how this 3/16 inch wall thickness pipe scales to a 30 foot diameter prototype, using Froude scaling laws.

The scaled wall thickness for a steel prototype CWP is 6.75 inches or 0.56 feet, and the required CWP wall density is 4.17 slugs per cubic feet. This density is much less than that of steel (15.2). The model CWP would therefore correspond to a steel CWP with added buoyancy. As an example, a 30 foot O.D. steel CWP with 0.56 foot wall thickness would require a 20 pound per cubic foot foam annulus with an inside diameter of 17.5 feet or a wall thickness of 6.25 feet (0.56 feet of steel and 5.69 feet of foam). It is unlikely that such a large quantity ot foam would be used in a prototype CWP.

Conclusions

The technology associated with theoretical and experimental analyses of OTEC CWPs and platforms is still evolving and conclusions based on current status, as described in this paper, may soon be outdated. Nonetheless, it seems appropriate to take note of several important conclusions to be drawn from this current status. These include:

1. Available theoretical methods for calculating the response of platform CWP systems to waves appear to be in general agreement; however, validation is in most cases limited. Many capabilities of the methods have received no validation. Further validation is clearly needed.

2. Tests of the largest scale ratio (about 1 to 75) possible in existing laboratory facilities are subject to various scale effects. These scale effects are most important for tests in combined waves and current. For tests in waves alone, or for tests of the CWP alone, roughness or velocity can be adjusted to avoid serious scale effects.

3. Analysis of the lateral CWP response reveals that both undistorted and distorted models of CWP's tested at relatively high frequencies should be useful in augmenting the results obtained from conventional small scale Froude models.

4. Laboratory facilities of sufficient depth to avoid all scale effects do not exist and hence large scale, at-sea tests such as the recent tests with the DOT X-1 platform are needed to validate laboratory experiments and theoretical methods. Such tests are probably too expensive to be used as a regular part of the design process.

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A LIMITING MECHANISM FOR THE HEAVE-INDUCED, MATHIEU-TYPE INSTABILITY OF A COLD WATER PIPE

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Abstract

When the hull of an OTEC plant is subjected to heave motion, the acceleration causes oscillating tension in the cold water pipe. If the pipe has a natural bending period close to two times the period of heave motion, unstable vibrations may be initiated. For the "small angle" approximation in general use for cold water pipe studies those vibrations are limited only by damping, and solutions may show excessive stresses before a limit cycle is reached. This paper shows that when the small angle approximation is refined to permit second order terms ($\cos \theta \neq 1$) an additional source of oscillating tension is recognized. The additional oscillating tension is out of phase with the heave induced component, and counteracts its destabilizing effects to make the limit cycle occur at, usually, moderate stress levels.

Introduction

In some recent response studies of a potential cold water pipe design, using the TRW time domain program for analysis, it was noted that in some conditions the calculated peak stress levels were disproportionately sensitive to hull motions. This effect was not reproduced in frequency domain (linear) analysis of the design.

Additional analysis and review was performed in efforts to verify the effect and to correct the time domain program if the results proved spurious. The review indicated that the apparent disparities were not spurious and were the result, primarily, of the high levels of hull heave and resultant time variation of tension in the pipe. This tension variation transforms the normal pipe equation into a generalized form of Mathieu's equation whose solution will, under some circumstances, be unstable.

Thus, the high stress levels calculated for this condition were judged to be qualitatively valid. However, further review of the results indicated that they were conservative because the "small angle" approximation used in the time domain program concealed an additional source of tension variation which would have stabilizing effects if included. A time domain program which does not employ small angle approximations will be necessary for full quantitative assessment.

Analysis

The bending equation for a cold water pipe is, as first derived in Reference 1:

$$)\frac{\partial^2 Y}{\partial t^2} + \frac{\partial^2}{\partial x^2} EI(x) \frac{\partial^2 Y}{\partial x^2} - \frac{\partial}{\partial x} P(x,t)\frac{\partial Y}{\partial x} = F_H(x,t) \quad (1)$$

$m(x)$	local mass/unit length
$EI(x)$	local flexural rigidity of pipe
$P(x,t)$	local instantaneous tension in pipe
$F_H(x,t)$	local hydrodynamic force/unit length

Provided m, EI, P are independent of x, and the boundary conditions of the pipe are pin-pin, the homogeneous portion of the equation is amenable to solution by separation of variables; the resulting ordinary differential equation for the time portion of Y is:

$$\ddot{T} + (\Omega_i^2 - \frac{p(t)}{m})T = 0 \quad (2)$$

$p(t)$ is the time-variable portion of P ($P = P_o + p(t)$)

Ω_i is the i^{th} natural period of the pipe in the absence of $p(t)$;

$$\Omega_i = (\frac{i\pi}{L})^2 \left(\frac{EI + \frac{P_o L^2}{(\pi i)^2}}{m} \right)^{1/2}$$

Equation 2 is recognized as Hill's equation or, if p(t) is a sinusoid, as Mathieu's equation.

The restrictions placed in the previous paragraph on m, P, EI, and boundary conditions would not apply to real cold water pipes and Equation 1 cannot, in general, be reduced to Mathieu's equation. Nevertheless, the widely available stability studies of Mathieu's equation (Reference 2, among many) provide valuable insight to possible pipe problems resulting from tension variation, and warrant discussion here.

Figure 1, adapted from Reference 3, is the well-known stability diagram of a system with variable elasticity. The abcissa is the ratio of a modal frequency, Ω_i, to ω, the frequency of p(t). The ordinate is the abcissa multiplied by the root of the ratio of the magnitude of p(t) to $\hat{P}$, the total steady modal restoring force,

$$P_o + EI(\frac{i^2 \pi^2}{L^2}) = \hat{P}_i$$

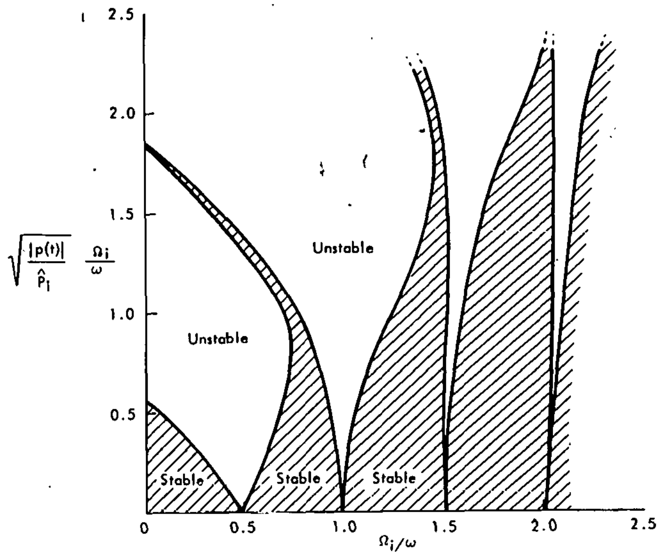

Figure 1
Mathieu Stability Diagram

The shaded regions indicate that the solutions to Mathieu's equation are stable, while the clear regions indicate unstable solutions.

It is seen that there are a large range of combinations of ω, Ω_i, p(t), $\hat{P}$ where unstable solutions are predicted. The most important of those regions (important because of their breadth, indicating that the instability is relatively insensitive to changes in the parameters) are those near Ω_i/ω = .5, 1. Thus, an ideal pipe with one of its natural periods near 16 seconds would, in the absence of damping, incur unstable motion if the tension varied even a small amount at a period near 8 seconds or 16 seconds.

Damping will ameliorate that finding importantly, so that even small amounts of damping will cause motions to be stable for small variations of tension, such as would be associated with a steel pipe subjected to even severe heave, or with a lighter wet-weight pipe subjected to moderate heave. With some pipes being considered, however, major tension variations result from even small heave accelerations and there is cause for concern if the idealizations necessary to force Equation 1 into the form of Equation 2 are approximately justified.

Application to Non-Ideal Pipes

To investigate the applicability of Mathieu's results to non-uniform pipes with applicable cold water pipe-appropriate boundary conditions (pin-free) TRW recently ran a case of a polyethelene (slightly positively buoyant) pipe with sufficient weight at the bottom to hold the pipe against small currents.

The analysis tool used was the TRW HULPIPE program, which is a time domain program developed specifically for analysis of cold water pipe-hull coupled motions. The program contains several features which cannot be included in the (necessarily linearized) frequency domain programs that are in wide use in the OTEC community, with the most important such features being velocity-squared drag and the time varying tension which is discussed here.

The pipe with weight had mass of almost 10^6 lbs, but wet-weight of only 30,000 lbs. Because of the low tension, high mass (including mass of water within pipe and added mass) and the low modulus of elasticity of such a polyethylene pipe, pipe natural periods had very high modal density, with three natural periods estimated to occur between 14.5 sec. and 17.8 sec. The pipe was subjected to heave motion at a period of 8 seconds. Motions were found to be negligibly small for heave amplitude of 2ft. (.04g) but to indicate instability limited only by the high damping associated with velocity-squared drag and high velocities, when heave amplitude was increased to 10ft. (intermediate levels were not investigated). The motion was plainly dominated by the 16 sec. component, (Ω_i = .5ω), with the only 8 sec. content (Ω_i = ω) present being of the order expected from the creation of harmonics by velocity-squared drag.

To remove the final approximation required to bring Equation 1 into the form of Mathieu's equation, the problem was run again using the hull motions associated with a random sea. For the hull and sea being considered, there was rms heave of 4.4 ft., with most of the energy concentrated at periods between 6 and 10 sec. (Figure 2). Peak heave accelerations of more than .25g were developed, but the heave acceleration was less than .16g for 86% of the time, and less than .08g 40% of the time. The peak motions were found to be much lower than those associated with the single component heave motion, but were approximately twice as great as the calculations made with otherwise identical conditions but no heave. This is interpreted as indicating that there was some instability, but its effects were limited by velocity-squared drag at a moderate and acceptable level.

Concern was not eliminated by this quantitative finding, because it was recognized that if drag coefficients were lower than the nominal ones employed, or if the seas or hull provided more heave than the assumed, or if the significant period of heave acceleration were slightly different in an unfavorable direction, significantly higher stresses could result. True bounding conditions were sought, and simultaneously a review of the validity of the TRW HULPIPE program technique for including time varying tension was initiated.

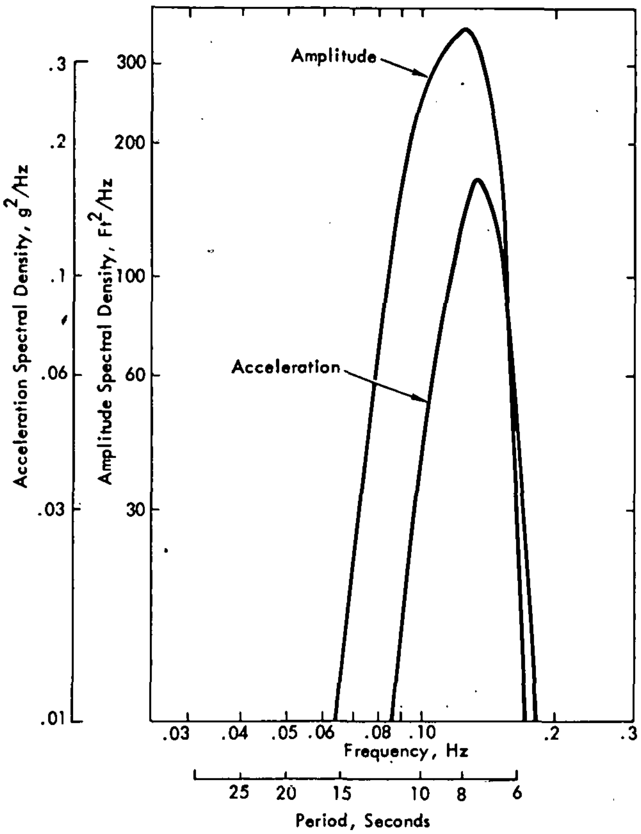

Figure 2
Heave Spectrum, 16 Ft. Significant Wave

Calculation and Refined Calculation of Local Tension

HULPIPE converts Equation 1 to an implicit finite difference formulation. $P(x,t)$ is recalculated at each time step, and the structural coefficients of Equation 1 are changed accordingly for the next time calculation. The method of solution was judged valid, and attention was shifted to the calculation of $P(x,t)$.

The program calculates the tension by:

$$P(x,t) = W_0 + M_0 \cdot \ddot{H}(t) + \int_x^L a(s)\left[\rho_p(s) \cdot \ddot{H}(t) + (\rho_p(s)-\rho_w)g\right] ds \qquad (3)$$

$\ddot{H}(t)$ instantaneous heave acceleration
W_0 weight at bottom
M_0 mass at bottom
$\rho_p(s)$ local density of pipe material

ρ_w sea water density
g acceleration of gravity

Equation 3 is an approximation to the solution of:

$$\frac{\partial}{x} Ea(x) \frac{\partial X}{\partial x} = m(x) \frac{\partial^2 X}{\partial t^2} \qquad (4)$$

subject to the boundary condition of hull heave motion. The approximation is an excellent one provided the axial natural periods are short in comparison to the periods of hull heave motion. For the pipe being considered the approximation was judged to be very slightly non-conservative, and sufficiently good as to not justify a higher order solution. However, it was realized that it was necessary to reexamine the validity of Equation 4.

Equation 4 and its approximate solution, Equation 3, was derived using the small angle theory ($\sin\theta \approx \theta$; $\cos\theta \approx 1$) that is implicit in Equation 1 and that is similarly assumed in all current competing cold water pipe programs. The time domain solution of Equation 1, however, was found to be yielding oscillating gimbal angles in the neighborhood of ±.2 rad. For that angle, the sine is within less than 1% of the angle, and the cosine is 0.98. These results would, in most cases, indicate that adequate accuracy is being achieved from small angle approximations. Nevertheless, on a long pipe non-zero angles result in the bottom of the pipe being elevated above its static position by:

$$\delta = L - \int_0^L \cos\theta ds \approx L\theta_0^2/4 \qquad (5)$$

For a 2000 ft. long pipe with gimbal angles of the order of .2 radians, δ is 20 ft! This is greater than the heave motion which induced the angle, and is in that sense, at least, not a higher order effect!

The shortening of the (projected) pipe length can be taken into account by adding to the vertical (heave) translation implicit in Equation 3, the shortening associated with the cosine of non-zero angles:

$$X(x) = H + \int_0^x \cos\left(\frac{\partial Y}{\partial s}\right) ds \qquad (6)$$

where H is heave amplitude.
To make an approximate evaluation of Equation 6 we assume that the pipe is oscillating in a limit cycle, so that the previously stipulated dictum against separation of the variables of Equation 1 has little weight. This permits approximating Y by:

$$Y \approx A \sin(kx) \sin(\Omega t) \qquad (7)$$

(Y will, usually, also contain a term in sinh(kx). The coefficient of the hyperbolic term is inevitably small, so that ignoring it creates an error only near the end of the pipe. For the effect being investigated, ignoring the sinh(kx) results in some conservatism. Because of space variation of tension, Y will also contain terms involving Bessel functions, which likewise may be ignored with little error.)

Differentiating Y in Equation 7, and replacing $\cos(\partial Y/\partial s)$ in Equation 6 by its second order approximation

$$\cos\left(\frac{\partial Y}{\partial s}\right) \simeq 1 - .5\left(\frac{\partial Y}{\partial s}\right)^2 = 1 - .5(Ak \sin\Omega t \cos ks)^2$$

permits Equation 6 to be written as

$$X(s) = H + s - .5\, A^2 k^2 \sin^2\Omega t \int_0^s \cos^2 k\zeta \, d\zeta \qquad (8)$$

wherein ζ is a dummy-variable of integration

The indicated integration may be performed, and the result differentiated in time to yield the local instantaneous vertical acceleration

$$\ddot{X}(s) = \ddot{H} - .5\, A^2 \Omega^2 k^2 \cos 2\Omega t \left(s - \frac{\sin 2ks}{2k}\right) \qquad (9)$$

$$= \ddot{H} - .5\theta_o^2\, \Omega^2 \cos 2\Omega \left(s - \frac{\sin 2ks}{2k}\right) \qquad (9')$$

Calculation of the local instantaneous tension can now be performed in the previous (Equation 3) manner, provided "$\ddot{H}$" in Equation 3 is replaced by $\ddot{X}(s)$.

Provided the limit cycle oscillation of Equation 7 is the result of periodic hull motion with a frequency double that of the response frequency (first Mathieu instability region; $\omega = 2\Omega_1$) the term added to $\ddot{H}$ has the same frequency as $\ddot{H}$ and therefore directly reduces its effects in Equation 3.

Conclusions and Recommendations

We claim, now, that the newly recognized term can be at least as important as the velocity-squared drag in placing limits on the levels of vibration reached in Mathieu (or Mathieu-type, when the governing differential equation cannot be formally reduced to Mathieu's Equation) unstable vibrations. Calculations made without taking this effect into account are conservative, and may be seriously and misleadingly conservative.

Quantitative prediction of the limiting amplitude is not permitted by the generally qualitative techniques and simplifications applied in this work, but it may reasonably be expected that, even in the absence of damping, limiting will occur when:

$$\theta_o^2\, \ell \simeq 4H \qquad (10)$$

Oscillating angles greater than 0.12 rad., approximately, may be expected to be very rare. Such angles are certainly not trivial, but the stresses associated with them for the low order mode ($\Omega = .5\omega$) of principal consideration will usually be small.

In closing, we urge that existing time domain programs be modified to permit accounting for this effect. New programs to be written should certainly include this effect.

Acknowledgement

The possible importance of the second order effects considered here was first suggested to the author, in a somewhat different context, by P. Kaman, of Morris Guralnick Assoc., Inc. His suggestion is acknowledged with gratitude.

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PRACTICAL SCHEMES FOR REDUCING COLD WATER PIPE LOADS

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Abstract

One of the key technological issues in the OTEC program of DOE is the ability to design, construct and deploy a cold water pipe which will survive the environmental conditions to which it will be exposed throughout its lifetime. Present studies related to the structural loading and response of the CWP indicate that unless some of the major environmental loads and structural responses are reduced through effective engineering design, the CWP may represent a serious technological barrier to the entire OTEC concept.

The thrust of this paper is twofold. First, a brief summary of the major environmental loads (e.g., wave and current induced) is presented along with results of studies conducted during the OTEC Commercial Plant Study and the 10/40 MWe Modular Applications Platform. The second part of the paper takes a look at various means of reducing the magnitude and/or frequency of the environmental loads with a view towards maintaining the CWP stresses below the tolerable limits. Included here are schemes such as hinges, vortex suppressors, the use of protective material coatings, tension mooring, platform motion stabilization and optimum CWP structural design. In each case, the scheme is assessed in terms of its effectiveness in reducing the loads and stresses and the cost of implementing it.

Introduction

As the OTEC program moves from the R&D phase, through feasibility conceptual and preliminary design studies to detail and contract design, leading to hardware construction and large scale testing, the need for practical engineering becomes more and more demanding. Perhaps the most challenging area where design innovation is needed is in the design of the cold water pipe. The structural integrity of the CWP has been for a long time one of the key technological risk items in the design, construction and deployment of the OTEC platforms. With life requirements of up to 30 years, operating in sites with severe environmental loadings due to waves and currents, it is imperative that the CWP designs incorporate practical schemes for reducing the resultant cyclic and extreme structural loads. This is not only necessary from operability and survivability considerations but is just as critical to minimize the structural loads during the deployment of the CWP.

Several areas in the design of the CWP offer room for improvement regarding structural load reduction. In fact, over the past three years a number of studies have been funded by the Department of Energy aimed at predicting the magnitude of the structural loads and developing ways for reducing them. References (1-6) illustrate some of the findings and recommendations made by the different organizations involved in such studies.

At the present time, most of the CWP design efforts, with the exception of the OTEC-1 platform, are centered around the OTEC 10/40 MWe Modular Applications Platform. Consequently, the bulk of the proposed schemes for reducing structural loads on the CWP have been designed for pipes with diameters in the 15-to-30 feet range and a length of 3,000 feet. In view of this, it is important to keep in mind that the schemes for reducing loads as described in this paper are tailored to pipes in the size range indicated. Any intent to extrapolate these designs to the larger CWP's needed for the larger OTEC commercial plants could lead to serious errors in estimating the effectiveness, constructability, and cost of the load reducing scheme.

The intent of this paper is twofold. First, to present to the OTEC community some of the ideas for CWP load reduction currently being pursued in the OTEC 10/40 MWe Modular Applications design studies. Second, to encourage the exchange of information between the various organizations involved in similar efforts with the aim of arriving at practical and cost-effective ways of reducing the CWP technical risks.

CWP Structural Loads

The CWP must be designed to withstand the worst combination of the loads listed below with proper consideration given to the phasing and probabilistic nature of the dynamic loads:

1. PRIMARY LOADS

   Static - Initial mooring tension (if applicable)
   - Net weight/buoyancy
   - Steady-state current bending loads

   Dynamic - Wave induced bending loads
   - Wave induced vertical loads
   - Unsteady current induced bending loads

2. SECONDARY LOADS

   Static - Head loss within the pipe
   - Density difference across the pipe wall
   - Steady-state current stagnation pressure

   Dynamic - Wave orbital velocity stagnation pressure and internal transients

Of the above listed loads the dynamic loads due to waves and currents are the most severe, and thus the need for incorporating load reduction schemes in the design of the CWP.

In order to select the appropriate load reduction system, it is necessary to understand first the dynamic characteristics of the CWP.

CWP Dynamic Response Properties

All of the previous and current efforts in the area of CWP design indicate the criticality of the system dynamics relative to CWP loads and responses. It is obvious that any system subjected to cyclic loading should be designed such that its natural frequency and the frequency of the forcing function do not coincide in order to avoid resonance. Therefore, it is necessary to review the factors which affect the frequency and amplitude of both the forcing function and the response of the CWP in an effort to minimize the loads to be supported by the CWP.

The response of the CWP is dependent on both the properties of the CWP itself and the boundary conditions acting to restrain or excite the pipe.

CWP Properties

The properties of the CWP which affect its bending response are the stiffness of the structure and the mass of the CWP plus the entrained water. The identifiable characteristics of the pipe which contribute to these properties of mass and stiffness in bending are:

1. Modulus of the material
2. Mass density of the material
3. Diameter
4. Thickness
5. Length
6. Displaced weight of pipe structure
7. Added and entrained mass of right circular cylinder of fluid

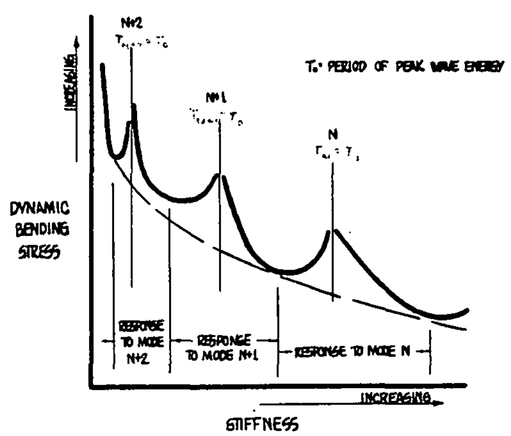

Figure 1. Structural Response of the CWP as a Function of Stiffness and Natural Mode

The response of the CWP can be generalized as depicted in Figure (1). In this figure, response to a cyclic load, as measured by dynamic bending stress, is presented as a function of the pipe stiffness. Pipe stiffness is represented as a primary property of the pipe although similar plots can be produced by systematically varying any one of the identifiable characteristics listed above. The localized resonant peaks represent pipe stiffness/mass combinations which result in coincidence of the frequency of the forcing function and one of the harmonic natural frequencies of the pipe.

Since the length and diameter for the 10/40 MWe CWP are defined as 3,000 feet and 30 feet or 15 feet (40 and 10 MW respectively), the only variable parameters affecting the dynamic response of the pipe are the modulus and density of the material, and the equivalent wall thickness of the pipe wall.

Boundary Conditions

The boundary conditions affecting the CWP are the hydrodynamic forces and mass of the platform, the method of attachment between the platform and CWP, and the method of restraint, if any, acting on the bottom or at any intermediate point along the CWP.

(a) Hydrodynamic Characteristics of the Platform - When a platform is exposed to waves of a given frequency, forces are developed which, unless constrained by a mooring system, result in motion. The relationship between these forces and the frequency of the waves can be depicted as a Response Amplitude Operator (RAO). Inspection of a typical motion RAO curve for the platform will show that the response of the platform can approach a resonance condition. Thus, even if the CWP is properly designed to avoid resonance, the natural period of the platform must also be tuned to avoid the frequency of peak energy, or the platform response can be altered by the addition of damping (motion stabilization). The first method would shift the peak RAP and the second would decrease the peak-value. Either approach will result in decreased loads to be supported by the CWP. The specific application of one or both of these methods depends on the geometry of the platform. For a spar platform with a small waterplane area and underwater symmetry, three degrees of motion are redundant and the natural periods of the remaining motions are long enough to effectively detune the platform from any critical sea spectrum.

(b) Mass of the Platform - Another effect which the platform exerts on the response of the pipe system is a lowering of the natural frequency by increasing the mass. The mass plus added mass of the platform is on the same order of magnitude as the mass plus the added mass of the CWP and entrained water; therefore, the effect is significant. The displacement of the hull is the minimum which is consistent with volumetric and configuration requirements, and cannot therefore be reduced significantly. Similarly, to increase the mass of the platform would require a major alteration of the existing configuration. The virtual mass of the platform cannot therefore be considered to be a variable in the relationship of mass, stiffness, and natural frequency.

(c) Method of Attachment of CWP and Platform - By addressing all six degrees of freedom (or three degrees for the symmetric spar), it is implied that all motions of the platform and CWP are coupled. If, however, the CWP is allowed to move relative to the platform in one or more degrees of freedom, these motions become uncoupled and thereby do not affect the CWP response. This is the reason for the use of a universal or ball joint at the platform-CWP interface. Such a mechanism decouples the CWP from platform roll and pitch motions. It should, however, be noted that a resonant condition in the uncoupled degree of freedom should still be avoided as any device which allows relative motion (angular or linear) must have defined limits past which it is ineffective.

(d) Mooring System - The remaining boundary conditions affecting the CWP are the constraints imposed by a mooring system. Because of the problems involved with fatigue loadings and limiting motion and acceleration criteria related to the transmission riser cable, it is desirable to maintain positive control of the platform. This can be accomplished by a tension leg mooring system as discussed in Reference (3). The use of such a system is, however, limited to platforms with a relatively small water plane area in order to minimize the variation in tensile forces which the elements of the system must support as waves pass the platform. Thus, the spar configuration lends itself well to a tension moor. The use of the CWP as an element of the tension moor system presents the apparent advantage of decreasing the length, and therefore cost, of the mooring lines. This savings in mooring system cost must be traded off against any change in the cost of the CWP resulting from the increased tensile and bending loads.

The response of the CWP is sensitive to two characteristics of the mooring system: the steady-state tension and the system spring constant. The mooring tension is the major contribution to the design load which, with the allowable stress, determines the required cross sectional area of the system. The system spring constant is directly proportional to this cross-sectional area and the element modulus of elasticity. The values of the mooring tension and the spring constant are therefore related. The effects which they have on the response of the CWP are basically two. The line spring constant affects the heave response of the platform and hence, the vertical loads imposed on the CWP. The line tension affects the natural frequency of the CWP and hence, the extent to which the CWP response approaches resonance conditions in the presence of waves and currents.

CWP Natural Frequencies

From the discussions of CWP properties and the boundary conditions it becomes clear that the first step in properly tuning the CWP must be the determination of its natural frequencies. Several options are available for computing the natural frequencies of an OTEC cold water pipe. Two computer simulations which can be used to do this are those developed by Paulling (Reference 7) and by Barr, et al (Reference 8). However, a less involved and less expensive way of quickly estimating the CWP natural frequencies is presented in References (9), (10), and (11). Based on this work, the natural frequencies ω_i, for a CWP with line tension T acting at the bottom of the pipe are given by:

$$\omega_i^2 = \frac{\left(\frac{i\pi}{\ell}\right)^2 \left[EI\ell \left(\frac{i\pi}{\ell}\right)^2 + \frac{S\ell^2}{2} + \frac{T\ell}{4}\right]}{m\ell} \qquad (1)$$

where
ℓ = CWP length
E = material modulus of elasticity
I = CWP moment of inertia
S = weight/unit length of pipe
m = mass/unit length of pipe and entrained water
T = mooring line tension
i = index corresponding to CWP modal shape, i = 1,2,3 ... etc.

The effect of the platform at the top of the CWP is not included in equation (1). However, it may become significant in that it adds an inertial term to the equation of motion. For the third and higher modes of CWP vibration, the inertial effects are less important and the structural properties of the CWP tend to dominate its dynamic response. An inspection of the analysis given in Reference (10) demonstrates that the solution becomes more complicated as a result of the platform mass coupling the modal responses.

The natural frequencies of the CWP for the first and second modes including the effects of line tension T and platform mass M is given by:

$$\omega_i^2 = \frac{\frac{1}{2}\left(\frac{i\pi}{\ell}\right)^2 \left[EI\ell\left(\frac{i\pi}{\ell}\right)^2 + \frac{S\ell^2}{2} + \frac{T\ell}{4}\right]}{m\ell + nM} \qquad (2)$$

$$n = 3 \text{ for } i = 1 \quad ; \quad n = 1 \text{ for } i = 2$$

An algorithm for estimating the pipe natural frequencies was developed in Reference (11) for the case of a CWP with multiple hinges along its length. The equation was derived from a tensioned string equation with individual masses and the natural frequency equation for a CWP as presented in Reference (10). The analysis of Reference (10) was modified for no structural rigidity and pinned boundary conditions. It should be noted that the analysis did not take structural response into account; therefore, the maximum mode occurs at the number of hinges on the CWP. The resulting equation is of the form

$$\omega_n = K_m \sqrt{\frac{T}{M\ell}} \qquad (3)$$

where
ω_n = natural frequency in rad./sec.
M = mass of a segment between hinges, including added mass, and the mass of the entrained water
ℓ = total length of the pipe
T = tension of the mooring plus 7/20 of the weight of the pipe in water (from Reference 12)
K_m = the modal coefficient which is a function of the number of hinges and the modal response. Reference (11) presents curves for estimating the value of K_m.

Environmental Dynamic Loads

(a) Wave Induced Loads - Wave induced loads cannot be reduced to a regular sinusoidal force of a single amplitude and frequency. Rather, each sea state associated with a specific geographic site must be represented as a frequency-energy spectrum. Additionally, the frequency and duration of occurrence of each sea state are both probabilistic in nature. Consequently, the design of any system to be exposed to such conditions must reflect the probabilistic nature of the loads which it must support.

For a given sea state, the natural frequency which the system should be designed to avoid is the frequency of peak wave energy. Consider, for example, the design sea spectrum for the OTEC 10/40 MWe Spar Platform to be located off Punta Tuna, Puerto Rico. This spectrum with a significant wave height of 44.2 feet and a period of peak wave energy of 13.8 seconds, corresponds to a 100-year event at this site. Over the projected 30-year life of the platform this event will occur with a probability of .26 (i.e., $1-(1-.01)^{30}$). The duration of such conditions, however, are extremely short. Furthermore, although the extreme event has a greater amount of total energy (i.e., the area under the spectral curve) than any other more frequently occurring sea state, the corresponding frequency of maximum wave energy is lower. Thus, if the CWP system is designed such that one of its natural frequencies is slightly greater than the frequency of maximum wave energy of the hundred-year storm, a lower sea state may result in a greater response because of the more nearly coincident forcing and natural frequencies.

Another related problem concerns the imprecision of prediction of characteristics of the design sea state. The highly tuned nature of the pipe response can be significantly altered by the shift of the period of peak wave energy by as little as five percent. A recommended approach to address this uncertainty is discussed in Reference (3).

(b) Current Induced Loads - In the presence of certain combinations of configurational and hydrodynamic flow characteristics, the flow around a right circular cylinder can be characterized by well defined alternate shedding vortices. The orderly fashion in which the vortices are shed leads to the development of periodic lateral forces. If the frequency at which these alternating forces occur is near a natural frequency of the pipe, the response of the pipe may approach a resonant condition.

The flow characteristics which lead to the formation of these alternating vortices occur in specific Reynolds number regimes. As indicated in Reference (4), the synchronized formation of vortices occurs for Reynolds number below 2×10^5 and above 3×10^6. Within these regimes the frequency at which the vortices are shed is also dependent on Reynolds number. Thus, the occurrence and frequency are sensitive to the pipe diameter, the fluid velocity, and the fluid viscosity.

The diameter of the CWP is uniform along its length. The viscosity and velocity of the fluid, however, vary along the entire length of the CWP. This results in Reynolds numbers ranging from $.74 \times 10^6$ at the bottom of the CWP to 12.5×10^6 at the top. Therefore, the proper conditions for synchronous shedding exist over the majority of the length of the pipe. The analysis of Reference (4) shows the variation of current velocity and, hence, Reynolds number along the length of the CWP for normal and extreme conditions, indicating the regions for potential synchronous vortex shedding. Within those regions the corresponding Strouhal number is taken as between .25 and .31. Thus, the potential frequencies of shedding correspond to these Strouhal numbers. The resulting periodic lateral forces (lift) occur at this frequency, and the oscillatory portion of the in-line (drag) force occurs at twice this frequency. Reference (4) describes a methodology for conversion of this profile into a series of dynamic loads each with a specific frequency.

Dynamic Response

(a) Wave Induced Response - The results of a number of simulations conducted for a continuous pipe are presented in Reference (4) for variations of modulus of elasticity, thickness, diameter, mooring tension, and mooring system spring constant. The analysis of Reference (4) presents these relationships in parametric form. Although these results are approximate and require the determination of additional values to fully define the functional relationships, they do demonstrate the sensitivity of critical loads in the CWP to key parameters which are characteristic of the related systems. Additionally, the use of the natural frequency relationships presented above can be used to determine values of each of the parameters which correspond to resonance. In this way the parametric data can be related to a local region of the idealized curve presented in Figure (1).

(b) Current Induced Response - The results presented in Reference (4) do not reflect an accurate simulation of the dynamic response of the CWP to current induced unsteady dynamic loads because of the unavailability of adequate analytical tools at the time of the investigation. However, they do provide some understanding of the nature of the problem. For a rigid pipe with thicknesses within the range investigated, resonant response to unsteady lateral forces is likely in the first and second modes of response of the CWP. An overly simplified static model for a continuous pipe is given in Reference (4). The stresses which result from this analysis are presented graphically for the normal current loading case as a function of wall thickness for 15 to 30 foot diameters. These results should only be used to develop an understanding for the potential criticality of the vortex shedding loads and the development of design methodologies to provide support for these loads. They should not be used for design purposes pending the availability of a structural response model for combined wave and currents currently under development.

Design Procedures

(a) Extreme Event Analysis - The CWP must be designed to support all of the loads listed earlier for both the yielding and non-yielding criteria. The former case considers the probability of stressing the entire cross section of the pipe to yield thereby forming a plastic hinge. The pipe is only considered to be adequate if the probability of exceeding the wave induced load which results in stressing the entire cross-section to yield is less than .001 over the life of the platform. This corresponds to a wave height 3.3 times the RMS, that is 73.7 feet (double amplitude). The second criteria imposes all loads, primary and secondary using the RMS dynamic bending stress, and the extreme current profile and factors of safety of 1.25, 1.5, and 1.33 for tension, column buckling and panel buckling, respectively. These designs will be validated after further investigation into the response of the CWP to current induced dynamic loads.

(b) Fatigue Analysis - The fatigue analysis considers the wave and current induced loads discussed previously. Reference (4) presents preliminary quantitative results based on a simplified static model. As mentioned before, these values should not be used for design purposes. However, the results based on these estimations demonstrate the significance of the vortex shedding loads relative to the fatigue life of the structure. Additionally, the analysis demonstrates the sensitivity of the fatigue life of the structure to the effectiveness of the corrosion protection system installed.

(c) Sensitivity Analysis - Once detailed scantlings for the pipe structure are chosen, a sensitivity analysis must be performed to clearly verify the adequacy of the design for off-design characteristics. Since the current forces, if applicable, are imposed at the various natural frequencies of the pipe, no sensitivity analysis is required for the unsteady lift and drag forces. For the wave induced loads, minor shifts in period of peak wave energy must not cause the pipe to be overstressed.

If stresses resulting from shifts of ± 15% of the period of peak wave energy are acceptable, the additional constraint that the mode of response must not change within that range must be satisfied to preclude the possibility of resonant response within that range.

To appreciate the behavior of the pipe and its sensitivity to the various properties previously discussed, parametric relationships between these properties and the natural frequencies of the CWP are extremely useful. The simplified calculations using equations (1-3) to estimate the pipe natural frequencies provide a cost-effective selection of preliminary values for each parameter. This approach eliminates the need for conducting expensive parametric analysis using one of the established CWP dynamics computer simulations (References 7 and 8).

Reduction of CWP Structural Loads

From the discussion given above, it is evident that there are several options available to the designer for reducing the magnitude and/or frequency of occurrence of the CWP structural loads and responses. Basically, the options can be grouped into the following categories:

- Platform design for reduced motions and, hence, CWP loads and stresses
- CWP structural design aimed at properly tuning the CWP with respect to the environmental dynamic loads
- Use of motion and load reduction schemes such as motion stabilizers, vortex suppressors, corrosion protection, double shell design, bottom-mounted pipe, use of FRP sandwich structures, use of reinforced elastomers, etc.

In the next few paragraphs, brief discussions of these various alternatives are presented. The intent is to summarize the results of analytical and design studies which have been conducted so far in an attempt to define where we stand today with regard to minimizing CWP structural loads and, hence, the resulting impact on technical risk.

(a) Platform Design - The platform motions induced by the waves will exert forces and moments on the CWP through the hull/CWP connection. If the connection is rigid, then all six degrees of freedom of platform motion (roll, pitch, yaw, heave, sway, and surge) will result in forces and moments at the top of the CWP. If, on the other hand, the pipe is hinged at the top, then the principal forces to be considered are those associated with the heave, sway, and surge motions of the platform. However, other motions such as roll, pitch, and heave must still be kept at levels which will avoid contact of the upper CWP section with the surrounding hull structure and/or result in excessive vertical loadings on the CWP. Several CWP/hull connection designs have been proposed. For example, Figure (2) shows the design developed hy Gibbs & Cox, Inc., for the 10 and 40 MWe spar latforms.

he connection begins with a cruciform structure uilt from the pipe walls to the lower half of a center universal joint. The top of the universal joint is connected by a second cruciform structure

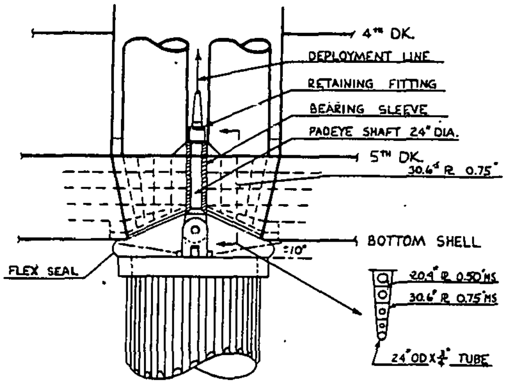

Figure 2. Hull/CWP Connection for the 10/40 MWe Modular Applications Spar Platform (Ref 3)

to the bottom shell and 5th deck of the platform. The cruciform structure is in line with the quartering bulkheads of the platform to provide structural continuity. A flexible seal is provided to allow ± 10 degress of motion while preventing mixing of the cold intake and warm surface water. The cost of this system is estimated at $0.6 million for the 10 MWe CWP and $1.02 million for the 40 MWe CWP (Reference 3). These costs represent about 5% of the total CWP cost.

Figure (3) shows the design of the CWP/hull connection for the APL Plantship (Paper 5.5 of these Proceedings by Jim George). The center of rotation of the CWP/platform connection is located near the C.G. of the platform. This minimizes kick loads to the pipe and reduces angular response at other joints. The 4 pumps are located above the CWP. The vanes will be feathered and locked when the OTEC plant is shut down during severe storms. The CWP rests on a stainless steel support ring which bears on a spherical stainless steel plate through teflon-coated elastomeric pads. Adequate clearance is provided between the CWP and platform support for angular motions in a head sea. Other CWP/hull connection designs are described in References (5) and (6).

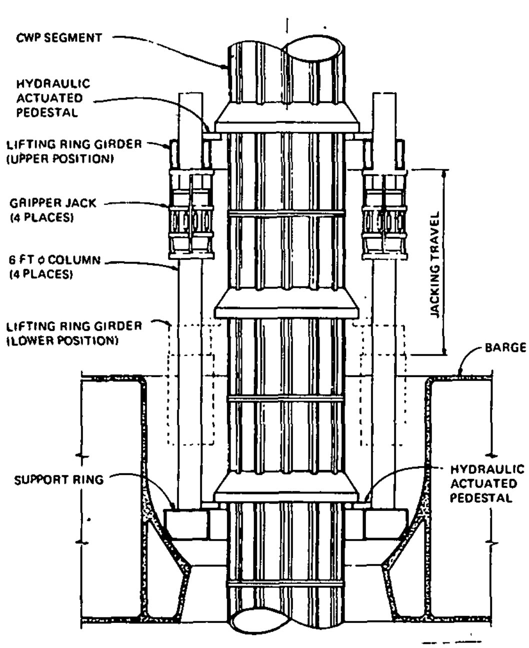

Figure 3. Hull/CWP Connection for the APL Plantship (Reference 13)

Platform motions and CWP loads can be reduced by 1) optimizing the hull configuration and weight distribution or 2) using a motion stabilization system. Effects of hull configuration on motions of the platform in the horizontal plane and bending stresses in the pipe are illustrated in Figs. 4 and 5 (from data in Refs. 4, 14 and 15). The motions and stresses for a spar platform are much lower than for a plantship when both have rigid, thick-walled steel pipes of 15-ft (for 10-MW_e plants) or 30-ft (for 40-MW_e plants) diameter. The estimated platform costs for the 10 and 40 MW_e spars are $21.5 million and $38.3 million respectively (Reference 3). The estimated platform cost for the APL Plantship is $32.8 million (Reference 20).

(b) CWP Structural Design - The structural design of the CWP is perhaps the most effective tool for maintaining the CWP structural loads & response at minimum levels. As pointed out above, the response of the CWP is dependent on both the properties of the CWP itself and the boundary conditions acting to restrain or excite the pipe. The case of the spar/CWP with a tension moor at the bottom of the CWP is a classical example of the extent of stress reduction which can be achieved by proper tuning of the pipe mooring system.

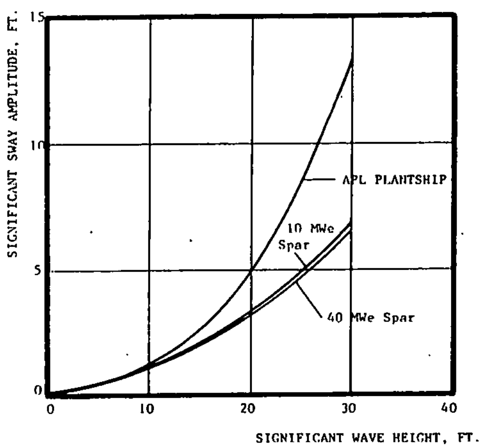

Figure 4. Comparison of Significant Sway Amplitude vs Wave Height (double amplitude, ft) for the 10/40 MWe Spar and the APL Plantship (References 14 and 15)

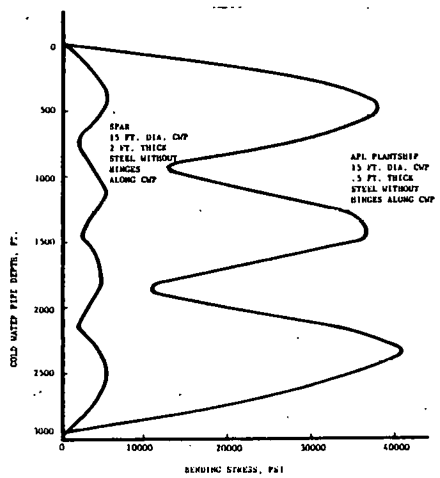

Figure 5. Comparison of CWP Dynamic Bending Stress for the APL Plantship & the 10/40 MWe Spar

The magnitude of the mooring tension will affect the pipe modal response (i.e., the natural frequencies). Consequently, a line tension must be selected which places the CWP natural frequencies outside the frequencies (or period) of peak wave energy and/or current induced oscillatory loads due to vortex shedding. Table (1) shows the effects of mooring line tension and stiffness on the spar heave motions and CWP stresses. These results illustrate the sensitivity of CWP dynamic bending stress to the mooring line tension. On the other hand, the line stiffness influences the CWP vertical dynamic stress and the platform heave.

Table 1. Platform Heave and CWP Dynamic Stresses as a Function of Mooring Line Tension and Line Stiffness (CWP Diameter = 30 ft.; CWP Thickness = 0.07 ft.; CWP Material = Mild Steel; CWP Length = 2,800 ft. (Reference 4)

T (lbs)	K_y (lbs/ft)	Maximum Bending Stress, psi	Maximum Vert. Stress, psi	Heave, ft.
3×10^6	80,000	8,000	108	3.13
	500,000	8,000	691	14.65
	970,000	8,011	5,480	61.8
10×10^6	80,000	12,770	108	3.13
	500,000	12,770	630	14.5
	970,000	12,770	5,478	61.8
22.7×10^6	16,800,000	14,488	4,824	0.72

The stiffness of the CWP will play a significant role in the dynamic response of the pipe to wave and current loading. Figure (1) illustrated the general relationship of CWP dynamic bending stress to pipe stiffness. As opposed to the conclusion which would be made from static considerations, Figure (1) indicates that the dynamic bending stress may decrease or increase with pipe stiffness depending on where the excitation frequency falls with respect to any of the pipe modal frequencies. These observations are to be considered as being of a preliminary nature pending the validation of existing analytical tools. However, they do point out the need to pay very close attention to the CWP as a dynamic system rather than a static one.

An effective scheme for load reduction in the structural design of the CWP is the use of multiple hinges. This approach has been used in the design of the APL Plantship concrete CWP and is currently being employed in the conceptual design of the CWP for the 10/40 MWe spar.

In general, the introduction of flexibility into the CWP in the form of hinges decreases the stiffness and thus the natural frequency of the CWP. Because of the discrete flexibility inherent in a linked pipe, the number of degrees of freedom (i.e., the highest mode of response possible) cannot exceed the number of intermediate hinges. Ideally, then the CWP should be designed with an adequate number of hinges in the system to lower the natural frequency, in the highest possible mode, below the lowest frequency excitation of concern. For example, with six hinges the natural frequency in the fifth mode should be less than the frequency of the lowest excitation frequency.

The concept of a hinged CWP evolved as a means of reducing construction and deployment cost and risk. During the Feasibility Studies, Reference (16), a number of pipe deployment schemes were evaluated, based on rigid pipes. One of the more promising schemes involved constructing the pipe on site, using the spar as a platform for the operation. The pipe would be fabricated from preassembled sections and would "grow down" through the center column of the spar until the desired 3,000 foot length was achieved.

Subsequent analysis of dynamic response as a function of length indicated the likelihood of resonances at intermediate length resulting in more severe CWP structural loads than when fully deployed. This, coupled with the adverse effects of possible spar motions, and the need for a large weather window for the deployment operation, led to the conclusion that this scheme would involve unacceptable risk. This resulted in the concept of a hinged pipe which could be constructed entirely ashore, floated out to the site and upended a section at a time in a rapid but controlled manner. At the same time, it became apparent that a virtually buoyant pipe would greatly enhance this operation as well as reducing CWP loads and spar buoyancy requirements.

As the design developed, it also became apparent that the optimum mooring system would be one which involved a tension leg moor attached to the bottom of the CWP. Thus, evolved the concept of an integrated, buoyant CWP/mooring system which could be towed from shore as a unit for installation in one continuous operation with essentially no underwater work.

It should be noted that, as of this writing, the proposed CWP/mooring system concept has not been validated due to the lack of a proven analytical tool which can properly model a hinged CWP with initial tension. Design features and structural requirements have been extrapolated from previous CWP and mooring system designs, backed up by approximate calculations such as those described earlier in this paper. However, the basic concept is considered valid, and the selected structure is considered conservative. Further studies are planned for the near future to validate the basic design as well as to optimize the location and number of hinges, materials and other features of the concept.

The proposed hinged buoyant CWP system is illustrated in Figure (6). The system consists of the following principal elements:

- Six (6) CWP hinged sections, nominally 460 feet long. The lowest section is 528 feet long and incorporates the inlet screen at a depth of 3,000 feet below the surface. Each section is neutrally buoyant and consists of an inner tube, an outer shell, and a collar.
- Five (5) hinged mooring links, each 200 feet long, suspended below the CWP inlet.
- Deadweight anchor.
- Buoyancy tanks secured to the upper CWP section to support the weight of the anchor and mooring links during deployment
- CWP/hull interface.

Each of these elements is described in detail in Reference (3).

Figure (7) shows the results of the dynamic analysis of the hinged CWP concept using the Paulling program (Reference 7). For the 100-year storm condition in Puerto Rico ($H_{1/3}$ = 44 feet) the

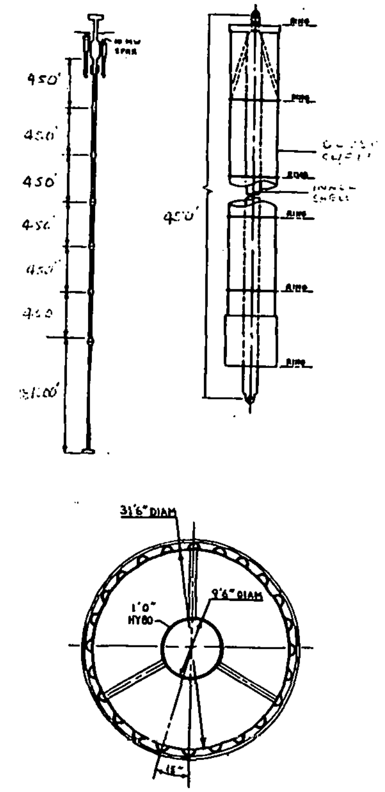

Figure 6. Hinged Steel CWP for the 40 MWe Modular Applications Spar (Reference 3)

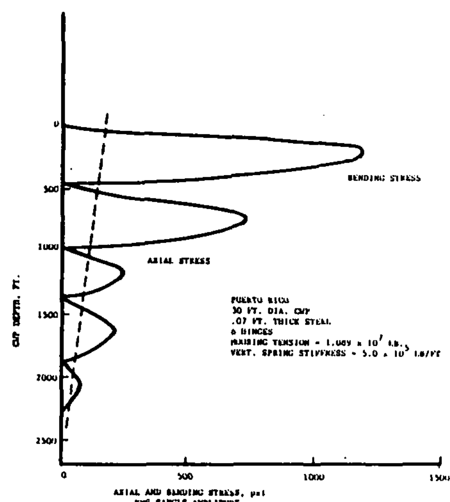

Figure 7. Dynamic Bending and Axial Stress for the 40 MWe MAP Spar CWP

rms wave induced dynamic bending stress is about 1,200 psi. This stress is four (4) times smaller than the stress shown in Figure (5) for a rigid pipe of comparable dimensions and material attached to a similar spar platform.

One of the drawbacks to the multiple-hinge concept is its lack of redundancy. For a standard CWP, this might be acceptable, since loss of the pipe would not lead to loss of the platform. In the case of a tension leg moor, however, failure of any of the hinge connections or other potential "weak links" in the system could lead to loss of the platform if it drifted ashore. Therefore, future studies will concentrate on developing some type of "fail-safe" concept or device to permit controlled failure of the CWP/hull connection in the event of extreme overload, with the potential for restoring the the connection to its original configuration.

One possible method for accomplishing this is to provide a hydraulic piston at the top of the padeye shaft in lieu of the retaining sleeve. If the system is pressurized to a level providing an upward force equal to the design pretension, equilibrium would be established for normal conditions. Under extreme overloads a relief valve would immediately open spilling hydraulic fluid into a reservoir, and dropping the shaft to relieve the overload. After closing the valve the piston could be pumped up to the equilibrium level. This and other concepts will be investigated in the near future once the final load spectrum for the CWP is established. The estimated cost of the CWP incorporating these features is $22 million for a 40 MWe spar platform.

In summary then it is quite obvious that the structural design process is in itself a highly effective load reduction scheme. When combined with an equally effective platform design, the result can only be a reduction in the structural loads to be experienced by the CWP. Table (2) has been prepared as a summary of the role played by the CWP platform and mooring system characteristics in the structural design process.

(c) Specific Load Reduction Devices - A third group of load reduction schemes involves the use of devices or systems which are added to the OTEC platform or to the CWP in order to alter their response to the environmental excitations. Three representative schemes which have been looked at during the feasibility and conceptual design phases are discussed here. These include platform motion stabilization systems, CWP vortex suppressors, and CWP corrosion protection systems.

Platform Motion Stabilization Systems

Conceptual studies on motion stabilization have been carried out for the APL Plantship and the results are reported in References (17) and (18). Since the CWP is attached to the hull via a hinged or flexible connection, the impact of angular motions on the CWP loads and stresses in secondary. The main loading exerted on the pipe comes from the horizontal plane motions of the platform (surge, sway, and yaw). Still, a reduction in platform roll, pitch and heave are desirable in order to (1) alleviate the CWP deployment operations; (2) avoid physical contact of the top of the CWP with the surrounding structure at the connection; and (3) maintain the vertical loads on the CWP at acceptable levels.

Various types of stabilization systems were investigated. Active systems such as fin stabilizers were eliminated from consideration due to their requirement of forward velocity. Though the APL grazing plant does operate at a minimal speed, it is insufficient for such fins to be effective. In considering passive stabilization, bilge keels, free surface (Flume) tanks, U-Tube tanks, and the Sea-Tek "Slo-Rol" pneumatic system were investigated for roll reduction along with flat plate dampeners for pitch reduction. In addition to these systems, a variable heading moor was examined for its effectiveness in overall motions reduction.

Table (3) identifies the stabilization systems considered in the APL Plantship studies along with their principal dimensions. Figure (8) shows a possible arrangement of free surface passive anti-roll tanks on the APL Plantship.

Arrangements for the other anti-roll systems are given in Reference (17).

Figure (9) presents comparisons of roll response for the unstabilized and stabilized APL Plantship. The roll tanks alone yielded approximately 50% roll reduction which is consistent with past experiments on other vessels. The bilge keels produced a low value of only 10% reduction. Though these keels are comparatively large (185 feet long by 10 feet in width), they do not share the hydrodynamic lift component which is available to keels fitted on ships which operate consistently at speeds above 10 knots.

This hydrodynamic component, unlike the added mass and drag components, cannot be utilized by the Plantship in its slow-moving grazing or moored modes.

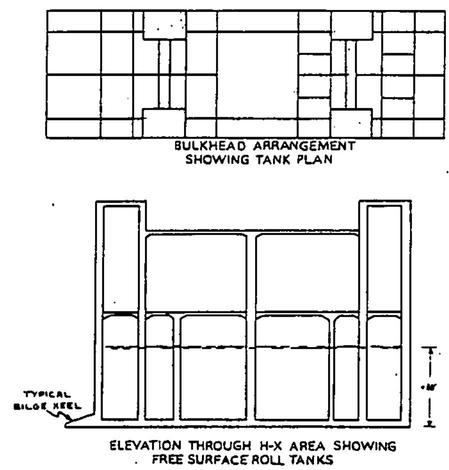

Figure 8. Arrangement of Free Surface Roll Tanks for the APL Plantship (Reference 17)

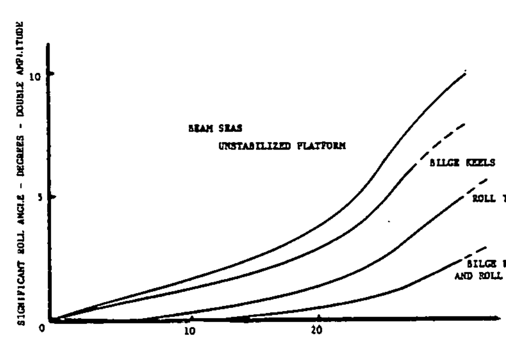

Figure 9. Significant Roll Angle for Stabilized and Unstabilized APL Plantship vs. Significant Wave Height for Beam Seas (Reference 18)

Combining both the roll tanks and bilge keels produced a surprising 60 - 70% reduction of roll motions. In examining the possible reasons for this high value of roll reduction, it was found that the two systems were "tuned" such that the maximum dampening action was produced for most sea state

Table 2. Role of CWP Platform and Mooring System Characteristics in the Structural Design of the CWP

	Bending Response			Vertical Response		
	Stiffness	Mass	Natural Frequency	Stiffness	Mass	Natural Frequency
CWP MATERIAL CHARACTERISTICS						
Modulus of Elasticity	D		D	D		D
Mass Density		D	I		D	I
CWP CONFIGURATIONAL CHARACTERISTICS						
Diameter	D		D	D		D
Thickness	D		D	D		D
Length	I		I	I		I
Displaced Weight	D		D			
Added Mass (horizontal)		D	I			
Added Mass (vertical)					D	I
Number of Hinges	I		I			
PLATFORM HYDRODYNAMICS						
Waterplane Area				(2)		(2)
Waterplane Mom. of Inertia	(1)		(1)			
PLATFORM MASS						
Mass		D	I			
Added Mass (horizontal)		D	I			
Added Mass (vertical)					D	I
METHOD OF ATTACHMENT	(1)		(1)	(2)		(2)
MOORING SYSTEM CHARACTERISTICS (3)						
Initial Tension	D		D			
Line Stiffness				D		D

D = Directly related I = Inversely related Blank indicated no effect

NOTES: 1. If rigid hull/CWP Interface - D
If free to rotate (roll, pitch) - No effect
2. If rigid hull/CWP Interface - D
If heave compensated - No effect
3. Assuming integrated CWP mooring system

Table 3. Stabilization System Dimensions for the APL Plantship

STABILIZATION SYSTEM	LENGTH	WIDTH	WATER DEPTH
Bilge Keels	185	10	---
F.S. Wing Wall Roll Tanks	44	121	65
F.S. Roll Tanks	40	121	30
U-Tube Roll Tanks	38	121	20
Slo-Rol System	378	10	10
Pitch Dampeners	30	121	---

(1) all dimensions in feet
(2) F.S. = free surface

conditions. This result is a frequency related phenomenon where poor tuning could have resulted in only a 40% reduction of motions just as easily.

The Sea-Tek "Slo-Rol" pneumatic system was particularly interesting in light of past results with jack-up platforms and drilling barges. Though existing installations feature tanks mounted external to the hull, in an effort to reduce refit cost the tanks proposed for the APL Plantship were located in ıe wing walls. Field and model tests indicate vorable results with roll amplitudes reduced from ı to 70 percent. Due to the dynamic nature of the system it was difficult to model the tanks at the feasibility design level; however, through discussions with the Sea-Tek technical staff, good estimates were obtained for rough costing and performance evaluation.

With the internal placement of the tanks, the tank size must increase relative to an external mounting for the same performance. Though there is a slight cost penalty for this installation, it must be balanced against the added drag of the external installation. Judging from past performance and estimates from Sea-Tek, it is expected that the "Slo-Rol" pneumatic system will have competitive cost and performance relative to standard anti-roll tank designs using water as the working fluid.

A flat plate dampener system was designed to reduce the pitch motions of the APL Plantship. These dampeners consisted of flat plates extending across both ends of the platform as shown in Figure (10). Varying sizes were tried in order to obtain a range of pitch reduction. As shown in Figure (11), all dampeners had a pronounced effect on pitch motions. A range of 30 - 60% reduction in pitch amplitudes was achieved with their use.

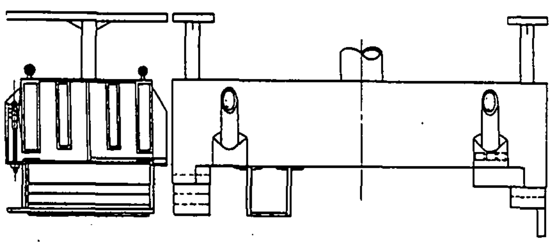

Figure 10. Illustration of 30' Pitch Dampeners Fitted to APL Plantship (Reference 17)

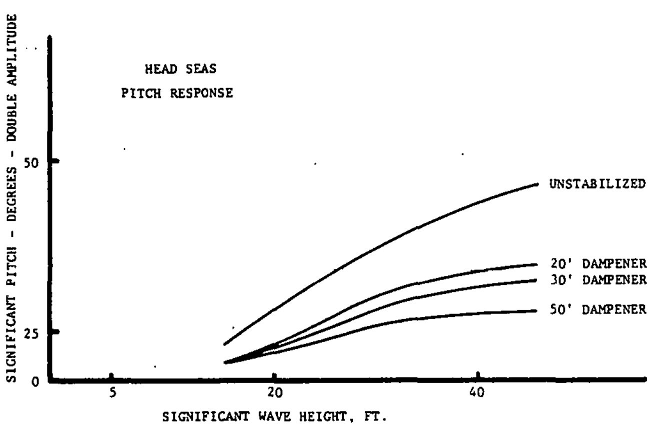

Figure 11. Significant Pitch Angle for Stabilized and Unstabilized APL Plantship vs. Significant Wave Height for Head Seas

A comparison was made of roll and pitch motions for the APL Plantship in unstabilized and stabilized modes along with the motions for a projected 400 MW OTEC Commercial Plant of the ship/barge configuration. The roll motions for the APL Plantship equipped with anti-roll tanks are less than those of the Commercial Plant up to the 16-foot wave height. In the extreme wave heights of 44.2 feet, the stabilized APL Plantship only exceeds the Commercial Plant motions by 2.5 degrees. The pitch motions for the Plantship equipped with the 50-foot pitch dampeners are less up to the 26-foot significant wave height. At the 44.2 wave height the Plantship values are 5 degrees greater than the Commercial Plant.

Regarding reductions in vertical motions and loads, beginning at the 20-foot significant wave height, the unstabilized platform begins to exceed 0.1g accelerations. Using the smallest size of pitch dampeners (20-foot length) the 0.1g threshold is delayed until 30-foot significant wave heights are encountered. The 30 and 50 foot length dampeners delay the exceedance of the same threshold to the higher wave heights. With some combinations of platform location and dampener size, the 0.1g acceleration level is not exceeded in the range of all examined wave heights.

An investigation into several types of mooring systems for the APL Plantship was also carried out. Among the types were the standard 8 point, turret, single point, and single point with yoke. All systems except the 8 point moor require thrusters to help maintain station and heading. In storm conditions the thrusters will require large amounts of parasitic power to keep the ship in the most favorable heading. Therefore, the standard 8 point moor was selected as the most feasible mooring system for the Plantship.

Costs were developed for each of the stabilization systems using F.Y. 1980 dollars. Size and weight combined with appropriate cost factors yielded the construction costs. All of the systems meshed closely with the present structure, hence concrete was the preferred material except for the bilge keels which could be easily fitted in either steel or concrete. In all cases, the total system construction cost is anticipated to be under $1 million. Figures (12) and (13) present variations of cost with percent pitch and roll reduction for different stabilization systems.

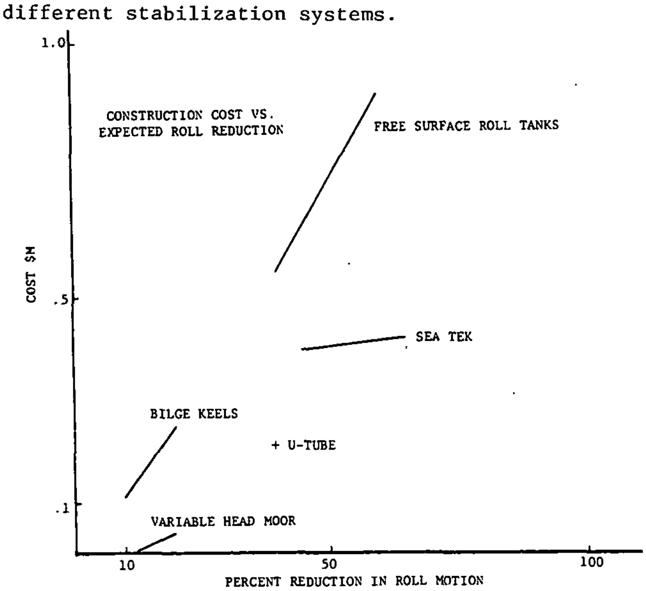

Figure 12. Variation of Cost with Percent Pitch Reduction for Various Stabilization Systems (Reference 17)

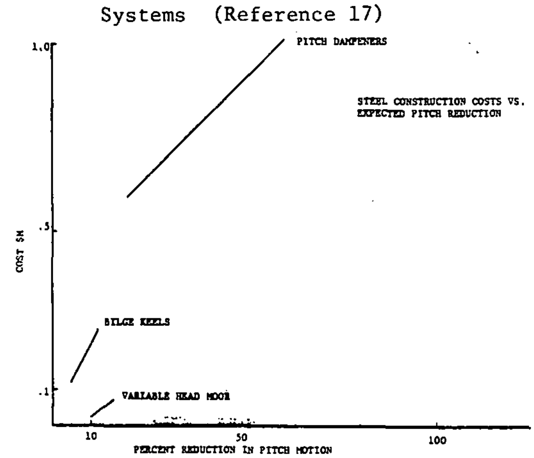

Figure 13. Variation of Cost with Percent Roll Reduction for Various Stabilization Systems

Vortex Suppressors

Preliminary investigations into the possible effects of vortex shedding on the response of the CWP have been reported in Reference (4). These studies indicate that the pipe may experience severe cyclic and extreme stresses due to the unsteady forces resulting from current induced vortex shedding. Reference (4) presents a series of cases where CWP's of different materials, wall thicknesses and diameters could go into lock-on or resonance as a result of vortex shedding. Lock-on is expected to occur when the vortex shedding frequency is within 20 - 30% of a CWP natural frequency. Table (4) shows the possible effect of the occurrence of lock-on on the fatigue life of the CWP. A cumulative damage factor of 1.0 derived from the Palmgren-Miner rule implies that the entire life of the structure has been used up and hence fatigue failure can occur. It can be seen that for the case of a steel CWP immersed in seawater or exposed to sea-spray the cumulative damage factors are much larger than 1.0.

Table 4. Summary of CWP Fatigue Analysis Using the Palmgren-Miner Rule for 17% Steel; Punta Tuna Site, 30 Years (Reference 4)

DIAMETER, FT.	WALL THICKNESS, FT.	CUMULATIVE DAMAGE DUE TO WAVES ONLY (D FACTOR)	CUMULATIVE DAMAGE DUE TO WAVES AND CURRENTS, WITH LOCK-ON (D FACTOR)
30[1]	0.07	.0047	81.075
	0.1	.0028	29.81
	0.15	.0142	9.58
15[1]	.07	.016	>100
	0.1	.0028	>100
30[2]	.07	3.4×10^{-8}	.00134
30[2]	.15	3.99×10^{-7}	4.3×10^{-7}
15[2]	.07	2.3×10^{-11}	>100

1. Steel in seawater or sea-spray
2. Steel in air

Several vortex suppressor designs have been looked at for potential application to OTEC CWP's. One system for which experimental data is available is the helical strake concept produced by Fathom Oceanology, Inc., of Ontario, Canada. However, for CWP's with 15 to 100 feet diameters, Fathom has concluded that helical strakes may not be appropriate. These suppressors have been used successfully by the U.S. Coast Guard in reducing vortex shedding by 90% on 20-24 inch pylons used as supports for navigational aids in San Pablo Bay, California (Reference 21). However, in the case of the OTEC CWP's they would add 20% to the drag load and, at 10% of diameter in height, the strakes would not be simple devices to construct. Fathom also concluded that articulated fairings to streamline large pipes such as OTEC CWP's do not look particularly feasible or cost effective.

Fathom's current thoughts favor the use of the rforated shroud concept shown in Figure (14) which s been successfully used in smoke stack applications (Reference 22). In fact, Figure (14) compares the effect of a stack with and without shrouds of different dimensions on the maximum amplitude of cross wind oscillations. Figure (15) compares the drag coefficient of strakes, smooth cylinders and shrouds versus Reynolds number. Such a shroud could be constructed from a number of identical man-handleable modules fitted to standoffs projecting from the pipe surface. The standoff distance should be on the order of 0.1 x D (D = diameter of CWP). At least 10% of the pipe should be fitted and the location of the shroud should be centered about the point of greatest deflection.

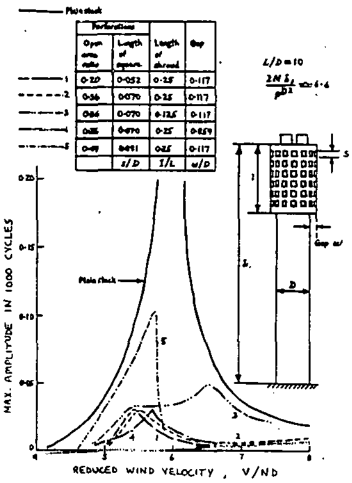

Figure 14. Effect on the Cross Wind Oscillations of Shrouds Fitted to a Model Stack (Ref.22)

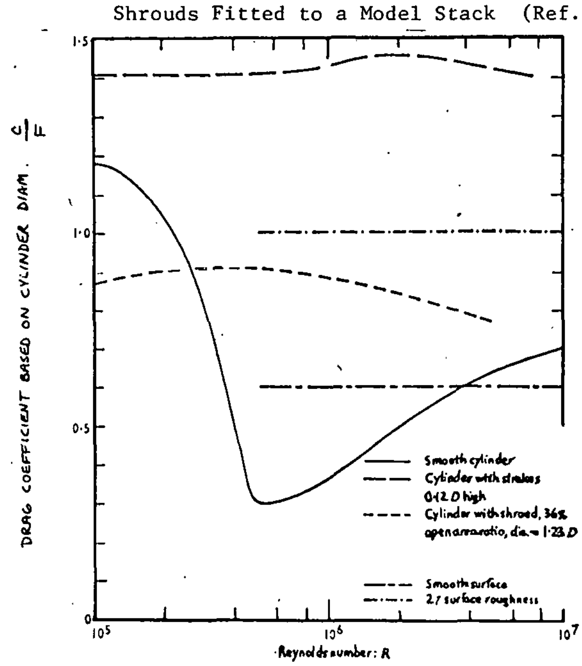

Figure 15. Drag Coefficient for an Infinite Circular Section Cylinder of Infinite Length to Diameter Ratio (Reference 22)

Detailed design of the shroud concept will depend upon the selected pipe, and shroud coefficients must be allowed for. Fathom favors the use of thermoplastics due to their neutral buoyancy and light weight in handling. GRP may also be a good candidate. The design will also be governed by the deployment techniques selected, i.e. at what stage does one assemble the shroud on the pipe? Cost-effective solutions to these problems should emerge from design studies and trade-off analyses at the appropriate time.

Another alternative or complementary solution to the vortex shedding excitation may well be the segmented pipe concept described earlier. For a segmented pipe, the criticality of vortex shedding may be eliminated if the highest degree of freedom natural frequency is less than the frequency of the worst frequency excitation force. For the 30-foot diameter CWP of the 40 MW plant, a periodic force with a frequency of .014 Hz (period of 73.4 seconds) represents the lowest frequency of excitation. If the use of six hinges as described before results in a fifth mode natural frequency of less than .014 Hz, the pipe may not be subjected to oscillatory lift forces.

It must be emphasized that, although vortex shedding may induce severe loadings and stresses on the CWP, the problem can be readily solved through proper design practice. For years the offshore oil and gas drilling industry has had to drill in areas of strong tides and currents. From experience gained so far, the industry appears to be considering a 2-knot water flow limit as a result of vortex shedding. In such cases, appropriate vortex suppressors are installed on risers and other similar structures. At the present time, there exist widely accepted design rules which cover the vortex shedding problem. This is exemplified by the Rules for the Construction and Inspection of Offshore Structures issued by Det Norske Veritas (Reference 23). Thus, the solution to the vortex shedding problem is well within the state-of-the-art and should be regarded as so in current and future OTEC CWP design efforts.

Corrosion Protection

It is a well-known fact that most of the structural materials used in ocean applications will experience deterioration due to corrosion effects. This is particularly true of materials such as steel whose fatigue life is drastically reduced when immersed in seawater or exposed to sea-spray as shown in the S-N curves of Figure (16). The fatigue analyses discussed above for the OTEC 10/40 MWe spar platform (Reference 4) indicate that the life of the CWP may suffer severely in the presence of large cyclic loads caused by waves and unsteady current loading. An inspection of Figure (16) indicates that a solution to the problem would be to use some type of anti-corrosion device which would shift the material endurance limit upward. This could be done via cathodic protection and/or the use of protective coatings. In either case, the net effect would be to approach the "in air" environment and hence, increase the ability of the structural material to withstand the long term fatigue loads. This can be easily concluded from the results presented in Table (4) where the cumulative damage factors for steel in air are considerably below unity, hence implying a safe structure from a fatigue standpoint. A recent study (Reference 6) includes bio/corrosion control systems as part of the design of steel CWP's. The system involves the use of cathodic protection and the removal of biofouling materials from the upper 300 feet of the CWP by a high pressure jet. Such a bio/corrosion control system is estimated to cost $1 to 1.2 million. It is suggested in Reference (6) that the effectiveness of the cathodic protection system be checked within a year after installation, and at least annually thereafter to insure adequate protection.

Conclusions

The ideas presented in this paper for reducing the CWP structural loads reflect the results of feasibility, conceptual and preliminary design studies which have been conducted for the Department of Energy in the last three years. Many other concepts have been proposed recently as outlined in References (5) and (6). These include the use of FRP Sandwich structures, bottom-mounted CWP concepts, polyethylene multiple-pipe concepts, reinforced elastomer pipes and many others. These concepts and their relative merits are discussed in more detail in separate papers being presented at this conference. In most cases, the ultimate objective of the design is to reduce the structural loading and response of the CWP and hence, to increase the level of confidence assigned to the structural integrity and long term survivability of the OTEC CWP. This can be achieved through the effective application of practical engineering analysis and design verification through small and large scale testing of the CWP and its components.

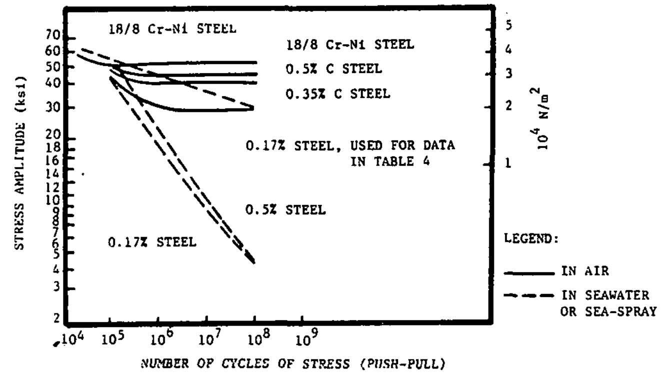

Figure 16. S-N Curve for Steels in Air and in a Marine Environment (Reference 4)

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DEVELOPMENT OF A LIGHTWEIGHT CONCRETE FOR OTEC COLD WATER PIPES

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Abstract

Development of a lightweight concrete suitable for use in cold water pipes of OTEC tropical grazing plantships is described. Target properties for the concrete include a minimum design compressive strength of 4000 psi and a maximum unit weight of 95 pcf.

Numerous trial mixes were prepared and tested to evaluate concrete properties obtained with available aggregates. Variables included very lightweight fine aggregates and newly developed admixtures. Tests were conducted to determine compressive, flexural, and tensile strengths as well as elastic and long term deformation characteristics. Water absorption was investigated under conditions simulating depths up to 3000 ft.

The investigation led to a lightweight concrete with an air dry unit weight of 77 pcf and compressive strength in excess of 4000 psi. Tests are continuing to determine long-term strength, deformation, and absorption characteristics, and properties under combined stress conditions.

Introduction

This paper describes part of an investigation to develop a lightweight concrete cold water pipe (CWP) for ocean thermal energy conversion (OTEC) plantships. The investigation is based on requirements established by the Johns Hopkins University Applied Physics Laboratory (APL). The OTEC program is sponsored by the U.S. Department of Energy.

The primary objective of research undertaken by the Portland Cement Association Construction Technology Laboratories (CTL) has been to design a lightweight concrete suitable for use in the pipe. Work on materials evaluation was also conducted at Concrete Technology Associates, Tacoma, Washington. Design of the plantship and cold water pipe is being conducted by ABAM Engineers Inc., Tacoma, Washington. This paper covers work performed at CTL.

The OTEC mission addressed by APL employs a floating plantship that produces energy-intensive products, such as ammonia, at sea for shipment back to the U.S. The plantship's CWP may be as large as 60 ft or more in diameter and 3000 ft in length. For the pilot plantship that will be evaluated prior to full-scale production, the CWP has an inner diameter of 30 ft and a length of 3000 ft. Weight of the CWP is of considerable importance because of deployment and ship buoyancy problems. Therefore, it was considered essential to obtain unit weights significantly lower than the 110 to 120 pcf range commonly used for structural lightweight concrete.

Establishment of target properties for the lightweight concrete to be used in the pipe was an evolutionary process. Initial properties established by APL included a minimum design compressive strength of 3000 psi, a modulus of elasticity of from 500,000 to 1,000,000 psi, and a unit weight in the range of 65 to 70 pcf. As work on materials evaluation and pipe design progressed it became apparent that the initial target properties required modification to accommodate design of the pipe and hinges. Target properties were therefore adjusted to include a minimum design compressive strength of 4000 psi and a unit weight in the range of 85 to 95 pcf. The design unit weight is for the pipe in service. Therefore, absorption of seawater by the lightweight concrete is included.

Lightweight concretes can be produced in a range of unit weights from 15 to 120 pcf.[1,2] This compares to normal weight concrete with a unit weight of about 145 pcf. A general classification of lightweight aggregates is shown in Fig. 1.[1] Lower unit weights indicate lower compressive strengths.

Lightweight aggregate concretes with unit weights ranging from 85 to 120 pcf are classified as structural concrete. For this classification 28 day compressive strengths must exceed 2500 psi.[1] Strengths commonly achieved range from 2500 to 6000 psi.[1,2] However, it is possible to obtain compressive strengths of 6000 to 12,000 psi with concretes having unit weights of 105 to 120 pcf.[2,3]

Lightweight concretes with unit weights ranging from 15 to 50 pcf are classified as insulating concretes. Compressive strengths for these materials range from 100 to 800 psi.[4] They are primarily used for their high thermal insulating characteristics.

Lightweight aggregate concretes with unit weights ranging from 50 to 85 pcf are classified as fill concretes. Concretes in this range have not had widespread use. This is because their strengths are not as high as structural lightweight concrete, and their insulating characteristics are not as good as lower density concretes.

Guides for fill concrete have been developed.[5] However, systematic efforts to upgrade structural

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properties of lightweight concretes in this range have not been reported.

The primary objective of the materials research at CTL has been to design a lightweight concrete within the weight range classified as fill concrete, but having strengths comparable to structural lightweight concrete. In evaluating strength versus weight characteristics, it was necessary to account for the fact that target properties were for the pipe in service. Therefore, weight gained by concrete from water absorption under hydrostatic pressures was an important consideration.

Research Approach

The basic research approach used to obtain target properties consisted of the following steps:

1. Search for candidate materials.
2. Tests of candidate materials.
3. Detailed tests of selected materials.

Search for Candidate Materials

The search for candidate materials was intended to isolate those that would have a high probability of producing the desired lightweight concrete without excessive cost. Several approaches were considered. These included optional replacement of the fine fraction of expanded lightweight aggregate particles with newly developed very lightweight small size particles, and use of special admixtures.

Review of previous investigations of structural lightweight concretes indicated that it was essential to consider expanded clay, shale or slate aggregates to obtain the required strength. However, such materials normally produce concrete in a higher weight range than desired for the cold water pipe. Therefore, a search was made to locate an aggregate with a higher than normal strength-to-weight ratio.

Based on manufacturers' data two aggregates were selected for tests. Available information indicated that the use of these aggregates would permit unit weight reductions of 15 to 25% of that obtained with most structural lightweight aggregates. To further reduce unit weight, consideration was given to replacement of fine aggregate particles with lighter weight materials.

Bulk unit weight of lightweight aggregates varies with size of aggregate particles. Fine fraction particles generally have a unit weight 25 to 45% higher than the coarse fraction.[6] Therefore, tests were made using special lightweight fines consisting of expanded clays, hollow fly ash, and hollow glass particles.

In addition to investigating different aggregates, various admixtures were considered to improve properties of the plastic and hardened concrete. In general admixtures were selected to improve workability and strength, and to reduce absorption.

Performance of selected admixtures was evaluated based on their effects on fresh and hardened concrete properties. Improvements in performance were also correlated with cost of the concrete mixes. Types of admixtures investigated included air-entraining agents, water reducers, superplasticizers, epoxy emulsions, and latex emulsions.

Tests of Candidate Materials

Once candidate materials were selected trial batches were made to obtain a mix design that would meet desired unit weight and strength requirements. Data were obtained on properties of fresh and hardened concrete. Properties measured included slump, air content, fresh unit weight, workability, compressive strength, and modulus of elasticity.

Results of tests for all candidate materials are summarized in Figure 2. It is apparent that

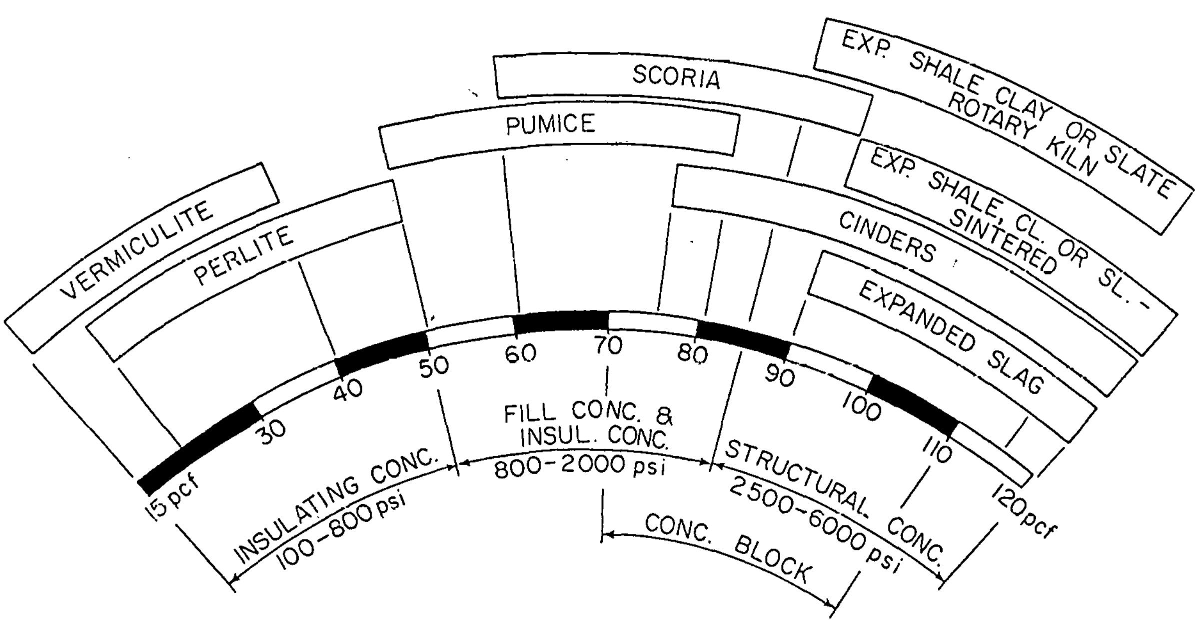

Fig. 1 Classification of Lightweight Aggregate Concretes (After Ref. 1)

compressive strength increased significantly with increased unit weight. Based on the tests of candidate materials, three mixes were selected for detailed testing. These were designated OTEC A, OTEC B, and OTEC C.

In this paper results are given for the OTEC C mix only. Properties of OTEC C concrete most closely represent those established as target values. Properties of other mixes and a more detailed description of the test program are given elsewhere.[7]

Detailed Tests of Selected Materials

The objective of the detailed tests was to more thoroughly define properties relevant to the proposed use of the concrete. Primary emphasis was given to strength and absorption characteristics.

Specimens for detailed tests were made from four six cubic foot batches. Procedures used for tests of the fresh and hardened concrete are listed in Table 1. In general, tests were in accordance with procedures of the American Society for Testing and Materials (ASTM).[8]

A standard test was not available to evaluate water absorption of concrete under hydrostatic pressure. Therefore, the following procedure was adopted. Tests were made in 5-in. diameter, 28-in. long steel cylindrical pressure tanks. Each tank held three 4x8-in. concrete cylinders. After cylinders were inserted, the tanks were filled with fresh water. Hydrostatic pressure was applied using compressed nitrogen gas. Pressurization and depressurization was at a rate of approximately 300 psi per hour.

Prior to insertion in the pressure chambers, cylinders were moist cured for 28 days. Using this method tests were made at hydrostatic pressures of 700 and 1400 psi. These pressures correspond to ocean depths of approximately 1500 and 3000 ft, respectively. Companion specimens to those in the pressure tanks were also placed in water at atmospheric pressure to provide a reference for absorption data.

Properties of OTEC Lightweight Concrete

Mix proportions for the OTEC C lightweight concrete are shown in Table 2. The mix consisted of Type I cement, fly ash, Livlite aggregate, water, WRDA, and vinsol resin.

Type I portland cement used in the mix met requirements of ASTM Designation: C150 "Standard Specification for Portland Cement."[8] In addition, the tricalcium aluminate content of the selected cement was approximately 7%, an amount considered satisfactory for use in marine environment.[9] The tricalcium aluminate content is related to resistance of the concrete to sulphate attack. Fly ash was used to improve strength and sulphate resistance as well as to reduce absorption and improve accelerated curing properties.

Livlite aggregate is manufactured by Tombigbee Lightweight Aggregate Corporation, Livingston, Alabama. Livlite aggregate was selected for its high strength-to-weight characteristics. It is an expanded clay with a 5/8-in. maximum aggregate size. Loose unit weights of the fine, medium, and coarse material were 39, 38, and 32 pcf, respectively. Moisture contents of as-received material ranged from 6% for the fines, to 0.5% for the coarse.

Livlite aggregate was selected based on tests conducted in this program. Since it was impossible to evaluate all lightweight aggregates, it is possible that equivalent properties can be obtained using other aggregate sources.

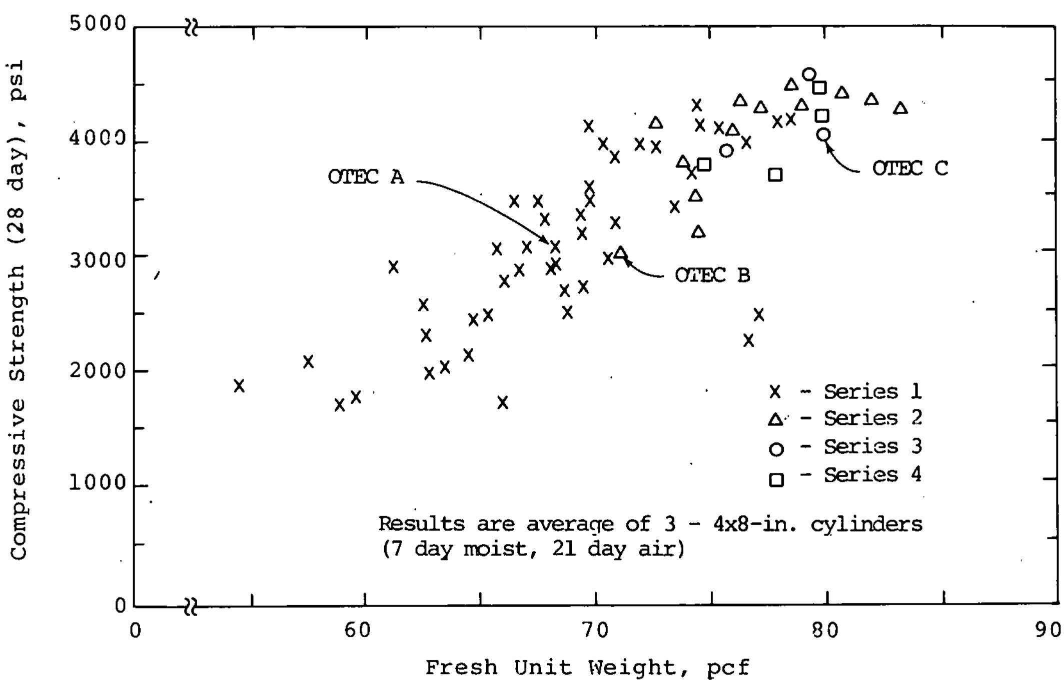

Fig. 2 Strength Versus Unit Weight for Candidate Materials

Liquid admixtures used in the mix included vinsol resin, an air entraining agent, and WRDA, a water reducer. The air entraining agent was used to improve workability and to decrease segregation and bleeding. The water reducer was used to improve strength without reducing workability.

Properties of Fresh Concrete

Slump, air content, fresh unit weight, and workability were used to evaluate properties of the fresh concrete. Slumps varied from approximately 2.5 to 2.9 in. Air contents ranged from 3.5 to 5.5%. Fresh unit weights varied from 78 to 81 pcf.

Workability of the mix was reasonably good. There was no evidence of segregation or bleeding. The concrete consolidated satisfactorily under external vibration. Limited tests made using internal vibration indicated that the concrete could be adequately placed. The mix tended to be "harsh" and was not easily finished. Therefore, it is not considered suitable for flat work where extensive finishing operations are required. This characteristic, however, does not limit its use in the pipe.

Variations of properties of the fresh mixes were similar to those commonly obtained for structural lightweight concrete.

Properties of Hardened Concrete

Tests of the hardened concrete were made to determine unit weight, strength, deformation, volume change, and absorption. Several of these tests are being continued to determine properties of the concrete over a one-year period.

TABLE 1 - DETAILED TESTS OF SELECTED MATERIALS

Property	Applicable Standard(8)
Aggregate Properties	ASTM Desig: C33
Slump	ASTM Desig: C143
Air Content	ASTM Desig: C173
Fresh Unit Weight	ASTM Desig: C138
Workability, Bleeding, etc.	Visual
Compressive Strength*	ASTM Desig: C39
Modulus of Elasticity*	ASTM Desig: C469
Flexural Strength	ASTM Desig: C78
Splitting Tensile Strength*	ASTM Desig: C496
Unit Weight, air dried concrete	ASTM Desig: C567
Unit Weight, pressure saturated concrete**	Pressure Chambers
Drying Shrinkage*	ASTM Desig: C157
Creep*	ASTM Desig: C512

*6x12-in. cylinders
**4x8-in. cylinders

Strength and Deformation. Results of tests for compressive strength, modulus of elasticity, Poissons ratio, tensile splitting strength, and modulus of rupture are summarized in Table 3. All specimens were moist cured a 100% RH and 73F for 7 days. After 7 days specimens were either moist cured or air dried. Moist cured specimens were maintained at 100% and 73F. Air dried specimens were maintained at 50% RH and 73F.

Compressive strengths exceeded 4000 psi at 14 days for both wet and dry specimens. Values of modulus of elasticity ranged from 1,130,000 to 1,670,000 psi. The relationship between measured values of compressive strength, unit weight, and modulus of elasticity was consistent with what would be expected for lightweight aggregate concretes.[1]

Values of Poissons ratio ranged from 0.17 to 0.21. These values are consistent with published data for structural lightweight concretes.[1]

Tensile strength data indicate that curing conditions had a significant effect. In particular, drying of the concrete reduced tensile strength. This would be expected because differential shrinkage stresses are induced as concrete dries.

Differential shrinkage stresses also changed the relationship between splitting tensile strength, a measure of direct tension, and modulus of rupture, a measure of tension in the presence of a strain gradient.

For moist cured specimens, splitting tensile strength was approximately 60% of the modulus of rupture. Tests of dry specimens at 14 and 28 days indicated splitting tensile strengths and modulus of rupture values were approximately equal. However, for dry specimens tested at 90 days, splitting tensile strengths were approximately 75% of the modulus of rupture. In addition, magnitude of the tensile strength values increased considerably. This may be an indication of a reduction in differential internal stresses as a result of more uniform moisture conditions over the longer drying period.

TABLE 2 - MIX PROPORTIONS FOR OTEC C*

Type I Cement	611 lb
Fly Ash	97 lb
Livlite Coarse	350 lb
Livlite Medium	416 lb
Livlite Fine	314 lb
Water	346 lb
WRDA**	56 oz
Vinsol Resin**	44 oz

*Quantities per cubic yard
**Proprietary product used in this investigation. Equivalent products are available from other sources.

Creep and Shrinkage. Creep tests are being run at stress levels of 1000 and 1500 psi. Results over a period of about four months indicate that creep will be greater than that of normal weight concrete.

Absorption. Absorption data are summarized in Figure 3. In this Figure, unit weight of the concrete is plotted as a function of time for three test conditions: 0 psi (atmospheric), 700 psi (1500 ft depth), and 1400 psi (3000 ft depth).

Data indicate that a significant part of the absorption takes place within the first month after hydrostatic pressure is applied. In addition, there is a difference in gain of unit weight depending on the level of hydrostatic pressure. Data must be obtained over a longer period before these results can be considered conclusive.

Similar tests on other concretes indicated that although mixes had fresh unit weights that varied from 68 to 80 pcf, their unit weights after pressurization at 1400 psi were approximately the same.

Construction Procedures

Mixing, transportation, and placing procedures for the OTEC lightweight concrete should be in accordance with recommended practices for structural lightweight aggregate concrete.[1,10]

Batching, Mixing, Transporting and Placing

Effective quality control procedures for batching and mixing the lightweight concrete will be required to insure a uniform product. In particular strict monitoring and control of moisture will be required to obtain uniform slumps and desired fresh unit weights. It is recommended that fresh unit weight, as well as air content and slump, be monitored as part of the quality control operations during production.

Although variations in gradation of the Livlite aggregate used in this project caused no problems, it would be desirable to have more uniform aggregate gradation. Installation of a small screening system at the batch plant may be desirable.

The lightweight concrete should not require special handling during transporting and placing. However, pumping lightweight concrete requires special consideration. Therefore, if pumping is considered for use in construction of pipe segments, trials should be made with the OTEC lightweight concrete.

Because of its light weight, reduction in capacity of equipment required to transport concrete should be possible.

Curing Procedures

Specific construction procedures for OTEC cold water pipes had not been determined when detailed tests were initiated. Therefore, curing conditions were investigated. Tests bracketed possible curing conditions that might be used in the field. Table 4 contains data on effects of curing conditions on properties of the mix. Curing conditions investigated included use of steam as well as ordinary moist curing.

Steam cured specimens were maintained at 160F for 18 hours in a steam cabinet. They were then stored in laboratory air at 73F and 50% RH. Moist cured specimens were stored at 73F and 100% RH. In all cases, air drying took place at 73F and 50% RH.

Results indicate that the mix is suitable for steam curing. Also accelerated curing using heated forms is feasible.

Handling and Deployment

Handling and deployment procedures for the cold water pipe must accommodate changes in unit weight

TABLE 3 - PROPERTIES OF HARDENED CONCRETE

Property	OTEC C					
	7 Days	14 Days		28 Days		90 Days
	Wet	Wet	Dry*	Wet	Dry*	Dry*
Unit Weight of Compressive Specimens Tested, pcf	81.3	80.6	78.3	80.7	77.3	76.4
Compressive Strength, psi	3,700	4,070	4,330	4,460	4,030	4,300
Modulus of Elasticity, ksi	1,400	1,600	1,650	1,670	1,300	1,130
Poisson's Ratio	0.20	0.21	0.21	0.21	0.21	0.17
Tensile Splitting Strength, psi	325	365	200	385	190	265
Modulus of Rupture, psi	480	640	200	570	115	345

*Air dried after 7 days moist

of the concrete that occur during curing, storage, and positioning of the pipe. Equipment requirements for handling the pipe in air will be reduced by use of lightweight concrete as compared to normal weight concrete.

As the pipe is deployed in the ocean, consideration must be given to increases in unit weight that occur as water is absorbed. This is evident from Fig. 3. Tests of absorption under hydrostatic pressure are not yet complete. As additional data are obtained, a more complete analysis of effects of hydrostatic pressure on weight of the pipe will be possible.

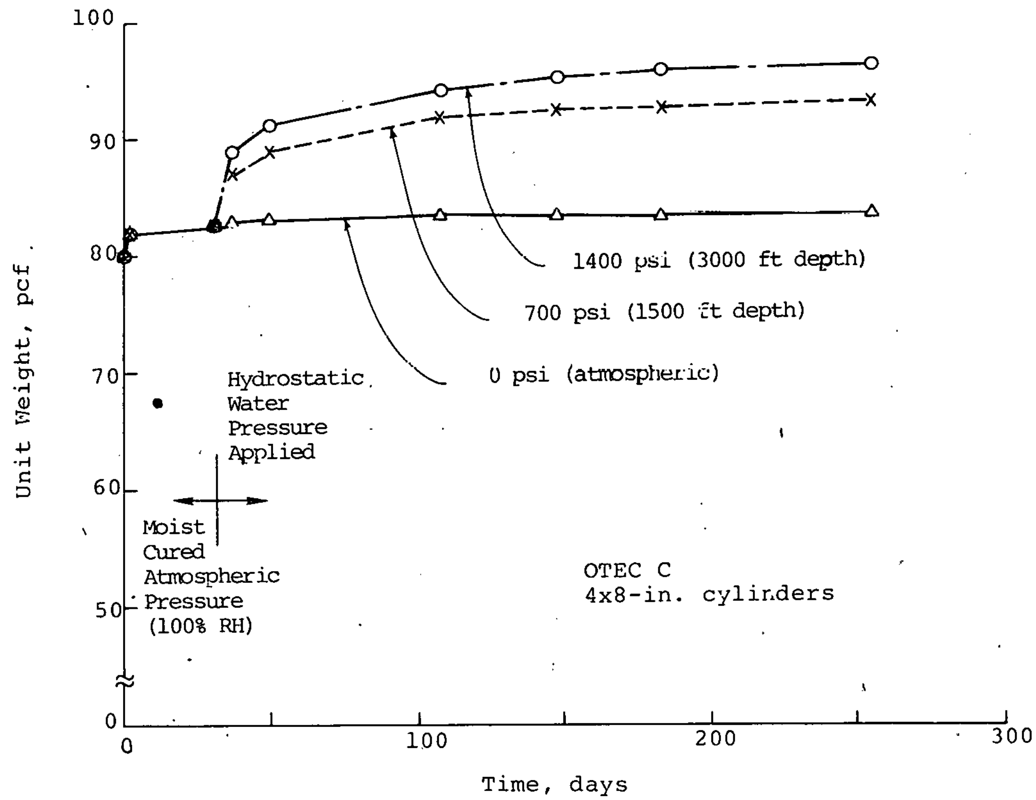

Fig. 3 Absorption Data

TABLE 4 - EFFECT OF CURING CONDITIONS

Curing Condition	Unit Weight, pcf	Compressive Strength, psi	Modulus of Elasticity, ksi	Poisson's Ratio	Tensile Splitting, psi	Modulus of Rupture, psi
Steamed - 1 Day	80.7	3,910	1,500	0.22	--	--
Steamed - 1 Day Dry - 27 Days	78.4	4,390	1,370	0.19	--	--
Steamed - 1 Day Dry - 89 Days	77.8	4,250	1,300	0.20	--	--
Moist Cured 28 Days	80.7	4,460	1,670	0.21	385	570
Moist - 7 Days Dry - 21 Days	77.3	4,030	1,300	0.21	190	115
Moist - 10 Days Dry - 80 Days	76.4	4,300	1,130	0.17	265	345

Post-Tensioning

Two aspects with regard to post-tensioning need special consideration. First, initial data indicate that creep of lightweight concrete is higher than that for normal weight concrete. This must be considered in developing post-tensioning procedures. Effects of creep are offset to some extent because lower weight of the lightweight concrete pipe reduces post-tensioning requirements.

The second aspect of post-tensioning that needs attention concerns development of anchorage zone stresses. Strength data indicate that tensile resistance varies as the concrete dries. This finding needs to be evaluated so that proper reinforcing details and post-tensioning procedures can be used to avoid anchorage cracking.

Marine Applications

Specific tests to determine suitability of lightweight concrete for use in marine environments were not part of the initial program. However, all tests to date indicate that properties of the lightweight concrete developed for the OTEC program are similar to those for structural lightweight concretes. Investigations of use of structural lightweight aggregate concrete for marine and offshore applications indicate that such concretes are suitable as long as proper design and construction procedures are used.[9,11,12,13,14]

Information is currently being developed on the long term strength, deformation, and absorption characteristics of OTEC lightweight concrete. In addition, tests will be conducted to determine strength characteristics under combined stress conditions similar to those that might occur in the submerged pipe. These tests will provide data for final design.

Several aspects need to be considered with regard to long term durability of the concrete in the marine environment. Since the pipe remains submerged throughout its life, wetting and drying conditions that lead to build-up of chloride ions will not exist. However, in the zone near the surface, extra precautions against corrosion may be considered. Possible precautions include special coatings on concrete or reinforcement, or use of high density concrete near the surface.

For sections of pipe at greater depths, corrosion is not expected to be as great a problem. This is because the concentration of chloride ions cannot exceed that caused by complete saturation. Also, availability of oxygen is limited.[13]

Even though the OTEC concrete has a low unit weight, it has a relatively high cement content and low net ratio of water to cementitious material. This is required for good durability.

Summary and Conclusions

An investigation was conducted to develop a lightweight concrete for use in cold water pipes of OTEC plants. The investigation consisted of a search for candidate materials, tests of candidate materials, and detailed tests of selected materials. Physical tests were conducted to determine compressive, flexural, and tensile strengths. Deformation characteristics were evaluated under instantaneous and sustained loads.

An important characteristic dictated by service requirements for the pipe is the tendency of hardened concrete to absorb water. This was investigated under conditions simulating depths to 3000 ft.

The investigation led to development of a lightweight concrete with the following nominal properties:

Fresh Unit Weight = 80 pcf
Air Dry Unit Weight = 77 pcf
Unit Weight After Absorption
(224 days, 1400 psi) = 97 pcf
(224 days, 700 psi) = 94 pcf
(224 days, 0 psi) = 84 pcf
Compressive Strength, f'_c (90 days) = 4300 psi.

Additional mixes with 3000 psi compressive strengths and lower unit weights were also tested in detail. Tests are continuing to determine long term strength, deformation, and absorption characteristics.

Acknowledgements

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Mr. E. A. Valko was in charge of fabrication and testing of specimens. Clerical and editorial assistance was provided by E. Ringquist.

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OSCILLATIONS OF AND DRIFT FORCE ACTING ON AN OTEC-CWP STRUCTURE

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Abstract

This model test deals with the oscillations of and the drift force acting on the structure of ocean thermal energy conversion plants to investigate the effect of the type of cold water pipe (CWP) and its attachment to the hull. A rigid, semi-flexible, or flexible CWP is connected to a circular floating hull by using a rigid, universal-joint, or slide-type attachment. The oscillations and drift force are found to be greater for the universal-joint and slide-type attachments than for the rigid type. The floatig system with flexible CWP exhibited smaller oscillations and drift forces than those with rigid type. Although these results are indicative, they should be carefully applied for the actual prototype structure subjected to real environmental conditions, since this study deals with regular waves under limited laboratory conditions. Further investigation is needed with larger models under more representative environmental conditions.

Symbols

The symbols used in this study are as follows:

- L_o: wave length ($= g.T^2/2\pi$)
- T : wave period
- H : incident wave height
- ζ : amount of heaving (twice amplitude)
- ξ : amount of surging (twice amplitude)
- ξ': amount of surging (twice amplitude measured on the top of the center pipe)
- θ : amount of pitching (twice amplitude)
- ϕ : a factor for non-dimensional conversion ($=\pi.H/L_o$)
- F_d: drift force
- ρ : fluid density
- g : acceleration of gravity

Introduction

The dependance of Japan's energy upon imported oil is so substantial to cause her incessant anxiety over the economical and political situation of the world. It is, therefore, especially a matter of importance for Japan, surrounded by the warm currents of the ocean, to utilize ocean thermal energy conversion (OTEC) plants.

In Japan, the major OTEC study has been conducted under the "Sunshine Project" promoted by the Agency of Industrial Science and Technology, Ministry of International Trade and Industry (MITI). A committee for investigating OTEC plants was established early in fiscal year 1974 to start with system analysis, design, environmental and resource assessment and development of OTEC power plants. One of the major works of the committee so far is the conceptual design of a commercial 100-MW_e power plant.

Starting with the 1975 design of a hull of the rectangular barge type, recent activity covers a submerged cylinder type with the experimental studies on the motions in waves. The ability to predict the motions of the hull coupled with the CWP is one of the important objectives of the OTEC program.

Some examples of CWP's introduced are a rigid pipe made of concrete,[1] a flexible pipe made of a frame covered with nylon-fiber-reinforced neoprene,[2] a composite pipe made of steel rings linked with vertical ropes and circular membranes,[3] etc.

With respect to the attachment of CWP to the hull, a moderately flexible attachment is recommended for the balance of CWP and uniform distribution of the bending moment over the length of the pipe.[4] In this paper, CWP's made of aluminium, glass-reinforced plastic (GRP), and concrete with flexible joints are discussed.

The analytical and experimental studies on CWP/hull motions and CWP stresses have been conducted and applied to the design.[5-8]

The flexibility of the pipe itself and its attachment to the hull play an influential rôle in the behavior of the structure. Although much research has been performed, it is necessary at this stage of activity to compare the representative types of CWP's and the attachments to the hull in terms of motions of the structure. For this reason, this experimental study has been performed.

Experimental Details

Model

A plant having a 12-m diameter, 500-m long CWP fixed to a cylindrical hull of 100-m diameter, 31-m high, was reduced to a model with a scale of 1:200. Due to limited testing facilities, the CWP with the length of 2.5-m (500-m/200) was again reduced to 0.6-m by suitably adjusting the weight to equate the position of the center of gravity with that of prototype. The particulars are shown in Fig.1.

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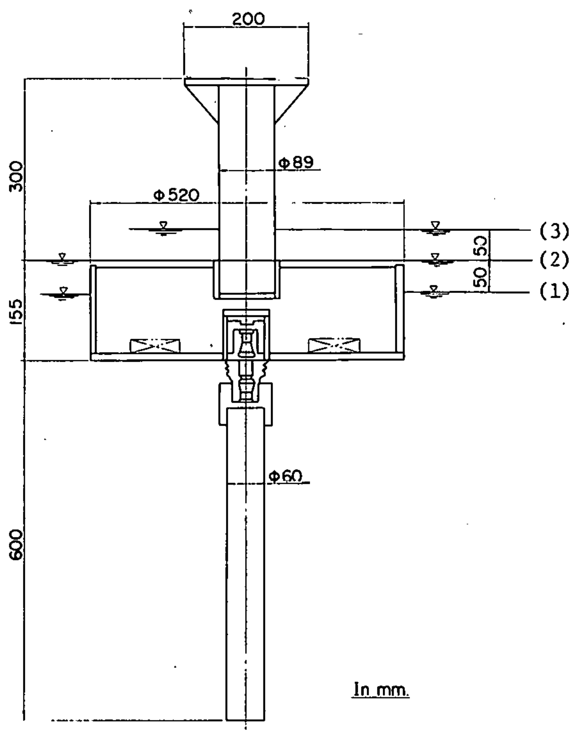


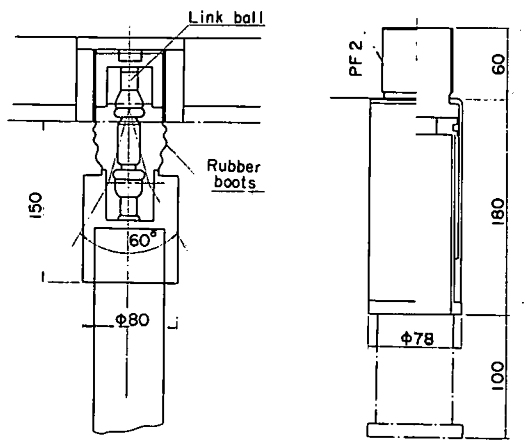
Fig.1 Model of OTEC-CWP structure.

CWP Attachment Types

The following attachment types, with their symbols are discussed in this study:

R : Rigid type
U : Universal type free for pitching
S_1: Slide type free for heaving with spring coefficient of 5.0 kg/cm
S_2: Slide type free for heaving with spring coefficient of 16.8 kg/cm

Figure 2 illustrates the latter 3 types.



a) U type b) S_1 and S_2 types

Fig.2 Types of CWP attachments.

CWP Types

The following three types of CWP are used:

R : Rigid type made of vinyl chloride
F_1: Semi-flexible type made of vinyl chloride and rubber (see Fig.3)
F_2: Flexible type made of rubber and reinforced with steel rings (see Fig.3)

The weight of the model is given in Table 1.

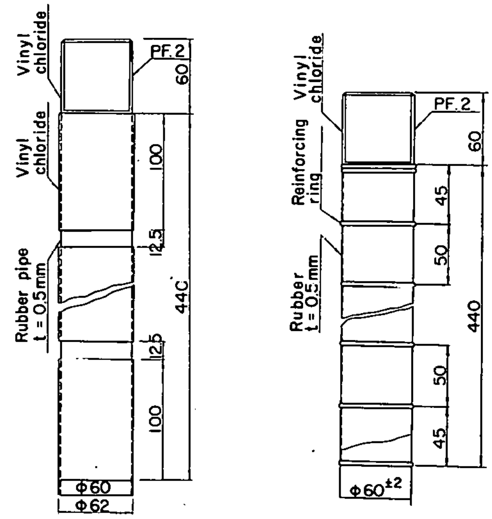


Fig.3 The F_1(left) and F_2(right) types of CWP's.

Table 1 Weight of model.

Parts of model	Weight(kg)
Center pipe	1.7
Cylindrical hull	14.1
Attachment of CWP (U)	2.7
Attachment of CWP (S_1,S_2)	2.5
CWP (R)	13.8
CWP (F_1)	0.5
CWP (F_2)	0.4

The Experimental Set-up and Method

The details of the wave basin are shown in Table 2. Wave height and period are measured with capacitance wavemeters.

Oscillations of the model in the wave basin are recorded with 3-component motion measurement apparatus to obtain amplitude of heaving, pitching and surging, separately. Fundamental characteristics of oscillations of the model are measured with inclining test and damping vibration test.

As shown in Fig.4, a pulley system with wire and spring is used to make heaving, pitching and surging oscillations free and the drift force acting in the direction of wave propagation is measured by counterbalancing with weight.

Table 2 Wave basin characteristics.

Dimensions	
Length	10.75 m
Width	4.35 m
Depth	1.05 m
Wave generator	Plunger type
Wave absorber	Beach type

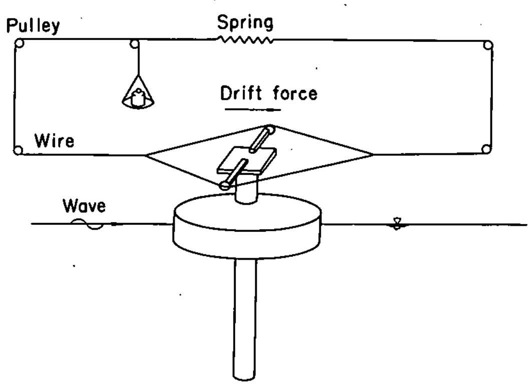


Fig.4 Drift force measuring device.

Experimental Cases and Parameters

The models and drafts taken in the experiment are shown in Table 3. The numbers (1), (2) and (3) refer to draft conditions, namely, the floating state with free board, the state with the water up to the top of the structure, and the submerged state, respectively.

Regular waves with maximum wave height of 8-cm (16-m in prototype) and wave period ranging from 0.4-sec to 1.7-sec (about 6-sec to 24-sec in prototype) are generated. Wave propagation is fixed in one direction.

Results and Conclusions

Heaving (Fig.5)

In case of R type of CWP and joint with draft condition (1), heaving is very large at resonance and ζ/H approaches unity for long period waves. In other words, as the wave period increases, the body tends to oscillate with the same amplitude as that of incident wave. As the draft increases, heaving decreases. This is probably due to the fact that the natural period in heave becomes long and the eddy and wave formation cause a nonlinear damping force when the hull moves up and down around the water level. Accordingly, even when the draft increases for draft condition (1) and the wave splashes on the top of hull, the effect stated above has to be considered. The response, there-

Table 3 Cases of model tests.

Symbol	Attachment	CWP	Free-board (cm)	Weight†) (kg)	Center of gravity††) (cm)	GM (cm)	Natural frequency Heave (sec)	Natural frequency Pitch (sec)
R-(1)	Rigid	Steel pipe	5	30.6	28	43.1	1.00	1.20
R-(2)	Rigid	Steel pipe	0	35.0	25	16.5	0.96	1.48
R-(3)	Rigid	Steel pipe	-5	36.2	25	13.0	9.00	1.90
U-(1)	Universal joint	Steel pipe	5	30.6	28	33.8	1.05	0.68
U-(2)	Universal joint	Steel pipe	0	35.0	25	6.9	1.03	0.70
U-(3)	Universal joint	Steel pipe	-5	36.2	25	4.7	9.32	2.12
S_1-(1)	Slide 1	Steel pipe	5	30.4	28	43.4	1.05	1.34
S_1-(2)	Slide 1	Steel pipe	0	34.8	25	16.5	1.18	1.40
S_1-(3)	Slide 1	Steel pipe	-5	36.0	25	13.1	9.20	1.65
S_2-(1)	Slide 2	Steel pipe	5	30.4	28	43.4	1.01	1.34
F_1-(1)	Rigid	Short pipes of vinyl chloride connected by rubber couplings	5	17.3	12	23.3	0.96	0.89
F_2-(1)	Rigid	Rubber reinforced with spacings of steel rings	5	17.3	13	23.4	1.00	0.95

†) This includes weight for draft adjusting. ††) Measured from the top of the hull

fore, depends not only on wave period but also on the splashing conditions of the wave, namely, the draft and wave height (see Fig.5 a)).

The response with the universal-joint is greater than that with rigid-joint, probably due to the coupling of heaving and pitching oscillations.

For types S_1 and S_2, which allow only heaving, the springs do not have a capacity for such large heaving. Consequently, they show response curves similar to that for the rigid type. The natural periods of CWP's supported with springs in the attachments S_1 and S_2 are 0.7-sec and 0.5-sec, respectively. But the peaks are not found around these periods but rather around the natural period of the model itself.

The F_1 and F_2 types with flexible pipes show responses similar to a pontoon type of structure with least coupling to other modes of motions.

Above mentioned types of CWP and attachment are compared in Fig.5 b). The universal-joint shows especially large oscillation around resonance period, and other types, almost the same resonance characteristics. Under normal environmental conditions for operation, heaving will be smaller.

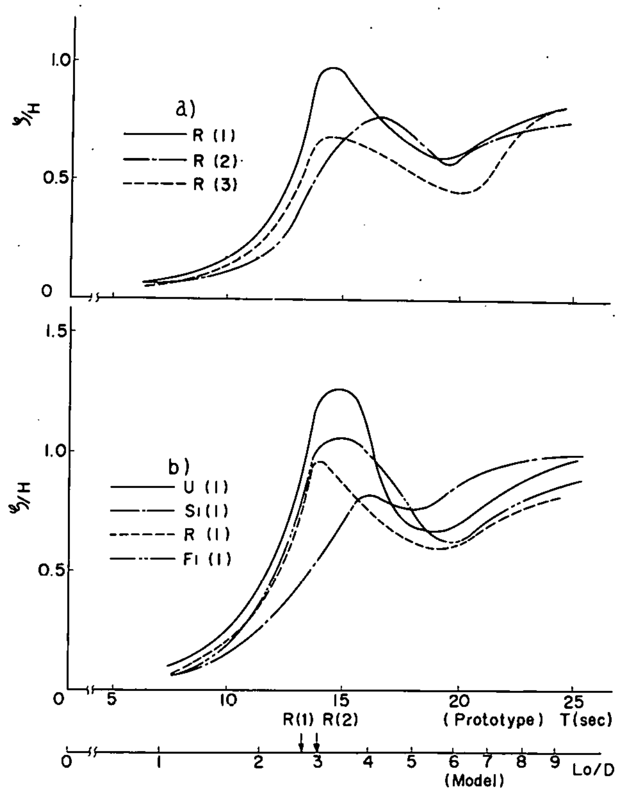


Fig. 5 Heaving characteristics a) of R type (arrows indicate L_o/D at natural period) and b) for draft condition (1).

Pitching (Fig.6)

Pitching is especially large for the R type of CWP and attachment around the resonance period. But for draft condition (2), there is some shift of resonance from the natural period. This is probably due to a nonlinear damping force as explained before, and/or to some error in the measurement. Again there is a small peak in the region less than the resonance period. This also may be attributed to nonlinear damping (see Fig.6 a)).

For the U joint, for both draft condition (1) and (2), the natural period is small but the resonance period in both cases is much larger, with considerable oscillation, probably due to the coupled motions of the CWP and the hull.

Since the spring systems in slide types do not affect pitching motion, the resonance curves for rigid and slide types are similar. But in the S_2 case with draft condition (1), response increases even beyond the resonance period.

In F_1 and F_2 cases, the response is generally small, and $\theta/2\phi$ does not exceed 2.0 even around resonance period. These cases show smaller pitching for all wave periods than other cases due to the lightness of the CWP. For draft condition (3) (submerged state), the natural period is very long, and we rarely come across such long period waves in nature. There is no fear of resonance in the prototype (see Fig.6 b)).

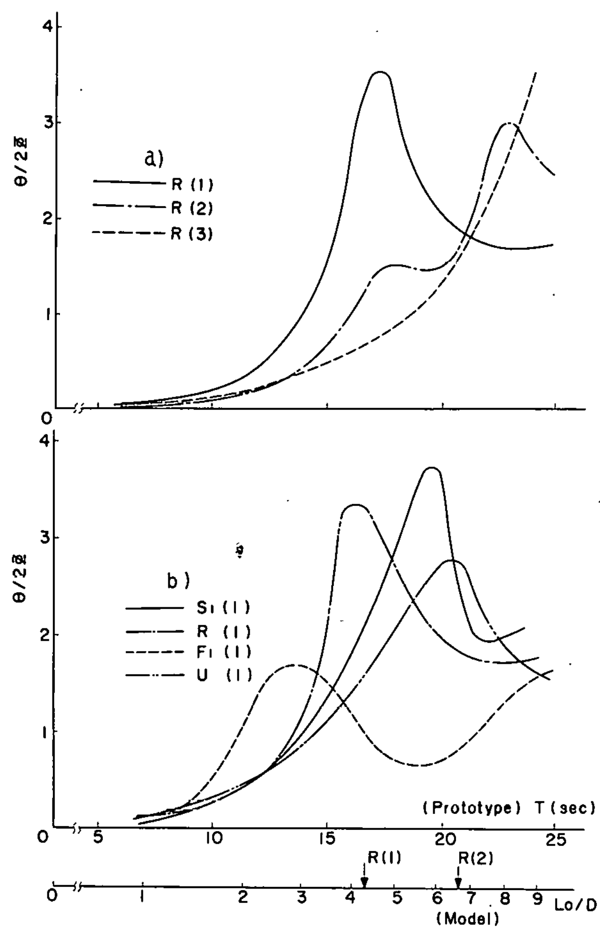


Fig.6 Pitching characteristics a) of R type (arrows indicate L_o/D at natural period) and b) for draft condition (1).

Surging (Fig.7)

For draft condition (1), the measuring device for surge motion is placed on the top of the center pipe. Accordingly, to and fro motion due to pitching above the center of gravity has to be deducted from the measured value to get the actual value of surging. Due to this manipulation, the measurements seem to be scattered, especially in case U, where the coupled motions of pipe and hull are considerable, and it is very difficult to assess the exact amount of pitching. The conclusions on surging characteristics are drawn with the error due to the above reason in view.

In R-(1) case, there is one peak of response at $L_o/D = 3$, and this falls in around $L_o/D = 6$, converging then to $\xi/H = 1.0$ for long period. The response in R-(2) case is smaller than that in R-(1) and it is much smaller in R-(3) (see Fig.7 a)).

The response curves for slide types are similar to those for the rigid type, since the horizontal movement has no relation to the sliding motion of the attachment.

In F_1 and F_2 cases, the response is much smaller than that in other cases.

Figure 7 b) compares surging motions among these cases for draft condition (1).

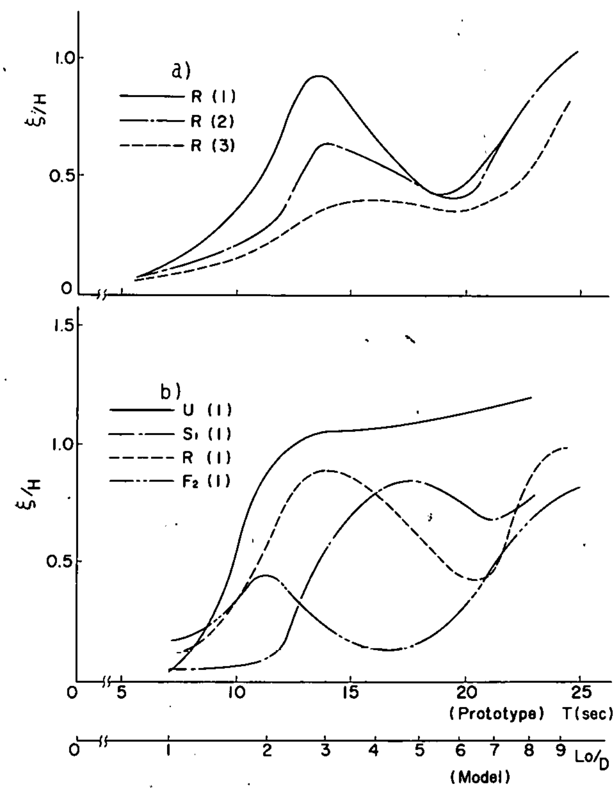


Fig.7 Surging characteristics a) of R type and b) for draft condition (1).

Drift Force Characteristics (Fig.8)

As shown in Fig.8, the drift force is nearly zero for long period waves in the R-(1) case. But for short period waves, the data are scattered when the period is around 0.5-sec due to the splashing of waves on the top of the hull, which causes a nonlinear wave force. For draft conditions (2) and (3), the drift force is smaller than for draft condition (1). The drift force of the submerged structure is smaller than that of the floating structure.

For the U joint, the drift force is considerable around $L_o/D = 2$ for all draft conditions, probably due to the movement of the hull, particularly the effect of the phase difference between heaving and pitching motions.

In the S_1 and S_2 cases, the drift force remains similar to case R except for draft condition (1), near $L_o/D = 7$, where the drift forces are greater than that in R type, probably due to the peak of the pitching response existing near this value and the phase difference between heaving and pitching oscillations.

On the other hand, in cases F_1 and F_2 when for $L_o/D > 4$, there is no drift force. For smaller periods, the drift force is smaller than that of other types. This phenomenon is probably due to the fact that motions except heaving are much less in this case, with almost the same amount of heaving response as that of other types. Consequently, the phase differences between different modes of motion and the position of the center of gravity may be responsible for this phenomenon.

Comparing these cases, the drift force of case F is small for a wide range of wave periods and is decreased more for the submerged state. On the other hand, the drift force of case U is always large.

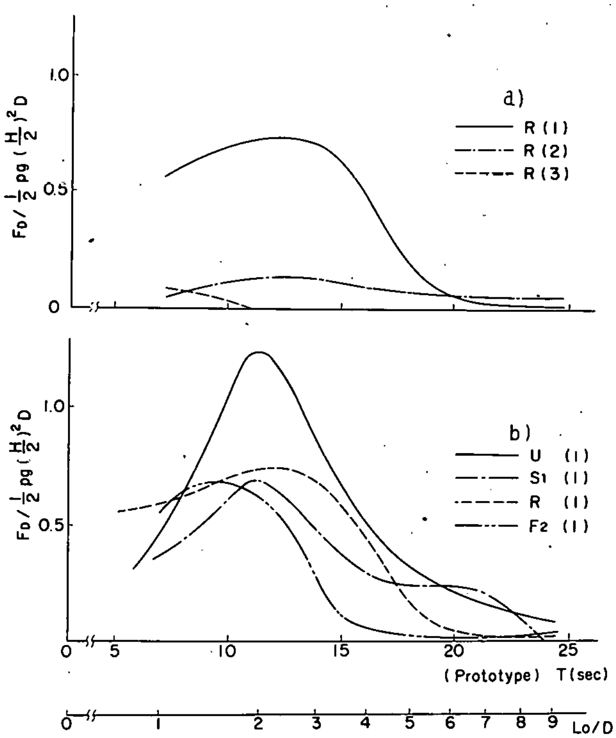


Fig.8 Characteristics of drift force a) of R type and b) for draft condition (1).

Conclusions

The conclusions obtained in this experimental study are summarized as follows:

1) With respect to heaving except that at resonance, all cases have almost the same characteristics. But at resonance, heaving is more in case U than that in other cases. There is not much difference in hull heaving between the rigid and flexible types of CWP. Heaving decreases as the draft increases.

2) The variation of pitching is more pronounced for different flexibilities of the CWP than for differences in the type of attachment. When the light flexible CWP is used, the pitching is smaller for a wide range of periods than those in other cases. In the submerged state, the natural period is too long to cause resonance in nature. This type shows, therefore, little motion, being effective for offshore structures.

3) Although an exact analysis of the results is not possible because of data scatter, no special advantage is observed in any type of attachment with respect to surging motion. The largest response is shown by the rigid CWP; the smallest, by the flexible type.

4) The drift force is greater for the U joint than for other attachments. The drift force for the flexible CWP is smaller than that of the rigid type. The drift force is smallest for the submerged type.

In summary, oscillations and drift forces are more pronounced for the U joint than in other cases of a rigid CWP. Next comes the slide attachment, the smallest being rigid type. When the attachment is allowed to move, the oscillations and the drift force increase. For a light flexible CWP, oscillations and drift force are small.

However, it is rather difficult to reach definite conclusions for the CWP on the basis of these experimental results. As the size increases up to prototype, and when moorings are applied under natural environmental conditions, many unforeseen factors may come into play, and there will be a need to investigate these cases again in more detail, with large-scale models, under more representative natural conditions.

Acknowledgements

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DISCUSSION

T. McGuinness, NOAA: I do not understand your conclusions. If I look through your various curves the results show consistently that, for pipes attached to a platform via a universal joint, the heave and roll and drift force amplifications were higher than for a pipe rigidly attached to the platform, which in my mind smacks against the laws of physics. I don't understand why the amplification would be higher with a decoupling, at least to some degree of freedom by use of a universal joint.

H. Kobayashi: At this point, we also have some doubt and we are trying to do some experiments with larger scale models. But one thing I can say is that when you deal with a universal-joint type the motion of the structure itself, and at the same time that of cold water pipe, should be considered. This motion of the pipe might cause some larger amplifications at certain conditions. But this also is in doubt, and the most interesting cases, which we are going to try next, are with a cold water pipe of a flexible material, then a cold water pipe attached with that coupling.

T. McGuinness: May I suggest that there has been a study by Lockheed of commercial-size platforms showing that the problem is not so much with the universal joint versus a rigid connection, rather where the attachment point of the pipe is relative to the center of rotation of the platform. Basically, what you are doing is creating a moment arm if you do not have the pipe attached close to the center of rotation of the platform. And varying that distance drastically changes the platform pipe coupling situation versus a universal joint change. I suggest that if you consider further experiments, first you should investigate the Lockheed studies, and perhaps vary your parameters relative to location of those connection points. I am not a dynamics expert, but it seems to me that anytime we talk about dynamic systems and the modeling systems used, we have to understand the dynamic response of a system. It would be well to present the data in a undimensional form relative to, say, the resonance conditions. Large amplitudes may simply indicate lock-on to a natural resonance of the system or nearness to a natural resonance of the system.

DYNAMIC ANALYSIS OF COLD-WATER-PIPE SYSTEMS FOR OCEAN THERMAL POWER GENERATION PLANTS

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Abstract

The dynamic analysis of cold-water intake pipes made of FRP (fiberglass-reinforced plastic) presented ~~herein~~ provides the criteria for the structural design in the stability sense. The mathematical models for both the cantilever and the simply-supported beam are developed, and the approximate transfer functions are obtained through Laplace transforms and Galerkin's 5th-order polynominal fitting. The design criteria are obtained as functions of stiffness, fluid velocity, material properties, and pipe dimensions.

Nomenclature

ρ Density of sea water, 1.03 t/m^3, where t represents a metric ton

ρ_F Density of pipe material (FRP), 1.1 t/m^3

ω Rotation angle, rad

A_a Section area inside pipe, 4.15 m^2

A_F Section area of pipe wall, 0.22 m^2

E Young's modulus; for FRP, 4.99×10^5 t/m^2

I Moment of inertia of pipe, 0.15 m^4

v Velocity of fluid (ρ is constant)

L Pipe length, m

C Viscous damping coefficient, t·s/m^2

R Dimension of amplitude

t time, s

α_0 $A_a \rho v^2 L^2/EI$

α_1 $2\, A_a \rho v\, C/ZEI$

α_2 CL^2/ZEI

Z^2 $(\rho A_a + \rho_F A_F)EI$

m $\rho A_a + \rho_F A_F$

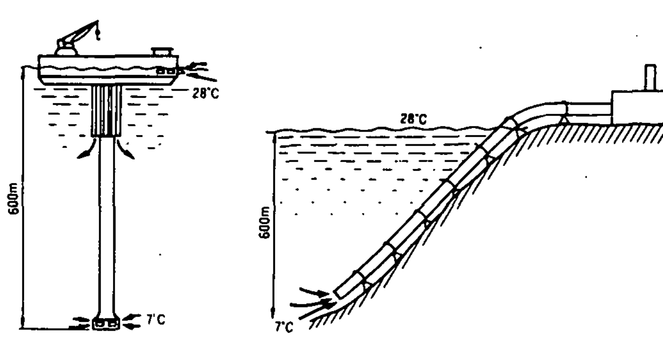
Introduction

An Ocean Thermal Energy Conversion (OTEC) system uses the temperature difference between the sea-surface temperature and the deep seawater temperature to generate power. In the Sunshine Project, promoted by the Ministry of Trade and Industry of Japan, many investigations have been conducted on OTEC during the past five years. We have addressed the feasibility of using fiberglass-reinforced plastic (FRP) as the material for the cold-water intake pipes. The required properties depend on the methods used to install the CWP and to fix it. Hence it is reasonable to treat separately the investigations of two aspects: (a) required properties to permit the installation of the CWP and (b) required properties during the operation of the power plant including structural safety, insulation ability, durability, and economy.

We have developed the structural analysis covering the following external loads and have established the fundamental design criteria for determining the structural dimensions of FRP pipes:

1. Static loads induced by the coastal current, the static water pressure, the buoyancy, etc.;
2. The dynamic load imposed by wave turbulence; and
3. The load imposed by the internal flow and the impact.

The subject of this paper is a part of the dynamic stability analysis of the self-vibration that would be induced by the seawater flowing through the pipe, whose structural design had been already completed. We first built the mathematical model of the beam vibrations with two types of the boundary conditions: (a) cantilevers for floating OTEC plants (Fig. 1a) and (b) simply-supported beams for onshore OTEC plants on tropical islands (Fig. 1b).



a) Floating plant b) Fixed onshore plant

Fig. 1 Types of OTEC plants considered.

Through the Galerkin method of integral residuals with respect to the displacement of the beam[1] and the Laplace transform to time[2] applied to the differential equations of the model, we obtained the solution and the transfer functions of the beam in a compact form, which are approximations in the total energy sense like the buckling model. The Hurwitz criteria of stability[3] are also applied to the transfer functions, and the stability zones of the combination of physical parameters are successfully obtained in the form of inequalities involving the stiffness, EI, scale dimensions, L,D, velocity, v, the densities of the material, ρ_F, and the seawater, ρ.

FRP as the Pipe Material

Steel pipelines have been commonly used for fuel gas and for petroleum transportation under seawater, often as a composite structure with coal tar-epoxy or epoxy-polyethylene. It was reported by the transportation authority in the U.S. that 46% of the failures or accidents were caused by corrosion.[4] A potential material for coping with these problems is FRP, which has been widely investigated in recent years. FRP has the following basic features: (a) a higher ratio of failure strength/density (steel: 6.4, FRP: 10), (b) anti-corrosiveness, (c) insulation ability, and (d) a lower and controllable (variable) density (steel: 7.8; FRP: 1.1 to 1.8).

Showa Denko K.K., one of the leading FRP manufacturers in Japan, with the cooperation of the Tokyo Electric Service Co. (TEPSCO), has been concerned with FRP development.

Mathematical Models

A cylindrical CWP is subject to the lateral inertia force posed by the internal seawater flow at constant velocity; i.e., the vibrational displacement of the pipe changes the direction of the vectoral inner flow resulting from the lateral inertia force. Furthermore, the inertia force of the inner mass of fluid and the Coriolis inertia force, caused by the flowing fluid in the moving reference, must be considered in the stability problem.[5]

The free motion of any span of the beam is given by a differential equation of the following type (see Appendix):

$$m\frac{\partial^2 y}{\partial t^2} + 2\rho v\frac{\partial^2 y}{\partial t\partial x} + \rho v^2\frac{\partial^2 y}{\partial x^2} + EI\frac{\partial^4 y}{\partial x^4} + C\frac{\partial y}{\partial t} = 0 \quad (1)$$

where m is the mass per unit length of FRP and the inner confined fluid, and C is the viscous damping coefficient.

The nondimensional form of Eq. 1 is

$$\frac{\partial^2\eta}{\partial t^2} + \alpha_1\frac{\partial^2\eta}{\partial t\partial\xi} + \alpha_0\frac{\partial^2\eta}{\partial\xi^2} + \frac{\partial^4\eta}{\partial\xi^4} + \alpha_2\frac{\partial\eta}{\partial t} = 0 \quad (2)$$

where

$$\eta \equiv y/R,\ \xi \equiv x/L,\ \tau = t/ZL^2$$

Boundary Conditions

Case 1 --- Cantilever type (see Fig. 2a)

$$\text{at } \xi = 1,\ \eta = Am\sin\omega\tau,\ \partial\eta/\partial\xi = 0$$

$$\text{at } \xi = 0,\ \partial^2\eta/\partial\xi^2 = 0, \partial^3\eta/\partial\xi = 0$$

where Am is related to the forcing function.

Case 2 --- Simply-supported beam type (see Fig. 2b)

$$\text{at } \xi = 1,\ \eta = 0,\ \partial^2\eta/\partial\xi^2 = 0$$

$$\text{at } \xi = 0,\ \eta = 0,\ \partial^2\eta/\partial\xi^2 = 0$$

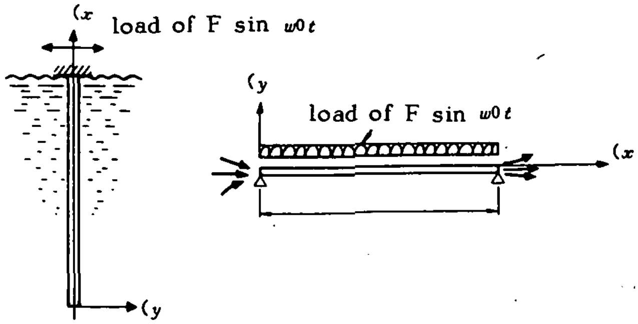


Fig. 2 Notation for the two types shown in Fig. 1.

Solutions (Case 1)

After obtaining the Laplace transform of Eq. 2 and using the weighting functions, Eqs. 3 and 4, the 5th-order polynomial solution of the Galerkin approximation is expressed as Eq. 5:

Weighting functions

$$\delta v_1 = 3 - 4\xi + \xi^4 \quad (3)$$

$$\delta v_2 = 4 - 5\xi + \xi^5 \quad (4)$$

$$\overline{\eta}(\xi,\tau) = A_4(3 - 4\xi + \xi^4) + A_5(4 - 5\xi + \xi^5) + Am\sin\omega\tau \quad (5)$$

where A_4 and A_5 are determined from the boundary conditions. The solution in Laplace space S is expressed as

$$\overline{\eta}(\xi,S) = \frac{Am\omega}{S^2+\omega^2}\Big[\frac{S^2+\alpha_2 S}{P(S)}\{-5.27 + 2.82\alpha_0 - 1.24\alpha_1 S - 23.3(S^2+\alpha_2 S)\}(3 - 4\xi + \xi^4) + \frac{S^2+\alpha_2 S}{P(S)}(0.47 - 1.91\alpha_0 + 0.82\alpha_1 S)(4 - 5\xi + \xi^5) + 1\Big] \quad (6)$$

where

$$P(S) = 23.3S^4 + (-45.3\alpha_1 + 46.6\alpha_2)S^3 + (23.3\alpha_2^2 - 45.3\alpha_1\alpha_2 - 0.2\alpha_1^2 + 16.9\alpha_0 + 295.0)S^2 + (16.9\alpha_0\alpha_2 - 0.4\alpha_0\alpha_1 - 7.8\alpha_1 + 295.0\alpha_2)S + (57.6 + 1.26\alpha_0 - 0.01\alpha_0^2) \quad (7)$$

Stability Criteria[3]

The denominator $P(S)$ (of Eq. 6) provides a very important clue for stability criteria.

Let us define the roots of $P(S) = 0$ as S_i's. Even if one root among S_i has a positive real part, the whole system will reach the instability zone which means induced self-vibration. We use Hurwitz Determinants to define the stability zone. For brevity, we rewrite Eq. 7 as the following:

$$P = a_0S^4 + a_1S^3 + a_2S^2 + a_3S + a_4 \tag{8}$$

The Hurwitz criteria are:

$$H_1 = |a_0| > 0$$

$$H_2 = \begin{vmatrix} a_1 & a_3 \\ a_0 & a_2 \end{vmatrix} > 0$$

$$H_2 = \begin{vmatrix} a_1 & a_3 & 0 \\ a_0 & a_2 & a_4 \\ 0 & a_1 & a_3 \end{vmatrix} > 0$$

The beam will remain stable when all a_i's in Eq. 8 and H_1, H_2, and H_3 are positive.

Figure 3 and Table 1 present the critical values of the combination of pipe length, L, and fluid velocity, v, for the values of FRP properties and pipe dimensions given in the Nomenclature.

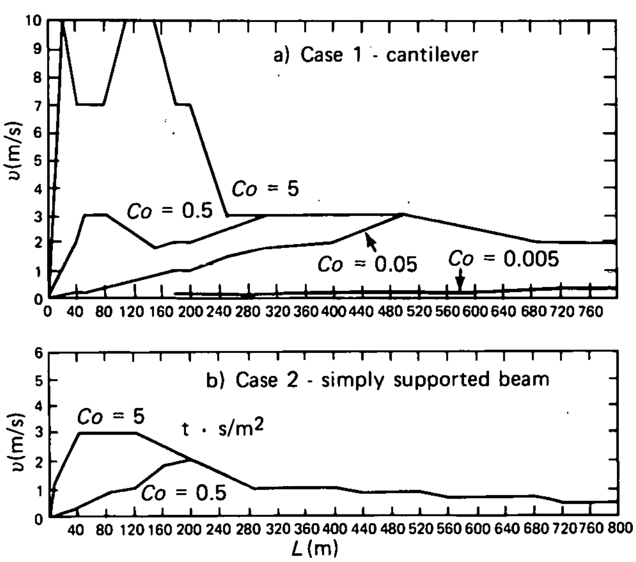


Fig. 3 Upper limits of stability.

Summary

The zones of stability for both the cantilever and the simply-supported beam representations of FRP pipes have been obtained and are powerful tools for checking the structural design, which had been done previously for other sources of loads such as wave forces, current forces, etc.

The solution presented here is obtained from the viewpoint of the total energy, which is different from the usual one that contains the proper frequencies.

If we wish to include external vibrational forces, such as waves, in this model, this kind of energy method cannot be applied. As for the viscous damping coefficients, C, we should take empirical values determined by experiments.

Table 1 Length-velocity solutions for $C = 0.5$ t·s/m ^{2}

L (m)	v, m/s Cantilever Case 1	Simply Supported Beam, Case 2
15	0.6	0.1
25	2.5	0.3
50	3.0	0.5
80	3.0	0.8
100	2.5	0.9
200	2.0	2.0
300	3.0	1.0
400	3.0	1.0
500	3.0	0.8
600	2.5	0.6
700	2.0	0.4
800	2.0	0.4

Acknowledgement

This study was sponsored by Tokyo Electric Power Service Co., and the permission of manager, Mr. Koike, is gratefully acknowledged. We also thank Professor Masaru Hoshiya, who helped us substantially.

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Appendix

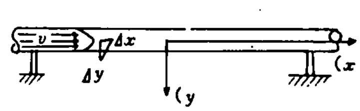

Fig. A-1 Notation sketch.

As an element of fluid flows along the pipe it has velocity v tangent to the pipe axis and a velocity $\dot{y}=\partial y/\partial t$ in a vertical direction. Assuming small displacements of the pipe, the x - component of the velocity is v, and the y-component is

$$\dot{y} + vy' = \frac{\partial y}{\partial t} + v\frac{\partial y}{\partial x} \tag{A-1}$$

The kinetic energy in a length ∂x of the pipe is that of the pipe plus that of the fluid, which is

$$dT = \{\frac{1}{2}(m - \rho)\dot{y}^2 + \frac{1}{2}\rho[v^2 + (\dot{y} + vy')^2]\}dx \qquad \text{(A-2)}$$

The first term on the right side represents the kinetic energy of the pipe shell, and the second represents the kinetic energy of the fluid. The strain energy in the length dx is, according to beam theory

$$dV = \frac{1}{2} EI(y'')^2 \qquad \text{(A-3)}$$

Hamilton's principle states that the variation of $\int_{t_1}^{t_2}(T-V)dt$ is equal to zero, which for this problem is

$$\delta\int_{t_1}^{t_2}\int_{x_1}^{x_2}\{\frac{1}{2}(m - \rho)\dot{y}^2 + \frac{1}{2}\rho[v^2 + (\dot{y} + vy')^2]$$

$$- \frac{1}{2} EI(y'')^2\} \, dxdt = 0 \qquad \text{(A-4)}$$

Performing the variation gives

$$\int_{t_1}^{t_2}\int_{x_1}^{x_2}\{(m - \rho)\dot{y}\delta\dot{y} + \rho(\dot{y} + vy)(\delta\dot{y} + v\delta y')$$

$$- EIy'\delta y''\}dxdt = 0 \qquad \text{(A-5)}$$

Since

$$\delta\dot{y} = \frac{\partial}{\partial t}(\delta y), \; \delta y' = \frac{\partial}{\partial z}(\delta y), \delta y'' = \frac{\partial^2}{\partial x^2}(\delta y)$$

each term may be integrated by parts so as to eliminate the various derivatives of δy.

When this is done, there is obtained

$$\int_{t_1}^{t_2}\int_{x_1}^{x_2}(m\ddot{y} + 2\rho v\dot{y}' + \rho v^2 y'' + EIy'''') \, \delta y dxdt = 0 \qquad \text{(A-6)}$$

where all the integrated terms have disappeared because of the boundary conditions.

Since δy is arbitrary, the only way for the integral to be zero is for the expression in parentheses in Eq. (A-6) to be identically zero. Equating this to zero gives the differential equation

$$\rho v^2\frac{\partial^2 y}{\partial x^2} + 2\rho v\frac{\partial^2 y}{\partial t\partial x} + m\frac{\partial^2 y}{\partial t^2} + EI\frac{\partial^4 y}{\partial x^4} = 0 \qquad \text{(A-7)}$$

This equation states that the beam is acted upon by three different inertia forces. The first term represents the inertia force associated with the change in direction of v, enforced by the curvature of the beam; that is, the fluid experiences an acceleration because it travels along a curved path. The second term is the inertia force associated with velocity v relative to the pipe, while the pipe itself has an angular velocity $(\partial^2 y)/(\partial t\partial x)$ at any point along its length. The third term represents the inertia force associated with the vertical acceleration of the pipe. As the pipe's action has a damping force, we add a damping force term to Eq. (A-7). This results in the differential equation (1).

VORTEX EXCITED OSCILLATIONS OF MARINE STRUCTURES WITH APPLICATION TO THE OTEC COLD WATER PIPE

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Abstract

Vortex-excited oscillations of marine structures result in reduced fatigue life, amplified hydrodynamic forces, and sometimes lead to structural damage and to premature failure. The cold water pipe of an OTEC plant is nominally a bluff, flexible cylinder with a large aspect ratio (L/D = length/diameter), and is likely to be susceptible to resonant vortex-excited oscillations. The purpose of this paper is to survey recent results pertaining to the vortex-excited oscillations of structures in general and to consider the application of these results to the OTEC cold water pipe. These results are compared to design methods which have been developed for the dynamic analysis of marine structures and cable systems.

Introduction

It is often found that unstreamlined, or bluff, structures display some form of oscillatory instability when exposed to wind or current flow past them. One common mechanism for generating these resonant, flow-excited oscillations is the organized and periodic shedding of vortices as the flow separates in an alternating fashion from opposite sides of a long, bluff body. The flow field exhibits a dominant periodicity and the body is acted upon by time-dependent pressure loads which result in steady and unsteady drag forces in-line with the flow as well as unsteady lift or side forces in the cross flow direction. If the structure is flexible and lightly damped, then resonant oscillations can be excited normal or parallel to the incident flow direction. For the more common cross flow oscillations, the body and wake have the same frequency near one of the natural frequencies of the structure, and near the Strouhal frequency at which pairs of vortices would be shed if the structure were stationary. This phenomenon is commonly termed "lock-on."

Vortex-excited oscillations of marine structures and cables result in reduced fatigue life, amplified hydrodynamic forces, and sometimes lead to structural damage and to premature failure. Flow-excited oscillations very often are a critical factor in the design of drilling risers and offshore platforms, since these complex structures usually have bluff cylindrical elements which are prone to vortex shedding in an incident current. The cold water pipe of an OTEC plant is typically a bluff, flexible cylinder with a large aspect ratio (L/D = length/diameter), and it is likely to be susceptible to current-induced vortex excitation. Thus an understanding of the nature of the fluid-structure interaction and the resultant flow-induced loads is an important consideration in the reliable design of the OTEC cold water pipe and its associated mooring and riser cable systems.

The purpose of this paper is to survey recent results pertaining to the vortex-excited oscillations of bluff structures in general and to consider applications to the OTEC cold water pipe. The physical scales of the experiments to be discussed range from small-scale laboratory studies to full-scale measurements on marine structures. These results are compared to design methods which have been developed for the dynamic analysis of offshore structures and cable systems.

Oscillations In-Line with the Flow

In the context of vortex-induced oscillations one usually thinks of vibrations lateral to the incident flow since the shedding of vortices produces periodic forces which are greater in the crossflow direction at a frequency near the Strouhal value for vortices of the same sign. The in-line component of the periodic force, which occurs at twice the Strouhal frequency, is typically an order of magnitude less than the cross flow component and, in wind, rarely excites the structure due to the small density of air. With the increased utilization of lightly-damped bluff structures in water, the likelihood of in-line oscillations has been increased because of the larger fluid density and because of the lower velocities at which the vibrations can be initiated.

Investigations of the in-line oscillations of bluff cylinders began as a result of the troublesome and damaging vortex-induced oscillations of pilings during the construction of an oil terminal in England during the late 1960's. A description of the problems encountered at the construction site has been given by Sainsbury and King (1) and a discussion of subsequent full-scale experiments to ascertain the causes of the vibrations has been reported by Wootton, Warner, Sainsbury and Cooper (2).

A cylindrical bluff body is excited into crossflow oscillations when the reduced velocity $V_r = V/f_n D \doteq 3.5$ to 5. This critical velocity is the reciprocal of the Strouhal number ($St = f_s D/V = 0.20$ to 0.27) evaluated at the condition $f_s \doteq f_n$, where f_n is a natural frequency of the body. In-line oscillations might be expected to occur at $V_r = 2.5$ since the periodicity in the drag fluctuations occurs at $2f_s$, or twice the Strouhal frequency. However, the in-line oscillations occur over a range $V_r = 1.5$ to 4.0 and in two distinct resonant instability regimes separated by $V_r \lesssim 2.5$. The response in each of the two regions depends strongly on the non-dimensional parameter

$$k_s = \frac{2m\delta}{\rho D^2} \qquad (1)$$

often called the "reduced damping", which is the product of the logarithmic decrement δ of the structural damping and the ratio of the mass of the structure to the displaced fluid mass. Typical laboratory-scale measurements of the dependence of the in-line displacement amplitude upon both V_r and k_s are plotted in Figure 1. An analogous dependence upon k_s for the cross flow displacement amplitude of oscillation in both air and water is well-known for a wide variety of bluff cylindrical structures. The in-line oscillations reach displacements of only about 0.4D peak-to-peak even at very low damping, whereas cross flow amplitudes of oscillation are known to reach 2 to 3D when the damping is small (see Figure 9). The first three successive instability regions for both in-line and cross flow motions are shown schematically in Figure 2.

Two distinct flow patterns have been observed for the two resonant regions of the in-line response. For $V_r < 2.5$ the flow is characterized by the shedding of two vortices of opposing sign from opposite sides of the cylinder during one motion cycle, while for $V_r > 2.5$ a single vortex is shed during each motion cycle and a street of alternately rotating vortices is formed downstream of the cylinder. Two such flow regimes had been postulated by Wootton et al (2) in trying to explain the appearance of two in-line instability regions during the full-scale field experiments at Immingham. The amplitude and frequency regions over which the in-line vibrations control the vortex shedding are plotted in Figure 3, taken from Griffin and Ramberg (5). The unsteady force coefficients that result from the in-line vibrations have been measured by King, and some typical results are plotted in Figure 4.

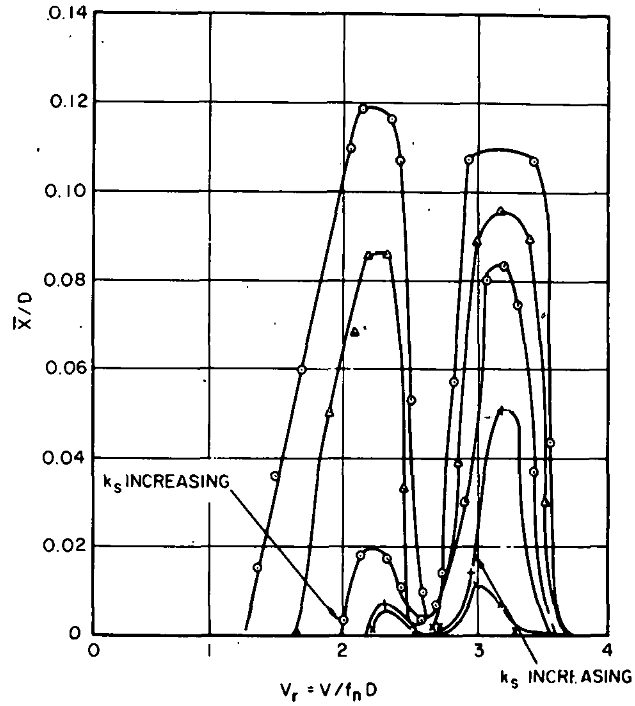

Figure 1. Vortex-excited displacement of a flexible cantilever (from equilibrium) in the in-line direction plotted against the reduced velocity $V_r = V/f_n D$. The reduced damping of the structure is denoted by k_s; from King (3).

with k_s (model) = 0.30 and k_s (full-scale) = 0.32. Hardwick and Wootton (6) also obtained good agreement between model experiments and the Immingham field test data.

It is important from an application standpoint to note that the vortex-excited instability in-line with the flow is initiated at a flow speed ($V/f_n D = 1.2$) that is about *one-fourth* of the flow speed ($V/f_n D = 3.5$ to 5) at which the cross flow instability is initiated.

Cross Flow Oscillations

Measurements of the frequencies, displacements and forces which result from cross flow oscillations have been obtained by many investigators from experiments both in air and in water. In this section, the most recent of these experiments are summarized. Typical measurements for three spring-mounted cylinders in water are presented in Figure 6. The results obtained are generally the same in both air and water; as the incident flow speed V, or V_r as in Figure 6, is increased the displacement amplitude first builds up to a maximum, after which it begins to decrease as the upper limit of the lock-on range between the vortex and vibration frequencies is approached. For the example in the figure the limits between which the vibration displacements are above their resonant threshold are given by reduced velocities V_r = 5.5 and 7.5, with the maximum amplitude occuring at $V_r \sim 6$. Dean, Milligan and Wootton (8), while studying experimentally the response of flexible model offshore structures in flowing water, obtained limits of V_r between 4 and 8, with the maximum crossflow response measured near $V_{r,MAX} = 6$. This is typical of measurements in water at *all* Reynolds numbers.

The inverse of the reduced velocity V_r or the Strouhal number corresponding to the peak vortex-excited displacement is plotted as the solid line in Figure 7, adapted from the results of Wootton (9). The large Reynolds numbers are typical of the range in which an OTEC cold water pipe might be expected to operate. For the results in the figure the shedding frequency f_s was locked onto the natural frequency f_n of the cylinder, and the dashed line in the figure represents the value of St corresponding to the initiation of lock-on. The cross flow motions of several full-scale cylinders are plotted in Figure 8, from the results of Sainsbury and King (1). The data were taken from visual observations but they

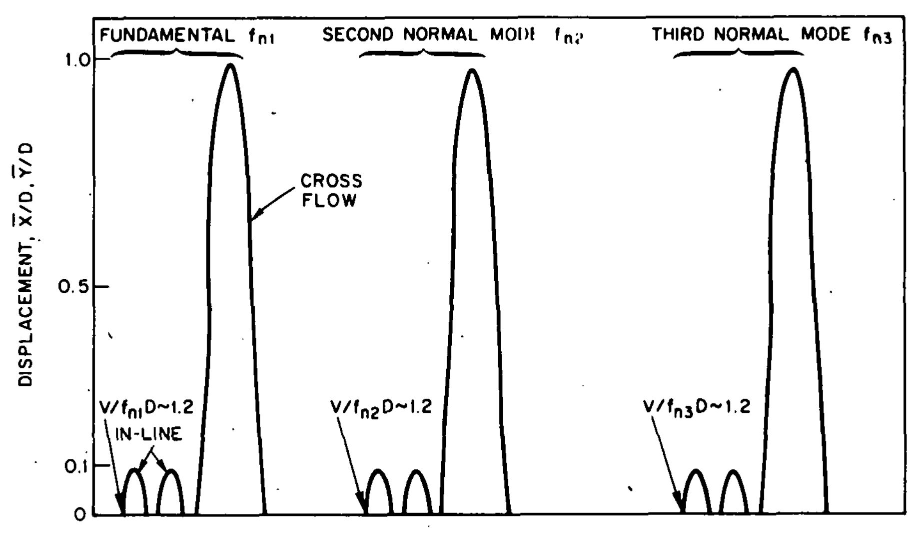

Figure 2. Composite stability diagram for the first three normal modes of a flexible cantilevered circular cylinder as a funtion of the incident water velocity. Both in-line (small displacement) and cross flow (large displacement) instabilities are shown; from King (4).

King obtained excellent agreement between measurements on a 1/27-scale model and a full-scale marine pile 457 mm (18 in) diameter which was characterized by a Reynolds number of $6(10^5)$. These experiments are compared in Figure 5. Both cylinders had very nearly the same combination of mass and damping parameters, clearly show the large displacement amplitudes that can be expected in water for long flexible cylinders. The Reynolds numbers for the experiments in Figure 8 were somewhat larger than 10^6, which approaches the range of OTEC applications. Cross flow oscillations greater than $\overline{Y}/D$=0.1 were initiated at V_r = 3.5 to 4, which agrees with the results in Figure 7. These large-amplitude oscillations were found to occur even though the current varied in magnitude from 2.4 m/s to 0.6 m/s and in horizontal direction by as much as 40° over the immersed length of the cylinder (18 to 20 m). As noted recently by Dean and Wootton (10), full-scale data on the cross flow response of cylinders are quite limited because the large-amplitude motions usually lead to failure within a few cycles of the oscillation.

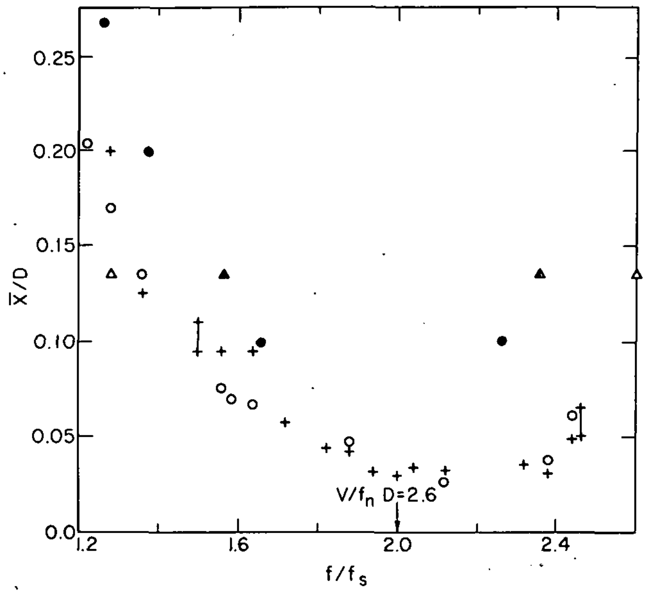

Figure 3. Displacement amplitude and frequency ranges over which the in-line vibrations of a cylinder control the vortex shedding (lock-on boundaries). The measurements at frequencies near twice the Strouhal frequency correspond to V/f_nD = 2 to 4.4; from Griffin and Ramberg (5).

In the previous section it was seen that the peak in-line displacement amplitude is a function primarily of a response or reduced damping parameter of the form

$$k_s = \frac{2m\delta}{\rho D^2}. \tag{1}$$

This formulation of the reduced damping can be written in the analogous form

$$\zeta_s/\mu = 2\pi\ St^2 k_s \tag{3}$$

when the damping is small and $\zeta_s = \delta/2\pi$. The importance of the reduced damping follows directly from resonant force and energy balances on the vibrating structure, as shown by Griffin (7) and Sarpkaya (12), for example. Moreover, the relation between Y_{MAX} and k_s holds equally well for flexible cylindrical structures with normal mode shapes given by $\psi_i(z)$, for the ith mode. If the cross flow displacement (from equilibrium) is written as

$$y_i = Y\ \psi_i\ (z)\ \sin\ \omega t$$

at each spanwise location z, then the peak displacement is scaled by the factor (7)

$$Y_{EFF,MAX} = Y I_i^{1/2} / |\psi_i(z)|_{MAX} \tag{4a}$$

$$I_i = \frac{\int_0^L \psi_i^4\ (z)\ dz}{\int_0^L \psi_i^2\ (z)\ dz} \tag{4b}$$

Experimental data for Y_{EFF} as a function of ξ_s/μ are plotted in Figure 9. These results encompass a wide range of single cylinders of various configurations at Reynolds numbers from 300 to 10^6. The various types of structures represented by the data points are given in the figure caption. As a typical example, King (4, 13) reported measurements on a flexible cantilever in the fundamental mode. Peak-to-peak displacements up to 2 to 4 diameters were measured for length/diameter ratios of 20 to 30. The latter are typical of the OTEC cold water pipe. All available experiments to date indicate that the limiting unsteady displacement for a flexi-

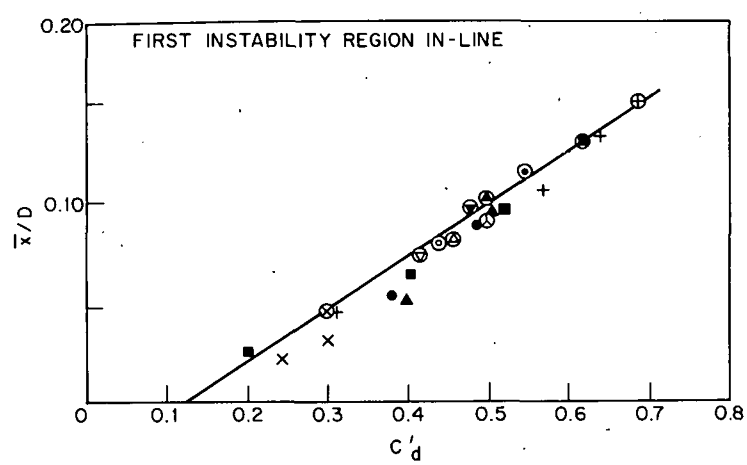

Figure 4. Flucutating drag coefficient C_d' (Force/½$\rho V^2 D$) plotted against in-line displacement amplitude $\bar{x}/D$ (from equilibrium), for the fundamental and second normal modes of a flexible cantilevered circular cylinder. The circles denote the maxima of C_d' for a given run; from King (4).

When Reynolds and Froude number effects are minimal, the peak vortex-excited displacement in the cross flow direction can be expressed from dimensional analysis as being dependent on three quantities, viz.

$$Y_{MAX} = f\left[\frac{\omega_s}{\omega_n},\ \zeta_s,\ \mu\right]. \tag{2}$$

Here ω_s/ω_n is the ratio of the Strouhal and structural frequencies $\omega_s = 2\pi St\ V/D$ and ω_n, respectively; and ζ_s is the *structural* damping ratio. The parameter μ is a mass ratio, defined by $\mu = \rho\ D^2/8\pi^2 St^2 m$, which also results from the normalization of the force coefficients in the governing equation of structural motion as shown by Griffin and Koopmann (11) and Sarpkaya (12).

ble, circular cylindrical structure is about $2Y_{EFF} \approx 2$ to 3. This result has been obtained both in air and in water, even though the *mass ratios of vibrating structures in the two media differ by two* orders of magnitude. For typical structures vibrating in water the mass ratio $\frac{2m}{\rho D^2}$ varies from slightly greater than 2 to about 10; in air the mass ratios corresponding to Figure 9 typically vary from $\frac{2m}{\rho D^2}$ = 50 to 1000. The maximum unsteady displacement to which a given system can be excited is a function of the reduced damping, the product of the mass ratio and the structural log decrement or damping ratio. Sarpkaya (14) has demonstrated that a similar type of dependence between $Y_{EFF,MAX}$ and k_s or ζ_s/μ is obtained for flexible structures vibrating in periodic flows such as ocean waves as well.

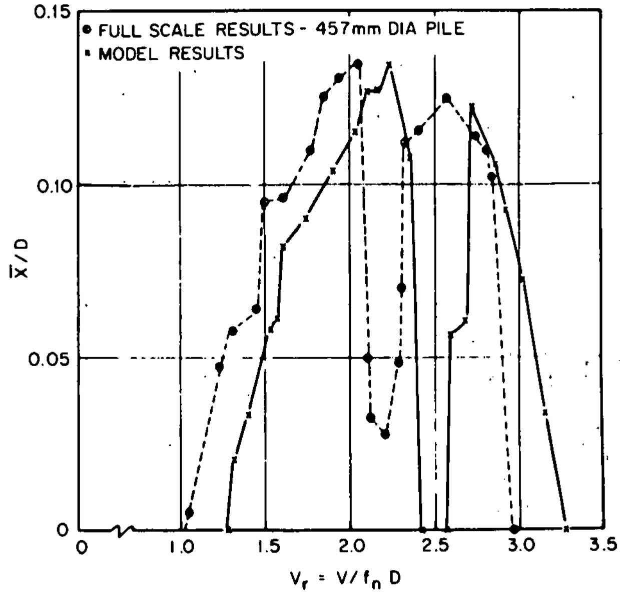

Figure 5. A comparison of the full scale and 1/27-scale model results for vortex-excited in-line oscillations of a flexible, cantilevered circular cylinder. The full-scale Reynolds number was about $6(10^5)$ and $k_s = 0.3$ for both cases; from King (3).

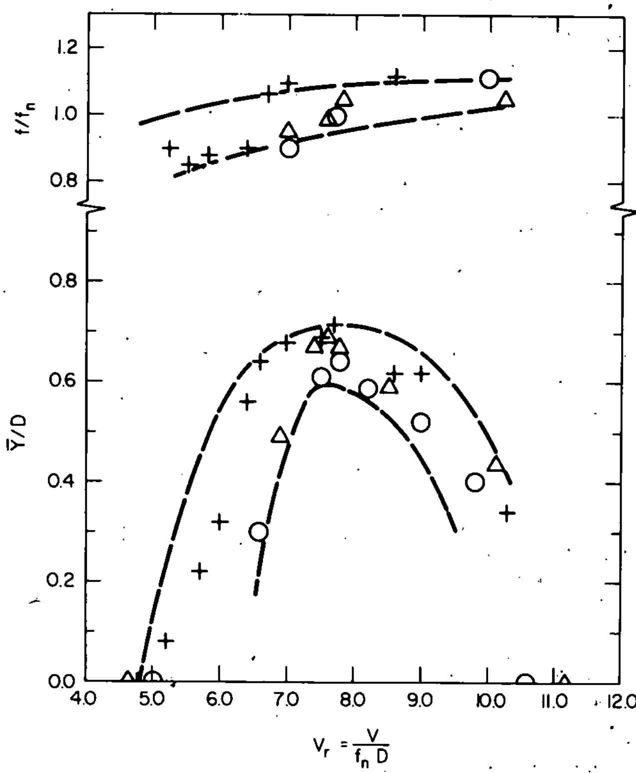

Figure 6. Cross flow-induced displacement amplitude $\bar{Y}/D$ and frequency f (scaled by the natural frequency f_n) of three circular cylinders in water, plotted against the reduced velocity V_r; from Griffin (7).

Legend for Data Points
Glass (1970)

	L/D	f_n (Hz)	$2m/\rho D^2$
0	8.5	2.4	7.6
+	11.8	1.8	7.6
Δ	15.4	1.3	7.6

When a cylindrical body resonantly vibrates due to vortex shedding, the periodic motion is accompanied by increased coherence of the vortex shedding lengthwise along the body and an amplification of the unsteady fluid forces. Though a number of measurements of the forces have been made, only recently has attention been paid to the mechanisms by which this fluid-structural interaction force is generated (15,16), and how results may be scaled to large Reynolds numbers (10,17). Further insight into the behavior of the unsteady fluid loading due to vortex shedding is important to the designer of offshore structures and cable systems from a practical standpoint, and from a basic standpoint it is important to further understand the complex nonlinear interactions between a vibrating bluff body and its wake.

The fluid forces induced on a resonantly vibrating, cylindrical structure by vortex shedding have been characterized recently (7, 11) and the various components of the total hydrodynamic force are:

- the exciting force component, by which energy is transferred to the structure;
- the reaction, or damping force, which is exactly out-of-phase with the structure's velocity;
- the "added mass" force, which is exactly out-of-phase with the structure's acceleration; and
- the fluid inertia force.

The various components can be deduced from the total hydrodynamic force as measured, say, by Sarpkaya (12) or the various components can be measured individually as shown by Griffin and Koopmann (11).

The excitation component of the total fluid dynamic lift is defined as

$$C_{LE} = C_L \sin \phi \tag{5}$$

and, as just mentioned, is important because it is this component of the fluid force system that transfers energy to the structure. A number of measurements of C_{LE} by various means are plotted in Figure 10, and Table 1 describes the various conditions under which the experimental results were obtained. Several important characteristics of the unsteady lift and pressure forces that accompany vortex-excited oscillations are clear from the results. First there is a *maximum* of the exciting force coefficient at a peak-to-peak displacement between 0.6 and 1 diameters for all the cases shown in the figure. Second, the maximum of the force coefficient is approximately C_{LE} = 0.5 to 0.6 for all but one case, the sole exception being the result at C_{LE} = 0.75. The coefficient C_{LE} then decreases toward zero in all cases and results in a limiting effective displacement of 2 to 3 diameters. This limit is clearly shown by the results at low reduced damping in Figure 9.

Not only are the unsteady forces amplified as shown in Figure 10, but the steady drag loads also are increased substantially as a result of vortex-excited oscillations. Sarpkaya (12) has measured steady drag coefficients as high as $C_D = 3.1$ for a cylinder vibrating in water at a displacement of $2Y_{MAX} = 1.7$. This represents an increase of nearly a factor of 3 from the drag on a stationary cylinder; i.e. $C_{DO} = 1.1$. Griffin, Skop and Koopmann (17) found that the drag coefficient was increased by as much as a factor of 1.8 from the stationary cylinder case ($C_{DO} = 0.9$) for their experiments plotted in Figures 9 and 10.

Other factors also influence the hydrodynamic forces that result from vortex-excited oscillations. Among these are surface roughness and shear gradients, which have been discussed for stationary structures by Hove, Shih and Albano (18). Sarpkaya (Private communication, 1979) has measured the hydrodynamic forces on sand-roughened vibrating cylinders and has compared his measurements to comparable experiments with smooth cylinders. Some typical results for the *total* hydrodynamic force coefficient

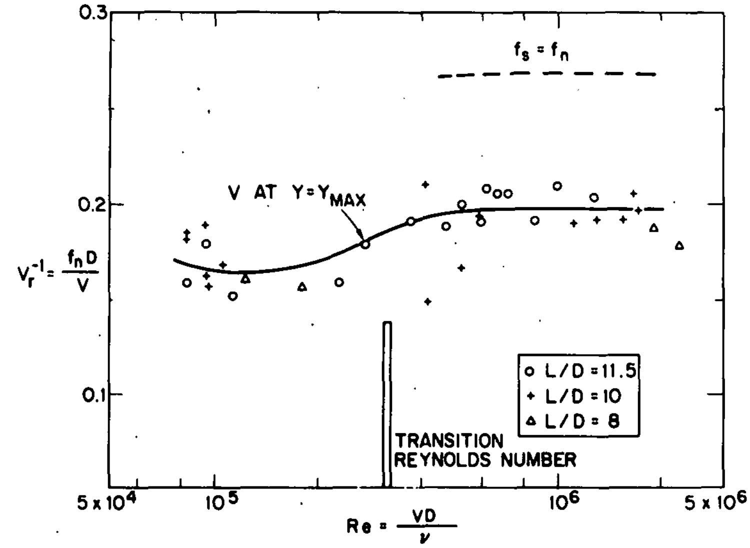

Figure 7. Inverse reduced velocity V_r^{-1} for maximum displacement amplitudes plotted against the Reynolds number for roughened cylinders. Maximum peak displacement ____; onset of vortex-excited oscillations ---; from Wootton (9).

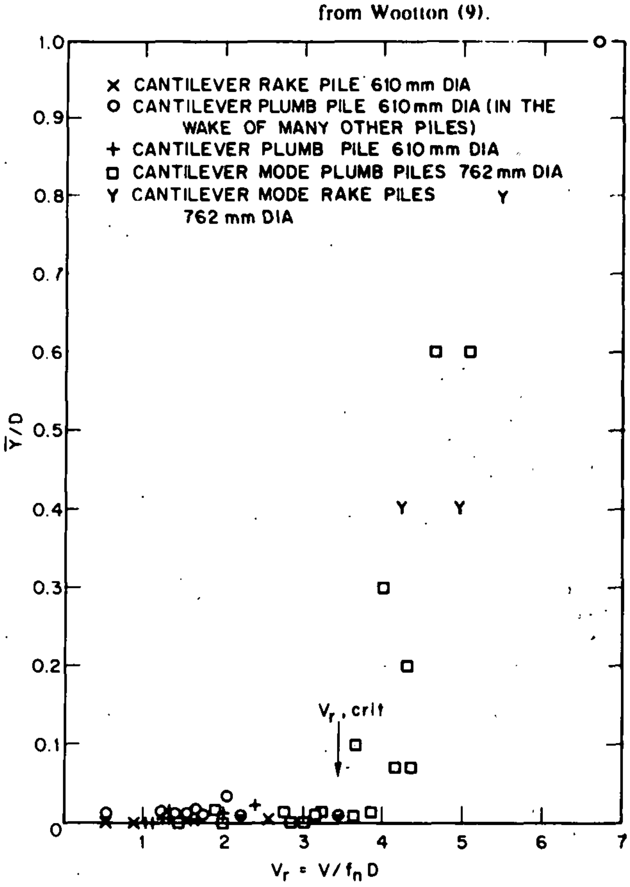

Figure 8. Observed cross flow displacement amplitude $\bar{Y}/D$ (from equilibrium) plotted against reduced velocity V_r for full-scale marine piles; from Sainsbury and King (1).

$C_{T,MAX}$ are plotted in Figure 11. Substantial increases are apparent in the total hydrodynamic force coefficient for the rough cylinder. Nakamura (19) has measured the steady drag forces and Strouhal frequencies on rough circular cylinders at supercritical Reynolds numbers, and has observed strong regular vortex shedding at $Re \approx 5(10^6)$ and above. In this Reynolds number range the vortex-excited cross flow displacement amplitude of a rough cylinder increased substantially from the corresponding smooth cylinder experiment. This finding would tend to confirm Sarpkaya's measurements of amplified hydrodynamic forces due to surface roughness.

The effects of velocity gradients are difficult to quantify on the basis of available evidence, especially for structures which are vibrating. However, the sparse information that is available suggests that a cylindrical structure will vibrate at large displacement amplitudes even in the presence of non-uniform flow effects if the reduced damping is sufficiently small and the critical reduced velocity is exceeded (see Figure 8).

OTEC Cold Water Pipe Applications

Two primary non-dimensional groups of relevant parameters must be considered in assessing the potential severity of vortex-excited oscillations.

These are: the REDUCED VELOCITY, $V_r = \dfrac{V}{f_n D}$

and the REDUCED DAMPING, $k_s = \dfrac{2m\delta}{\rho D^2}$ or $\zeta_s/\mu = 2\pi St^2 k_s$.

The latter is a combination of the mass ratio $m/\rho D^2$ and the structural damping ratio ζ_s or the log decrement δ. *Both* the reduced velocity and the reduced damping must be considered in a complete assessment.

The results in the preceding sections suggest criteria for ascertaining the critical incident flow velocities for the onset of vortex-excited motions. They are:

$$V_{crit} = (f_n D)\, V_{r,crit}$$

where $V_{r,crit} = 1.2$ for in-line oscillations and $V_{r,crit} = 3.5$ for cross flow oscillations at Reynolds numbers greater than about $5(10^5)$, in the range of OTEC applications, c.f. Figure 7.

An increase in the reduced damping will result in smaller amplitudes of oscillation and at large enough values of ζ_s/μ or k_s the motion can be suppressed completely. Reference to Figure 9 suggests that oscillations are effectively suppressed at $\zeta_s/\mu > 4$ or $k_s > 16$, but cylindrical structures in water fall well toward the left-hand portion of the figure. The measurements of in-line oscillations by King (4) have shown that vortex-excited motions in that direction are effectively suppressed for $k_s > 1$. The results obtained by Dean, Milligan and Wootton (8) indicate that the reduced damping can increase from $\zeta_s/\mu = 0.03$ to 0.5 (a factor of sixteen) and the peak-to-peak displacement amplitude is decreased only from 2 diameters to 1 diameter (a factor of two). At the small

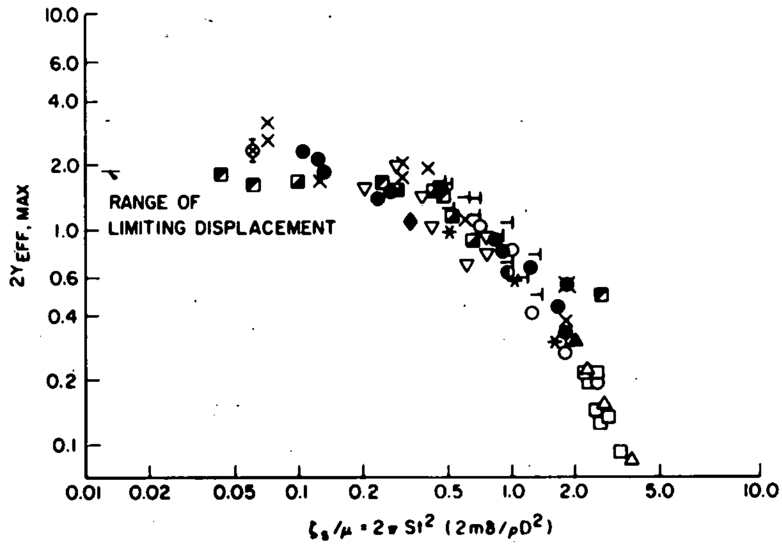

Figure 9. Maximum vortex-excited cross flow displacement amplitude $2Y_{EFF,MAX}$ of circular cylinders, scaled as in equation (4a), plotted against the reduced damping $\zeta_s/\mu = 2\pi St^2\ (2m\delta/\rho D^2)$.

Legend for Data Points

Type of cross-section and mounting; medium	Symbol
From Griffin (7):	
Spring-mounted rigid cylinder; air	× ◐ + ■ ○
Spring-mounted rigid cylinder; water	◆
Cantilevered flexible circular cylinder; air	◬
Cantilevered flexible circular cylinder; water	× ▽ ⊗
Pivoted rigid circular rod; air	⊥ ▲
Pivoted rigid circular rod; water	●
From Dean, Milligan and Wootton (8):	
Spring-mounted rigid cylinder; water	◩
Flexible circular cylinder ($L/D = 72$); water	◪

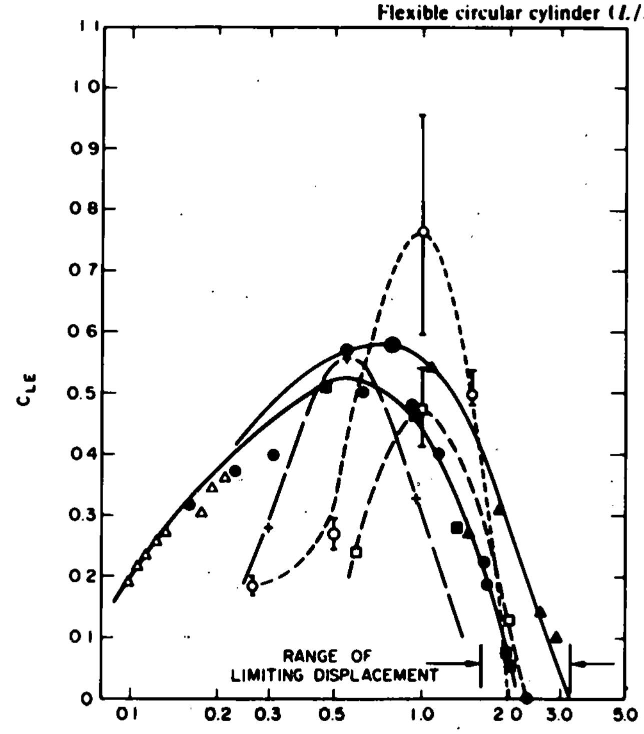

Figure 10. The excitation component C_{LE} of the lift force, equation (5), plotted against the vortex-excited cross flow displacement $2Y_{EFF,MAX}$ (peak-to-peak), as in equation (4a). The legend for the data points is given in Table 1.

mass ratios and structural damping ratios that are typical of light, flexible structures in water the hydrodynamic forces predominate and it is difficult to reduce or suppress the oscillations by means of mass and damping control.

Vortex excited oscillations sometimes can be reduced and suppressed by the installation of external devices that modify the flow field about the structure. Helical fins, porous shrouds and streamlined fairings have been used with some success (20,21), but in general it is preferable to design the structure itself to avoid the vortex-excited oscillations if possible.

It is useful to consider briefly the vortex shedding characteristics of a representative OTEC cold water pipe design. As an example the TRW fiberglass pipe is chosen for convenience. The design parameters of the pipe are given in reference 22 and are listed in Table 2. The natural periods of the first four bending modes of the fiberglass pipe are listed in Table 2. From these the in line and cross flow critical velocities can be determined and the results are listed in the table. A typical OTEC operating condition employed in recent OTEC studies is the offshore Brazil region "grazing" mode which yields a flow of 1 knot (0.51 m/s) incident over the pipe length neglecting relative currents. From the critical velocities in Table 2 it is clear that for these conditions the 9.2m (30 ft) diameter pipe would be in the critical velocity range for in-line oscillations in the first two modes and for cross flow oscillations in the fundamental mode.

The reduced damping of the structure must also be considered in order to assess the potential severity of vortex-excited oscillations. A thin-walled, water filled circular cylinder with a high slenderness ratio (in one OTEC case an L/D = 83) typically will

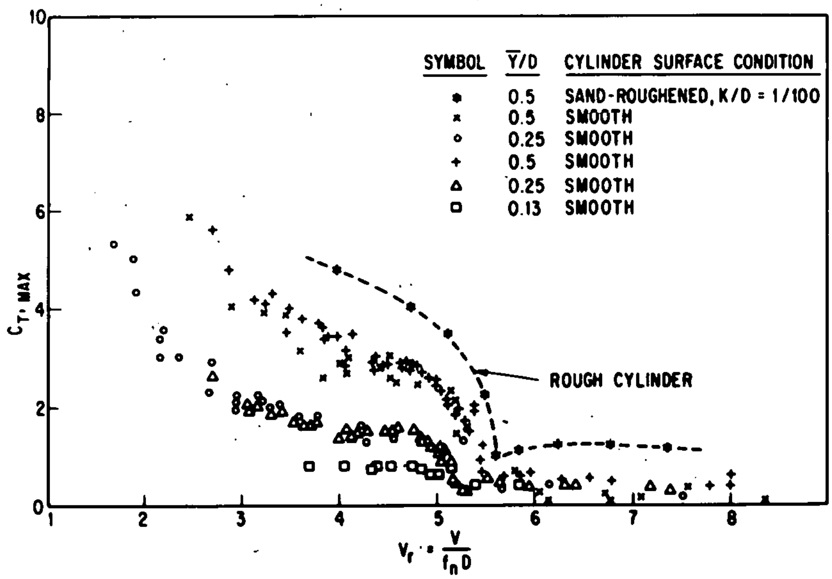

Figure 11. Total hydrodynamic force coefficient (Force/½$\rho V^2 D$), $C_{T,MAX}$, measured in uniform flow on smooth and rough circular cylinders vibrating in the cross flow direction, plotted against the reduced velocity V_r; from Sarpkaya (Private communication, 1979).

Table 1. The Exciting Forces on Vibrating Bluff Cylinders
Description of the Data in Figure 10

Symbol	Type of cylinder	Medium	Cylinder material	Investigator(s)
Δ	Flexible cantilever	Water	PVC PVC Aluminum Stainless steel	King (1977)
●	Pivoted rigid cylinder	Water & Air	Brass	Vickery and Watkins (1964)
+	Spring-mounted rigid cylinder	Air	Aluminum tubing	Griffin and Koopmann (1977)
□	Rigid cylinder, forced oscillations	Water	Stainless steel	Mercier (1973)
O	Rigid cylinder, forced oscillations	Water	Aluminum tubing	Sarpkaya (1978)
Δ	Flexible cantilever	Air	Aluminum	Hartlen, Baines and Currie (1968)

Table 2. The Effects of Vortex Shedding on an OTEC Cold Water Pipe
Pipe parameters from reference 22.
Cylindrical fiberglass pipe, 50mm thick;
Diameter = 9.2m; Length = 762m. Length/Diameter = 83

Mode of oscillation	Natural period (sec)	Critical velocity V_{crit} (m/s)	
		In-line	Cross flow
Fund.	121.0	0.092	0.265
2	43.2	0.253	0.741
3	21.4	0.512	1.49
4	12.6	0.854	2.54
$V_{crit} = V_{r,crit}\ (D/T_{NAT})$		$V_{R,crit}$ = 1.2 (in-line); 3.5 (cross flow).	

have a mass ratio in the range $\frac{m}{\rho D^2}$ = 1.2 to 2. When this is combined with typical values of the structural log decrement δ for such a configuration (21), δ = 0.1 to 0.2 for the fundamental and second modes, a range of reduced damping

$$k_s = \frac{2m\delta}{\rho D^2} = 0.2 \text{ to } 0.8 \text{ or } \zeta_s/\mu = 0.1 \text{ to } 0.35$$

is estimated. In terms of the cross flow displacement amplitudes in Figure 9, one might expect such a pipe to experience oscillations at a level between 1.2 and 2 to 3 diameters peak-to-peak. In this range of k_s in-line oscillations between 0.1 and 0.3 diameters can occur.

Step-by-step procedures for determining the deflections that result from vortex-excited oscillations have been developed by Skop, Griffin and Ramberg (23), by King (4) and by Mes (24). The steps to be taken are explained in detail in these references and should follow the sequence:

- Compute/measure vibration properties of the structure (natural frequencies or periods, normal modes, modal scaling factors, etc.)
- Compute Strouhal frequencies and test for critical velocities, V_{crit} (in-line and cross flow), based upon the incident flow environment.
- Test for reduced damping, k_s, based upon the structural damping and mass characteristics of the pipe.

If the structure is susceptible to vortex shedding, then

- Determine steady-state deflections based upon steady drag augmentation according to the methods of reference (23).
- Determine new stress distributions based upon the new steady-state deflection and the superimposed unsteady forces, displacements and accelerations due to vortex shedding.
- Assess the severity of the augmented stress levels relative to fatigue life, critical stresses, etc.

Concluding Remarks

Problems associated with the shedding of vortices often have been overlooked in the past in relation to the design of offshore structures, largely because reliable experimental data and design methods have been unavailable. However, the dynamic analysis of underwater structures has become increasingly important in order to predict stress distributions and fatigue life in the ocean environment. These factors are particularly relevant to the OTEC cold water pipe, which must be designed to survive and operate in the ocean environment over a long (20 to 30 years approx.) time period.

This paper has summarized the present state-of-knowledge relative to the vortex-excited oscillations of marine structures. Reliable experimental data now are in hand for the dynamic response of and flow-induced forces on model-scale structures, and based upon these experiments semi-empirical prediction models have been developed and tested. Many of these existing results and design methods are applicable in the Reynolds number range that characterizes the OTEC cold water pipe.

Prototype and full-scale test data are essential for the effects of vortex shedding and vortex-excited oscillations to be confidently included in the design of ocean structures such as steel jacketed platforms, marine risers and the OTEC cold water pipe.

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DISCUSSION

R. Lyon, Oak Ridge: In computing your natural frequencies, you include the mass of the water inside the pipe; I wonder if you also include the effect of the velocity of the water through the pipe — the mass flow rate through the pipe, which I think would affect the natural frequency quite drastically.

O. Griffin: None of the experiments were done in conditions where water was flowing through the pipe. That would obviously have to be taken into account in terms of OTEC applications.

T. McGuinness, NOAA: You have calculated the effect on natural frequency of the fluid flowing in the pipe haven't you, Arnold? Can you give us some idea of the magnitude of the effect?

A. Galef, TRW: Yes. The effect on natural frecies is simply the centrifugal force, and that is equal to the mass flow rate multiplied by the velocity and this effectively subtracts from the tension at all points. For the TRW pipes it essentially subtracts 60,000 pounds of tension. Depending on what the tension without the flow rate is, the effect is large or small. If we had a couple of million pounds of tension, it had negligible affect. If we had 100,000 pounds of tension, it would have a very big affect. The frictional force effect is assumed to be small and is neglected.

Question: Have you ever observed, other than in cable experience, the amplification factors for higher-mode isolations?

D. Hove, SAI: The only experimental evidence I have ever seen on vortex lock-on in application is for the fundamental mode. In fact, mostly in cantilever beams. Some small-scale experiments in water have gone up to second mode. The cylinder was oscillating at large enough amplitudes that they were almost equal to the fundamental modes. You would get the same type of force amplification. We attempted to do such an experiment recently under NOAA's sponsorship, and it is very difficult to reach a second mode with any kind of thin wall pipe.

Question: In your work for NOAA, do you intend to address probably the most uncertain question, that is, shear-flow lock-on regions?

D. Hove: As far as anybody can; there isn't much evidence around.

J. Vadus, NOAA: With regard to the problem of vortex shedding in the presence of the shear flow or velocity profile like we have in typical OTEC sites, the paper at this conference by Dr. Rooney, at Virginia Polytechnic Institute addresses the problem by wind-tunnel measurements.

J. Coyle, Navy: You have been discussing the measurement of circumstances in which these various oscillations occur. Has there been any thought given to designing a pipe that would be somehow tuneable so that you could dynamically change it, perhaps inflating or deflating parts of it, to detune whatever resonance may occur due to a change in ocean current?

T. McGuinness: Clearly, the practical approach would be to design around it like a friend of our's, Ralph Blevins from APL, contends, and I agree it's a design problem. However, before you can deal with a design problem, you have to have some understanding of the phenomenon and the means by which you may eliminate it. In terms of tuning the pipe, let me relate a bit of information that was delivered to me from the oil industry. They generally do not worry too much about shedding because generally they have bottom-mounted structures, tension-leg moored where you can put a significant amount of tension in a vertical riser system. Therefore, the frequencies where they would lock-on would be a very high velocity, and, in fact, failures have been noted when velocities go above those of the design and shedding occurs at frequencies that were not predicted. Your point is well made, and I think that is the approach that will win out and is in the direction of this program — to design around the problem.

G. Wachtell, Franklin Res. Center: In line with the previous question, I wonder whether you might wrap a projection helically around the surface to prevent large vortices from being shed periodically.

T. McGuinness: Our Office of Engineering is involved in issuing a solicitation, within about two months, to test cold water pipes in various configurations. One phase of the program will be to build and test a pipe about 1 ft in diameter by 100 ft long in a situation where it would shed and then to test various devices such as this helically wound vortex-shedding-suppresser or others to assess their effectiveness in currents, in shear flows, and with and without biofouling. So, we are very much going in that direction and we appreciate your suggestions.

A. Galef: I would like to suggest that some of the oil industry experience with what they think is vortex shedding may be misleading. The oil industry riser will invariably be of noncircular cross-section, because you have the kill and choke lines running down and you have some kind of conduit for them; thus you have the potential for galloping, which is a lift in the direction opposite to the place where you expect it to go. It's effectively a stall flutter situation. I've run some sections

DISCUSSION

in the Aerojet water tunnel some years ago, and I was amazed that such small deviations from circular sections could induce galloping. I believe a lot of the oil industry experiences that they think are vortices are really that instead.

J. Vadus: Perhaps someone will make a comment. We discussed briefly the problem of vortex shedding when a cylinder becomes non-symmetrical, noncircular. Would anyone like to get up and make a comment on that?

Comment: Galloping is a problem for noncircular sections. Another problem is what people call ovalling or breathing-like oscillations, and there has been some recent evidence to suggest also that they are not vortex-induced either. They come from some as yet unknown origin. So that would be another one of your noncircular sections, because the pipe is typically deformed when it is ovalling or breathing. You have to be careful in terms of assessing those, which is essentially what the OTEC people are doing in terms of looking at the shell model for the pipe.

CURRENT-WAVE COUPLING AND HYDRODYNAMICS

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Abstract

The coupling of a horizontal current U(y) having a vertical gradient with surface waves is studied by two approaches. The first uses the nonlinear irrotational theory of waves of permanent form and adjusts the resulting flow field to account for the current. The second approach is a linear perturbation with the current regarded as the basic flow. This approach can be used to analyze waves that are colinear with the current as well as those that propagate at an angle to the current.

It is found that the two approaches give similar results for small amplitude waves, with the linear approach overestimating the magnitude of the velocities at larger amplitudes.

The first approach was used to calculate the velocity and acceleration fields for the maximum wave expected at four potential OTEC locations. The results are presented graphically in the form of curves of constant horizontal velocity and acceleration. These curves can be used to find the forces on vertical members provided that diffraction effects can be neglected.

Introduction

The expected forces on the components of the OTEC platform and cold water pipe are important in determining the proper structural design of these elements. The hydrodynamic forces arise from the fluid velocity adjacent to the elements. This velocity field is determined by the wave and current environment in the vicinity of the platform.

Bretschneider[1] has calculated the hurricane design wind, wave, and current conditions for four potential OTEC sites: Keahole Point, Hawaii; Punta Tuna, Puerto Rico; New Orleans, Louisiana; and the West Coast of Florida. A summary of these results appears in Appendix A.

This paper combines the current and wave velocities obtained from Bretschneider's data to arrive at the velocity field due to the combined effect of currents and waves.

Summary of the Method

The given data for a particular location are: the current U(y) as a function of depth, the maximum wave height H and the significant period T_s. It will be assumed here that the maximum forces on the structure can be obtained by calculating the forces associated with a regular plane wave having a height H and period T_s. It is further assumed that the height and period given are those that obtain at the site and do not have to be modified as would be the case if the wave characteristics referred to a region where the currents are absent. (See Phillips[2] p. 56, for the corrections which would apply in such a case).

The wave is not assumed to be linear, but an ad-hoc procedure is used to combine the current and wave velocity fields. The procedure is based on the effect of superposing a uniform current on an irrotational plane wave. The procedure will be justified by comparing the results with those obtained using a linear rotational theory, which is discussed later.

In the absence of a current, a plane wave in deep water will propagate with a speed c_o which is a function of the wave number k_o and the wave steepness $a = k_oH/2$. Associated with this wave will be a velocity field described by its components $u(x,y,t)$, $v(x,y,t)$. If this wave is observed in a frame of reference moving to the left with a speed U, the result will be the same as superposing a uniform current U in the positive x direction. The period of the wave will be unchanged but the speed of propagation relative to the moving frame is $c = U + c_o$ and the wavenumber is thus $2\pi/(U + c_o)T$, or $k = k_oc_o/(U + c_o)$. If the current U, the period T_s and the waveheight H are given, the speed of propagation is not immediately known, and it is necessary to solve the above equations for k_o. Once this is determined, the flow in the presence of a current is given by $u' = u(x-Ut,y,t) + U$, $v' = v(x-Ut,y,t)$.

For the case of a uniform current, this procedure is exact. It also is correct if the wavelength is short compared to the depth required for there to be a significant change in the current, since the fact that the current changes at lower depths, where the wave is not felt, would not affect the propagation of the wave.

The procedure adopted here is as follows: First, the surface value of the current, U_s, is used in determining the wave speed as if a uniform current U_s was involved. The second step is to combine the actual current U(y) with the velocity field obtained in the first step. We proceed by analogy with the case of a uniform current. Suppose that a current -c is superposed on a wave of wavenumber k. The net effect is to stop the wave completely so that the flow is then steady. The flow can then be represented by a streamfunction Ψ with streamlines $y(\Psi,x)$. These streamlines are wavy lines oscillating about a mean value $\overline{y}(\Psi)$. The assumption made here is that the horizontal fluid

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velocity $u(x,y,t)$ at $y = y(\Psi,x)$ is augmented by the current at the mean location of the streamline, that is, by $U(y)$ evaluated at $y = \bar{y}(\Psi)$. The substitution is thus

$$u' = u(x - U_s t,\ y,\ t) + U(\bar{y}). \quad (1)$$

In this manner, velocity and acceleration fields were obtained for the four OTEC locations.

Further Details of the Method

The theory of waves of permanent form was given by Stokes[3,4], who outlined a method whereby the variables of interest could be expanded in a power series involving a small parameter. In recent years, Stokes' procedure has been formulated in a systematic way so that the perturbation expansion can be carried out by computer. Both Schwartz[5] and Cokelet[6] have presented such computer expansions, using Stokes' second formulation of the problem in which the spatial coordinates are taken to be the independent variables, and are regarded as functions of the velocity potential and streamfunction. Schwartz carried out his expansion using the steepness as the expansion parameter. Cokelet used a different expansion parameter, which he found to have better convergence properties. Both authors accelerated the convergence of the power series by the use of Pade approximants,[7] that is, rational fraction approximations to the power series.

In the work reported here, Cokelet's approach was used to generate the coefficients in the various expansions and sums were extrapolated by converting the series into continued fractions, a method closely related to that of Pade approximants.

For a wave of permanent form in deep water, as seen in a frame of reference in which the wave surface is stationary, the flow is specified by the equations

$$X(\Phi,\Psi,\varepsilon) = -\frac{\Phi}{c} - \sum_{n=1}^{\infty} \frac{a_n(\varepsilon)}{n} e^{-n\Psi/c} \sin\frac{n\Phi}{c} \quad (2)$$

$$Y(\Phi,\Psi,\varepsilon) = -\frac{\Psi}{c} + \sum_{n=1}^{\infty} \frac{a_n(\varepsilon)}{n} e^{-n\Psi/c} \cos\frac{n\Phi}{c} \quad (3)$$

where X and Y are the coordinates of a point and Φ and Ψ are the velocity potential and streamfunction at that point, with the free surface at $\Psi = 0$. The parameter ε is a parameter which measures the steepness of the wave, and in Cokelet's expansion it is defined by

$$\varepsilon^2 = 1 - \frac{q^2_{crest}\, q^2_{trough}}{c^4} \quad (4)$$

where q is the velocity of the fluid at the point in question, measured in the frame of reference in which the wave surface is stationary. This parameter has the property that $\varepsilon \to 0$ for a linear wave and $\varepsilon \to 1$ for a wave of maximum steepness.

Cokelet has given a method for expanding the Fourier coefficients in powers of ε, and in this report an expansion to order ε^{50} was used, which combined with the continued fraction approach can be regarded as giving exact results. The wave speed c and the steepness $a = kH/2$ are also found as functions of ε.

The steps required to solve for the velocity components are thus the following:

1) Assuming the flow to be that of a wave of a given height H and frequency ω on a uniform current $U_s = U(0)$, the speed of the wave referred to a rest frame is

$$c^* = U_s + c(\varepsilon,\ k) = \omega/k \quad (5)$$

and the steepness is

$$a(\varepsilon) = kH/2 \quad (6)$$

This gives a pair of equations for the unknowns ε and k.

2) Once ε is known, the Fourier coefficients $a_n(\varepsilon)$ can be found and the velocity components can be calculated as functions of Φ and Ψ.

3) To find the velocity components at a specific location $(X_0,\ Y_0)$, the pair of equations

$$X(\Phi,\Psi,\varepsilon) = X_0 \quad (7)$$

$$Y(\Phi,\Psi,\varepsilon) = Y_0 \quad (8)$$

must be solved for Φ and Ψ. This can be done by means of Newton's method.

4) The current velocity is then added to the wave velocity in the manner described in the preceding section.

In this manner, results were obtained for the four OTEC locations and are presented graphically in figures 1-16. These show the lines of constant u and constant $\partial u/\partial t$ (horizontal velocity and acceleration) for the four locations with currents acting in the same direction as the waves and in opposition.

Discussion of the Results

Taking the Keahole Point results as an example, it is seen that when the current is in opposition to the wave, the waves become steeper. This is reflected in the shorter wavelength, which is 218.8 meters when the currents oppose the waves and 243.5 when they run in the same direction. To assess the effect of wave amplitude on the results, these runs were repeated with a waveheight of 1.51 meters instead of the design height of 15.1 meters. For this lower height, the wavelengths were found to be 208.6 meters and 235.25 meters, respectively. Since the speed of the wave relative to the current increases with steepness, and since the wavelength increases linearly with this speed, the steeper wave has the larger change.

The value of the parameter ε is also a measure of the nonlinear effects. For the Keahole Point data, ε is 0.515 for currents in opposition to the waves and 0.465 for currents in the same direction. The Keahole Point waves are the least steep and those for the West Coast of Florida are the steepest.

Alternative Approach to the Problem

An alternative formulation of the problem that does not superpose the current on an irrotational wave is also possible. In this approach the flow is regarded as rotational, since the current has shear. This type of coupling was first explored by Rayleigh[8] and Taylor.[9]

In this formulation, the waves are regarded as a perturbation on the uniform flow, and the linear equations at least are simple. Perturbing about a main stream U(y), the equations for plane waves travelling in the x direction are:

$$u_x + v_y = 0 \tag{9}$$

$$u_t + Uu_x + vU_y = -\frac{p_x}{\rho} \tag{10}$$

and

$$v_t + Uv_x = -g - \frac{p_y}{\rho} \tag{11}$$

If the streamfunction ψ is introduced for the components u and v of the fluid velocity relative to the stream, the equations reduce to

$$\nabla^2\psi_t + U\nabla^2\psi_x - U''\psi_x = 0 \tag{12}$$

and if ψ is a function of (x - ct),

$$(U - c)\ \nabla^2\psi - U''\psi = 0. \tag{13}$$

Simple formulations are also possible for waves travelling in a direction at an angle to the current, although the simplification that results from expressing the velocity components in terms of the streamfunction is not possible in this more general case.

The above equation was first derived by Rayleigh, and Taylor[10] has discussed two piecewise linear currents, while Abdullah[11] has investigated the interaction of waves with an exponentially decaying current. In the first case discussed by Taylor the current is uniform in a layer and zero outside it. The U" term is thus zero within each layer and the flow is irrotational except for the presence of a vortex sheet at the interface. In Taylor's second model the current decreases linearly to zero so the flow is rotational, as it is in the case considered by Abdullah.

An extension of Taylor's second model was devised to approximate an arbitrary current: the fluid was divided into an arbitrary number of layers, with the current varying linearly in each layer, and being continuous from layer to layer. The equations of motion are solved in each layer, and the pressure and normal velocity are made continuous at each interface.

In principle this calculation can also be done for the non-linear case, but in practice the perturbation procedure is very complicated beyond the second order.

The linear equations of this multi-layer model were solved numerically to compare with the results obtained by the method discussed in the preceding sections.

Results and Discussion

The results obtained by using the two approaches are surprisingly close considering that the first regards the current as a small perturbation on the wave and the second the wave as a small perturbation on the current. For the Keahole Point data, the second method gives wavelengths of 205.6 and 233.0 meters for the two cases. These values are very close to those found by the first approach when the waveheight was reduced to one tenth of its actual value. The velocities computed using the second model are somewhat larger than those obtained using the first, but when the values are compared for the reduced waveheight, the two approaches give practically identical results. Thus, it would appear that the first approach is validated by its agreement with the second method in the regime where the second method would be expected to be applicable.

Acknowledgment

This work was supported by Department of Energy Contract No. EY-76-S-03-0235, "Current-Wave Coupling Project."

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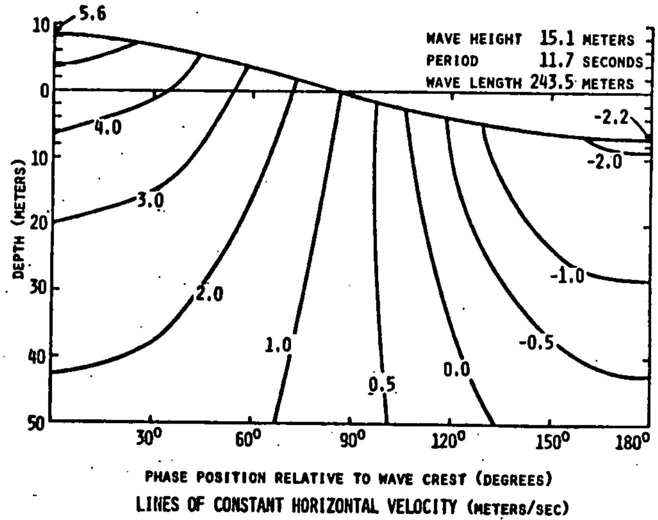


Figure 1

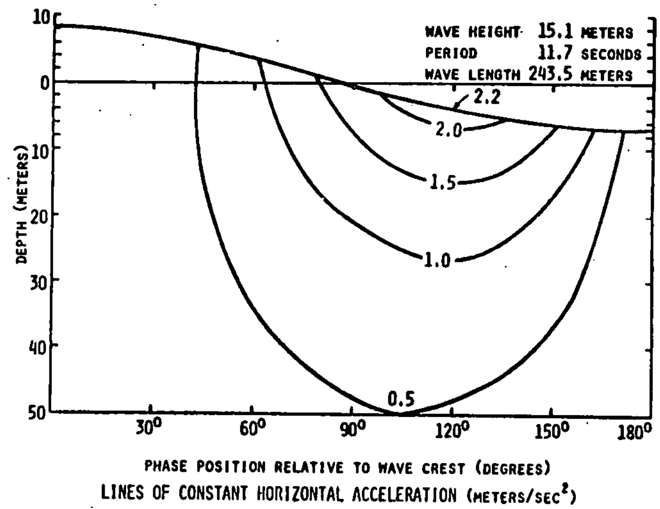


Figure 2

KEAHOLE POINT, HAWAII OTEC SITE
Currents in direction of wave propagation

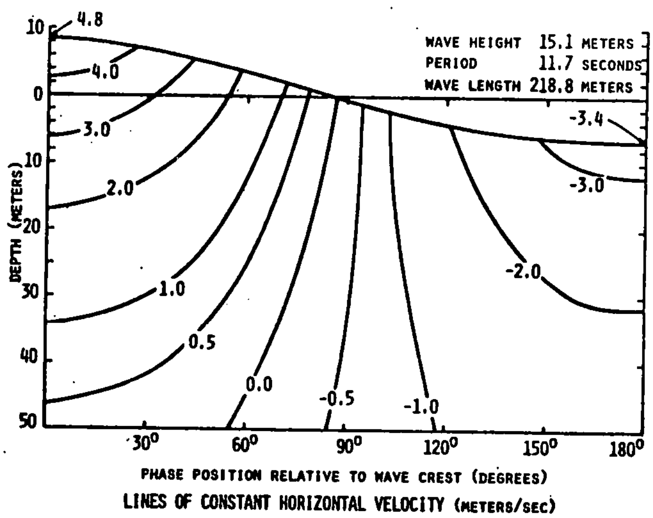


Figure 3

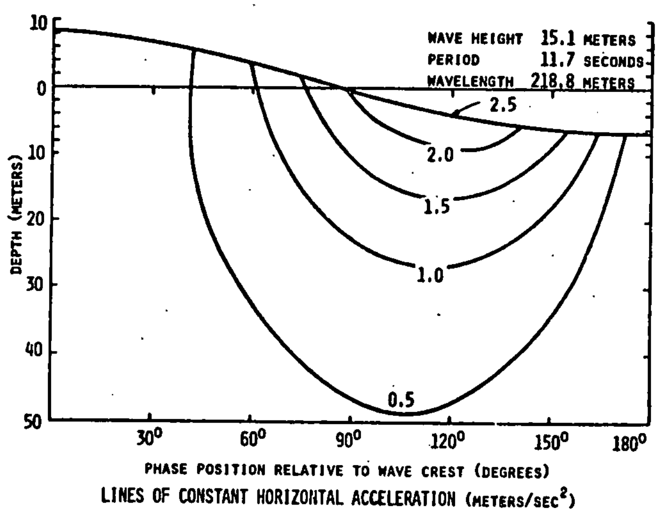


Figure 4

KEAHOLE POINT, HAWAII OTEC SITE
Currents in direction opposite to wave propagation

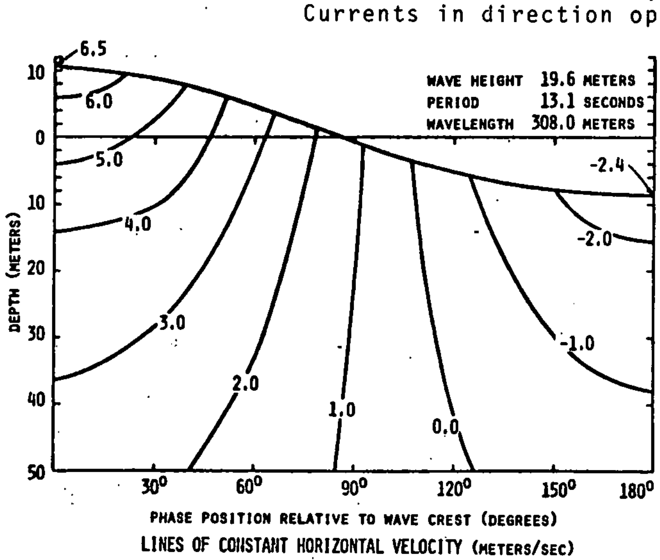


Figure 5

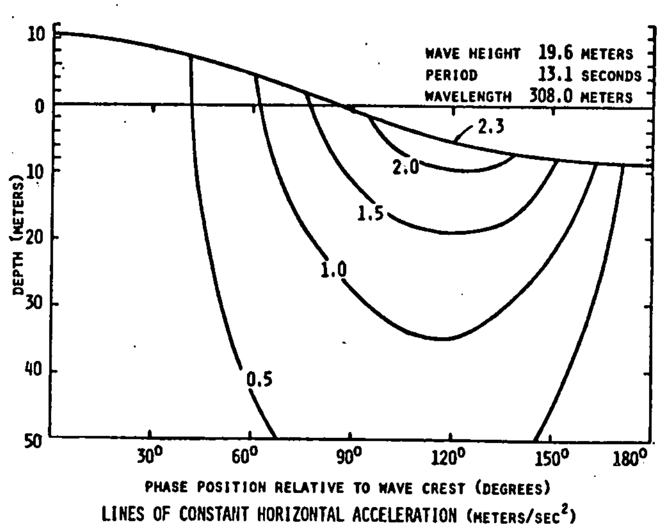
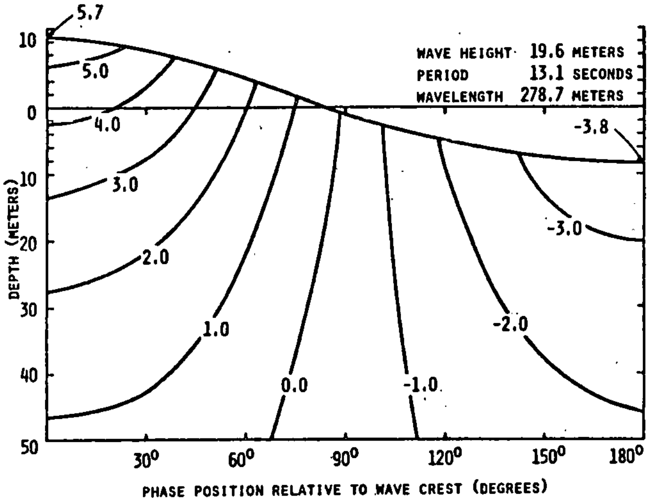


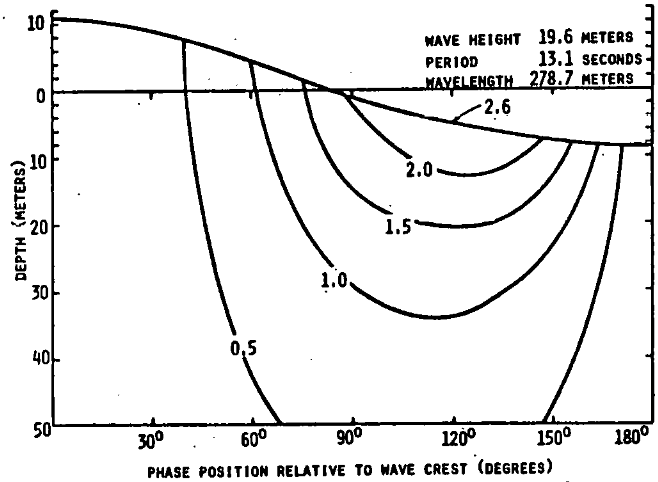
Figure 6

PUNTA TUNA, PUERTO RICO OTEC SITE
Currents in direction of wave propagation



LINES OF CONSTANT HORIZONTAL VELOCITY (METERS/SEC)

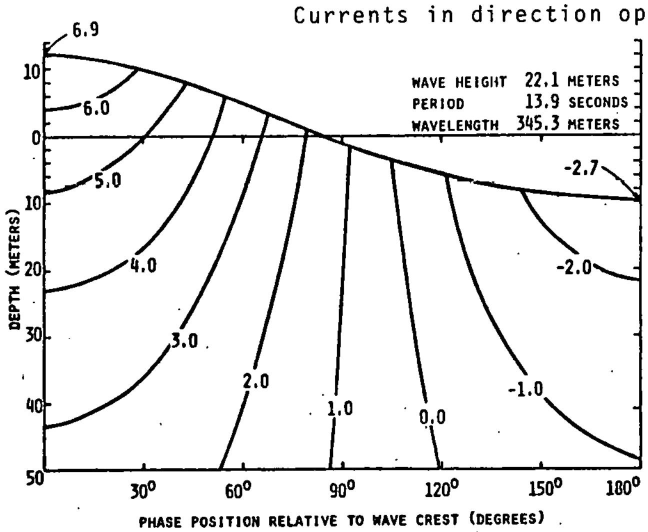
Figure 7



LINES OF CONSTANT HORIZONTAL ACCELERATION (METERS/SEC^2)

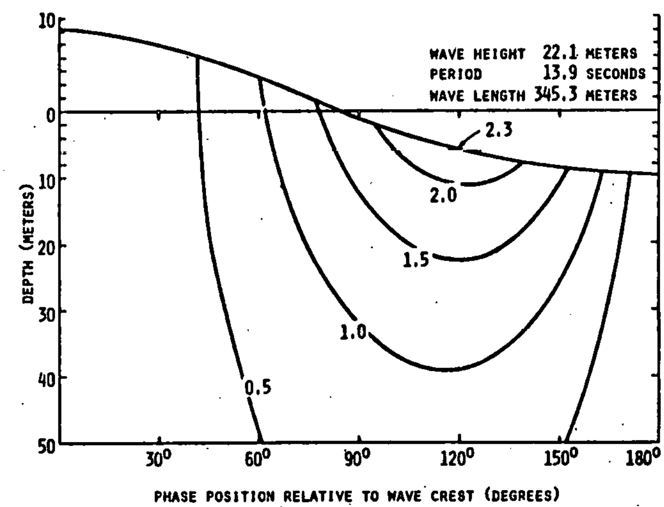
Figure 8

PUNTA TUNA, PUERTO RICO OTEC SITE
Currents in direction opposite to wave propagation



LINES OF CONSTANT HORIZONTAL VELOCITY (METERS/SEC)

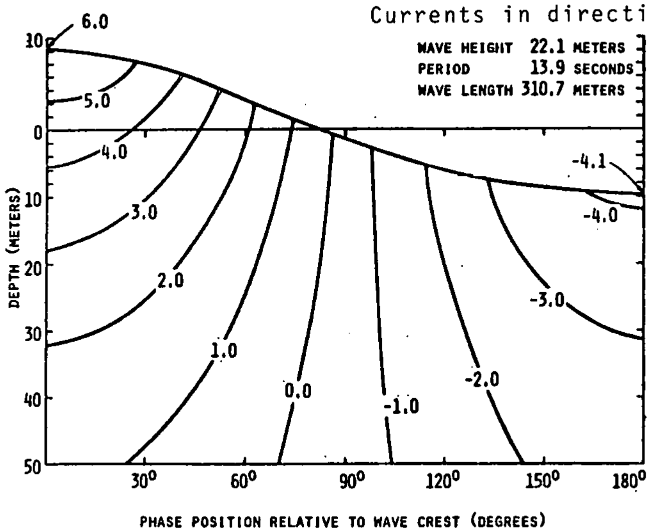
Figure 9



LINES OF CONSTANT HORIZONTAL ACCELERATION (METERS/SEC^2)

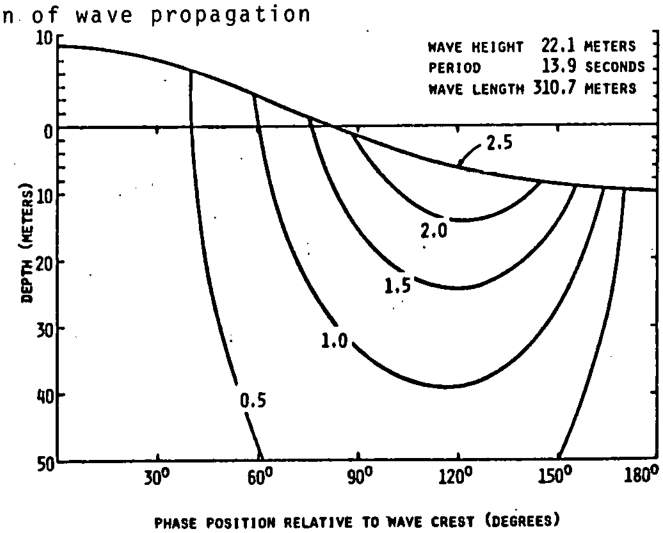
Figure 10

NEW ORLEANS, LOUISIANA OTEC SITE
Currents in direction of wave propagation



LINES OF CONSTANT HORIZONTAL VELOCITY (METERS/SEC)

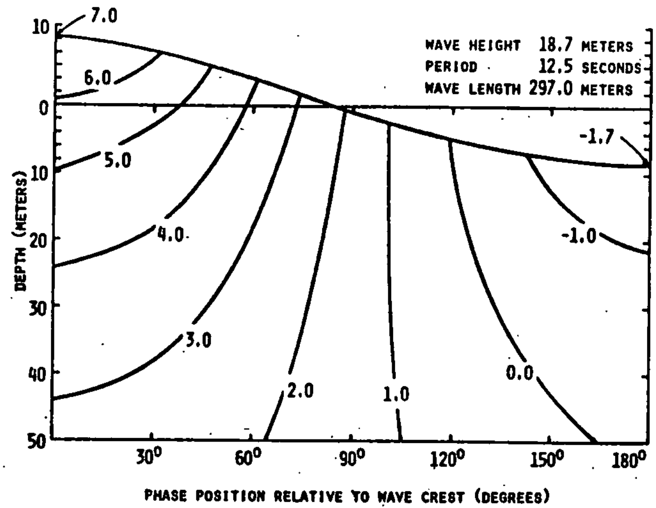
Figure 11



LINES OF CONSTANT HORIZONTAL ACCELERATION (METERS/SEC^2)

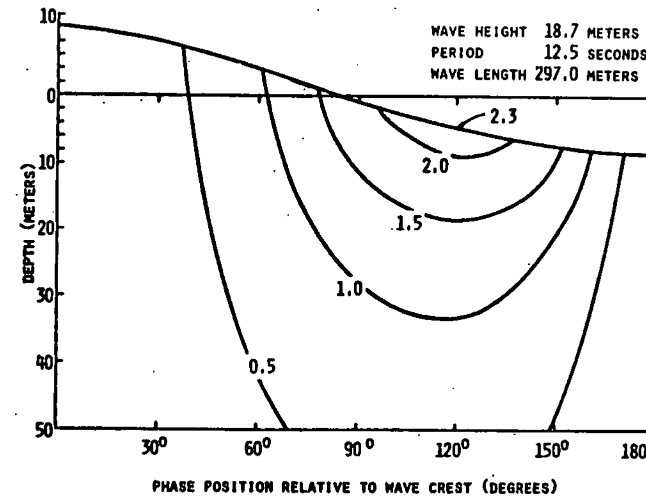
Figure 12

NEW ORLEANS, LOUISIANA OTEC SITE
Currents in direction opposite to wave propagation



LINES OF CONSTANT HORIZONTAL VELOCITY (METERS/SEC)

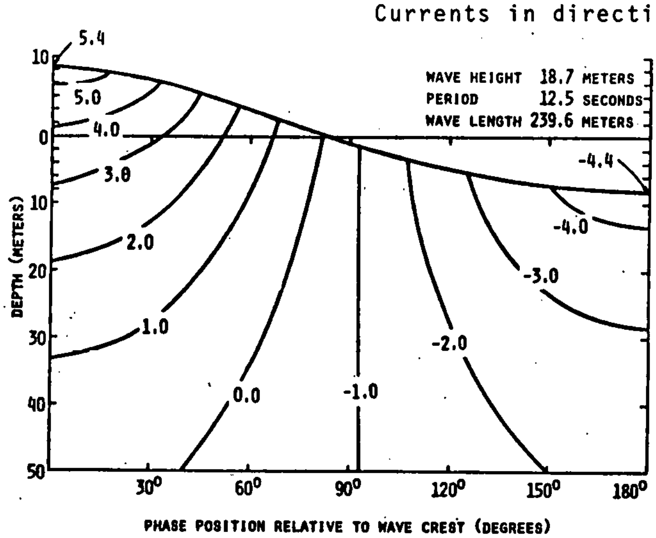
Figure 13



LINES OF CONSTANT HORIZONTAL ACCELERATION (METERS/SEC^2)

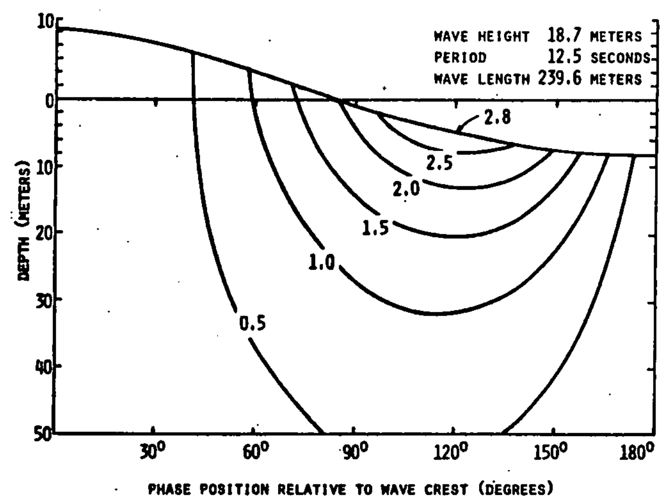
Figure 14

WEST COAST OF FLORIDA OTEC SITE
Currents in direction of wave propagation



LINES OF CONSTANT HORIZONTAL VELOCITY (METERS/SEC)

Figure 15



LINES OF CONSTANT HORIZONTAL ACCELERATION (METERS/SEC^2)

Figure 16

WEST COAST OF FLORIDA OTEC SITE
Currents in direction opposite to wave propagation

APPENDIX A

100-YEAR DESIGN HURRICANE WAVE CRITERIA FOR POTENTIAL OTEC SITES

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Abstract

The 100-year design wind and wave criteria are given (see Table A-I) and are used in the main section of the paper. A non-dimensional hurricane wave model was developed by Bretschneider[1,2] and applied to the U.S. standard project hurricanes of Graham and Nunn[3]. We have also used the hurricane wind equations of Myers[4]. We now call this method the two-direction significant wave forecasting model, after Bretschneider[5]. The standard project hurricanes have been used to determine the 100-year hurricane for design criteria for potential OTEC sites in the Gulf of Mexico. Similarly, model hurricanes have been used for the Keahole Point, Hawaii, and Punta Tuna, Puerto Rico, OTEC sites. This two-direction wave forecasting model has been verified by Bretschneider and Tamaye[6] as corrected for scale of map by Bretschneider[7], for the maximum values of the significant wave heights measured at six ODGP stations for Hurricane Camille as reported by Cardone, Pierson and Ward[8] (see Figs. A-1, A-2 and A-3).

Constant Wind Speed and Constant Wind Direction

The basic equations for the one-direction significant wave forecasting model, reproduced from Bretschneider (1970,1979), for constant wind speed and direction are as follows:

$$\frac{gH}{U^2} = 0.283 \tanh\left[.0125\left(\frac{gF}{U^2}\right)^{0.42}\right] \tag{A-1}$$

$$\frac{C_o}{U} = \frac{gT}{2\pi U} = 1.2 \tanh\left[0.077\left(\frac{gF}{U^2}\right)^{0.25}\right] \tag{A-2}$$

and

$$t_{min} = 2\int_{0}^{F_{min}} \frac{1}{C_o}\,dx \tag{A-3}$$

where H, T, U, F and t are significant wave height, significant wave period, ten minute average sustained wind speed at 10 meter anemometer level, fetch length and wind duration.

The U.S. Army Corps of Engineers Coastal Engineering Research Center[9], Shore Protection Manual, determined a solution of eq. A-3 as follows:

$$\frac{gt}{U} = K \exp\left\{\left[A\left(\ln\left(\frac{gF}{U^2}\right)\right)^2 - B\ln\left(\frac{gF}{U^2}\right) + C\right]^{1/2} + D\ln\left(\frac{gF}{U^2}\right)\right\} \tag{A-4}$$

where the various constants are: K = 6.5882, A = 0.0161, B = 0.3692, C = 2.2024 and D = 0.8798, and $\ln = \log_e$.

Constant Wind Speed at an Angle to Fetch Direction

The two-direction significant wave forecasting relations can be obtained by replacing the wind speed squared (U^2) in the X-Y domain with the corresponding wind stress components proportional to UU_x and UU_y, where $UU_x = U^2 \cos\theta$, $UU_y = U^2 \sin\theta$ and θ = the angle of wind direction measured from the x- axis. This leads to two sets of complementary equations, one set corresponding to each of the four equations given above. From these complementary equations, it can be found that

$$\frac{gF_x}{UU_x} = \frac{gF_y}{UU_y} = \frac{gF}{U^2}, \text{ where } F^2 = F_x^{\,2} + F_y^{\,2} \tag{A-5}$$

$$\frac{gH_x}{UU_x} = \frac{gH_y}{UU_y} = \frac{gH}{U^2}, \text{ where } H^2 = H_x^{\,2} + H_y^{\,2} \tag{A-6}$$

$$\frac{g^2T_x^{\,2}}{UU_x} = \frac{g^2T_y^{\,2}}{UU_y} = \frac{g^2T^2}{U^2}, \text{ where } T^4 = T_x^{\,4} + T_y^{\,4} \tag{A-7}$$

$$\frac{g^2t_x^{\,2}}{UU_x} = \frac{g^2t_y^{\,2}}{UU_y} = \frac{g^2t^2}{U^2}, \text{ where } t^4 = t_x^{\,4} + t_y^{\,4} \tag{A-8}$$

Variable Wind Speed and Variable Wind Direction

Finally the above complementary equations can be put into differential form and by numerical integration means can be used to generate wave fields due to winds of variable magnitude and variable direction. This method of application is known as the two-direction significant wave forecasting method, and can easily be applied to hurricane wind fields, which do in fact consist of variable wind speed and variable wind direction.

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* Chairman and Professor

The Hurricane Wave Model

Using this scheme, Bretschneider[1] developed a non-dimensional stationary hurricane wave model, based on the U.S. Weather Service, Graham and Nunn[3] hurricane wind model. This gives a hurricane wave field coupled with the hurricane wind field. Any change in the wind field will result in a change in the wave field. For example, when the hurricane moves forward at a speed, say 10 to 20 knots, there will be a change in the wind field. The winds on the right side of the hurricane will increase and on the left side will decrease. These are usually small corrections in the wind field, which correspond to small corrections in the wave field. Because the hurricane moves forward there will also be a change in the effective fetch length, which will cause an additional small increase in the wave height on the right side of the hurricane center and a corresponding decrease on the left side. It is found that the combined incremental change in wave height can be obtained from

$$H + \Delta H = H \left[1 + \frac{\Delta U}{U}\right]^2 \qquad \text{(A-9)}$$

where

H = the wave height for the stationary hurricane at location of wind speed U

ΔH = change in wave height due to
ΔU = change in wind speed.

Since ΔU can be positive or negative ΔH will be accordingly positive or negative.

Significant Wave Hindcasts for Hurricane Camille

Using the methods developed by Bretschneider[1], we have reconstructed the hurricane wind and wave field for Hurricane Camille (1969), and compared the results with the measured significant wave heights from six (6) offshore stations of the Ocean Data Gathering Program (ODGP) reported by Cardone, Pierson and Ward[8]. Figure A-1 shows the path of Hurricane Camille (1969).

Figure A-2 shows the reconstructed (predicted) hurricane wind and wave fields. At the bottom of Figure A-2 are shown the six ODGP wave recording stations. The hurricane (Fig. A-2) wave field was moved across the six ODGP stations to obtain the maximum values of the significant wave height. Figure A-3 shows a comparison between the predicted and the measured significant wave heights for the six ODGP stations.

Application to the 100-Year Design Wind and Wave Criteria

From the standard project hurricane wave criteria, given by Bretschneider[2], we have determined the 100-year design hurricane wave criteria for the New Orleans and West Coast of Florida OTEC sites.

There are no standard project hurricanes available for Keahole Point, Hawaii, or Punta Tuna, Puerto Rico. However, we have still used the model hurricane technique presented in the paper to estimate the 100-year hurricane wind and wave criteria. This was done by use of literature search to obtain reasonable estimates of the three hurricane parameters required, namely: R, the radius of maximum wind; ΔP, the central pressure reduction from normal; and V_F the forward speed. The results of the 100-year design hurricane parameters for the four OTEC sites and also for Hurricane Camille are given in Table A-I.

For the deterministic wave theory approach, we use $H_{max} = 1.8\, H_s$, and $f_o^{-1} = \sqrt[4]{5/4}\, T_s$ for maximum wave height and wave period, in the determination of the hydrodynamic wave pressures.

Acknowledgment

This work was partially supported by Department of Energy Contract No. EY-76-S-03-0235, "Current-Wave Coupling Project."

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TABLE A-1 SUMMARY 100-YEAR DESIGN HURRICANE PARAMETERS FOR VARIOUS OTEC SITES

PARAMETER	OCEAN THERMAL ENERGY CONVERSION (OTEC) SITE				HINDCASTS FOR HURRICANE CAMILLE (1969)
LOCATION	KEAHOLE POINT HAWAII	PUNTA TUNA PUERTO RICO	NEW ORLEANS LOUISIANA	WEST COAST FLORIDA	
LATITUDE	19°44'N	17°57'N	27°59'N	26°00'N	∿29°N
LONGITUDE	156°06'W	65°52'W	88°55'W	84°54'W	∿89°W
DISTANCE FROM LAND N. MILES	∿2.0	∿3.0	∿68	∿150	
R, *radius of max wind* (N. miles)	12	10	14	8	10
ΔP, inches of mercury	1.1	2.19	2.34	2.56	3.1
V_F, forward speed (knots)	10	10	11	10	10
f, Coriolis parameter (rad/hr)	0.177	0.162	0.246	0.230	0.255
K, (coefficient)	67	67	66	67	66
k^*, *(coefficient)*	0.796	0.813	0.886	0.886	0.886
$U_R = K\sqrt{\Delta P} - 0.5\ fR$ (knots)	69.21	98.34	99.24	106.3	114.93
$U_{Rs} = k^* U_R$ (knots)	55.09	79.95	87.93	94.16	101.8
$U_s = U_R + \frac{1}{2} V_F$ (knots)	60.09	84.95	93.43	99.16	106.8
fR/U_R	.0307	.0165	.0347	.0173	0.0222
K', (coefficient) (Table II)	6.38	6.80	6.26	6.78	6.63
$H_R = K'\sqrt{R\Delta P}$ (ft) (at r = R)	23.18	31.82	35.83	30.68	36.91
$H_s = H_R\left[1 + \frac{0.5\ V_F}{U_{RS}}\right]^2$ (ft)	27.58	35.92	40.17	34.02	40.62
$T_s = 0.4\ U_s \tanh\{[\text{arc tanh}\ \frac{40H_s}{U_s^2}]^{0.6}\}$ (sec)	11.12	12.41	13.06	11.79	12.91

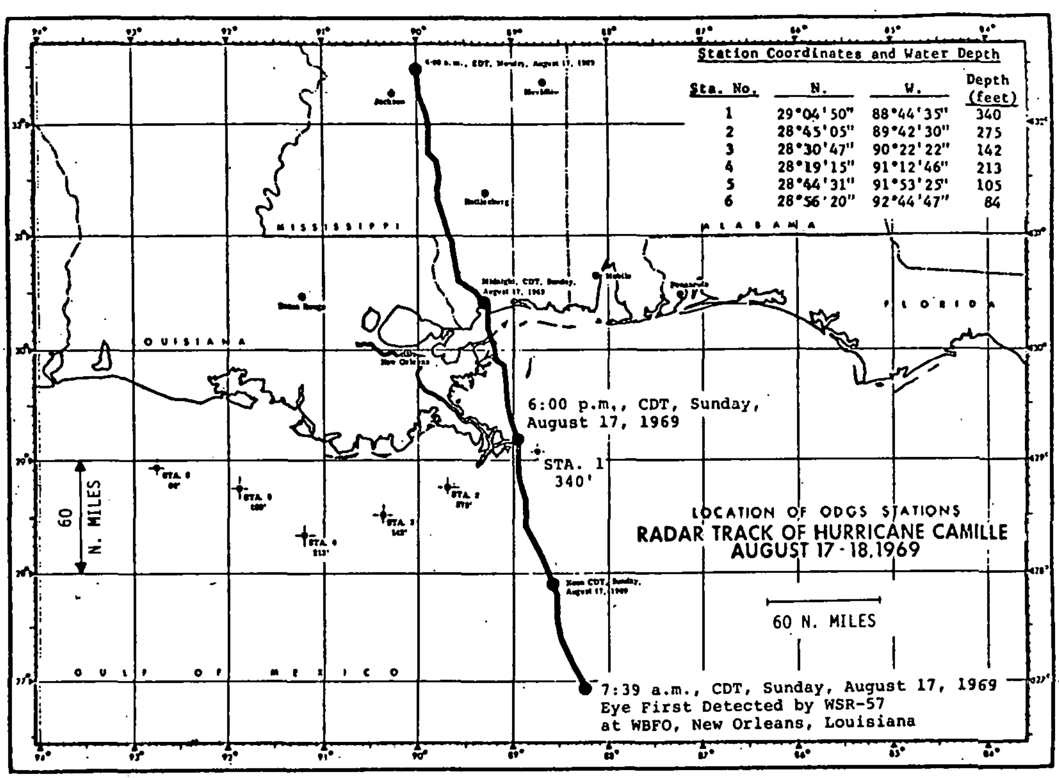

FIG. A-1 PATH OF HURRICANE CAMILLE (1969) THRU SIX MEASURING STATIONS-- OF SHELL'S OCEAN DATA GATHERING PROGRAM--IN THE GULF OF MEXICO

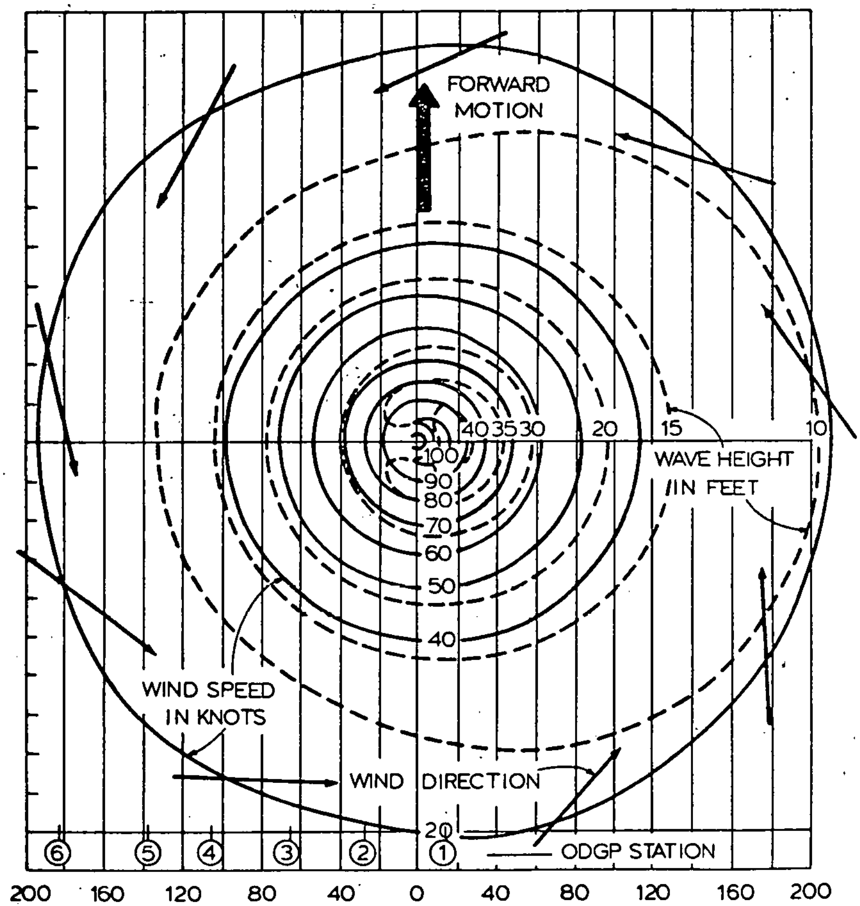

FIG. A-2 WIND AND WAVE FIELD PREDICTED FOR HURRICANE CAMILLE (1969)
(About ± 20 to 25° in change of forward direction is allowable.)

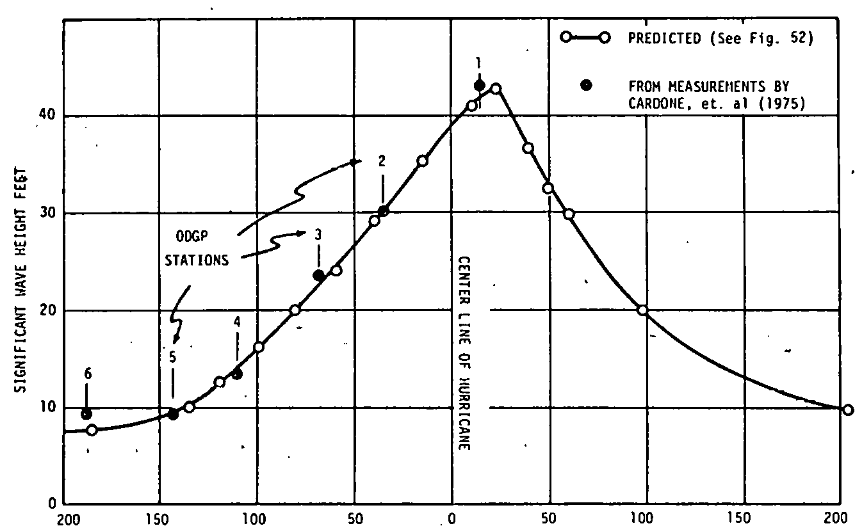

FIG. A-3 PREDICTED MAXIMUM SIGNIFICANT WAVE HEIGHT OF MODEL HURRICANE
FOR HURRICANE CAMILLE R, ΔP, & V_F PARAMETERS

THE EFFECTS OF SHEAR ON VORTEX SHEDDING PATTERNS IN HIGH REYNOLDS NUMBER FLOW: AN EXPERIMENTAL STUDY

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Abstract

Model tests were performed in a wind tunnel to determine vortex shedding patterns induced around a cylinder by spanwise shear in high Re flow. In addition, mean and fluctuating surface pressure measurements were made at various points along the cylinder. The introduction of shear in the upstream flow generated two distinct cells of vortex frequencies behind the cylinder, similar to the pattern observed in low Re flow. Further tests with different shear levels and aspect ratios will produce valuable information for the design of the Ocean Thermal Energy Conversion Plant Cold Water Pipe.

Introduction

Vortex shedding has been recognized to be a potential source of loading on the Ocean Thermal Energy Conversion (OTEC) Plant Cold Water Pipe (CWP). In an effort to understand the phenomena of the vortex shedding on the CWP, ORI, Inc was commissioned by NOAA's Office of Ocean Engineering to investigate the effects of shear on vortex shedding in high Reynolds number (R_e 10^5) flow. Virginia Polytechnic Institute and State University (VPI & SU) subcontracted with ORI, Inc. to perform the work outlined in the task order. This report documents the results and analysis of the tests conducted at VPI & SU investigating the effects of shear on vortex shedding.

The primary impact of vortex shedding occurs when the eddy shedding frequency (Strouhal frequency) of the Karman vortex street is at or near the natural frequency of the CWP. In this situation, amplification of the pipe motions to near resonant levels may occur. This is due to a shifting of or "locking-on of the pipe vibrational response to the Strouhal frequency. During lock-on, CWP vibrational stresses are magnified and the possibility of structural damage is increased.

The existence of vertical shear in ocean currents complicates the effects of vortex shedding along the CWP. It leads to the presence of discrete cells, distributed along the pipe. The relative frequencies and phasing of these vortex cells may affect the dynamics of the pipe. Therefore, knowledge of the effects of shear on vortex shedding is essential in design of the OTEC cold water pipe.

The literature pertaining to the subject of vortex shedding from pipes is rapidly increasing. An excellent, up-to-date review of work in the field is presented in a paper by Griffin[2]. Of particular relevance to the present report are studies of interactions of a sheared flow with stationary and vibrating bluff bodies.

Shaw and Starr[6] showed that a synchronized pattern of spanwise shedding frequencies existed, the length of which was a function of the flow velocity gradient. Further studies performed by Maull and Young[5] verified the existence of distinct shedding cells of constant Strouhal frequency. This was demonstrated by measuring the power spectral density of the eddy-shedding frequency for a bluff body model in shear flow. They stated that "the coherence of the shed vortices requires a constant frequency over the lengths." They also investigated the boundaries between these cells and found "longitudinal vortices lying in the free-stream direction" which they hypothesized were "due to a rolling up of the original distributed vorticity into concentrated trailing vortices at the base."

Davies[1] measured cell structures in a highly sheared flow field for Reynolds numbers up to the critical, at which point the eddies became indistinct. The effects of endplates and the spanwise distribution of vortices were considered by Mair and Stansby[4]. They found that long endplates produced sharper spectral density peaks of the eddy-shedding frequency and more well-defined cells of vortices than did short endplates. Through further experiments they found that elimination of endplates altogether reduced the number of cells present and that for fixed endplates the number and length of cells were insensitive to variations in shear and body aspect ratio.

Stansby[7] investigated the "lock-on" characteristics of cylinders in both uniform and shear flow for mechanically vibrated (at a specified frequency and amplitude) and stationary cylinders in a wind tunnel. He noted that, in general, the Strouhal number for eddy shedding in vibrating cylinders is a function of Reynolds number, shear, aspect ratio, cylinder vibration Strouhal number, and relative amplitude of vibration. Additionally, he observed that cell length increased spanwise as the relative amplitude of vibration increased, that usually only one of the generated cells would lock on to the vibrating frequency, and that peaks in the power spectra of adjoining cells became more clearly defined. With stationary cylinders in shear flow he found that generally, only two end cells were produced such that "at any given time each cell has a finite length and a single frequency and cells do not overlap."

The Reynolds number for flows in Stansby's studies were all below 1×10^4. For this regime, he computed the range of frequencies for primary lock-on in shear flow and developed an empirical

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relationship for determining the spanwise length of lock-on cells.

The purpose of this experimental study was to investigate the effects of shear flow on vortex shedding for both smooth and rough circular cylinders with R_e on the order of 5×10^5. The focus of the experiments was to isolate the effects of shear on the spanwise distribution of vortex cells and to determine magnitudes of unsteady pressures produced on stationary cylinders.

Experimental Equipment

Experiments were conducted in the 6 foot by 6 foot cross-section Stability Wind Tunnel Facility located in the Aerospace and Ocean Engineering Department at VPI & SU. A sheared cross-stream flow was achieved in the test facility by means of a series of wire screens of non-uniform cross-stream distribution, placed upstream of the model (Figure 1).

The OTEC cold water pipe was represented by a 56-inch long, 6-inch diameter aluminum cylinder, affixed midway between the floor and ceiling of the test section. The cylinder had 1/16-inch thick, 10-inch diameter steel plates attached to both ends to prevent tip vortices from interferring with the Karman vortex street, which was the primary object of study in the test program. Both ends of the cylinder were 8-inches from the test section wall, thus insuring that the wall boundary layer would not interfere with the free-stream flow around the model.

To obtain a rough pipe, the surface of the cylinder was knurled to a roughness K/D .001, where K is the depth of a roughness element and D is the diameter of the cylinder. The purpose of this was twofold. First, the maximum Reynolds number attainable with the smooth 6-inch diameter cylinder in the wind tunnel was approximately 3×10^5. In that range, the flow regime behind the cylinder exhibited so-called critical characteristics, notably a highly turbulent wake with weakened vortex shedding. It was thought that increasing the roughness would have the effect of changing the wake pattern to the transcritical regime, with a well-defined eddy shedding pattern. Secondly, the roughened surface more properly represents the condition of a cold water pipe subject to biofouling throughout its lifetime, as noted by Hove et. al.[3].

The cylinder itself was hollow, permitting tubing from the pressure-sensing equipment to be fed through the inside and out the endplates to recording instruments outside the tunnel. Mean surface pressure values were measured through a series of three rows of 28 ports of 1/16-inch diameter. These ports were spaced at 2-inch intervals along the length of the pipe, with each row 15° apart. Attached to the ports were 1/16-inch diameter tubes (84 in all) fitted to a Scanivalve pressure sensing unit.

In addition, four Kulite pressure transducers (Model CQH 125-5) of 1/4-inch diameter including their plexiglas casings were flush mounted in the cylinder. They were located 10, 22, 34, and 46-inches from the right-hand, or high velocity end of the cylinder. They were connected to a Tektronix TM 503 Amplifier, so that the output from the transducers could be differentially amplified, as well as filtered to eliminate very high frequency (300 Hz) noise. Output from the amplifiers was channeled into a Tektronix Type 561A oscilloscope for visual inspection of instantaneous pressure readings. Simultaneously, the signal was fed into an rms voltmeter which yielded the RMS value of the pressure fluctuations. Finally, the signal could be sent to a machine (discussed in conjunction with hot-wire equipment) which produced a power spectrum plot of the pressure fluctuations.

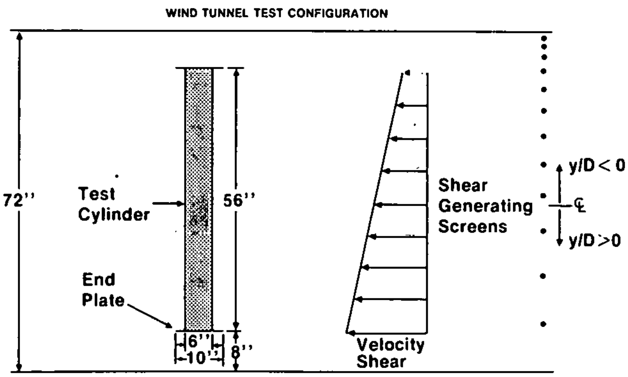

FIG. 1 WIND TUNNEL TEST CONFIGURATION

Free-stream turbulence measurements were determined from a single hotwire probe connected to a constant temperature linearized anemometer.

To examine the spectral characteristics of the flow behind the cylinder (in particular, to determine the eddy shedding frequency), the linearized, filtered anemometer signal was processed through a Zonic Technical Laboratories, Inc. Multichannel FFT Processor. To allow for immediate visual inspection of the signal, it was simultaneously monitored on an oscilloscope. The output, in the form of a power density spectrum (log amplitude versus frequency), was plotted on a graphics display terminal. A hard copy of the results was obtained by photographing the terminal screen with a camera mounted on a tripod.

Test Procedures

Before the effects of high Re shear on vortex shedding patterns could be investigated, tests had to be performed to determine and establish the desired inflow characteristics. Specifically, tests were required to determine the inflow velocity profiles, shear characteristics, and to verify that turbulence levels would be constant throughout the test investigation. The tests conducted to investigate the effects of shear on vortex shedding also measured mean pressures, rms, and peak-to-peak pressure fluctuations on the surface of the cylinder.

Figure 2 gives a typical upstream velocity profile of the different tunnel speeds and three sets of screens employed in the tests. In these figures, the vertical axis shows the normalized velocity (U/U_c) based on the centerline values of flow (U_c), is plotted as a function of the spanwise distance (y) from the center of the test cylinder as a function of its diameter (D). These velocity profiles were obtained by measuring the stagnation pressure at the 0° pressure points along the length of the cylinder. The cylinder was positioned carefully, insuring that the desired pressure port was within -0.25° of the direction of oncoming flow.

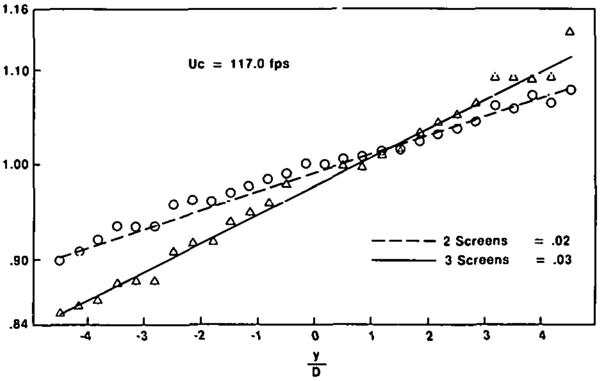

FIG. 2 FREESTREAM VELOCITY PROFILES

A series of upstream screens was employed to generate both freestream turbulence and shear. In order to produce turbulence in all the test configurations, it was necessary to employ at least one wire screen. With one screen in place, with wires of uniform density, no velocity shear was generated. With two screens of uneven wire distribution, the shear produced was characterized by a value of =0.02 and with three such screens took on a value of 0.03. The shear parameter is defined as (D/U_c) du/dy where: D is the test cylinder diameter, U_c is the inflow centerline velocity, u is inflow velocity at any longitudinal location, and y is the longitudinal distance from the cylinder center. The values of chosen were selected to represent typical ocean shears. However, the velocity profiles corresponding to of 0.02 and 0.03 are not markedly different and the greatest velocity variation over the length of the cylinder is approximately 30% of the centerline velocity.

Once the free-stream flow velocities were determined, measurement of free-stream turbulence could be performed. To insure experimental validity of the overall investigation it was desired to have a constant level of turbulence for all experimental configurations employed. Measurements were made of turbulence levels along the length of the cylinder as well as vertically above and beneath it. Three shear conditions with free-stream centerline velocities of 74, 104 and 117 ft/sec were investigated. A typical result, Figure 3, displays the level of turbulence for an individual mean velocity for the three shear conditions. Measurements were also performed one pipe diameter (6-inches) above the centerline of the pipe for the centerline wind velocity of 104 feet/second in all three shear conditions. This was done to assure that turbulence was not variable with elevation within the tunnel.

The turbulence level for unsheared flow was found to be virtually constant over the length of the cylinder at a level of approximately 0.25%. The addition of shear-generating screens raised the turbulence intensities to about 0.30% for = 0.02 and 0.35% for = 0.03 in the range of -3.0 y/D 3.0. Beyond these points turbulence readings increased greatly to 0.7% and even more. On the high velocity side of the cylinder the turbulence intensity reaches a peak near y/D= +4.0. This is a result of the abrupt mixing of the unobstructed flow near the wall and the shear flow through the wires. The high turbulence levels on the low velocity side of the tunnel are probably the result of a loose wire ing excessively in one of the shear-producing screens. Neither of these high values is attributable to the tunnel wall boundary layer.

In general, the turbulence levels registered were close enough to one another over the major portion of the cylinder to consider that the flow regime felt by the cylinder was the same for all the tests conducted.

Before rough pipe investigations commenced, experiments were performed on a smooth pipe model, but physical limitations of the wind tunnel facility presented a severe problem. The highest Re obtainable placed the flow behind the cylinder in the critical regime, where vortex shedding patterns were at best weakly defined. The results of these tests were not very conclusive or specifically applicable to the OTEC cold water pipe. Therefore, they are not included in this report. With a roughened pipe, however, the downstream wake pattern was altered enough to place the flow behind the cylinder in the transcritical range, where the vortices were clearly defined and could be measured.

The first tests to be performed for the experimental study in question were the ones to conduct an investigation of the surface pressure distribution on the cylinder. These tests were done in order to describe both the pressure distribution and to study possible connections between pressure fluctuations on the surface and vortex shedding. The surface pressure distribution was defined by taking mean pressure readings from the pressure ports and by measuring the RMS, peak-to-peak, and for a limited number of configurations the power spectra of the pressure fluctuations by means of the Kulite pressure transducers. (By comparing the results of the mean pressure readings to the range of values for the peak-to-peak readings a cross check of the pressure readings was available.)

Fluctuations were characterized in three ways: RMS fluctuation, peak-to-peak distance, and, for a limited number of configurations, the power spectra of the fluctuations. The transducer signal was fed into an RMS volt meter to determine the RMS value and monitored on an oscilloscope where the peak-to-peak distances were visually estimated. Measurements were taken at cylinder orientations of

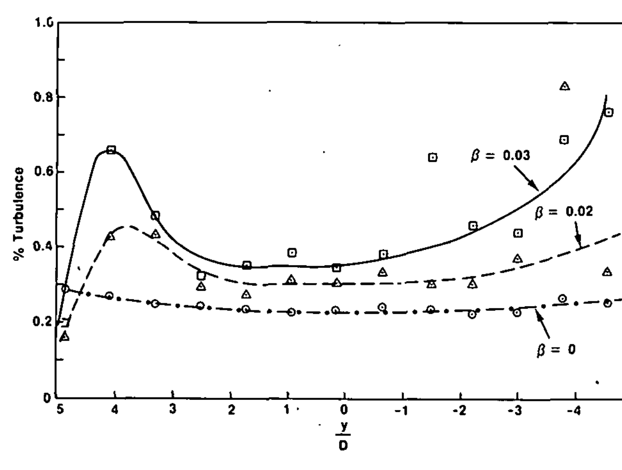

FIG. 3 TYPICAL FREESTREAM TURBULENCE PROFILE

0°, 45°, 90°, 135°, 225°, 270°, and 315° relative to the upstream flow. These readings were taken for flow velocities of 74, 104, and 117 feet/second at lateral locations of y/D equal to +3, -1, -3 for the three shear profiles. A transducer was installed at the location of y/D = +1, but data are not available at this location because it was irreparably damaged early in the test program. Measurements were also performed on the pressure fluctuations for the flow velocity of 154 feet/second. For these readings the transducer signal was sent through an FFT processor to obtain the power spectra of the turbulent pressure, which was then compared to spectra from the hot-wire probe.

The eddy shedding frequency was measured using a hot wire probe mounted on a traverse located four diameters (24-inches) downstream from the cylinder. Readings were taken every 4.7 inches. Vertically, the hot wire probe position was varied to measure the frequency of shedding where the strength of the vortices was greatest. The signal from the probe was transmitted to an FFT processor which was programmed to sample the signal every 5 msec and to average 25 spectra before generating the final power spectrum.

Discussion of Results

Surface Pressure Characteristics

Figure 4 presents the distribution of mean surface pressure over the circumference of the cylinder for the nine test configur- ations investigated. The vertical axes of each plot indicates the mean pressure in psi for each value of consider, while the horizontal axes give the circumferential location of the reading as defined on Figure 4. The three plots represent centerline velocities of 74, 104, and 117 feet/second, and spanwise positions y/D = -3.0.

A few observations can be made about the results for these and other values of y/D where measurements were taken. For low velocity upstream flow conditions (U_c = 74 ft/sec, Re_c = 2.5 x 10^5) the surface pressures generated by sheared flow are, in general, greater in magnitude than for unsheared flow. There is a definite asymmetry at a number of stations. This is largely a result of blockage in the tunnel. For medium velocity conditions (U_c = 104 ft/sec, Re_c = 3.5 x 10^5), the case shown is representative of most stations, with little difference in pressure readings for all flow profiles. Readings taken close to either end of the cylinder suffered distortion from endplate influence. The pressure pattern for the high velocity case (U_c = 117 ft/sec, Re_c= 3.9 x 10^5) appears to be entirely independent of upstream shear. An important point to be noted is that the asymmetry that existed at lower tunnel speeds was no longer present with high centerline velocities.

Figure 5 is a graph of the rms pressure fluctuations over the circumference of the cylinder. Again, each plot represents a single upstream centerline velocity, with measurements corresponding to y/D = -3.0. This particular value of y/D represented the most extreme examples of dependence of turbulence on shear levels. Values for each shear level are shown together on the three graphs. It is evident that increased shear caused turbulence levels to increase by two or three times relative to unsheared values. Maximum turbulence

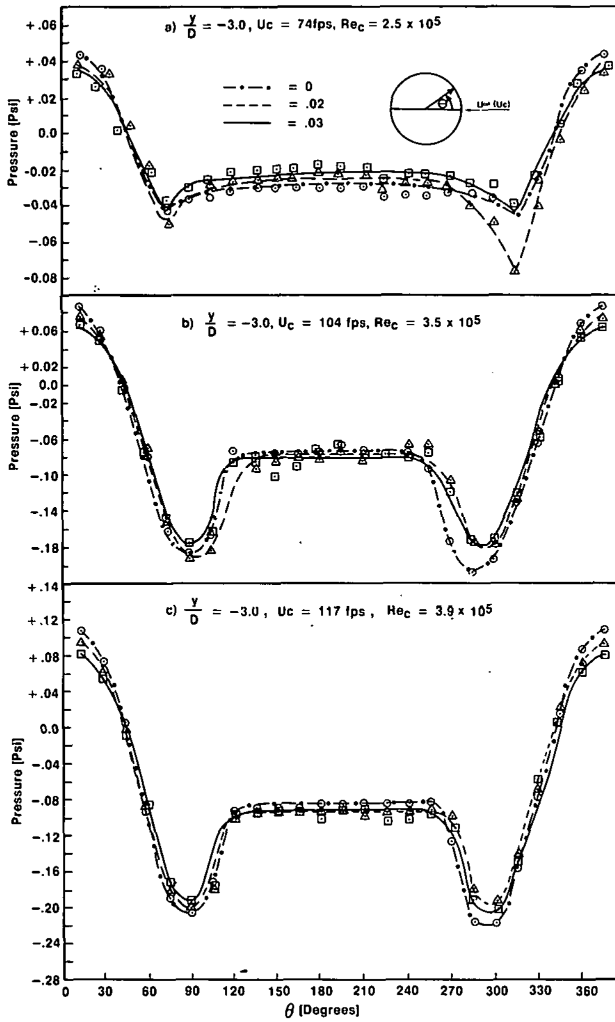

FIG. 4 MEAN PRESSURE DISTRIBUTION ON CYLINDER SURFACE

values occurred at 90° and 270° on the cylinder face, and were registered rms values of about 20% of the mean flow. At some other values of y/D, the turbulent intensities were comparable for all shear levels, and no discernible relation existed between y/D, and relative turbulence levels. However, the data shown here suggests that design specifications for the CWP should reflect the higher instantaneous pressures achievable in a sheared flow. Peak-to-peak measurements correlated well with the rms measurements.

Eddy Shedding Characteristics

Spectral characteristics of the downstream flow pattern for the low velocity case (U_c = 74 ft/sec of Re_c = 2.5 x 10^5) are indicated in Figures 6a and 6b. The power spectrum is plotted on a log amplitude scale versus frequency for a series of points at different y/D values. Both the two screen and three screen case (corresponding to of 0.02 and 0.03 respectively) show virtually identi- cal shedding patterns. The high velocity end is marked by a strong signal at 31 Hz, which persists beyond the midway point. As the probe was moved further to the low velocity side, a weaker, broader signal at 20

Hz emerged. The unsheared flow of the same centerline velocity generated a constant spanwise shedding frequency of 24 Hz.

Figures 6c and 6d depict the spectra measured at various points along the length of the cylinder for a centerline flow velocity of 117 ft/sec. A single peak at about 45 Hz is evident for the two screen case. For = 0.03 two peaks were evident. The sharper of the two was at about 49 Hz on the high velocity end of the cylinder. Near y/D = 2.0 a broader peak of 44 or 45 Hz appeared. The broad peak of 44 to 45 Hz was also evident in the associated spectra for unsheared flow. The spectral peaks measured for inflow centerline velocities of 117 ft/sec were less pronounced and disappeared considerably further from the endplates than in both the higher or lower velocity cases. It is possible that the change in effective Reynolds number produced in this case placed the flow in the critical regime where vortices are not well defined.

The spectra generated with a centerline flow velocity of 154 ft/sec are shown in Figures 6e and 6f. Both figures show definite two cell structures. For = 0.02 a peak at 66 Hz exists on the high velocity end. It persists beyond the midpoint, while a second peak at 61 Hz grows and gradually replaces it. The transition region in which both peaks appear occurs over a distance of approximately two cylinder diameters, which compares reasonably well with the findings of Maull and Young[5]. They attributed this dual signal to the probe picking up signals from the separate cells, as opposed to the existence of an oscillating cell boundary. When was changed to 0.03 the experimental findings were very similar to those for = 0.02 but the shedding frequencies were altered slightly; to values of 67 and 59 Hz.

The spectra of the eddy shedding frequency were compared to the spectra of the Kulite transducer readings of pressure fluctuations. The spectra for the Kulite transducer readings at 90° and 270° showed a good agreement with the hot wire spectral readings for vortex shedding frequency, demonstrating that eddy shedding is a chief source of such fluctuations. Additionally, spectra of upstream turbulence were recorded to determine if external inputs such as the propeller blade frequency, tunnel vibrations, or instrument noise could be the cause of any spectral peaks. No dominant peaks appeared in these spectra, so external inputs were ruled out as causes of the pressure or eddy frequency spectral peaks.

From the spectral analysis performed in this study, frequencies of vortex shedding were observed for the various velocities and shear conditions. Cylinder end effects and unsteady turbulence do not account for the frequency shift which is seen for the flows with Re_c = 2.5 and 5.2 x 10^5. Therefore, although the two cell structure does not permit correlation of cell length and shear, the experiment has produced a frequency shift or formation of a second cell at Re_c up to 5.2 x 10^5 which could only be the result of shear flow effects.

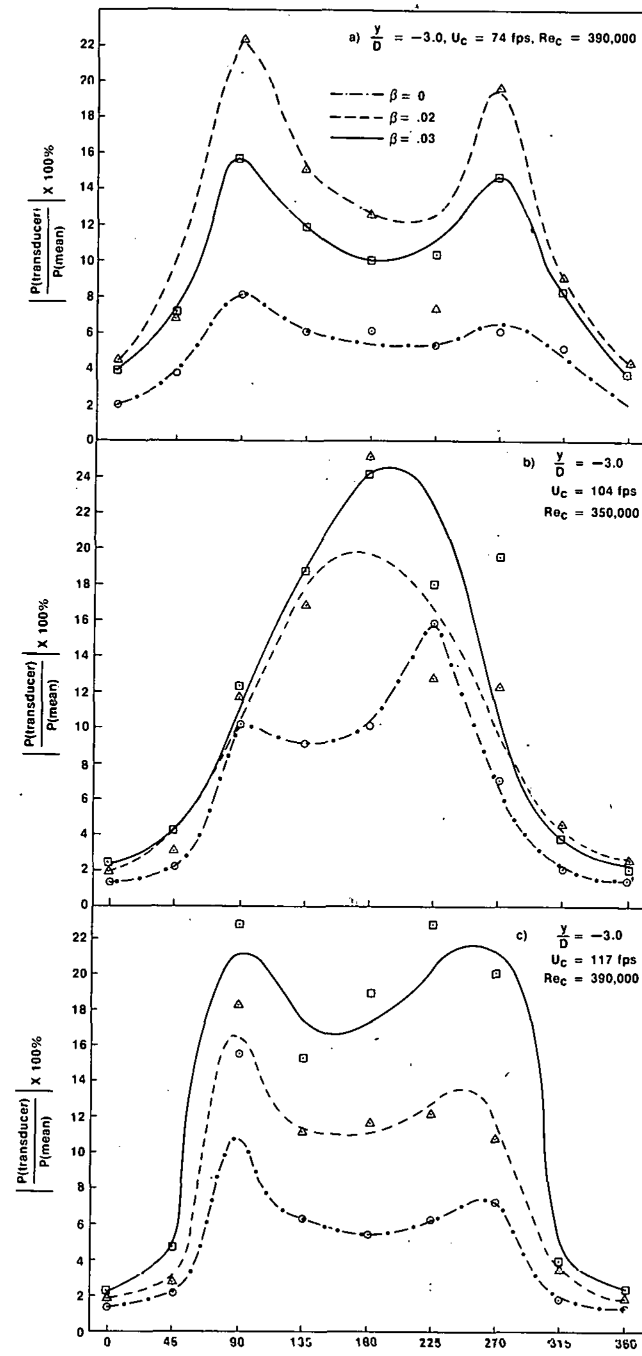

FIG. 5 RMS PRESSURE FLUCTUATIONS ON CYLINDER SURFACE

Conclusions and Recommendations

The report presented an experimental investigations into the effects of vertical shear on vortex shedding patterns for high Reynolds number (Re) flow around circular cylinders. This phenomenon was explored through tests to determine frequencies of the vortices shed by the cylinder and the characteristics of pressures on the cylinder. From these investigations it was determined that:

- Discrete cells of eddies are formed in high Re shear flow, but from the data measured, it is impossible to determine a correlation between sheer and cell length.

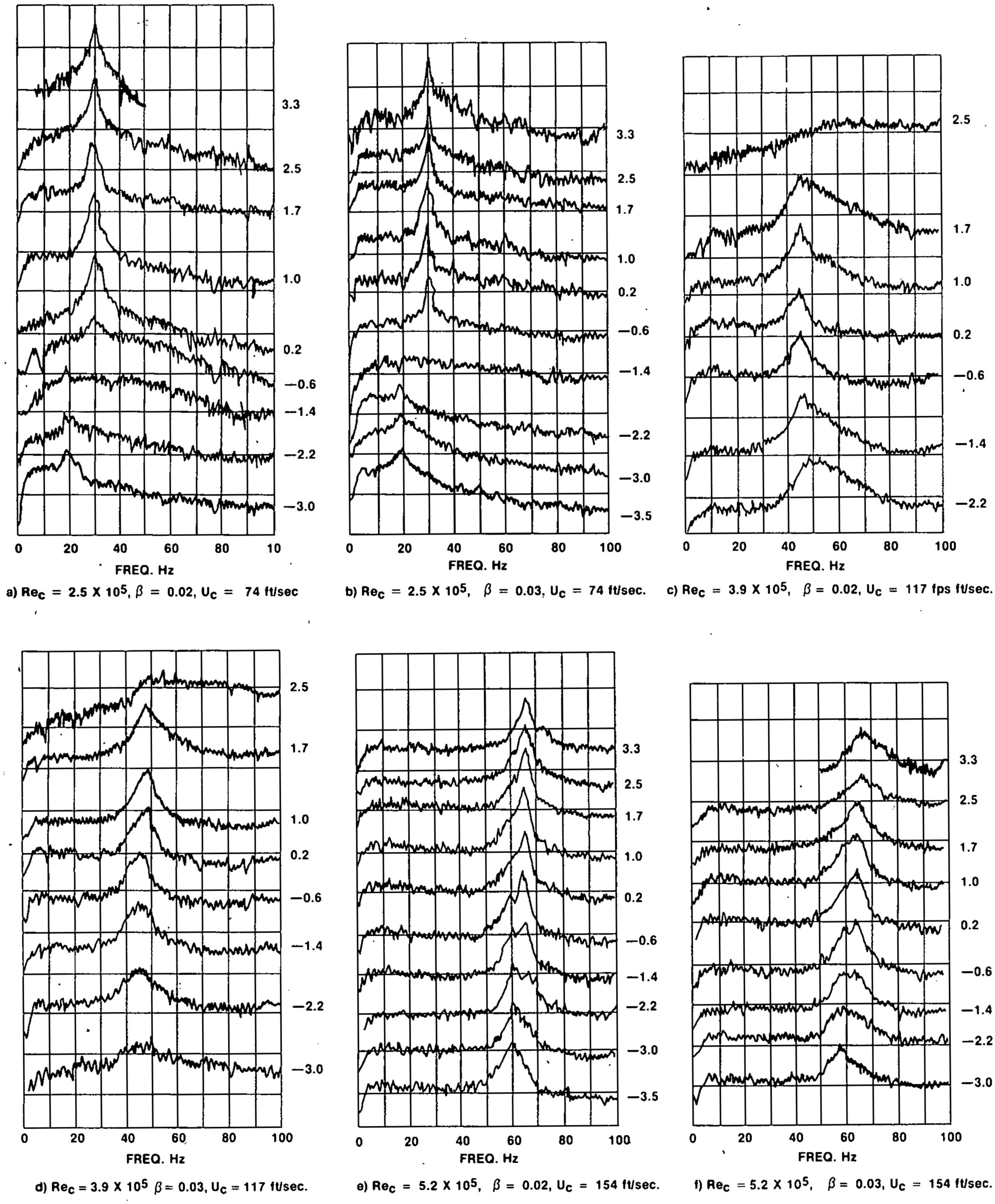

FIG. 6 POWER SPECTRA OF EDDY SHEDDING FREQUENCY OVER LENGTH OF CYLINDER

- For the aspect ratio studied, the eddy shedding phenomenon for high Re sheared flow was found to be essentially the same as for low Re sheared flow, with two distinct correlated eddies evident.

- The mean surface pressure distribution on the cylinder is independent of shear in the upstream flow, except where flow is in a transitional regime.

- The fluctuations in pressures on the surface of the cylinder can be as much as twice as high in high Re sheared flow than for similar unsheared flow, a factor which should be considered in determining CWP load levels.

In order to increase the understanding of the effects of shear on vortex shedding, the following points are recommended for further study.

- The analysis of this study should be extended to include a larger range of shear parameter, particularly for low shears to determine the minimum shear which produces a two cell eddy structure.

- A smaller cylinder (H/D 20) with much larger endplates, similar to models tested by Mair and Stansby[4] should be examined in high Reynolds number flows over a wide range of shears so that more cells may be observed.

- A cylinder subjected to forced vibration of varying amplitude, similar to models tested by Stansby[7], should be tested at high Reynolds numbers in a wide range of shears to investigate high Reynolds number formula for correlation lengths at lock-on.

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DISCUSSION

A. Galef, TRW: When we saw a cell of approximately 5/8 the length of the pipe, what sort of a current range did this encompass, was it ± 10%, or ± 5%?

D. Rooney: The current range is ± 15% from top to bottom; 5/8 would be closer to ± 10%.

A. Galef: Secondly, without a scale I couldn't tell whether those peaks were broad-band or narrow-band. Which were they?

D. Rooney: They were realtively broad because of end-plate effects due to the small length/diameter (L/D) ratio. I would expect the resolution to improve with a greater L/D. Previous results by others showed the cell length to be 5/8 of the pipe length at subcritical Reynolds numbers. We observed the same 5/8 value in our tests at supercritical Reynolds numbers.

DEVELOPMENT OF OCEAN CURRENT INFORMATION NEEDED TO ADDRESS THE PROBLEM OF VORTEX SHEDDING ON THE OTEC COLD-WATER PIPE

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Abstract

Recent information and experience indicate that cyclic stresses induced by vortex shedding may be a much more critical consideration in cold-water pipe (CWP) design than has previously been assumed. Because of this, it will become important to more thoroughly define and present operational information on ocean currents at potential OTEC sites, in a statistical format similar to the presentation of operational wave information. An example of cyclic stress calculations on the CWP, based on current information contained in the government furnished environmental package, is presented to show the inadequacy of this information. A conceptual plan for generating statistics on currents, based on hindcasting techniques is presented. It is shown that adequate wind information exists to generate a data base of operational currents. The data base is presented in terms of joint probability distributions of speed and direction as a function of depth, persistence of currents that would tend to induce critical vortex shedding frequencies and the distribution of directional coherence with depth. All of the above are of importance to the evaluation of the potential severity of vortex shedding loads for structural design. Scales of vortex correlation length will be considered relative to the vertical coherence of current direction and the wave lengths of the vibrational modes of the pipe. Finally, the authors will present a plan for developing statistical information to address these questions and will discuss the application of this data to CWP design.

Introduction

Of all the major components present in an OTEC power plant, probably the most crucial from an overall design methodology standpoint is the cold-water pipe (CWP). Because of its massive size (10-30 m. in diameter and nearly 1000 m. in length), costs per unit increase in design strength are enormous and generous levels of overdesign would be prohibitively expensive. Countering this is the relative importance of the reliability of the CWP to the overall success of the OTEC system. Structural failure of the pipe at any time after the initiation of deployment would more than likely result in the ultimate failure of the entire OTEC project. The same claims cannot be made for most other OTEC components which, by-and-large, may be replaced, repaired, and altered even after deployment. The cold-water pipe will be, in effect, a "one-shot deal."

The critical nature of the CWP as already been demonstrated by the loss of the pipe on the Deep Oil X-1, which experienced structural failure during deployment off Santa Catalina Island in the Pacific Ocean in December 1978.

Due to the nature of the problems associated with the CWP design and analysis, significant effort is being put into determination of the dynamic responses of the pipe to sea conditions. Two of the most widely known and most frequently exercised analytical tools for the determination of CWP dynamic response are the NOAA/DOE Dynamic Load and Stress Analysis Model[1], developed by Hydronautics, Inc., and ROTEC, the finite-element model developed by Dr. J.R. Paulling[2]. Both of the above models operate in the frequency-domain and can handle spectrally defined wave loads as well as steady and fluctuating current loads. A number of other analytical techniques, both time domain and frequency domain, have been developed for application to specific OTEC design cases. Some of these have been reviewed in the past by Gilbert Commonwealth Assoc.[3] and others were discussed at the OTEC Cold-Water Pipe Workshop held in January 1979.[4]

It is not the purpose of this paper to review or evaluate these analysis tools, however. The intent is rather to assess the available environmental input data provided by the Department of Energy (DOE) in the form of the Standard Environmental Data Package, and consider its applicability to CWP design problems. In particular, we are interested in the problems of vortex-shedding and other fluctuating current loads as they relate to fatigue analysis and other statistically definable phenomena.

Importance of Vortex-Shedding Loads

In general, our understanding of the dynamic response of large bluff bodies in the oceanic current environment is based on a combination of extrapolation from smaller scale phenomena, empirical evidence and physical intuition. A structure of the size of the OTEC cold-water pipe in many ways dwarfs some of the most impressive of engineering structures (Figure 1). In comparison, it is two and one-half times as high as the mammoth Sears Tower in Chicago, and nearly three times as large as the Cognac Drilling Rig. Complicating this situation is the very high

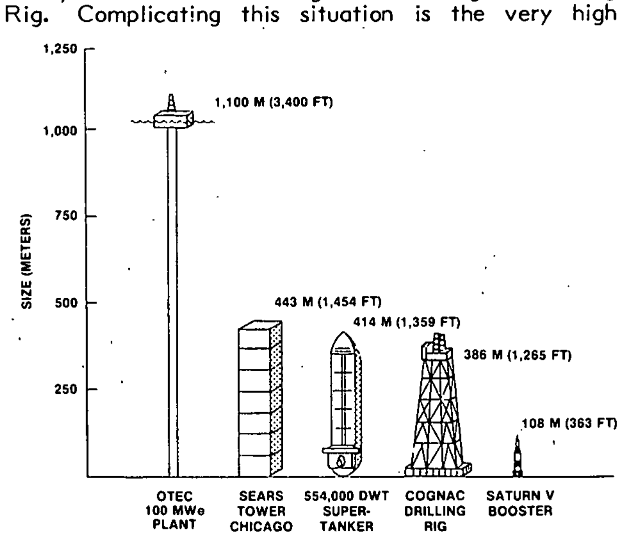

Fig. 1. Physical Relationship of OTEC CWP to Other Large Structures.

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aspect ratio (length/diameter) of proposed CWP designs, typically on the order of 100. Bodies of such aspect ratio tend to respond in a number of vibrational or "strumming" modes (Figure 2). Vibrational responses of significant magnitude imply cyclic stress reversals. This, in turn, warrants consideration of fatigue failure, particularly in the highly corrosive marine environment.

Recent research in the area of vortex induced loads applicable to the OTEC CWP has focused on problems related to amplified loadings due to "lock-on" of the pipe vibrational response to the vortex-shedding frequency[5], and the phenomenology of vortex shedding in shear flows[6].

In the case of "lock-on", a vortex-shedding frequency near one of the natural vibrational frequencies of the pipe will tend to induce sympathetic vibrations which will cause significant load amplification (Figure 3). As noted previously, vibrational responses of significant magnitude will cause cyclic stress reversals which make consideration of fatigue failure an important problem for the design engineer.

Studies of vortex shedding in the type of sheared current profiles frequently encountered in the ocean have concentrated on the definition of vortex correlation lengths (Figure 4) and their relationship to the current magnitude profile.

While these studies are adding a great deal to our knowledge of vortex shedding processes and consequent structural responses, it must be remembered that extrapolation of results from laboratory scales up to the size of an ocean structure of unprecedented size requires consideration of the limitations and caveats associated with such scaling.

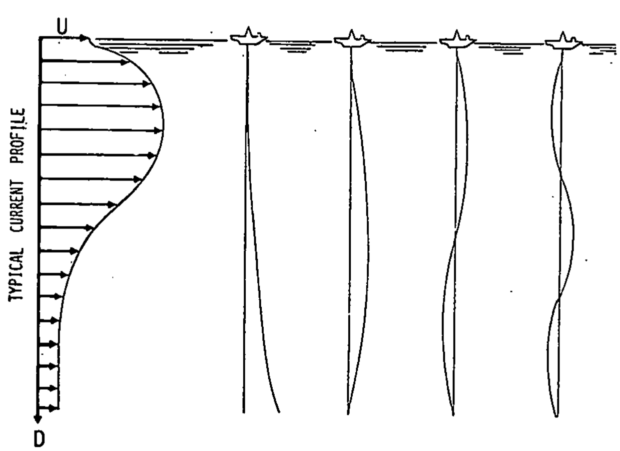

Fig. 2. Response Modes of OTEC Cold-Water Pipe.

Design Current Information

All of the above logically leads to consideration of the structure and variability of ocean currents. It has long been known that ocean currents of significant magnitude may exist down to great depths, induced by a number of geophysical processes such as:

- Direct driving by the wind,
- Inertial motion,
- Astronomical tides,
- Geostrophic adjustment.

All of the above processes can cause currents to vary in magnitude and direction over periods ranging from a few hours (tidal or inertial motion) to a few

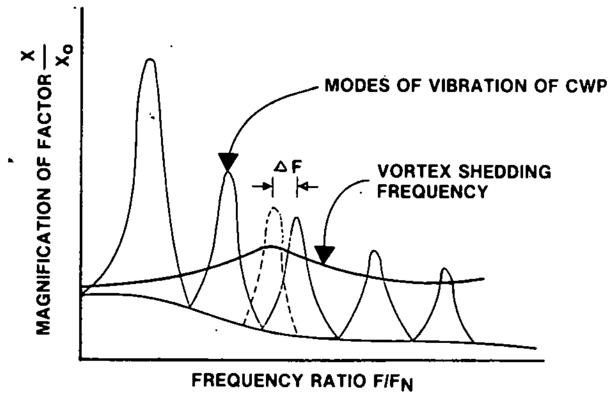

Fig. 3. Shift in Apparent Natural Frequency of CWP Correspondsing to Sympathetic Vibration Frequency of Vortex "lock on".

days or weeks (geostrophically driven motion). While some of these processes, in particular wind-driven flows, lend themselves easily to local modelling, others cannot be adequately simulated without resorting to global modelling of climatic/oceanic dynamics and interactions[7]. These mesoscale motions account for much of the seasonal climatological variability in oceanic flows.

How these interactions fit into the development of environmental design criteria for the cold-water pipe can best be explored by considering the present environmental information distributed by the Department of Energy (DOE) to the OTEC community and reviewing its application to one design concept that has been considered.

The OTEC design current information was originally developed by Bretschneider[8] and more recently revised by the Department of Energy[9]. The standard DOE environmental package contains surface currents for extreme and operational conditions and standard vertical current velocity profiles for four potential OTEC plant sites (Table I and Figure 5 respectively). Detailed referencing to the original data sources and prediction methods is contained in these two reports and will not be repeated here.

Of interest is the fact that the operational information consists of a single, unidirectional current profile, assumed to be invariant in time for the purposes of plant design. For many purposes, such

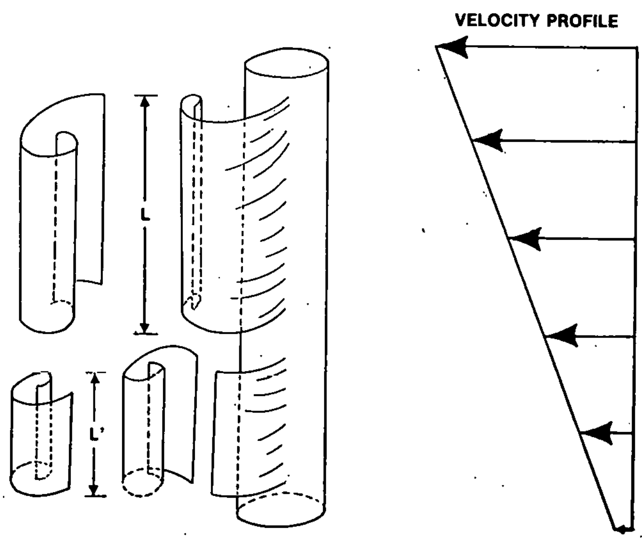

Fig. 4. Vortex Shedding Cells in a Shear Flow.

information is quite satisfactory. It will be shown in the succeeding example, however, that this is inadequate for CWP design calculations, particularly as from the standpoint of a cumulative damage fatigue analysis under the assumption of vortex "lock-on".

Example of Fatigue Analysis Using OTEC Current Profile

As noted in Hove, et al[10], current induced cyclic loads are associated with fluctuating lift forces perpendicular to the flow oscillating at the frequency of vortex shedding, and drag forces in-line with the flow, oscillating at twice the shedding frequency. This effect will be accentuated when "lock-on" occurs; that is, when the natural frequency of one of the response modes of the pipe "locks-on" to the vortex shedding frequency (see Figure 3).

Giannotti & Buck Assoc.[11] have performed preliminary analyses which indicate that a number of pipe configurations will experience "lock-on" in the second to fifth structural beam modes for the normal (operational) current profile for Punta Tuna. The configurations considered included a broad range of diameters, wall thicknesses and materials (steel, concrete and GRP) for use with the 10 MWe Modular Applications Spar Platform.

A further result of the Giannotti & Buck report is that for normal current condition, the stresses associated with the unsteady drag forces can be several times larger than the 100-year storm rms wave induced bending stresses. The analysis of these cyclic loads followed that proposed by Hove, et. al[10] with the following assumptions:

- the flow is assumed to be correlated over the entire length of the CWP,
- the velocities used to determine the shedding frequencies (Strouhal number) are the maximum velocities of the normal and extreme velocity profiles,
- the entire pipe will be excited if a local vortex shedding frequency falls within ±30% of a CWP natural modal frequency.

Of greatest interest in this discussion are the fatigue analysis procedures used. By applying the normal (operational) current profile to their stress model, the resultant cyclic drag forces over the entire lifetime of the structure were determined. This was applied to the Palmgren-Miner theory of cumulative damage for fatigue analysis[12]. According to this theory, the fraction of useable specimen life consumed at each load level during a structure's life history can be simply summed, and failure can be predicted when the sum of fractional damage is equal to one. The S-N curves for steel in air and in salt water are shown in Figure 6. This curve presents the number of cycles of stress reversal (tension to compression) that can be undergone by a material before fatigue failure as a function of stress magnitude. A structural element that undergoes a greater number of stress cycles in its lifetime than the number indicated in the S-N curve is assumed to fail at some time shorter than its design lifespan. The lifetime of the member is indicated by a parameter called the "D-factor" (Damage Factor) which is the ratio of the number of cycles determined to occur over the member's lifespan to the maximum number permissible on the S-N curve. Thus, a D-Factor of one or greater indicates a structural element that will probably fail in fatigue prior to its design lifespan. It is evident that, due to the corrosive nature of sea water, a much lower number of stress reversals is needed to cause failure in the marine environment than in air.

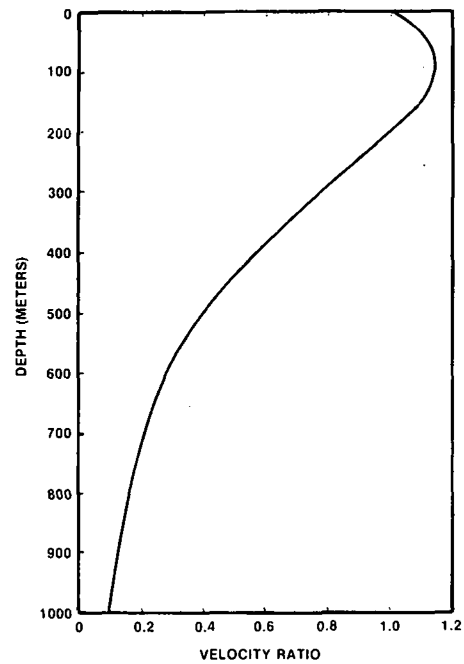

Fig. 5. Normalized OTEC Environmental Current Profile.

The results of the fatigue analysis for selected CWP designs are presented in Table 2 (after Giannotti & Buck Assoc.[11]). As can be seen from this table, cumulative damage, as indicated by the "D-factor" in this table, is much greater than one for a number of the cases considered. This would imply failure from corrosion fatigue for these cases. In the most extreme case, the cumulative damage number of greater than 100 implies a total lifetime of only 3 months, or less, assuming a planned lifetime of 30 years.

Table 1
Operational and Extreme Surface Current Magnitudes (cm./sec.) for Four Potential OTEC Sites

SITE	V_s (Operational)	V_s (Extreme)
Keahole Point	58.0	99.0
Punta Tuna	62.2	124.0
New Orleans	69.0	124.0
Florida	128.0	186.0

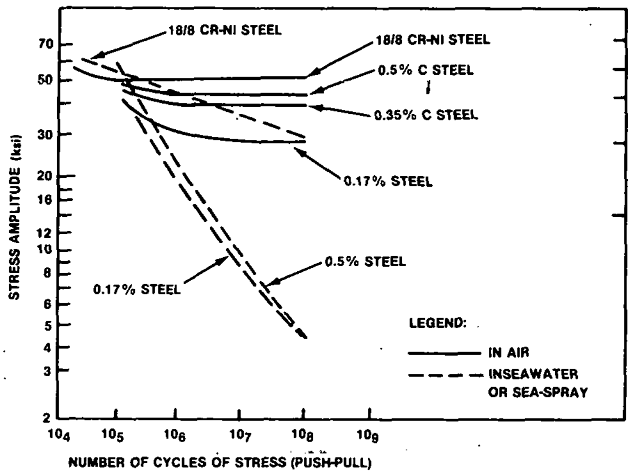

Fig. 6. S-N Curves for Steel on Air and in the Marine Environment.

Definition of Environmental Data Needs

We see, then, from the previous example, that indications are that unsteady lift and drag loads due to vortex shedding may cause catastrophic failure of the CWP within a very small portion of the expected lifetime of the structure due to corrosion fatigue. Admittedly, the validity of the preceeding analysis is qualified by a number of assumptions and generalizations relating to the environment and to the phenomenon of vortex shedding. But the assumption of a steady, unidirectional sheared current profile acting continuously throughout the life of the cold-water pipe is not the least of these. In order to make the current information reach at least the level of completeness of the opera- tional wave information presently available in the DOE environmental package, and allow it to approach the sophistication of the models to which it is being applied, the following information is a minimal necessity:

- Statistical distribution of currents at various depths,
- Information on directional variation of currents with depth, including statistical distributions.

For future developmental work, in particular, for application to the 3-D thin-shell model presently under development, more detailed information of the following type will be necessary:

- Persistence data on the occurrence of currents likely to induce intensified lift and drag due to vortex lock-on,
- Directional persistence information at various depths,
- Directional coherence of the currents as a function of depth.

It can be argued that such information is presently being developed through the OTEC Oceanographic Data Collection Programs run by Dr. R. Molinari and his associates at Atlantic Oceanographic and Meteorological Laboratories (AOML). However, while such information will indeed be of great importance in the future to model calibration and verification, it is not presently of sufficient volume or duration to address the needs posed herein. To fill these needs, a systematic approach combining hindcast analyses, use of available measured data and integration of the results of present research programs directed towards the better understanding of the vortex-shedding phenomenon is called for. As will be shown below, sufficient information presently exists as inputs to such a methodology that it can be accomplished within a time frame that is responsive to the needs of the OTEC community and can result in more reasonable, cost effective design decisions.

Table 2
Summary of CWP Fatigure Analysis Using Palmgren-Miner Rule for 17% Steel, Punta Tuna Site, 30 Year Expected Life
(After Giannotti & Buck Associ.[11])

DIAMETER, FT.	WALL THICKNESS, FT.	CUMULATIVE DAMAGE DUE TO WAVES ONLY (D FACTOR)	CUMULATIVE DAMAGE DUE TO WAVES AND CURRENTS, WITH LOCK-ON (D-FACTOR)
30[1]	0.07	.0047	81.075
	0.1	.0028	29.81
	0.15	.0142	9.58
50[1]	0.07	.016	>100
	0.1	.0028	>100
30[2]	0.07	3.4×10^{-8}	1.3×10^{-3}
30[2]	0.15	3.99×10^{-7}	4.1×10^{-7}
15	0.07	2.3×10^{-11}	>100

1. Steel in seawater or sea-spray   2. Steel in air

Proposed Approach

Other than the presently ongoing NOAA Oceanographic Data Collection Programs being carried out by AOML, the most extensive informational source of environmental data is the National Climatic Center (NCC) in Asheville, N.C. It is possible to obtain long, continuous time series of coastal wind records from meteorological stations for many locations in the U.S. and its possessions, including the Puerto Rican coast and the Hawaiian Islands. Not only do these time series present an opportunity to obtain meteorological forcing information for development of ocean current statistics, but they also may provide more reliable statistical wind information than that presently available in the DOE Environmental Package, which was developed from the U.S. Navy's Summaries of Synoptic Meteorological Observations (SSMO's) utilizing thousands of reports of wind speed and direction from ships in transit.

After obtaining this wind information, it may be applied to a wind-driven current model of the Ekman type[13]; that is, a single point model of pure wind drift. This model may be either a steady-state model such as that developed by Ekman or one of the more complex non-steady state models, such as that recently applied to drifting of spilled oil by Warner, et al[14]. The result of such an analysis will be a time series of wind-driven currents with both magnitudes and directions specified at a number of depths. A crucial part of the proposed approach is an assessment of the relative applicability of steady and non-steady state models to the development of operational current statistics for CWP design purposes.

But, as already discussed, the wind-driven component of the current is only one of a number of constituents generated by a complex group of geophysical processes.

Tidal current information is also reasonably straightforward to develop. In the absence of actual time series of current meter data at the proposed sites of sufficient duration to allow tidal harmonic analysis, it is possible to generate fairly reliable currents from simple one-dimensional models and coastal tide gage data.[15]

Longer term mesoscale motions, such as inertial motions, cannot be adequately predicted without resorting to large-scale oceanic/climatological interactions modelling. It is possible, however, to determine annual average or even seasonally averaged values for these currents utilizing the same hydrographic data base used by Bretschneider[8] in the development of the OTEC Environmental Data Package.

The information developed above may then be combined into an overall current time series from which cumulative probabilities of currents and persistence of currents of various critical magnitudes may be developed for the currents at various depths. This process is shown schematically in Figure 7, and the formats of presentation of the probability distribution and the persistence distributions are presented in Figures 8 and 9 respectively.

In Figure 8, the matrix elements would consist of frequencies (in percent) of current magnitudes at the indicated depths. In Figure 9, for the current speeds indicated across the horizontal axis and probabilities on the vertical axis, the matrix elements would consist of the number of hours over which the current magnitude would be less (or greater) than a certain lue. This information would be used to assess the k of currents conducive to vortex "lock-on" rsisting for a period of time long enough for the pipe respond. Before this determination could be made,

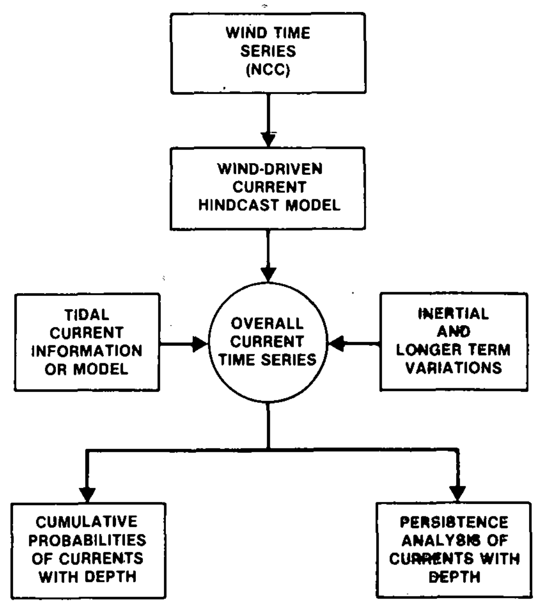

Fig. 7. Systematic Development of Statistical Ocean Current Information.

FREQUENCY DISTRIBUTION OF CURRENT SPEEDS

DEPTH (M) \ CURRENT SPEED (CM/S)	0-20	21-40	41-60	61-80	81-100	101-120
0-100						
100-200						
200-300						
300-400						
400-500						
500-600						
600-700						
700-800						
800-900						
900-1000						

Fig. 8. Matrix Form of Frequency Distribution of Current Speeds.

PERSISTENCE DISTRIBUTION OF CURRENT SPEEDS FAVORABLE TO VORTEX "LOCK-ON"

PROB (%) \ CURRENT SPEED (CM/S)	0-20	21-40	41-60	61-30	81-100	101-120
10						
20						
30						
40						
50						
60						
70						
80						
90						

Fig. 9. Matrix Form of Persistence Distribution of Current Speeds.

however, an assessment is needed both of the current magnitudes which would tend to induce "lock-on" and more detailed information on vortex correlation lengths in sheared flows on oceanic scales.

Statistical representations of directional variations of the current with depth have not yet been addressed. As will be discussed below, directional variations may be a critical limiting factor with respect to vortex correlation lengths and "lock-on" phenomena. Further research is needed in the determination of maximum levels of directional shear that can occur without destroying vortex correlation.

Some Unresolved Problems

The current information generated by the procedure described above would provide a great deal more information to the designers attempting to formulate rational CWP design information. A few questions on the phenomenology of vortex shedding in the actual oceanic environment directly relate to the application of this data, however.

The first of these concerns the response time of the pipe to increased vortex excited loads. A body of the immense size of the CWP, acting within a highly viscous medium, and constrained by structural and fluid damping, may not react to vortex "lock-on" in a time period short enough to allow full-scale resonant structural response. Some qualitative estimate of this response time could be developed by application of a time-domain model to a number of proposed CWP designs. From this, the susceptibility of the pipe to the magnified stress levels associated with "lock-on" may be at least qualitatively determined.

Although the question of vortex correlation length is being investigated in the laboratory, a great deal of care and consideration should be taken prior to attempting to apply the results developed on such small scale studies to the ocean environment.

Finally, serious consideration must be given to directional variation in ocean currents as a function of depth (Figure 10). It is not at all clear, for instance, how this variation will relate to the length scales of vortex correlation and of structural response modes. It could well be, based on physical reasoning, that amplified loads induced by vortex "lock-on" will be inhibited if directional shear increases beyond some critical level. Again, more caution in design assumptions and further research into the physical processes are called for.

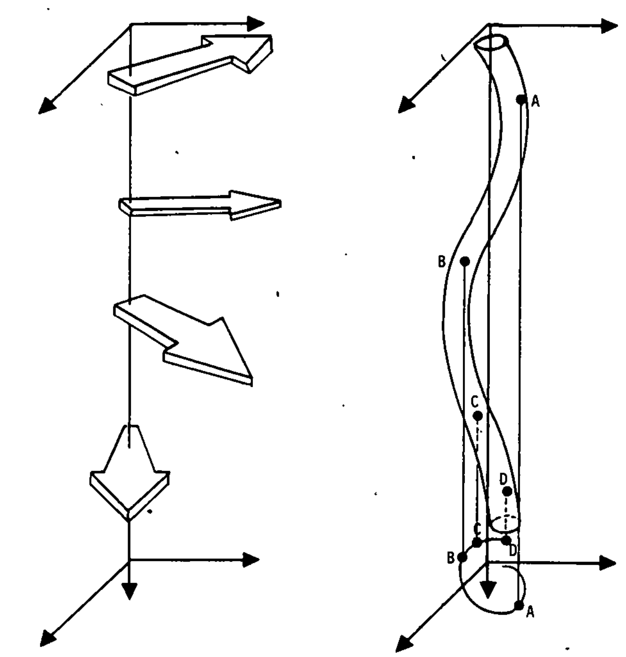

Fig. 10. Response of CWP to Complex Current Profile.

Summary and Conclusions

It has been shown that the current information in the presently available DOE environmental package for OTEC design is inadequate from the standpoint of cold water pipe design. At this point, it is feasible to consider developing a statistical basis of information, similar to the statistical wave height information in the present environmental package. Such information would provide an opportunity to assess the potential for critical vortex shedding loads in terms of their statistical distribution over the lifetime of the structure. It is also possible to develop persistence information from the hindcast analysis described in the previous section. This information would provide an assessment of the potential for magnified stress levels due to "lock-on" action of the vortices.

In summary, then, it is necessary to assess our ocean current information needs in terms of certain definable temporal and spatial scales of interest. The time scales of particular importance are scales over which the major variations in currents occur. These are:

- Tidal scales,
- Inertial Scales,
- Atmospheric System Scales,
- Scale of response time to CWP system "lock-on".

The spatial scales relate to the scales of pipe/fluid interaction, such as:

- Vortex correlation length,
- Modal Response wavelength,
- Scales of vertical current shear,
- Scales of vertical directional coherence of the currents.

Such information can be developed by utilizing presently available hindcasting techniques and can, on a seasonal basis, provide much more detailed environmental input information to CWP designers as well a those concerned with deployment and maintenance scenarios.

Acknowledgements

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DISCUSSION

D. Evans, Evans/Hamilton: In regard to the measurement of current data for the OTEC sites, many of you heard the paper given yesterday by Bob Molinari in which the current measurement program was outlined. Generally, there are data being acquired at every site at the Nobel Bay Area they are ... one year data at Tampa, about a half a year data in the South Atlantic [and] if they retrieve the mooring current array, they will have the ERA data. I agree with Mr. Pompa that there is a scarcity of data, but there is a program to acquire it. The environmental data package will reflect [the data] as soon as it is acquired and processed. Secondly, we don't anticipate that design type data can be obtained and measured in the time frame of 1 to 1½ years. This has been provided in the past by Dr. Bert Snyder. In a paper given by Dynalysis of Princeton, they indicated that a numerical hydrodynamic model has been set up of the Gulf of Mexico. It is a very integrated, 20-layered, sigma type model. They mention that they can force the model — using hurricane-design-type wind fields — and develop the multilayered velocity profile that would be used for the major data that are being acquired at the site. A lot of the data are in the mill and can be generated — it is a question of time. One of the questions for Mr. Pompa — one of the things that people measuring the data would like to know — is, at what frequency are data particularly needed in response to your particular problem of vortex shedding? At present the interval between measurements is 20 to 30 minutes.

J. Pompa: I did not quite understand one thing that you said. Did you say that you do or do not expect design information to come out of that program in the next 1 to 1½ years?

D. Evans: At present, it is not under contract — it would be an additional work effort.

J. Pompa: Addressing first the points that Mr. Evans makes about the present and future availability of actual ocean current measurements (as opposed to the hindcast information that I proposed in my presentation), I mentioned in my presentation that, while we are aware of the information being generated by Dr. Molinari's group, the present needs of the cold water pipe design teams are immediate and, it seems, on a time frame much shorter than this information is being made available to the community. Also, the current data derived by the program that I have proposed herein, because of the availability of longer term statistical climatological information, may have greater statistical generality than a year or 2 years of data. As a design information package, I feel that the present information is somewhat deficient and should be replaced with something more in keeping with the needs of the CWP design community.

To address the question about the multi-layered modelling efforts that are being undertaken in the Gulf of Mexico, two points need to be made. I'm sure that Mr. Evans is fully aware that such models are quite expensive to run and that, because of this limitation, they do not lend themselves to the question of long-term statistical information of the type that I have described. Further, the importance of the informational needs that I have described are not relevant to hurricane current generation, a very transitory phenomenon, but to the long-term operational currents, which may induce continuous cyclic loads and cause fatigue failure. Considering the planned 30-year life of the structure, this is a much more critical consideration than hurricane design currents.

7. OTEC POWER TRANSMISSION CABLES

A THEORETICAL STUDY OF TECHNICAL AND ECONOMICAL FEASIBILITY OF BOTTOM SUBMARINE CABLES FOR OTEC PLANTS

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Abstract

This theoretical investigation of the technical and economical feasibility of submarine cables connecting OTEC (Ocean Thermal Energy Conversion) plants to shore transmission grids began with a survey of the state of art in order to select bottom cables most suitable to transmit 250 MW per plant. For the longest links, consideration was given to transmitting 500 MW per plant and to the feasibility of cables converging from 12 (or 6) plants of 250 (or 500) MW each to an intermediate tie-point from which the aggregate power would be transmitted to shore by the minimum number of cables. Results of the first stage of the work enabled DOE to decide that solutions involving tie-points should be droppъd. Moreover, attention was focused on 400-MW plants off Tampa, Fla., and New Orleans, La., and on a 100-MW plant off Puerto Rico. The results indicate that the most suitable bottom cables are: self-contained, oil-filled cables at 138-345 kV ac for the short transmission links and paper-impregnated cables at ±250 kV dc for the long transmission links. Attention was also given to laying operations, including embedding of cables wherever the water depth is less than 300 ft. A list of critical issues which require experimental work in the ongoing program is presented.

Introduction

The transmission of power from Ocean Thermal Energy Conversion (OTEC) plants to shore-based grid facilities requires several significant advances in the state of art for submarine cables. Because of the unique requirements for the cable suspended from the platform to the ocean floor versus the ability to transmit power over long distances once the cable touches the bottom, the problem was split into two parts - riser cables and bottom cables. This paper covers only the work conducted for the bottom cable evaluation. The riser cable system is presented in another paper.[1]

After completing a literature survey and an assessment of the mechanical cable construction requirements for deep water submarine cables, it was tentatively determined that, for the power levels to be investigated, 345 kV ac and 138 kV ac voltages were suitable for 400 MW and 100 MW, respectively, over the short distances for island applications and ±250 kV dc voltage for 400 MW over longer distances. The representative sites for OTEC, where the necessary water depth of 900 - 1800 m (3000 - 6000 feet) can be obtained, were selected to be 3.7 km from shore for island applications (as Puerto Rico), and 150 - 280 km from shore for U.S. continental sites such as New Orleans and the West Coast of Florida.

Figure 1 indicates the present state of art for distance and depth and the specific needs of OTEC bottom cables. It is seen that much of the distance for the longer links is in waters not deeper than 90 m (300 feet). Because of the risk of cable damage due to trawler nets or ships anchors in shallow waters, it has been stipulated that the bottom cables be buried to a 2.5 m (8 ft) cover where the water depth is less than 90 m (300 ft).

Approach

A literature survey of existing submarine cable installations resulted in the following list of cable types:

High-pressure, oil-filled, pipe-type cable	(HPOF)
Compressed-gas-insulated cables	(CGI)
Extruded-insulation cables	(E)
Conventional paper-impregnated cables	(PI)
Self-contained, oil-filled cables	(OF)
Paper-impregnated, gas-pressure-assisted cables	(PG)
Synthetic-film-insulated cables	(S)

Based on the required operating characteristics, a set of criteria for consideration in a further more refined evaluation was developed. The following basic feasibilities were confirmed: operation at the ac voltages (≥ 138 kV) required for the short links, operation at the dc voltages (≥ ± 200 kV) required for the long links, and ability to lay, recover and repair when links of exceptional lengths and water depths are required.

Compressed-gas-insulated and synthetic-film-insulated cables were dropped because they were not sufficiently developed as yet and are not particularly suitable for the OTEC power levels. Paper-impregnated, gas-pressure-assisted cables were considered for the shallow water sections of long dc links[2] but were discarded because, for the conditions of the representative sites considered, they are less attractive than the simpler conventional paper-impregnated cables. High-pressure, oil-filled, pipe-type cables were also evaluated as an alternative to self-contained, low-pressure oil-filled cables, but were dropped because it was apparent that the pipe-type cables would require more R&D work without providing any significant technical or economical advantages.

Thus, the most suitable cable types are: self-contained, low-pressure, oil-filled (OF) cables and extruded-dielectric-insulated (E) cables at 138 - 345 kV ac (for powers 100 - 400 MW) and short transmission links; and conventional solid, paper-impregnated (PI) cables at ±250 kV dc for long transmission links.

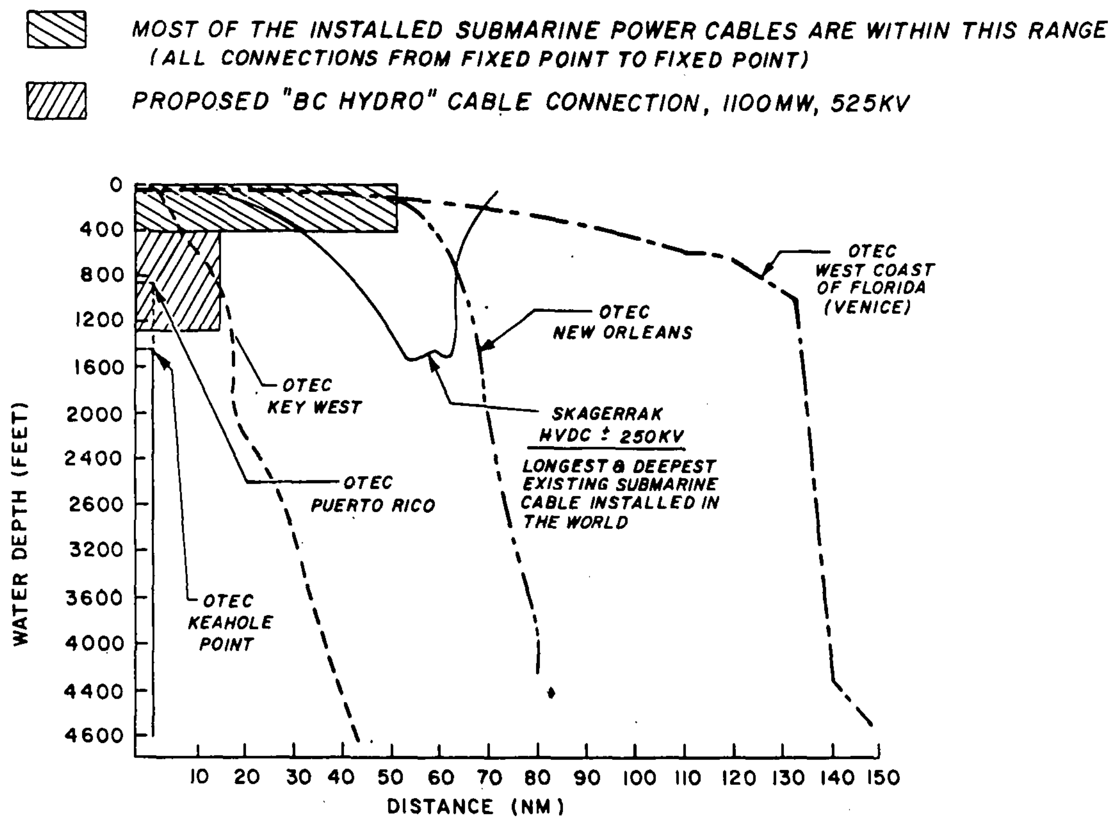

Figure 1. State of the art for submarine transmission cables.

Table 1 summarizes some basic data on the bottom cables considered for the various representative OTEC sites. No distinction is made for the conductor cross-section between embedded and non-buried portions of the route for the thermal capacity calculations. As a matter of fact for dc cables, the temperature drop across the insulation is the only concern of practical interest, whereas in the case of ac cables (which are only used in short links) it is not advisable to adopt a reduced conductor cross-section for the unburied portion of the cable and risk the complexity of introducting a flexible factory joint to mate dissimilar cable sections. In addition, one spare cable has been included in all the analyses in order to meet a minimum level of reliability for a single circuit connection to shore. From an electrical point of view, only extruded insulation cables up to 345 kV ac require R&D work. From a mechanical point of view, all candidate cables require R&D because of the exceptional depth of laying; in fact, as noted in Figure 1 no power submarine cables have been laid in waters deeper than 550 m (1800 ft).[3]

The main construction characteristics of all cables are given in Table 2; parameters and operating data in Table 3; and estimated costs in Table 4. These topics will be described in more detail in following paragraphs.

Bottom Cables for Puerto Rico

The thermal resource for OTEC plants near the island of Puerto Rico can be obtained extremely close to shore, i.e. within 3.7 km (2mi) corresponding to a water depth of 1200 m (4000 ft). Concerning the seabed for the Puerto Rico site, it was assumed to have a regular profile to be composed essentially of sand and clay, therefore this site offers favorable characteristics from the point of view of laying and embedding operations. In actual practice, a complete bottom survey of the selected route would be required to ascertain specific soil composition and possible abrasion conditions as well as an understanding of soil resistivity for ampacity calculations.

Case a: 400 MW Plant

Consideration was given first to 230 kV ac transmission, but it was not capable of transmitting the required 400 MW with a single circuit. Therefore, 345 kV ac was chosen for the 400 MW plant. At this voltage, two solutions, both involving a single cable circuit, are in principle possible: self-contained oil-filled (OF) cables, for which 525 kV ac underground and 400 kV ac submarine installations already exist; and cross-linked polyethelene (XLPE) cables, for which important research work up to 345 kV ac sponsored by the Electric Power Research Institute and DOE is in progress in U.S.[4]

The XLPE solution is based on the assumption that the aforementioned experimental work can be concluded successfully not only as far as the qualification tests are concerned, but also with reference to cable reliability (assessed using Weibull's statitistical criteria) and appropriately to the technology transfer of experimental results to submarine cables laid in deep waters. According to Ref. 4, the insulation wall of an XLPE cable operating at 345 kV ac must be 26.2 mm (1.03 in).

Table 1. Bottom cables suitable for representative OTEC sites

	Power MW	Distance km	Length to be buried km	Maximum depth,m	No. of cables[a]	kV	Case	Type	Conductor section, kcmil
Puerto Rico	400	3.7	1.5	1200	4	345 AC	a.1	OF	1400
							a.2	XLPE	1250
	100	3.7	1.5	1200	4	138 AC	b.1	OF	440
							b.2	EPR	1250
New Orleans	400	150	110	1200	3	250 DC		PI	1400
West Coast of Florida[b]	400	275	160	1200	3	250 DC		PI	1400

a. One circuit will be used with one cable added as a spare.

b. The two alternatives for 1800 and 1200 m differ only in the details of the armour.

Under these high electric stress conditions, it is mandatory that the insulation be protected from ingress of moisture by applying a hermetic lead sheath over the outermost insulation shield.

Since the OF cables offer high electrical safety margins, whereas the electrical reliability of XLPE cables at 345 kV has still to be assessed, and the OF cables can be manufactured without joints, whereas the industrial technology for 345 kV XLPE cables has not yet been established, the OF cables are preferred. Although the OF cables are more expensive, the cost difference is within the 10% error band of present estimating accuracy.

Case b. 100MW Plant

If only 100 MW must be transmitted, the most suitable voltage is 138 kV ac, and the submarine cables can be either the OF type or the extruded insulation type. In the latter case reference is made to ethylene propylene rubber (EPR) cables, which are routinely manufactured by several cable organizations for voltages up to 150 kV ac, in underground installations and, so far, up to 30 kV in submarine installations. It is estimated that EPR cables can safely operate in submarine installations at 138 kV without addition of a metallic sheath impervious to water provided the maximum electric stress is maintained lower than 5 kV/mm (127 kV/in). Here again the ratio between insulation and conductor diameters is designed to be near the optimum value.

Since EPR cables cost less than OF cables and have acceptable electrical reliability, they are attractive for 100 MW plants. However use of EPR cables is deemed unwise if a prototype 100 MW plant at Puerto Rico is considered to be a forerunner of commercial plants up to 400 MW capacity. In this case it is obviously advisable to use the same cable type (OF) as envisaged for the anticipated larger plant in order to obtain invaluable operating experience with them.

Bottom Cables for New Orleans

Due to the long distance involved (150 km), only dc cable solutions were considered. Based on the plant output of 400 MW, the most suitable transmission voltage is in the range of $\pm$250 kV dc.

Since HVDC extruded insulation cables are still in an early stage of research, they were not considered in the present analysis. In principle, the problem can be solved by means of conventional paper-impregnated (PI) cables; paper-impregnated, gas-pressure-assisted (PG) cables; or OF cables. However, PG cables face important mechanical problems in connection with the exceptional depth of laying. Similarly, OF cables encounter significant hydraulic limitations for the long length required. It follows that the most suitable solution, according to the present state of art, is PI cables.

Bottom Cables For West Coast Florida

For this representative site the conditions are similar to the New Orleans case, but the distance to be covered is much longer (275 km versus 150 km) and an alternative is required for maximum water depth of 1800 m (6000 ft) in addition to 1200 m (4000 ft). Power and voltage being unchanged, the dc cables described in the next-to-last colum of Table 2 can be used when the water depth is 1200 m. For the alternative concerning 1800 m water depth, the cable dimensions must be modified as shown in the last column of Table 2. The approximate costs of the West Coast Florida link are given in Table 4 for the 1800 m depth case; the 1200 m case is slightly less expensive.

General Considerations - Cable Designs

In the present analysis, it was assumed that the mechanical problems due to laying cables in water depths up to 1800 m can be solved by applying an antitwist armour to each cable consisting of two layers of flat steel wires (with high tensile strength) applied with opposite lay directions. This type or armouring is undergoing research work inside Industrie Pirelli Laboratories independently of OTEC applications.

Table 2. Main constructional characteristics of bottom cables in maximum water depth of 1200 m

	OF	XLPE	OF	EPR	PI	
	Puerto Rico 400 MW	Puerto Rico 400 MW	Puerto Rico 100 MW	Puerto Rico 100 MW	W. Florida 400 MW 1200m	1800m
Copper conductor: section, kcmil	1400	1400	440	1250	1400	
diameter, mm	32.8	30	21	30	34.3/28.4 (oval)	
Insulation thickness, mm	21	26.2	9.5	25.5	15.5	
Lead sheath thickness, mm	3	3.3	2.2	copper tape screen	2.75	
Antiwist armour in two layers of flat steel (high tensil strength) wires: thickness of each layer, mm	2.5	3	3	2.5	2.5	4
O.D. of cable, mm	115	125	77	115	101/95	107/101
Weight (in air), kg/m	36.7	40.5	18.2	26.5	89.9	35.2

The problem of flexible factory joints was considered only for the long links of New Orleans and West Coast Florida. In fact, the short links for Puerto Rico do not require flexible joints (other than for repairs) if the solution consists of self-contained oil-filled cables, whereas for XLPE cables it is necessary to wait for the relevant manufacturing technology to be developed. Referring to the long links and the existing manufacturing and impregnating tank facilities, it is possible to produce factory lengths in the range of 50 km. Examining Figure 1, if one starts from

Table 3. Parameters used and resulting cable operating ratings for each site.

	Puerto Rico 345 kV XLPE 400 MW	Puerto Rico 345 kV O.F. 400 MW	Puerto Rico 138 kV EPR 100 MW	Puerto Rico 138 kV O.F. 100 MW	N.Orleans ±250kVdc P.I. 400 MW	W.Florida ±250kVdc P.I. 400 MW
ASSUMED PARAMETERS						
Embedding depth of cable, m	2.5	2.5	2.5	2.5	2.5	2.5
Maximum ambient temperature, °C	25	25	25	25	24	24
Distance between cable centers, m	>10	>10	>10	>10	10	10
Thermal resistivity of soil, °Cm/W	0.7	0.7	0.7	0.7	0.7	0.7
Thermal resistivity of dielectric, °Cm/W	3.5	5	5	5	7	7
Thermal resistivity of anticorrosion sheath and jute, °Cm/W	4	4	4	4	4	4
DC resistance of the conductor at 20°C, ohm/km	0.0286	0.0256	0.0283	0.0815	0.0256	0.0256
Dielectric loss factor (tgδ)	0.0004	0.0025	0.01	0.0025	-	-
Dielectric relative permissivity (E)	2.5	3.5	2.9	3.5	3.8	3.8
Capacitance of cable, uF/km	0.14	0.24	0.169	0.31	-	-
Load factor	1	1	1	1	1	1
Maximum permissable conductor temperature, °C	90	90	90	90	50	50
OPERATING DATA						
Rated voltage, kV	345	345	138	138	±250	±250
Rated current, A	744	744	465	465	800	800
Frequency, Hz	60	60	60	60	-	-
Number of operating cables	3	3	3	3	2	2
Load factor	1	1	1	1	1	1
Conductor temperature, °C	78	85	51	73	50	50
Conductor losses, approx. W/m	21	18.8	7.6	21.3	18.3	18.3
Dielectric losses, approx. W/m	1	9	3.7	1.8	-	-
Sheath losses, approx. W/m	13.8	21.8	1	9.8	-	-
Armour losses, approx. W/m	33.3	25	18.3	18.3	-	-
Maximum electric stress, kV/mm	13.2	13.8	5	12.5	21	21

the platform and moves toward shore for 50 km, it becomes obvious that flexible factory joints will not be required in water depths in excess of 60 m to 245 m for New Orleans and the West Coast Florida, respectively. However, there is the possibility that a repair must be carried out in the event of cable failure in waters deeper than those mentioned above. Considering the two route profiles, one realizes that, if a spare cable 20-25 km long is available, repair joints can be confined to maximum depths of only 400 m (1300 ft) for the New Orleans and West Coast Florida sites. Presently, flexible repair joints have been experimentally developed for self-contained oil-filled cables for laying in depths upt to 400 m. The basic principle used for this joint can be applied to the flexible joints for the paper impregnated cables contemplated for the OTEC project. However, suitable experimental tests will be needed to demonstrate this point.

The problem of optimizing cable dimensions is affected by many factors at the time of design. In addition, any increase in cable size and weight can give rise to more difficult laying and/or recovery operations. It was, however, decided to investigate whether a reduction in cable power losses due to an increase of cable size could result in a more economical system. This analysis made it clear that with reference to present loss evaluation, the minimum conductor size capable of satisfying the continuous current requirements is likewise the optimum choice economically.

Table 4. Transmission Costs in Millions of dollars

	Cables	Assessories	Transport	Installation	Embedding	Total
Puerto Rico						
400 MW, OF	2.3	0.5	0.3	0.9	1.6	5.6
400 MW, XLPE	2.0	0.4	0.3	0.9	1.6	5.1
100 MW, OF	1.7	0.2	0.3	0.9	1.6	4.7
100 MW, EPR	1.0	<0.1	0.3	0.9	1.6	3.8
New Orleans						
400 MW, PI	38	0.1	2.5	2.5	39	82
West Florida						
400 MW, PI	70	0.1	2.9	2.7	56	132

General Considerations on Laying and Embedding

To lay cables in water depths in the range 1200-1800 m (4000-6000 ft) involves major design considerations for the cables and for the ship handling and laying equipment. The cables, as already mentioned, require a special antitwist (torque balance) armour. In case of conventional paper impregnated insulation, the conductor must be suitable ovalized to accommodate, along preferential directions, the volume expansion due to thermal cycles. Joints are potentially weak points even when flexible and manufactured according to the best techniques and it is essential to reduce their number by manufacturing the cables in the longest possible lengths.

Laying machines suitable for large power cables installed in water depths of 1200-1800 m must be developed in order to prevent damage to the insulation due to insufficient bending radii, excessive lateral pressures due to the fleeting rings, and so on.

Bending and torsional stresses of the cables during laying must remain within stipulated design limits, therefore proper choice of all bending and coiling diameters is essential. In certain cases, rotating platforms may be necessary to eliminate torsional stresses on the cable. The pulleys on which the cable pays out must be designed with exceptionally large diameter, and the braking equipment of the laying machine must be capable of tensions up to 70 tons (which is two to three times higher than present cable laying systems).

It must be borne in mind that any formation of kinks during cable laying and recovery operations is extremely detrimental to the safe operation of any type of paper-impregnated cable in that the paper tape separation at the sharp bend will precipitate an immediate electrical failure as soon as voltage is applied. In shallow waters, the problem comes not from laying out but from embedding operations and is more an economic than a technical problem. The lengths of cables to be embedded can be exceptional with a maximum in the range of 470 km for the three cables (one spare) envisaged for West Coast of Florida. At present, the embedding cost is extremely high and the time involved is very long. With one embedding machine, it would take years to bury the cables for the West Florida site based on an optimistic rate of one kilometer a day utilizing presently available techniques. Obviously, installation costs and duration will be reduced in the future because work is under way in several countries to develop more advanced systems. However, an accurate and precise survey of all possible cable routes must be carried out to identify preferred cable landings to minimize the impact of shipping lanes and trawler operations on cable damage. In any case, it is necessary that an exact knowledge of the physical characteristics of the sea bottom be available so that the correct choice of embedding procedures can be made. From the statistical data on the meterological conditions of the representative sites according to different seasons of the year, one can anticipate that in the January-May period it will, in principle, be possible to carry out the complex laying operations such as those involved in the OTEC Project.

Of course, it will be necessary to organize in advance all details of such operations and to rely upon accurate meterological forecasts.

Ongoing Program for the Bottom Cable

From the previous discussion it can be seen that the principal experimental work to be carried out in ensuing phases of the bottom cable are:

- Definition of the cable design details, with specific regard to the antitwist armour, so that the heavy mechanical stresses encountered during laying and recovery operations in very deep waters can safely be withstood.

- Development and test of flexible factory and repair joints, so that they can meet the laying and recovery operations at the anticipated water depths (less than 400 m (1300 feet)).

- Establishment of the requirements for the cable laying and recovery equipment, taking into account the exceptional water depths, the characteristics of cables and joints, etc.

- Investigation of the technological needs for recovery and repair operations, as well the design details of the relevant equipment.

- Development of the laying and recovery procedures for the special joint connecting bottom and riser cables (this experimentation is to be carried out jointly by manufacturers responsible for the two cables and the connecting joint).

After this experimental work has been concluded, a sea trial making use of a suitable cable length including factory and repair joints will be desirable in order to obtain full scale verification of all anticipated operations and mechanical stresses. Of course, what is said above applies to the particular cable and joints selected for each representative site which for all intents and purposes can be reduced to two cable types: self-contained, oil-filled, cellulose-insulated cables for ac and self-contained, paper-impregnated cables for dc applications. At this stage of the bottom cable program, the probability of success for the previously described experimental plan appears to be very high.

Substation Design

The landing points for the bottom cables will terminate in shore based electric substations. If the utility requirements were such that the load centers were somewhat remote from the shoreline, then over land transmission could be installed from the shore points to the load regions, and the substation facilities placed near ther. This additional cost element for grid connecting transmission line facilities on land has not been included in the various cost estimates. The shore substation for ac and dc requirements are well within the present state-of-art, and appropriate estimated costs are shown in Table 5.

While it is true that the shore based substation poses no unique development or engineering problems the facilities located on the OTEC platform must occupy as small a volume as practical so as to minimize space requirements on the vessel. For ac substation components, minimum volume can be achieved utilizing compressed SF_6 gas insulated equipment for buswork, switchgear, and instrumentation. The same cannot be said for the dc conversion equipment however. Compressed gas SF_6 equipment for converter valves, harmonic filters, smoothing reactors, and dc buswork is not commercially available, nor is this equipment likely to be developed in the private sector without some government stimulus to offset the R&D risks. Some R&D is currently in progress under sponsorship by the Electric Power Research Institute[3c], but this alone will not be sufficient to address all the major development work that will be required. In another related area, conventional power transformers are oil filled devices which can pose a significant fire hazard in a confined area, such as a floating vessel. The environmental ban on polychorinated biphenyl liquids for use as nonflammable fluids in transformers has left a gap because no acceptable substitutes have emerged even if these fluids could be used in the large power units required for commercial OTEC plants. A parallel project - employing compressed gas insulation, polymer film solid insulation, sheet windings, and separate ducted cooling for power transformers is currently in the early stages of R&D(4), but could significantly impact the OTEC program because of its inherent non-flammability.

Conclusions

Based on the analysis carried out during the Phase I study for OTEC bottom cables, the following conclusions can be drawn:

- From an electrical point of view, no major technical problems exist if use is made of ±250 kV dc paper impregnated cables for the long transmission links in the Gulf Coast region and 138-345 kV ac, self-contained, oil-filled cables for the short links needed for island plants. On the contrary, important R&D work is required if, for the short links, cross-linked polyethelene cables are chosen.

- From a mechanical point of view, the exceptional depths of laying require an extensive experimental evaluation of both cables and assessories, of laying procedures and equipment and of recovery and repair operations. The probability of success, however, is high.

- The costs incurred to protect the cables from external damage by means of embedment are high, as well as time duration involved in the burial operations. Although reductions are expected in the future as a result of development work under way in a number of other countries, it is important that the cable routes be chosen in such a way as to minimize the cable lengths requiring protection.

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Table 5. Substation Costs for Shore Connection to Electric Network

	Puerto Rico A. 345 kV OF	400 MW B. 345XLPE	Puerto Rico A. 138 kV OF	100 MW B. 138kV EPR	New Orleans 400 MW PI 250kV DC	W.C. Fla. 400 MW PI 250kV DC
System Voltage, kV	138	138	138	138	230	230
1. Landing Portion						
Site Preparation(1)	10	10	8	8	10	10
Manhole	15	10	12	8	12	12
Termination Structure	90	90	50	50	(2)	(2)
	115	110	70	66	22	22
2. Substation Portion						
Circuit Breaker(3)	275	275	100	100	N.A.	N.A.
Transformer	2000	2000	100	100	N.A.	N.A.
Other(4)	400	375	200	175	N.A.	N.A.
	2675	2650	400	375	-	-
3. Conversion Equipment(DC)						
Equipment(5)	N.A.	N.A.	N.A.	N.A.	12,800	12,800
Engineering	-	-	-	-	125	125
Subtotal					12,925	12,925
	2790	2760	470	441	12,947	12,947
4. Other						
Maintenance	2%/yr	1.5%/yr	2%/yr	1.5%/yr	6%/yr	6%/yr
IDC	16%	16%	12%	12%	32%	32%
Contingency	10%	15%	8%	8%	20%	20%

Notes
(1) Cost of land not included.
(2) For DC termination structure cost included in conversion equipment cost.
(3) Includes grounding switch and bus work and air break switch.
(4) Includes bus work, control, engineering, and miscellaneous.
(5) assumes unidirectional flow (ashore). All equipment indoors.

THE DEVELOPMENT OF RISER CABLE SYSTEMS FOR OTEC PLANTS

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Abstract

The riser cable system of an Ocean Thermal Energy Conversion (OTEC) plant links the platform to the high-voltage transmission cable on the sea bottom. This paper describes the results of the Phase I riser cable study and the development efforts to be performed during the next 4 years. Emphasis is placed on the design considerations for a 10-40 MW_e riser cable system including selection of cable type(s) for manufacture and testing, test criteria, production, suspension system design concepts and limitations, OTEC plant/riser cable interfacing, installation planning, and system design considerations. Also described are the overall plan and methodology for development of reliable riser cable systems for the 10-40 MW_e, 100 MW_e and 400 MW_e OTEC plants. Key elements of developmental testing are discussed in some detail. Specific costs are outlined for each of the proposed OTEC plant sites.

Introduction

The use of solar energy to provide a significant percentage of total power generated and supplied to shore for transmission through electric utility networks appears to be feasible through use of the OTEC concept. Since the Ocean Thermal Energy Conversion concept does not depend on diurnal cycles, it is the only major means of using solar energy without use of a man-made storage system to feed directly into a land-based power system. The electric power transmission riser cable system extending from the plant to the ocean floor is a key part in the OTEC transmission system to shore and is the subject of this paper.

In September, 1977, a Phase I study of the riser cable for OTEC was begun. This study was completed in January, 1979. A Phase II program was started on January 31, 1979 and is scheduled to be completed in January, 1983. This paper summarizes the results of the Phase I study and the first 4 months of the Phase II contract. For further detail the reader is referred to Refs. 1-4.

Phase I of the Riser Cable Development Program served primarily to identify crucial areas of design, chiefly for the ultimate, commercial-size OTEC systems. This work imposed bounds on the types of systems which could be employed for OTEC riser applications.

The major purpose of the Phase II program is to provide performance criteria for riser cable systems to be used for the 10-40 MW_e Modular Applications Plant. The riser cable system must comprise a highly reliable link in this first application of OTEC power delivery to shore.

Summary of Phase I Study Results

Work under the Phase I Study[3] identified important design considerations relating to OTEC riser cable systems generally, and with a concentration on plants for commercial applications. These OTEC systems will ultimately have a power rating of 400 MW_e. Seven major categories of work were addressed; each is summarized below.

Conditions Imposed on the System

The work focused on the identification of OTEC environmental characteristics that result in unusual requirements for the transmission cable riser. The study identified a number of key considerations. The mechanical loadings and motions imposed on OTEC riser cables are some of the primary design considerations. Mechanical loadings and motions in the cable come from basically four sources, including statics, current drag, strumming, and forcing by end motions.

Static tension in the OTEC riser cables may be in the range of 15 to 25 tons for cables weighing 6 to 8 pounds per foot in sea water. Current drag can cause the cable to swing and can increase tension in the cable by as much as 50% for the currents expected at OTEC sites. Strumming may increase drag by up to a factor of 5. Motions of the cable, such as caused by wave action on the plant, can result in substantial dynamic tensions in the cable, and will be responsible for most of the cycles of bending and twisting seen by the cable. These can be expected to be the greatest contributing factor to mechanical fatigue of the cable.

The growth of marine fouling may increase the mechanical loadings with time as sheer weight and drag are altered.

The terminations of the cable are prime locations for fatigue failure. Mechanical cycling causes concentrated loading at these points, therefore, terminations must be designed to limit these stresses.

The forces, in addition to axial tension, imposed on OTEC riser cables will result in bending and twisting of the cable structure. The cable will also be subjected to cyclic tension changes.

The importance of developing an understanding of the complex interactions that will occur on OTEC riser cables cannot be overstated. These cables will be subjected to the conditions summarized above in combination with thermal effects due to the electrical load and simultaneous subjection to high voltages that produce high electrical stress in the cable dielectric. These latter items (electrical stress and thermal effects due to the electrical current) are very important considerations. In fact, it is not mechanical fatigue of cable components that is the primary concern in the development of OTEC riser cables. Rather, it is the electrical/mechanical "fatigue" of components of the cable dielectric and shielding system that is the major consideration. Particularly, the phenomena that occur at the interfaces of dielectric and shielding layers during combined mechanical, thermal and electrical stressing are key items. In addition, the effect of these combined stresses on the integrity of the cable dielectric is a major consideration. This subject will be discussed more in later sections of this paper that deal with cable design and testing.

The environmental considerations for the OTEC riser cable system include corrosion and abrasion. The growth of marine organisms and/or the chemical effects on the outer cable components could lead to significant corrosion and are important items considered in the design of the cable. Likewise, the cable must either be designed to withstand abrasion at contact or touchdown points or the system must be designed to avoid degradation due to abrasion.

The installation, repair, and recovery of OTEC riser cables in 4000 to 6000 feet of water is a critical topic addressed in the Phase I study. Previously, no high-voltage power cables have been layed in depths greater than 1850 feet. The OTEC riser cables are expected to be from 3.5 to 6.5 in. in diameter and weight 6 to 8 lbs./ft. in sea water. The laying of submarine power cables is an art requiring experienced personnel, special equipment and vessels, and careful engineering and planning. Thus, due to the specialized nature of cable laying and the unprecedented nature of the OTEC riser installation, this is a key area requiring considerable planning. Sea trials may be required to perfect installation techniques and special installation equipment will be necessary. Tension imposed on the cable during installation, repair or recovery will likely exceed that occurring during normal operations. Bending radius of the cable during installation will almost certainly be smaller than that occurring during the operating life of the cable.

Suitability of Candidate Cable Systems

In a previous paper[1] we discussed the basic characteristics of four primary cable types:

Compressed-Gas Insulated (CGI)
Pipe-type High Pressure Oil-Filled (HPOF)
Self-contained Oil-or Gas-Filled (OF)
Extruded Dielectric (ED)

The results of the Phase I study showed that only the OF and ED types appeared to be viable candidates. The CGI and HPOF types employ a large diameter (8 in.) pipe which is either flooded with oil or gas. While both of these types are viable in land-based stationary systems, they both have significant drawbacks for use on OTEC. The ability of such systems to withstand the cyclic mechanical loading and flexing is doubtful. In addition it is evident that a rupture of the pipe caused by damage would be disastrous to either system in totality and would likely be irreparable. Installation of such systems to a moving OTEC platform is not considered feasible and would require a much more difficult procedure.

Self-contained oil-filled cables with laminated dielectrics are considered to be worthy of further evaluation for OTEC. The laminated dielectric may be either high grade paper and oil or might possibly be a synthetic material in oil. Another simpler version of this basic cable type, the solid-paper impregnated cable (without pressure) is a suitable candidate for DC systems having voltages up to $\pm$ 266 kVDC but is not considered suitable for AC voltages above 69 kV. The self-contained oil or gas filled cables have been used successfully in more traditional submarine cable crossings and possess basic capability to withstand operating voltages in excess of the 345 kVAC or up to $\pm$ 400 kVDC levels predicted for the 400 MW_e commercial OTEC plant. All laminated oil or gas-filled cables must have a hermetic sheath to prevent any ingress of water, since such water entry into the dielectric system will result in failure. A primary unknown concerning self-contained cables is the ability of the laminated dielectric system to withstand the cyclic loads and motions of OTEC. Significant realignment, creasing or breaking of the dielectric tapes in these cables could result in electrical failure of the cable. There is no substantial data available on electrical/mechanical testing of these cables that is directly applicable. Therefore, it is clear that these cables must be carefully evaluated by testing for their ability to withstand OTEC conditions.

Extruded dielectric power cables are considered to be worthy of further evaluation for OTEC riser cables. These types of cables utilize extruded polymeric layers, the primary dielectric polymers being cross-linked polyethylene (XLP) and ethylene propylene rubber (EPR). Extruded dielectric cables appear to be likely to perform satisfactorily in the severe OTEC environment; however, the effects of the cyclic loads and motions on these cable types is not yet well understood and can only be fully evaluated from extensive testing.

A major difference exists in the experience with laminated oil-filled cables versus the experience with extruded dielectric cables. Extruded cables are the new-comers to the extent that 15 years ago extruded cables rated higher than 35 kV were indeed a rarity. Today, 138 kV extruded cables are a commercial product and work has been carried out to develop these types of cables to 230 kV and 345 kV. These development efforts are continuing. Major improvements have been made in the past several years to improve the ability to supply ultra-clean polymeric compound to power cable manufacturers. Significant improvements have been made through research into the improvement of extruded dielectric compounds and understanding of the failure mechanisms has been extensively pursued. Improved equipment and innovative extruding and curing methods have resulted in production of extruded dielectric cables that are greatly improved over those attempted 10 to 15 years ago. Considerable work has also been carried out in the field of partial discharge (corona) detection resulting in improved testing procedures and quality control of manufactured cables.

A major design consideration for extruded dielectric power cables is the need for a hermetic sheath. It is generally accepted that such a sheath is desirable and is mandatory if average 60 Hz voltage stress in the dielectric exceeds about 50 volts per mil.

Laminated oil-filled cables, as contrasted to extruded cables have a long and excellent history of reliable performance in stationary installation. This is not to say that failures never occur but rather that the record for such cables is a very good one. In the U.S.A. comparatively little self-contained cable is in use but high-pressure-laminated oil-filled cable systems constitute about 90% of cable systems rated 69 kVAC and higher. The use of synthetic tapes for replacement of cellulose tapes is and has been a subject of considerable research. A primary advantage of the use of such synthetic tapes is the significant reduction of electrical losses in the dielectric. The possible use of such constructions for OTEC must not be abandoned but it is important to bear in mind that the great predominance of experience and proven performance has been with oil impregnated paper tape insulated cables under pressure.

One of the primary thoughts concerning the selection of riser cables for the OTEC application is the obvious desirability of using state-of-the-art designs and solutions wherever possible in order to minimize elapsed development time for the total project. Failure to use state-of-the-art methods, where suitably available, will result in a tremendous increase in development time and cost. For example, development of a totally new cable dielectric component package would typically require a research and development effort spanning a number of years including a very substantial period of developmental testing and prove-out. Therefore, the approach is to use either state-of-the-art technology or reasonable extensions of the SOA for most elements of the riser cable system development. However, there is no doubt that some design areas will require totally new and innovative solutions.

Three of the cable designs that the study has shown to be the most suitable candidates for OTEC applications are shown in Figures 1, 2 and 3. A tabulation of some of the characteristics of these 400 MW cable designs is shown in Table 1.

A major part of the Phase I study dealt with assessment of the means of bringing the cable from a point on the ocean bottom to the OTEC plant. At the time that this initial study was carried out, the watch circle limits of excursion for the OTEC plant were not defined and the cable support systems that were favored were those that would allow the greater amount of plant movement without imposing excessive loads, bending or twisting on the cable system. In that context, the most favorable system is the subsurface buoy supported cable (see Figure 4). The comparison of maximum extremes of allowable watch circle and of cable lift-off from the ocean bottom are shown in Table 2, for each of the 4 primary support system configurations shown in Figure 4. The scope of this paper does not allow a detailed discussion of this topic and the interested reader is referred to Reference 3. In summary, it should be emphasized that as plant mooring designs restrain plant watch circles to smaller and smaller percentages of water depth other support systems concepts, such as the direct riser and the cold-water-pipe attachment may become possible. However, the design considerations of the cable system interferring with the plant and the ocean bottom remain key points of concern. Attachment and anchoring of the cable system as well as installation, repair and recovery considerations remain dominant design parameters.

There is little doubt that there will be multiple riser power cables suspended from each OTEC plant. In the case of AC operation, the minimum number is 4 if a spare cable is provided. In the case of DC operation, the minimum number is 2. In all cases, at least some degree of redundancy must be provided. Therefore, a 3-phase AC system should have at least one spare cable. A DC system should have at least two total cables, thus if one is damaged or fails, the other can be designed to operate at 50% to 100% of full transmission power. Multiple cables are required due to the extremely large size of bundled cable configurations and the unfeasibility of manufacturing and installing such cables in long lengths. Also, it is most desirable that cable phases be physically isolated for independent maintenance and repair. Important design considerations are also brought into play by the use of multiple cables. Cables must not interfere with each other or other parts of the OTEC system.

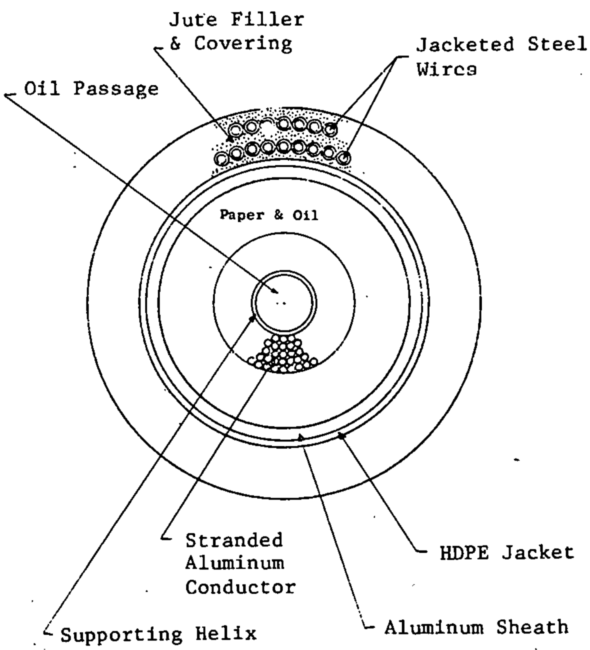

Fig. 1 Armored Self Contained OTEC Riser Cable

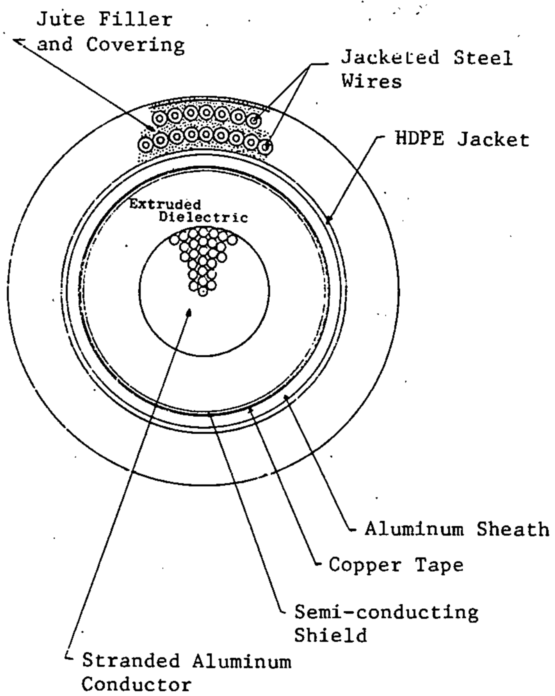

Fig. 2 Armored Extruded-Dielectric OTEC Riser Cable

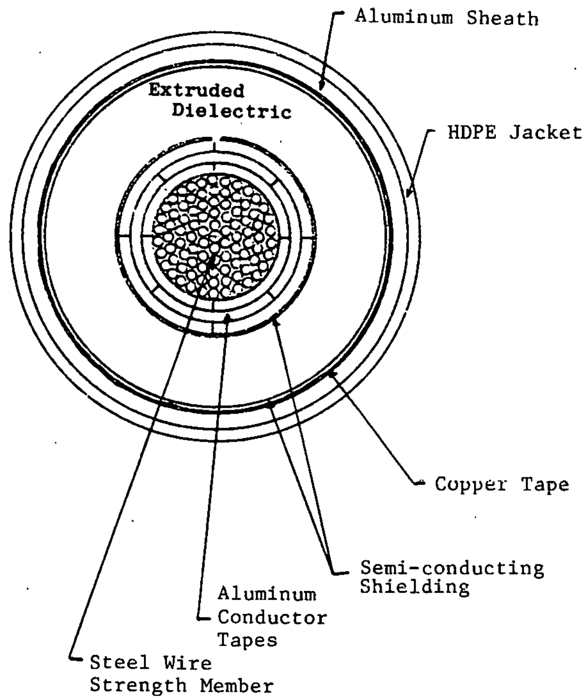

Fig. 3 Armorless Extruded-Dielectric OTEC Riser Cable

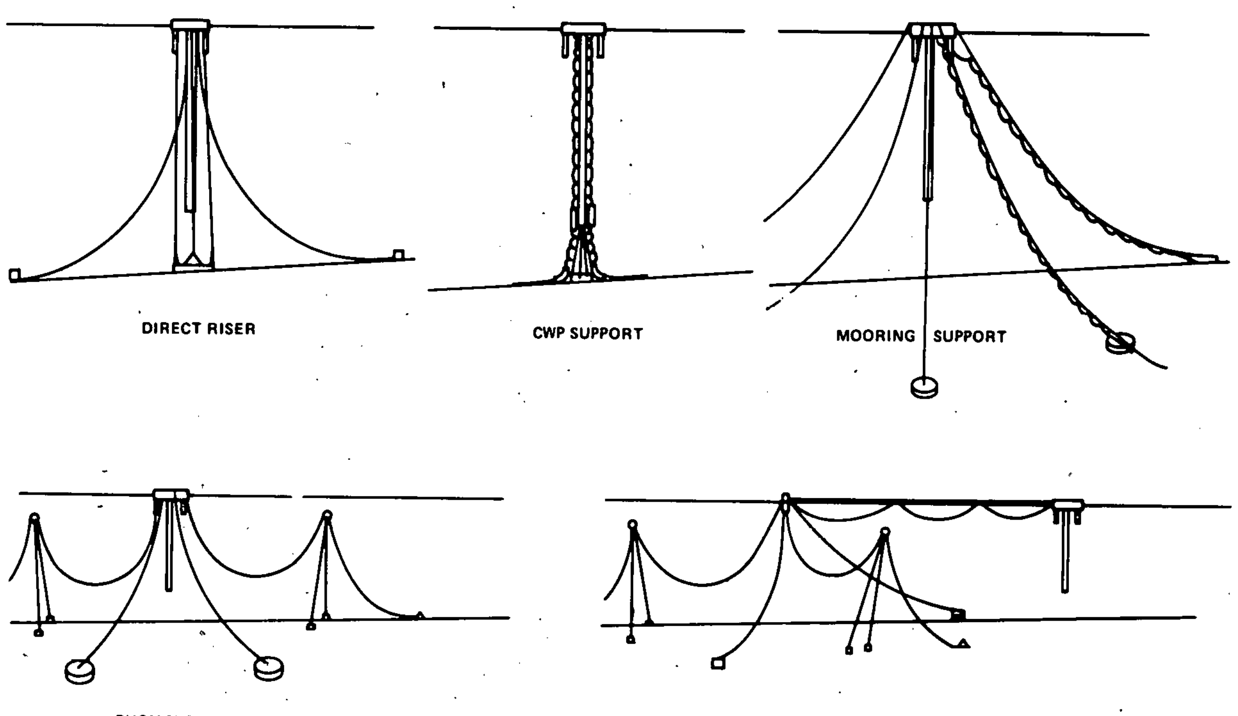

Fig. 4 Riser Cable Mooring System Concepts

Table 1 345 kVAC, 400 MW_e Riser Cable Characteristics

	Design No. 1*	2*	3*
Cable O.D., in.	6.01	8.01	6.24
Weight in air, lb/ft	18.7	26.3	19.4
Weight in water, lb/ft	8.1	7.3	5.8
Breaking strength, klb	308	310	215
Modulus, kft	37.7	42.6	37.1

* Reference Fig. 1, 2 and 3

Table 2 Comparison of Plant Motion Accommodation by Various Support Systems

(Distances and lengths as per cent of water depth)

Type of Support System	Max. radius of a watch circle	Max. length lifting off bottom
Direct Riser	20%	57%
Water Pipe Attachment	9%	34%
Mooring-Supported	20%	57%
Subsurface Buoy	40%	20%

Preliminary System Requirements

The cable must be designed to avoid fatigue of any individual component, interfaces of components, or groups of components. Fatigue in both bending and twisting must be considered. The minimum bend radius acceptable for the final cable design must be established through a physical test program. The relationship of mechanical cycling to electrical life must also be established through extensive testing.

The riser cable suspension system must be designed to:

Accept movements of the plant over an appropriate range.

Limit tensions in the cable to an acceptable maximum.

Limit bending of the cable to an acceptable minimum bend radius in all planes.

Avoid slack conditions which could subject the cable to hockling.

Allow appropriate installation of the cable on site.

Avoid damage to the cable from physical contact with other objects encountered during expected operation or from abrasion on the ocean floor.

Limit twisting of the cable to an acceptable maximum.

The motions of the OTEC plant due to changes in ocean current velocity will probably be greater than the response of the plant to wave action. Therefore, final performance criteria for the cable support system must be tied to performance criteria for the anchor system of the plant. All of the criteria listed above are inter-related and must be balanced against each other, and against additional factors relating to the physical layout of the plant and anchor system.

The riser cables must have both mechanical and electrical terminators. The mechanical terminators must provide a positive anchoring of the cable to the plant in such a way as to resist the forces (tension and torque) developed in the cable and must control cable bending so as to prevent cable damage from excessive bending. It must also protect the cable from abrasion from continual movement with respect to the plant. The electrical terminator must control electrical stress gradients while transmitting electrical power from the plant to the cable.

The subsea connector, if necessary, between riser and bottom cables must be capable of operating for the life of the cable system without human intervention or assistance. It must carry the cable voltage and current without electric failure, overheating, or other difficulty while subject to hydrostatic pressures of as much as 2700 PSI. During installation and possible later retrieval, it must withstand the tension, torque and bending. The subsea connectors must also transfer and balance the torsional characteristics of bottom and riser cables.

The basic electrical requirements for the entire OTEC riser cable system involve the areas of dielectric integrity, power transmission capability, compatability with bottom cable, electrical loss considerations, production test needs, transient overvoltage protection, short circuit protection, grounding, maintenance, fault location, and watertight integrity.

The installation requirements include the use of equipment that has capacity for storage and handling of these large, heavy and long cables. The cable must be controlled during installation at tensions of up to 65,000 pounds.

There are a number of prerequisites to a successful submarine power cable installation. These include:

A thorough bottom survey, including soil and sub-bottom conditions.

A pre-plot of the proposed cable route.

Complete and appropriate fitting out of the lay vessel to suit the cable to be laid and the operations involved.

Equipping of necessary support vessels.

Correct stowage of the cable with end preparations and other attachments to suit the planned operations.

Adequate crew training and proving trials.

A documented operations plan including cable laying procedure, equipment handling, navigation and communications.

Terminations and Submarine Connections

A commercial sulphur hexafluoride (SF6) cable terminator appears to be a reasonable device on which to base a design for the OTEC riser cables. It combines relatively small size and enclosed construction necessary for installation in confined spaces with attendant personnel safety. They are available for voltages to 345 kVAC and little or no modifications of the electrical termination are anticipated for the OTEC system.

Means to mechanically terminate the riser cable strength member and transfer the cable mechanical loads to an appropriate part of the OTEC plant structure are required. Methods for terminating armor wire include a slotted ring to provide an abutment to contact individual crimped sleeves on each armor wire, or a tapered annular cavity filled with structural epoxy into which the armor wires pass. Alternatively, various cable grips can be used. For cables with central strength members different means of mechanical termination are envisioned such as an electrically-isolated-from-ground support located above the electrical termination. Although the mechanical terminations for OTEC riser cables will be extensions of the present SOA, they do not represent formidable design problems.

The submarine connection could be either a cable-to-cable splice or could be a hardware device that is basically a specially designed enclosed chamber comprising two back-to-back cable terminators. The cable splice is judged to be the more desirable concept because of its smaller size, simpler design, greater ease of handling and much lower anticipated development cost. Either concept will require development and particularly due to the pressures of up to 2700 PSI and the great forces related to depths of installation to which it will be subjected, a substantial development effort is needed. At any rate, it is undesirable to attempt to join two high voltage power cable legs at sea and it is far more desirable to plan to join the lengths in the cable factory. However, splices may be necessary for repairs. In the event that bottom and riser cables can be of the exact same design, then it is of course most desirable to completely avoid such connection by making the circuit one continuous manufacturing length.

Reliability

The use of OTEC plants to provide baseload power to existing power grids on shore requires extremely high availability of the OTEC power generating and transmission system. If the transmission cable is to be a single circuit, then the reliability of the cable must be extremely high to maintain availability of power. Many unknowns associated with riser and bottom cable performance in the OTEC environments must be determined by thorough development tests. Although the study of the performance of certain cable designs in previously installed systems might yield indications of what cable designs might be most reliable for OTEC, it is not possible to determine what the true reliability of such cables would be without these important tests.

The future studies of cable system reliability will indicate the desirable number of spare cables to provide sufficient redundancy for quick restoration of transmission in the event of a fault in the cable system. The absolute minimum would be one spare cable. Both the total reliability of remaining operating cables and the time requirement for repair will influence the need for additional spare cables. Cost will be a dominant factor in the final decisions on planned redundancy.

From experience with conventional cables and with other cable systems in the ocean environment, certain factors can be defined which will affect riser cable reliability:

Cable manufacturing technology and quality control.

Inherent ability of the cable design and support system to withstand motions and forces in the OTEC environment without electrical and/or mechanical failure.

Splicing technology and reliability.

Transmission system electrical protection.

Susceptibility to external damage from vessels operating in the area.

Meteorological conditions including frequency and severity of hurricanes and severe storms.

Seismic activity.

OTEC plant mooring system reliability and ability to hold plant movement within design limits.

Experience and ability of handling, installation, testing, and operating personnel.

Suitability of cable handling and installing equipment.

Reparability and maintainability of OTEC riser cable systems.

Inherent ability of the cable design and support system to withstand corrosion in the OTEC environment.

It will be necessary to evaluate many of these factors during development testing of the riser cables, and to include these factors in reliability estimates for the cable in service. Ultimately, reliability information must be revised by experience with the first few OTEC plants.

Table 3. Specific Site Cost Estimates for 400 MW_e Riser Cable Systems in Millions of 1976 Dollars

Site	Cable	Suspension System	Platform Hardware	Submarine Connector	Cable Vessel (lease)	Other Vessels (lease)	Total
Hawaii[a]	2.1	1.3	0.4	1.0	4.1	1.0	9.9
Puerto Rico[a]	2.7	1.6	0.4	1.0	1.4	1.0	8.1
New Orleans[b]	12.2	7.2	2.2	4.5	4.8	4.5	35.4
Florida W. Coast[b]	14.1	8.2	2.2	4.5	4.8	5.1	38.9
Key West[a]	3.3	1.9	0.4	1.0	0.9	1.3	8.8

[a] DC plants with 4 single conductor cables

[b] AC plants with 3 single conductor cables

Cost Analysis

The cost of an OTEC riser cable system should be approached with the realization that it is a vital link in an electrical transmission system. Further, it is realized that the choice of basic system concepts, such as the number of spare cables to be provided, is related to the minimum acceptable riser cable system redundancy. It is necessary to develop accurate reliability figures for all OTEC transmission system components in order to determine the most desirable means of providing redundancy in the transmission system.

Specific site cost estimates for 400 MW plant riser cable systems are shown in Table 3.

The cost of energy losses in the cable system, the cost of maintaining a repair and maintenance capability for the cable system, and the cost of a dedicated OTEC cable production facility are subjects that are treated more completely in reference 3.

Further Development

The final task of the Phase I study was the definition of gaps existing between presently applicable SOA technology for OTEC riser cable systems and the technology deemed necessary. The resultant development plan that was created has as its prime objective the provision of OTEC riser cable system performance criteria to allow ordering of such systems by the Department of Energy. There are 40 parts to this program plan and they are divided into three basic tasks, as discussed in the following three subsections.

A. General System Studies pertains to both the 1040 MW and 100-400 MW cable developmental plans and are to be completed (Figure 5) during the first year of the Phase II work. These studies all relate to the installation of OTEC riser cable systems. One necessary study is the definition of detailed criteria for hydrographic surveys of potential OTEC sites as related to installation of cable systems. Studies of new deep water pipe-laying techniques for possible application to riser cable installation are also needed. Also, the need for deep-sea cable-laying trials must be studied and assessed due to the unique and unprecedented installation requirements for OTEC cables and is also necessary in order to plan cable installation.

B. Development of 10-40 MW_e riser cable systems is the immediate primary objective of the program and takes priority. The development activities under Task B are to be dome in the 2-year period indicated in Fig. 5 and briefly described below. Close interfacing with plant and plant mooring designers is planned throughout the 10-40 MW_e riser cable development.

Three interrelated parts of the Phase II work will be started within the first month and will provide the basis for design of tests, test equipment, and test candidate cable design(s). These three parts are the definition of criteria for developmental testing, identification of 10-40 MW riser cables, and implementation of a technical workshop for riser cable test design. This workshop will provide a unique opportunity for the presentation and discussion of matters vitally linked to the research and development required to prove-out OTEC riser cables. The comprehensive testing of riser cable properties and responses to mechanical forces is a key to the development effort. Figure 6 shows the basic relationships between the elements of cable design and testing.

The design and provision of test facilities for 10-40 MW cables is a necessity. The plan calls for completion of manufacture of test cable(s) to coincide with availability of test facilities. The developmental testing of the 10-40 MW cable(s) will proceed over a period of 30 weeks to supply early indications and data to show the ability of the design(s) to survive in the dynamic OTEC environment. Such testing will either verify the acceptability of the candidate design(s) or will indicate the need for redesign or modification. All efforts will be directed towards avoiding a second design/manufacturing/test cycle through use of cable design expertise to provide a successful first-round riser cable design. In spite of this comprehensive approach to design, actual testing may reveal the need for redesign or modification. While purely mechanical fatigue can be reasonably predicted for a given design, using extensions of existing technology, the combined effects of mechanical motions and forces on the electrical survivability of power cable insulation systems cannot be adequately predicted. These effects remain unknown and can only be ascertained by means of developmental tests.

Concurrently with the work described above, evaluation of the means of suspending the riser cables will proceed based upon provision by D.O.E. of semifinal or final plant mooring parameters. Conceptual riser cable suspension system designs will be developed in further detail for these more specific plant mooring designs. These conceptual designs will go through an evaluation and approval cycle with D.O.E. and other subsystem contractors. Detailed supension system design will then be carried out.

A detailed installation plan for the 10-40 MW riser cable system will be provided to D.O.E. and the development of this plan will be carried out concurrently with the detailed suspension system design. This important installation plan will use Phase I study results, input from Task A of Phase II, input from subsytem contractors, input from D.O.E., and input from industry experts and consultants. This document will cover all necessary information for deployment, operation, repair, and demobilization of 10-40 MW riser cable systems.

The final major output of Task B will be the provision of a detailed set of final purchase criteria for the complete 10-40 MW riser cable system. This document will allow the ordering and planning of all necessary parts of the riser cable system for the 10-40 MW OTEC system.

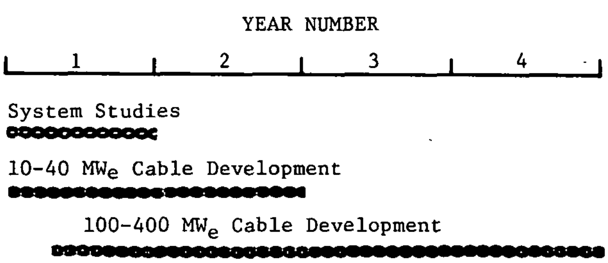

Fig. 5 Phase II Development Schedule by Year

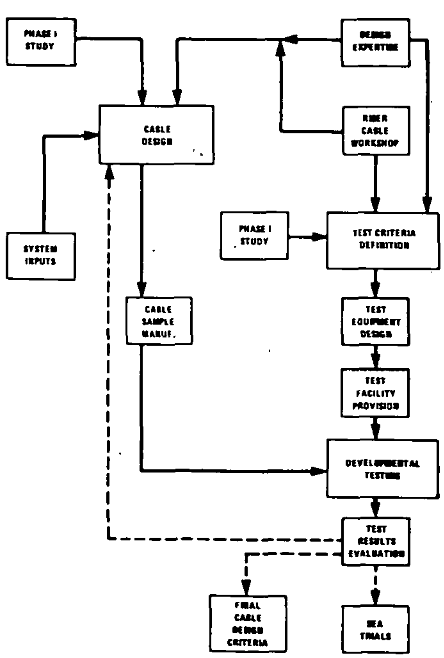

Fig. 6 Phase II Development Plan

C. Development of 100-400 MW riser cable systems is a longer range objective of the program but is very interrelated with the 10-40 development. The objective of this task is the provision of riser cable performance criteria for 100 MW and 400 MW OTEC plants. Information must be developed to allow ordering of all necessary riser cable system components for these systems. The preliminary study carried out in Phase I has defined the many considerations involved with such cables. These cables have developed requirements that exceed those for the 10-40 MW riser cable system. Not only are the voltages for 100-400 MW cables higher (in the 138 kV to 345 kV ac and + 250kV dc range) but the projected life requirements for these cables are to be in the range of 30-40 years. Cable(s) selected for 10-40 MW testing and evaluation may be of the same type as those evaluated for 100-400 MW development, but that task in itself will not be sufficient to satisfactorily predict a 30-40 year cable life at the electrical stresses that will occur in extra high voltage cable designs.

There will be many outputs from the previous task that will complement the 100-400 MW development and, in some cases, such as the provision of test facilities for 100-400 MW cables, the additional work required will be a further extension of work performed in the preceding (10-40 MW) task. Figure 5 shows the schedule for 100-400 MW riser cable development.

Throughout the 100-400 MW development, close technical interfaces with plant, plant mooring, and bottom cable designers are planned. These will be primarily by means of technical discussions and presentations.

The determination of developmental test criteria will be an extension of work performed in the preceding task. Provision of additional test facilities to accomodate the greater number of 100-400 MW samples requiring testing will also be an early part of this work.

A survey of existing manufacturing facilities will be carried out concurrently with a detailed analysis of the adequacy of existing quality control measures used for extra high voltage power cables. These will be immediately followed by a study of production test requirements for the very long cable lengths required for some OTEC systems. This latter study may be expected to clearly define what steps must be taken to assure that cables leaving OTEC cable production facilities possess the necessary quality to insure high reliability. The output from these manufacturing state-of-the-art studies will provide a major input to a final definition of the OTEC cable manufacturing facility. This latter information will be supplied in document form after the final cable design(s) for the 100-400 MW systems are selected.

Existing manufacturing facilities (probably multiple plants) for sample test lengths of 100-400 MW riser cable will be defined and test cable manufacturing plans will be formulated. Manufacture of test lengths of perhaps 1000 to 2000 feet of each of 4 to 6 different designs will be carried out following detailed design of candidate cables.

Developmental testing will be performed with these samples over a projected 105 week period. These tests will provide the basis for prediction of mechanical and electrical/mechanical fatigue life of candidate cable designs. Cable designs initially selected for test shall be refined during initial design to the greatest extent possible in order to reduce the need for redesign or modification.

Reliability studies will be carried out during this development. The first data on riser cable reliability will begin to become available during the early stages of test and as this effort progresses the effect of riser cable reliability will be determined.

The cost studies that will be included in many of the parts of this task will be integrated into one part and a report document of costs for 100-400 MW riser cable systems will be presented to D.O.E.

The design of the riser cable suspension system(s) will be finalized and detailed concurrently with the development of the cables. Definition of plant mooring and plant layout will allow detailed design of the riser cable suspension system to the point of providing the necessary criteria for ordering purposes.

Plant cable terminations, both electrical and mechanical, will be fully defined after candidate cable designs are resolved. Development requirements will be included in this effort.

The development of a submarine connection for joining riser and bottom cables will be carried out in conjunction with the riser cable sample manufacture and test. This capability is necesary in order to join riser and bottom cables.

An overall operational plan for installation of 100 MW and 400 MW riser cable systems will be prepared subsequent to formalizing of OTEC plant siting and configuration and the definition of candidate cable and suspension system designs. This plan will be an effective working document for installation through mobilization of manpower and equipment, sea trial, installation, final tests, contingency repair procedures and demobilization.

Comprehensive criteria will be developed for manufacture of riser cables. Preparation of the performance criteria will commence following final design definition of cable type as determined from analysis of electrical and physical test data on cable samples. The criteria will be based on data obtained during production of test cables and from a systematic investigation of current industry practice in materials procurement, process control and effective quality control techniques.

Designs and requirements for riser cable, subsea connections, plant terminations, cable support and mooring system, and installation criteria shall form the basis for an integrated riser cable system performance criteria final reort. This report will be completed as the final tep in Phase II and this will allow procurement f all components for a full scale riser cable system in the 100 MW-400 MW range.

Phase II Riser Cable Development Results

It is the purpose of this part of the paper to provide the reader with an overview of work carried out during the first four months of the Phase II development program which began on January 31, 1979. The last section of this part will describe priority actions to be carried out in the immediate future. The following were major areas of activity during this period.

Definition of Test Criteria for 10-40 MW OTEC Riser Cables

This key work has identified the specific tests to be performed. The need for a series of basically mechanical tests on OTEC riser cables has been defined in a progression from screening tests that are least costly and time consuming to more advanced and complex tests that combine various test parameters for simultaneous evaluation. More advanced mechanical tests must be performed with mechanical terminations that are of the same design as those envisioned for the actual cable system. Also, some basic tests are as follows:

- Tension
- Torque
- Bending
- Voltage
- Electrical load current

In addition, crushing and hydrostatic tests, and abrasion and corrosion tests will be performed. The variables will be evaluated for various cable designs in step-wise fashion in order to attempt to isolate cause and effect. The sequence of testing may be as follows:

- Tension
- Bending without tension
- Twist without tension
- Cyclic tension
- Cyclic tension and forced twisting
- Cyclic tension and bending
- Cyclic tension, bending and forced twisting
- Cyclic tension, bending and forced twisting combined with electrical load current and voltage. (step-by-step introduction of variables for isolation of cause and effect)

Cable candidates that enter the final series of tests and certain earlier tests shall be tested for basic electrical characteristics and inherent electrical withstand capability on AC, impulse, and DC. This shall provide control data for comparison with lengths that complete the test series.

Rate sensitivity is an important consideration in test design. The test must be designed to accurately duplicate performance in actual service. Care must be exercised in planning the tests so as not to make the test unrealistically severe or, on another extreme, to mask out true effects by unrealistic increase in frequency of cycling. At the same time, it is desirable to accelerate the testing as much as possible without invalidating the results.

There is no doubt that some special mechanical test equipment will be needed. In order to minimize the equipment and still obtain the level of understanding needed from the test program results we are in the process of setting the priority of the tests. From the total number and variety of tests that would ideally be performed, if cost and time were not the very real constraints that they are, the study is now focusing on defining the most important tests to be performed, and the sequence of performance.

The tests may be grouped in the following broad categories:

"Simple" mechanical tests with only one variable.

Mechanical tests with two or more variables.

Model tests in the ocean environment and in the laboratory

Electrical characteristic tests

Combined electrical/mechanical tests with two to five variables

Design of 10-40 MW Canditate OTEC Riser Cables

Based on the results of the Phase I 400 MW cable study, it is clear that downward extrapolations of the 400 MW laminated dielectric and extruded dielectric designs are suitable for the lower output systems. However, there are some basic approaches to the design that first had to be determined. It is possible to carry the 10 to 40 MW power levels with cables rated as low as 35 kV AC. It is even possible, within the constraints of sheer cable size and manufacturing/handling capability, to design a 35 kV AC three-conductor single cable (see Fig. 7) that theoretically will perform for load levels of as much 40 MW. The desirability of using this approach is subject to two basic considerations:

Such a cable would be about 6.5" in diameter and would weigh about 4.5 lbs. per foot. While conceptually possible, such a cable is subject to greater manufacturing risks and much greater difficulty in joining the phases of such a cable, should a factory splice be necessary.

A three-conductor single cable would not be representative of later commercial-scale cable designs that will necessarily be single conductor cables. The mechanical behavior of such a cable is undoubtedly much different.

What this means is that the results of a developmental program for 3 conductor 35 kV AC (40 MW) cables would have limited relevance to later single conductor 138 kV to 345 kV AC(100-400 MW) cable development. It does not appear economically desirable to obtain extensive test results that apply only to the 10-40 MW plants and not to later commercial size plants. Therefore we have made the assumption that cables used for the 10-40 MW plants will in fact be the same cable designs proposed for the 100 MW commercial plants. It is anticipated then that the results of tests on 100 MW cables will also be applicable to later cables for 400 MW plants, while not totally eliminating the need for additional testing. Therefore, the design efforts have focused on both 35 kV AC three conductor designs and 138 kV AC single conductor designs with the emphasis on the latter. The preliminary 10-40 MW cable candidates have been designed and Table 4 summarizes some of the characteristics. The construction of designs 2, 3, and 4 is similar (scaled-down) to that shown earlier in Figures 1, 2 and 3.

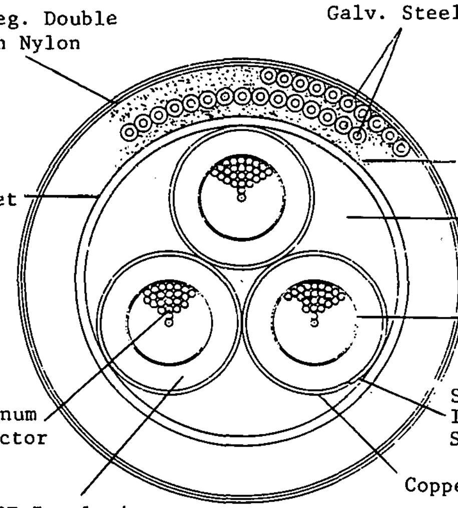

Fig. 7 Three-Conductor Torque-Balanced OTEC Riser Cable

Table 4 Summary of Preliminary 10/40 MW_e Riser Cable Design Characteristics

	Design No.[a]			
Parameter	1	2	3	4
O.D., in.	6.6	4.7	3.9	3.8
Weight in air, lb/ft	17.5	11.3	8.1	8.5
Weight in sea water, lb/ft	4.6	4.8	3.8	3.5
Breaking strength, klb	308	285	225	206
Cable modulus, kft	67	59.6	59	59.6

[a] #1: 3 conductor, 35 kV
#2: Single conductor. 138 kV, extruded
#3: Single conductor, 138 kV, laminated
#4: Single conductor, 138 kV, extruded with center strength member

Riser Cable Workshop

On May 22 and 23, 1979, a technical workshop was held in Portsmouth, N.H. to address key areas of the design, testing and mathematical analysis of OTEC riser cable systems. This workshop brought together some 55 technical experts in the fields of power cable design, ocean engineering, cable testing, mathematical analysis, materials, corrosion, installation of ocean cables and other fields. The workshop yielded many valuable inputs to the OTEC riser cable development program.

The results of this workshop are now being condensed into a report that will be used directly in the further work.

Study of Riser Cable System Constraints Imposed by Tentative OTEC Plant and Plant Mooring Designs

This study has recently been focused upon the possible use of riser cable attachment to the cold water pipe (CWP) of a 10-40 MW OTEC plant restrained within a watch circle radius equal to 10% of water depth. The study has also looked in more detail into the implications for the cable of using a number of variations of support systems in various combinations with possible means of mooring the plant. The mooring system approach has not yet been resolved by the government thus making final choice of preferred riser cable support systems not possible until some later date.

Other areas of activity in the program during the first four months of Phase II will be mentioned here but not described due to limitation of space. Definition of criteria for cable route hydrographic surveys for potential OTEC sites has been an active project as has been a study of new deep-water pipelaying technology as applicable to OTEC riser cables. The search for test facilities that have some or all of the requirements for the OTEC riser cable testing was begun during the first month of Phase II and is an active part of the study. Technical meetings with other 10-40 MW system designers have been carried out and will continue to be pursued. Work has been performed on conceptual design of the 10-40 MW plant riser cable system.

The above is intended to provide the reader with an overview of the Phase II work carried out to date. Due to the obvious space limitations we are unable to treat these areas in greater depth here.

Future Phase II Development Priorities

In the immediate few months ahead work will focus on performing initial mechnaical tests on power cables to begin to produce test results to increase the understanding of mechanical phenomena in cables and to determine the validity of some mathematical predictions. This first level of testing will be performed on specially-obtained scaled down model cables that shall be instrumented to provide measurement capability. This effort will also serve to refine some of the testing techniques.

Concurrently, many other parts of the program will continue to progress. Foremost among these will be the 10-40 MW riser cable development tasks including final definition of cable test criteria and plan, final design of cable candidates, design and manufacture of special test equipment, arrangement for test facilities, test cable manufacture, initial testing, and design of the 10-40 MW riser cable system. Another area of development that will be pursued is the possibility of suspending an instrumented model OTEC riser cable from an ocean test platform and obtaining some early experimental data on motions and mechanical performance of the cable. This data would be used to determine validity of mathematical predictions.

Conclusion

The key considerations and results of past studies of the OTEC riser cable system have been summarized in the foregoing discussions. In summary, it may be said that the technology has not yet progressed to the point where it can be said that a given cable system design will positively perform reliably as an OTEC riser. Only through test-length production of these specially designed cables followed by extensive developmental testing can viable cable candidates be truly determined. Previously, no high voltage power cable has ever been required to extend to more than 45% of the 4000 foot depths planned nor have high voltage power cables been operated in a dynamic surrounding that even comes close to the severity of the OTEC environment. As in the past, it is anticipated that through concerted effort, specially designed systems can be thoroughly evaluated, modified as needed based on test results, and finally proven viable. For the OTEC riser cable system we believe this to be true.

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LOAD CRITERIA FOR OTEC RISER CABLE DESIGN

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Abstract

The practical success of the Offshore Thermal Energy Conversion Plant rests with its ability to transfer power to where it is needed. The OTEC electrical riser cable is a critical element in the power transmission system. Consequently, satisfactory design of the riser cable has become one of the key technological issues in the OTEC Program. Particular emphasis must be placed on the riser cable design because the dynamic conditions and hostile environment to which it is subjected are unique and unprecedented for any insulated high voltage power transmission system.

Modern approaches to engineering design emphasize the need for a rational and scientific determination of the loads that a structure will experience. This paper proposes a method for the definition of the extreme values and fatigue loading spectra which the cable will experience in its lifetime.

Introduction

The requirements for transmission of electrical power from an OTEC platform demand bold extrapolations be made from the accumulated experience in high voltage electromechanical cables. To this end, the pragmatic rules formulated from past successes and failures cannot always be used with a sufficiently high level of confidence, particularly when the requirements for new design strongly departs from the proven capabilities of the old design. In such a case, the new design must evolve from explicit and justifiable rationale. Sometimes referred to as "rational design," this concept involves the complete determination of all loads on the basis of scientific rather than empirical procedures, so that uncertainties can be reduced to a minimum.

This procedure also assumes that the response of the cable can be accurately determined and that large factors of safety or "factors of ignorance" can be avoided. Instead of ensuring that a simple calculated design stress is below the ultimate strength of the cable by an arbitrary factor of safety, an attempt is made to predict the demand of all loads acting on the structure and the capability of the structure to withstand the load without failure. This report deals only with the demand, or loading, on the riser cable.

The random nature of both seaway-induced and flow-induced loads on the cable requires a probabilistic approach. Such an approach is consistent with the "probability of failure" criterion inherent to rational design. Furthermore, this procedure is particularly responsive to the cost and risk perspectives of OTEC Project Management.

The use of probability of failure as a criterion for design requires a definition of failure. Failure may result from exceeding either the "ultimate" strength of the cable or fatigue limits in tension, bending, or torsion. Cyclical loadings, as well as extreme lifetime values must be considered in the definition of the cable's load history. Flow induced structural vibrations are particularly critical to the fatigue aspects of load prediction, and methods of vibration prevention or reduction have an important bearing on the problem.

It is not intended to minimize the value of the conventional empirical methods used in electromechanical cable design which have served the designers and manufacturers well through the years. However, there is an obvious need for a fully rational approach to riser cable design in light of repeated emphasis on the unprecedented nature of the problem.

The procedures described in this paper are conceptually applicable to any riser cable configuration. However, the catenary configuration as proposed by SIMPLEX Wire and Cable Company has been used as the "baseline" configuration. Some approximate calculations have been included, and are based on a sample case with the following particulars:

Cable Type: 345 KVAC armored self-contained
Platform: APL OTEC Plantship

Length of Catenary:	10,000 ft
Cable Wet Weight:	8.1 lbs/ft
Cable Dry Weight:	18.7 lbs/ft
Cable Outside Diameter:	6.01 inches
Breaking Strength:	308,000 lbs
Plantship Location:	Puerto Rico
Cable Attachment Point:	Centerline/bow/keel
Distance between Plant and Subsurface buoy:	
Case 1A	8,000 ft
Case 1B	4,000 ft
Extreme Conditions:	"100-year storm"

Additionally, consideration is given primarily to tensile type loads. The procedure is equally appropriate for torsional, bending, or combined loads.

Sources of Loading on the Riser Cable

The loads to which the riser cable will be subjected can be classified within two broad categories--steady and unsteady. Steady loads result from:

- cable weight and buoyancy
- steady current
- hydrostatic pressure and ocean temperature gradient
- configuration

Unsteady loads include:

- platform-motion induced
- (subsurface) wave force
- flow-induced vibration
- unsteady currents

- internal waves
- thermal stresses
- dynamic "snap" loads

Other factors that may impact on the loading conditions are:

- creep
- corrosion
- abrasion and chafing
- fishbite
- marine organisms

To get a general idea of the forces involved, some preliminary calculations were performed. There are several areas for which no "numbers" are available and which require further attention or data from current studies. These areas are noted.

Static Tensions. Determination of static tensions is based on catenary equations. The first step is to determine the coordinates of the catenary and its angle with the horizontal. This is accomplished using the well known equation for a catenary:

$$\frac{y}{a} = \cosh \frac{x}{a} - 1 \qquad (1)$$

where x and y are the coordinates using the lowest point on the half-catenary as the origin; and a is the catenary parameter which is found by solving the following transcendental equation for a:

$$\frac{s}{a} = \sinh \frac{x}{a} \qquad (2)$$

where s is the length of half the cable; and x is half the distance between the platform and the buoy.

The angle of any point on the cable with reference to the horizontal is:

$$\tan \alpha = \frac{y_{n+1}-y_n}{x_{n+1}-x_n} \text{ .. and solve for } \alpha. \qquad (3)$$

This coordinate system is then transformed to a platform-referenced coordinate system. The catenary parameters are 3383 and 784 for Case 1A and 1B respectively. The static tension in the cable results from net forces of cable weight, buoyancy, and configuration geometry. The static tension can be calculated at any point along the catenary using the following equation:

$$T = a\,w\,\cosh \frac{x}{a} \qquad (4)$$

where w = lb/ft (wet weight)
T is in lbs.

The point of greatest static tension on the cable is at the platform attachment point. For Case 1A, T = 48,895 lbs; and for Case 1B, T = 40,954 lbs.

Steady Current. Reference 1 estimates that the increase in tension due to current will not exceed 50% of the static tension. This should be a conservatively high assumption, assuming no localized flow induced vibration. Thus, the tension would be on the order of 25,000 lbs or less.

Hydrostatic Pressure. At 4,000 ft, the cable will experience about 1,800 psi.

Installed Loads. The induced stresses and residual stresses that result from cable manufacture, handling, and installation are presently unknown and difficult to determine. Installation loads are necessary to determine because they may be the extreme lifetime loads.

Platform Induced Motions. Reference 2 determined significant absolute accelerations for the APL Plantship for various sea conditions and locations. Depending on the attachment point of the riser cable, acceleration and the induced forces can vary a great deal. A representative case has the following parameters:

Type of Seas - Head, 0.0 knots
Significant Height - 29 feet
Distance from Amidships - 100 feet
Distance from Centerline - 0 feet

Conservatively assuming 50% of the cable's mass is accelerated, the dynamic significant force at the terminal would be about 17,166 lbs. The highest expected amplitude in 100 excursions would be 26,000 lbs and in 1,000 excursions would be 32,000 lbs.

Wave Induced Loads. If the cable is not connected to the plant near the keel or lower drafts, then it may experience substantial wave impact loads near the sea surface. These have not been computed for this report, but the significance of this load source is important for near-surface terminations.

Flow Induced Vibrations. Preliminary investigations of the vortex-induced structural vibrations of the OTEC Riser Cable are described in Reference 3. Strumming oscillations lead to a virtual increase in the drag coefficient of the cable resulting in higher tensile loading. Early fatigue failure can result from the accelerated cyclic loading. Actual probabilities, magnitude, and frequencies associated with strumming remain to be determined. The catastrophic effects of strumming on cable terminations and end fittings have been emphasized in many published papers and articles.

Unsteady Currents. These currents can be "blocked" into stationary processes and will be applied using long-term conditional probabilities. The force magnitudes, however, will be no greater than steady currents.

Internal Waves. Little analysis has been performed. Their contribution is presently assumed negligible.

Thermally Induced Stresses from Electrical Load Cycling. Few calculations have been performed in this area. The temperatures can be calculated by the cable designer for the various candidate designs. The significance of this factor is not presently known.

Snap Loads. High wave heights can cause the platform to impart to the cable instantaneous variations in tension. Kinking/hockling can result in the effectively "slack" portions of the cable. The cable is designed against kinking/hockling as much as possible. However, high impulse loadings from the platform in larger sea states may be significant and could result in terminal loadings exceeding 50,000 lbs.

Time-Dependent Mechanical Deformation. Although rate of strain may be low, deformations in cable materials, particularly the strength members, may result in a condition where the electrical components are subject to load-bearing conditions. There are no known high-voltage configurations where very long lengths of self-supported cable must operate for an extended period under relatively high forces. This is an area that has received very little attention. Further analysis and investigation is imperative.

Chafing and Abrasion. The adverse effects of chafing and abrasion should be precluded by effective design of the subsurface buoy and seabottom "touch-down" area.

Corrosion. Corrosive effects on the type of materials used for the fittings and strength member are known. The disasterous impact of sea water on fatigue properties of metals emphasize that corrosion in any form cannot be tolerated.

Marine Organisms. Definitive information relating to bio-fouling is necessary to determine impact on cable weight, cable displacement, cable drag coefficients, shedding frequencies, natural frequencies, jacket deterioration, and heat transfer.

Determination of Long-Term Distributions For Each Load Source

Reference 4 summarizes and applies the efforts of the last two decades toward "rational" structural design. The basis of this work is that all major components of the demand on the structure (as well as those of its capability) are random to some extent. Within this context, the objective on the demand or load side of the problem is the synthesis of a "long-term" probability density function of loading. For proper application to the riser cable problem, the sources of loading discussed previously will be reduced and classified according to Table 1.

Load Source		f(Random Variables)	
L_1	static tension	R_1	plant-buoy separation distance cable weight and buoyancy
L_2	platform motions	R_2	sea state, heading
L_3	current drag	R_3	current profile, current/cable angle
L_4	flow induced vibration	R_4	$f(R_1, R_2, R_3)$
L_5	waves	R_5	sea state, heading, cable attachment location
L_6	radial-type	R_6	electrical load-thermal hydrostatic pressure

Table 1. Sources of Loading

Each of the load sources enumerated is considered to be a function of other random variables which define the long-term environment and operational characteristics of the OTEC system. Thus, the problem of synthesis of the long-term probability density of the total load (L_t), $f(L_t)$ say, concerns the synthesis of the joint long-term density of the six load components and the random variables of which they are functions:

$$f(L_1,L_2,L_3,L_4,L_5,L_6,R_1,R_2,R_3,R_4,R_5,R_6)$$

where R_1,R_2, etc. are the sets of random variables defining the long-term environment and operation, and the marginal joint density of $L_1 \ldots L_6$ is found by integrating out the R's. To some extent, all components of loading are in some way influenced by all environmental and operational parameters. Thus, the ideal problem of determining the marginal density of all L's becomes one of unmanageable dimensions. The simplification of this synthesis problem requires further analysis. For the purposes of this presentation, the problem can be made tractable by reasonable engineering arguments. Before confronting the synthesis problem, the determination of the long term probability density functions for each load will be discussed.

Static Tension $\{L_1\}$

Reference 1 recommends that the platform must remain between 4,000 and 8,000 feet from the subsurface buoy. Although the type of the station keeping system has not yet been chosen, certain assumptions will be made which should generally apply to any station keeping system. It will be assumed that there is a 0.001 probability of going out of this recommended watch circle on one side or the other (total probability = 0.002). It will also be assumed that the platforms position will be normally distributed with a mean value of 6,000 feet from the buoy. Based on the .001 exceedance probabilities, the distance between the mean and the watch circle limits is 3.09σ, or σ = 647.25 feet. The long term probability distribution representing the relative location of the plant and subsurface buoy is:

$$f_1(x) = \frac{1}{\sqrt{2\pi\sigma^2}}\, e^{-\frac{(x-m)^2}{2\sigma^2}} \qquad (5)$$

This distribution is shown in Figure 1.

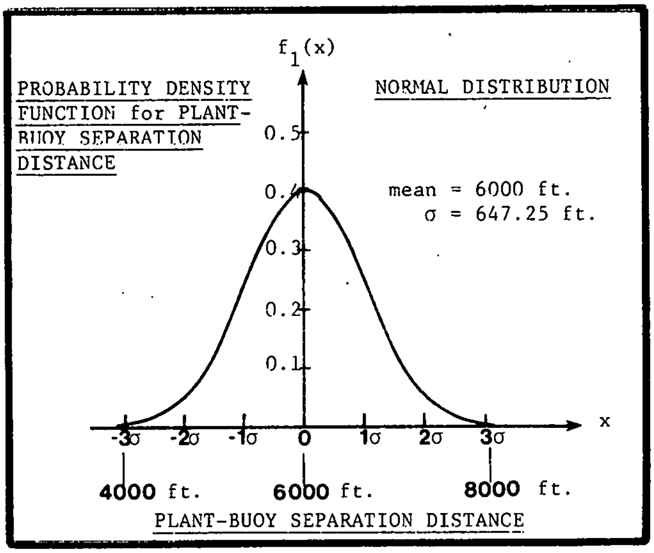

Figure 1. Probability Density Function for Plant-Buoy Separation Distance.

The static tension (L_1) at the cable-plant connection is a function of the plant-buoy separation distance. Equations (2) and (4) are evaluated for various distances. A curve, T(x), can be developed representing the static tension as a function of distance. Therefore, the long-term probability density function (p.d.f.) for static tension is:

$$f(L_1) = f_1\{T(x)\} \tag{6}$$

This long term p.d.f. is exemplified in Figure 2.

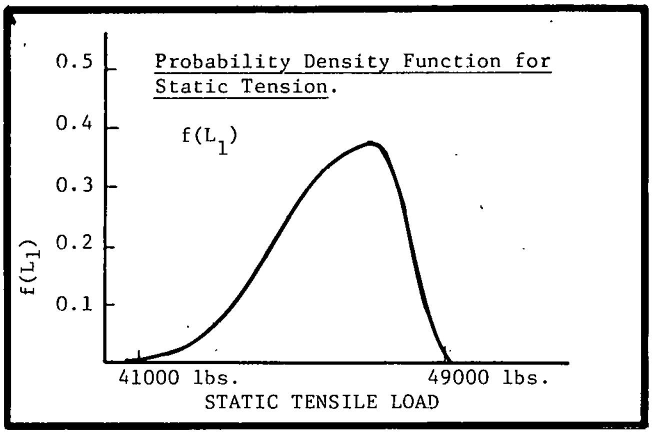

Figure 2. Probability Density Function for Static Tension. (Sample)

Wave Induced Load $\{L_2\}$

If the probability of a given wave induced load at a sea state s, at a heading ψ, is $f_{s\psi}(x)$, then the probability density function applicable to the long term response can be written as follows:

$$f(L_2) = \frac{\sum_{\psi}\sum_{s} n_{\psi s}\, p_s\, p_\psi\, f_{\psi s}(x)}{\sum_{\psi}\sum_{s} n_{\psi s}\, p_s\, p_\psi} \tag{7}$$

where
- $f_{s\psi}$ = probability density function for a short-term response for given s and ψ
- $n_{s\psi}$ = average number of responses per unit time of short-term response
- $(m_0)_{s\psi}$ = area under short-term response spectrum
- $(m_2)_{s\psi}$ = second moment of short-term response spectrum
- p_s = weighting factor for sea condition
- p_ψ = weighting factor for heading to waves in a given sea

The total number of responses expected in the lifetime of a marine system due to wave motions becomes:

$$n = \left(\sum_{\psi}\sum_{s} n_{\psi s}\, p_s\, p_\psi\right) \times T \times (60)^2 \tag{7a}$$

where T is the total exposure time to the sea.

Necessary information for Equation (7) must be derived from analysis of the platform motion in irregular seas for each sea state and heading to be considered. There are several computer programs used for this purpose. The spectral formulation presently being applied to the OTEC studies was developed by Bretschneider:

$$S(f) = 5\left(\frac{H}{4}\right)^2 f_o^{-1} \left(\frac{f}{f_o}\right)^{-5} e^{-\frac{5}{4}\left(\frac{f}{f_o}\right)^{-4}} \tag{8}$$

For each sea state s, and for each heading, ψ, the response spectrum and characteristic-statistical values of the platform are calculated by the program. Ideally, a 3-dimensional analytical cable model can be used to compute loads. Otherwise, the load on the cable termination can be computed using basic principles. The motion response spectrum is converted to an acceleration response spectrum, transformed to the coordinates of the cable-plant connection. Knowing the vertical acceleration response, the load can be calculated on the cable end. From these calculations, a characteristic value for wave induced dynamic loading is obtained for each case. The short-term probability density function can be described by a Rayleigh distribution. This distribution has been found by many investigators to represent observed short-term distributions of wave heights and related phenomena, such as ship motions. Therefore,

$$f_{\psi s}(x) = \left(\frac{2x}{E_{\psi s}}\right) e^{-\left(\frac{x^2}{E_{\psi s}}\right)} \tag{9}$$

Determination of $n_{s\psi}$ is made from the formulation:

$$n_{s\psi} = \frac{1}{2\pi}\sqrt{\frac{(m_2)_{s\psi}}{(m_0)_{s\psi}}} \tag{10}$$

From the short-term response spectra calculated by the motions program the area (m_0) and second moment (m_2) can be determined.

The weighting factors for the sea conditions are obtained from environmental data specifically prepared for the OTEC sites (Reference 5). Table 2 lists the pertinent data for Puerto Rico.

Significant Wave Height	Frequency of Occurrence (p_s)
<1	.045
1 - 2	.254
3 - 4	.385
5 - 6	.195
7	.078
8 - 9	.026
10 - 11	.010
12	.003
13 - 16	.003
17 - 19	.0005
20 - 22	.0005
100-year storm	.001 (ea. yr.)

Table 2

The weighting factors for the headings into the seas must be estimated pending better data. The distribution of headings can be greatly affected by the type of station keeping systems and operational decisions made by the plant's master (i.e., "keep the bow headed into the seas in waves greater than 8 ft."). A typical assumption made is that there is an equal probability for all headings. For computational purposes, the headings are generally grouped, as shown in Figure 3.

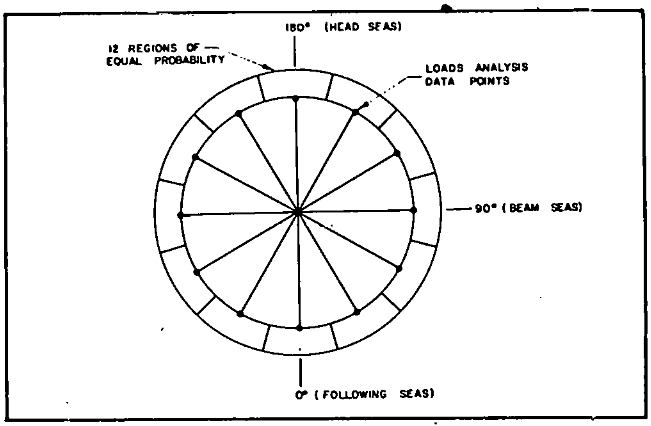

Figure 3. Assumed distribution of ship heading to wave direction.

Current Induced Drag $\{L_3\}$

A significant amount of current data for OTEC design is contained in Reference 5. The design currents include geostrophic and tide current in phase and added to the Eckman wind current for each sea state. Two design current cases are given for each sea state: 1) The surface wind current in line with the geostrophic current; and, 2) The surface wind current rotated 90° counterclockwise with the geostrophic current. Obviously, these are only two of an infinite number of possibilities, since the wind and sea state can be of any direction. For design purposes, a set of design current profiles could be formulated which describe all significant current conditions. For example, eight specific current profiles, each with its own frequency of occurrence, should be developed. Thus, current loads can be applied in the same fashion as the wave loads. Figure 4 illustrates this classification technique.

Over a short-term occurrence of a particular current profile, the cable response and induced load will be steady, assuming no flow induced vibration is initiated. The angle between the vertical plane extending through the platform and buoy and a characteristic direction associated with the current profile must also be considered. Thus, the long-term probability distribution for current induced load on the cable is written:

$$f(L_3) = \sum_c \sum_\alpha \sum_b p_b\, p_\alpha\, p_c\, f_*(x) \qquad (11)$$

where p_c = weighting factor for current profile
p_α = weighting factor for angle between catenary and current direction
p_b = biofouling weighting factor
$f_*(x)$ = increased load at cable termination with a particular current profile c, and angle α, and biofouling, b.

The calculation of $f_*(x)$ is complicated by the fact that with a particular design current profile,

Θ	P_c	PROFILE	Θ	P_c	PROFILE
025	.11		185	.18	
065	.18		253	.16	
095	.11		296	.01	
165	.22		340	.03	

Θ - characteristic direction of current
P_c - probability of occurrence

Figure 4. Sample of Design Current Profile Classification Scheme

the magnitude and direction of the current change with depth, and the cable no longer hangs within a plane. There are a number of 3-dimensional analytical cable models (References 6, 7, 8) with which to attack the problem. However, none have yet been comprehensively applied to the riser cable configuration. Reference 1 proposes an approach to determine increased tension due to a current perpendicular to the plane of the catenary, and indicates that the increase in tension will not exceed 50% of the static tension. This is conservatively high and should not be used in the rational determination of current induced loads. When the current is more nearly parallel to the catenary plane, Reference 1 indicates that the increased tension is comparatively less than that with the perpendicular aspect. In keeping with the rational approach proposed in this paper, a comprehensive analytical cable model should be used to determine the increased tension due to each current case $f_*(x)$. Figure 5 illustrates the p.d.f. in the long term of L_3.

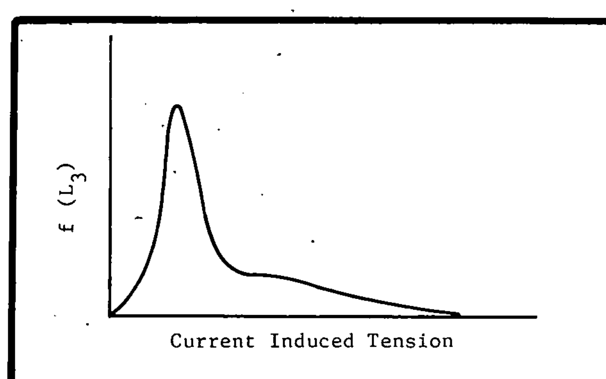

Figure 5. Long Term Probability Density Function for Current Induced Loads

Flow Induced Vibration $\{L_4\}$

It has been found (Reference 3) that the shedding frequencies along the riser cable correspond to the 4th through 30th modal frequencies of the cable. Shedding frequency is a function of cable diameter and normal current velocity:

$$f_s = \frac{SU_n}{D} \tag{12}$$

S = Strouhal Number (0.21 for riser cable)
f_s = Shedding frequency
U_n = Normal current velocity
D = Cable diameter

Several approaches to computing the natural frequencies of the cable are discussed in Reference 3. Shedding frequencies and natural frequencies have been computed for the riser cable for the following case:

- normal current profile for Puerto Rico
- 8,000 feet separation distance between platform and subsurface buoy
- current direction perpendicular to catenary plane

Figure 6 clearly indicates that various vortex shedding frequencies along the cable are close to many of the natural frequencies, particularly when a lock-in band of 30-50% on each side of each f_s is considered.

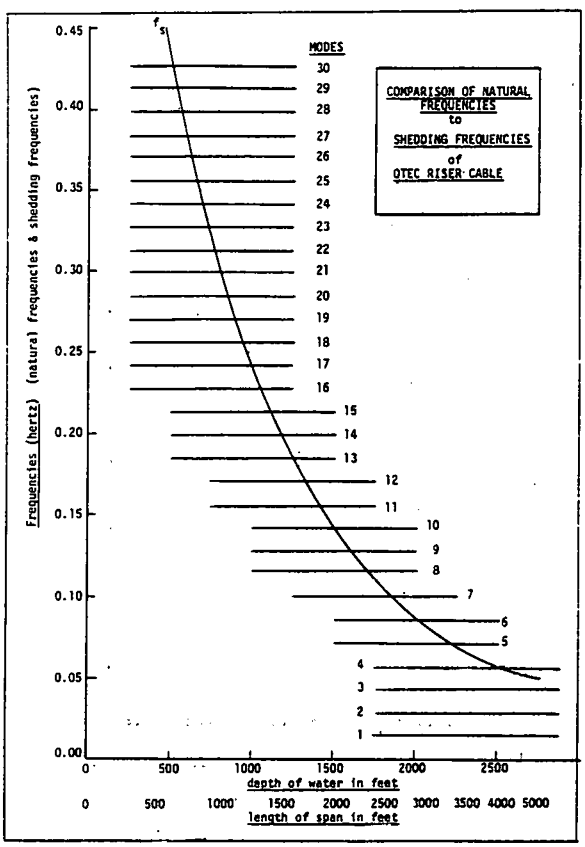

Figure 6. Comparison of f_s and f_n

At which of these frequencies or a combination of frequencies will the riser cable actually vibrate is not known at this time. Various sections of the cable may oscillate at different discrete frequencies.

Griffen, et. al. (References 9, 10, 11) has extensively investigated vortex-induced structural vibrations of cables and provides a great deal of insight into the problem. Unfortunately, some unique qualities of the riser cable sub-system complicate the rational analysis of an already extremely complex phenomena. The riser cable problem departs from cable experiments for which data has been gathered in several ways:

- The flow is not uniform--it varies tremendously in magnitude and direction along the cable.
- The riser cable diameter is up to six times larger than experimental cables.
- The riser cable possesses greater structural damping than experimental cables.
- The axis of the cable varies from being perpendicular to the flow to being parallel.

It is not uncommon for the vortex shedding to excite the cable in very high modes, i.e., 27 or 28. For lower modes, response is pure (vibration in one mode only). However, at higher modes of vibration, several modes participate in the response. Little is known about this kind of behavior. Because determination of the frequency of vibration is particularly important in the fatigue analysis, further investigation is required.

The strumming phenomena as applied to the riser cable problem is related to many variables:

D = cable diameter
δ = cable structural damping
L = cable length
d = plant-buoy separation distance
C_i = current profile
α_j = angle between cable and current
S_i = sea state
ψ_j = platform heading into seas

The fact that the last five elements of this list are random variables, represented by some probability distribution, suggests that the strumming effect on cable loading can also be approached in a probabilistic manner. To account for flow-induced vibration effects on the virtual increase of current drag, Equation (11) must be modified:

$$f(L_3|L_4) = \sum_{\psi}\sum_{S}\sum_{\alpha}\sum_{c}\sum_{d} P_d P_c P_\alpha P_S P_\psi f_*(x) \tag{13}$$

where $P_d, P_c, P_\alpha, P_S, P_\psi$ are the weighting factors for plant-buoy distance, current profile, current-cable angle, sea state, and platform heading, respectively

$f_*(x)$ - short-term probability distribution of increased tension due to current and drag increase due to strumming.

The determination of $f_*(x)$ is complex and the nature of $f_*(x)$ is not known. However, as implied by the notation, it seems appropriate that $f_*(x)$ is not a single value, i.e., 45,000 lbs, but rather has a probability distribution. This seems particularly true in light of the random nature of many of the important variables.

The primary effect on total loading from strumming results from increase in current drag. The secondary effect is fluctuation due to the vibrations. Although these higher frequency load changes are orders of magnitude less than the current drag, fatigue considerations require that L_4 must be accounted for:

$$f(L_4) = \frac{\sum_{\psi}\sum_{s}\sum_{\alpha}\sum_{c}\sum_{d} n_* p_d\, p_c\, p_\alpha\, p_s\, p_\psi\, f_{*v}(x)}{\sum_{\psi}\sum_{s}\sum_{\alpha}\sum_{c}\sum_{d} n_* p_d\, p_c\, p_\alpha\, p_s\, p_\psi} \qquad (14)$$

where n_* = predominant vibration frequency (hz) for a particular short-term case

$f_{*v}(x)$ = short term probability distribution of augmented tension due to vibratory displacements

A drastic simplification to this process can be made if strumming suppression techniques are applied to the problem. Reference 12 investigated three methods of strumming suppression. The first method utilizes a trailing edge fairing which interferes with the vortex interaction in the near wake and disrupts the vortex formation length. The second method utilizes a helically wrapped fairing or ridge which tends to break up the spanwise coherence of vortices by causing a variable location of the separation point. The fairings referred to can be put in five categories: fringe (bunched tufts of strands of flexible material); hair (individual strands of flexible material); ribbons; ridges; and, airfoil shaped modules. The materials used for the fringe, hair, and ribbon fairings are usually nylon, polypropylene, polyurethane, or polyvinyl chloride. The third method is designed to suppress the regular periodicity of the vortex formation by placing a perforated cylinder, or shroud around a plain cylinder (cable).

Each of the three devices successfully reduce or prevent strumming vibration. Two other criterion must be considered, however. The device must work in spite of the large variance in flow directions and incident angles associated with the riser cable problem. Also, since periodic retrieval of the cable is not envisioned, the device should be maintenance free. The first criteria eliminates trailing edge fairings as candidates. For the second reason, as well as higher acquisition costs, the airfoil shaped modules are not recommended. Consequently, the helically wrapped ridges or strakes, and perforated shrouds are the leading candidates for strumming suppression. Both have advantages and disadvantages with respect to each other. The advantage of the shroud over the strake is lower drag. The advantage of the strake over the shroud is lower cost, easier construction and deployment, and lower maintenance. Stability has been obtained by placing these anti-strumming devices at the top 25% to 35% of the cylinder, and in the region of the anti-nodes.

Tests have been performed on helically wrapped fairings at several locations. Results from General Electric, U. S. Navy, Massachusetts Institute of Technology, Naval Undersea Center, National Physical Laboratory, and Woods Hole Oceanographic Institute show reduced strumming but increased drag for various configurations. The increase in drag is on the order of 50 - 100%. Parameters varied in these tests include number of ridges, pitch to diameter ratio, fairing height to diameter ratio, fairing type, angle of attack, and Reynolds number. Typical curves of percent strumming reduction for various pitch to diameter ratios are shown in Figure 7.

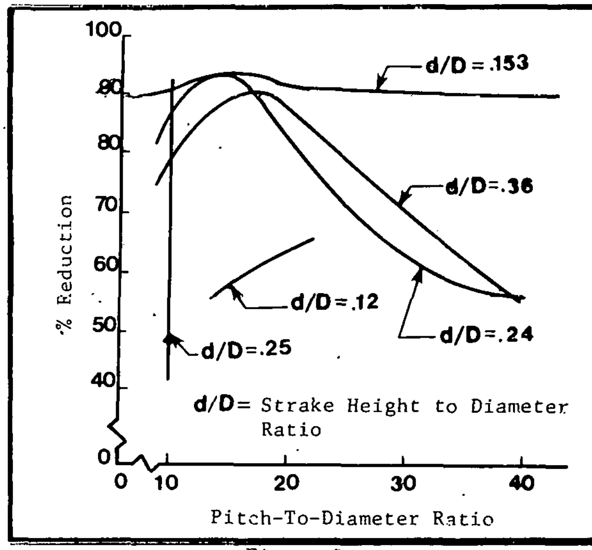

Figure 7.

It is seen that pitch to diameter ratio in the area of 15 seems to be the optimum design. Figure 8 shows pitch to diameter ratio plotted against fairing height to diameter ratio for tests which reduced strumming by 90 percent or greater. This gives an idea of the fairing heights involved. Sharp-edged rectangular strakes have shown to be more effective than circular or rounded section strakes.

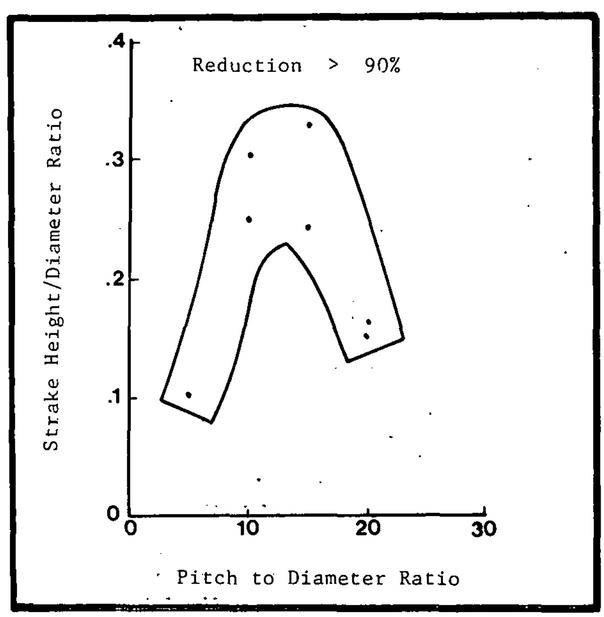

Figure 8.

Perforated shrouds fitted around a cylinder and separated by a gap have been investigated and have shown to be effective against regular vortex shedding. Neither the gap width nor the open-area ratio seem to be critical in reducing strumming, but the best results have shown that a gap width of .12D and an open-area ratio between 20% and 36%

are the most effective.

Should the riser cable be designed with devices to prevent strumming then the determination of the long-term distribution of vibration increased current induced loads $p(L_4|L_5)$ can be dispensed with. The comparatively simpler Equation (11) can be used.

Wave Induced Loads $\{L_5\}$

Depending on the method and location of mechanical termination at the platform, the riser cable will experience inertial and drag forces from the oscillatory flow of the seaway. If the selected plant-cable configuration results in wave induced loading on the cable, the resultant forces can be handled probabilistically:

$$f(L_5) = \frac{\Sigma\Sigma n_* p_s p_\beta f_*(x)}{\Sigma n_* p_s p_\beta} \quad (15)$$

where
p_s = weighting factor for sea state
p_β = angle between seaway and catenary plane
n_* = average number of responses per unit time of short term response
$f_*(x)$ = short term response p.d.f. for given sea state and angle of incidence

If the cable attachment method and location are known, $f_*(x)$ can be determined using procedures detailed in numerous references (13, 14, 15).

Radial Forces $\{L_6\}$

Variations in the temperature of the cable associated with the transmission of electrical power cause variations in the internal stress of the cable micro-structure. The effect of this type of loading is strictly within the domain of the cable designer and shall not be discussed here. However, this mechanical loading can be described as a function of the electrical loading, which in turn can be described by some form of probability distribution or weighting factors.

Another source of radial-type loading is hydrostatic pressure. This is a function of depth, and the lowest point on the catenary can rise and fall between 2,000 - 4,500 feet.

These types of forces are not predominant and may or may not be important to extreme load or cyclic loading criterion. In any case, a probability density function can be developed for each.

Application to Design

The cable designer is concerned with the different ways that the structure can suffer damage or fail. Two types of damage are of particular interest: 1) Exceedance of an ultimate strength limit that initiates electrical or mechanical failure; and, 2) Failure due to cyclic fatigue. To preclude the first mode of failure, the cable designer must know the maximum extreme load to which the cable will be subjected in its lifetime. To prevent fatigue failure, the designer must have a load history for the cable's lifetime, i.e., number of cycles for each level of stress.

Determination of Extreme Loads

Ochi (Ref. 16) has demonstrated that the computations for extreme value for design considerations estimated from the long term prediction method agrees with that estimated from the short term prediction method. The estimation procedure of the extreme values through the short term prediction approach, however, is extremely simple in comparison with that through the long term approach. Therefore, the short term approach will be used here.

Determination of extreme value for design must take into account all sea severities, all varieties of wave spectral shapes, headings to waves, etc., expected in the OTEC platform's lifetime, weighted by the frequency of occurrence. Additionally, the concept of a risk parameter, α, is used. The risk parameter represents the probability that the extreme response in a given sea will exceed the estimated design load. The selection of the value of the risk parameter is at the discretion of the designer. It has been suggested that a value of $\alpha = 0.001$ is appropriate for OTEC related system designs.

By taking these various factors which affect the magnitude of extreme responses into consideration, the probable extreme value and the design extreme value for a sea of specified severity can be evaluated by the following formulas:

Probable extreme value (amplitude):

$$\bar{y}_n = \sqrt{2 \ln \left\{\frac{(60)^2 T}{2\pi}\sqrt{\frac{m_2}{m_0}}\right\}}\sqrt{m_0} \quad (16)$$

Design extreme value (amplitude):

$$\hat{y}_n(\alpha) = \sqrt{2 \ln \left\{\frac{(60)^2 T}{2\pi(\alpha/k)}\sqrt{\frac{m_2}{m_0}}\right\}}\sqrt{m_0} \quad (17)$$

where
T = longest duration of specified sea in hours
m_o = area under response spectrum
m_2 = 2nd moment of response spectrum
α = risk parameter
k = number of encounters with a specified sea in ship's (or marine structure's) lifetime

Figure 9 illustrates the application of this approach. The maximum value is connected with the "100-year" storm, the significant height of which is H_s = 44.2 feet. This value results from loads caused by platform motions. To this value, design extreme values for each of the other loading sources must be calculated. The long term probability distributions f(x) determined for each load source are transformed into cumulative distributions, F(X), where

$$F(X) = \int_{X_{min}}^{X} f(\chi)dX \quad (18)$$

If we have selected a risk parameter, $\alpha = 0.001$ for example, then the design extreme value (X) for that load source will be the value corresponding to $F(X) = (1-\alpha) = 0.999$. In this manner, the design extreme value can be calculated taking into account all load sources.

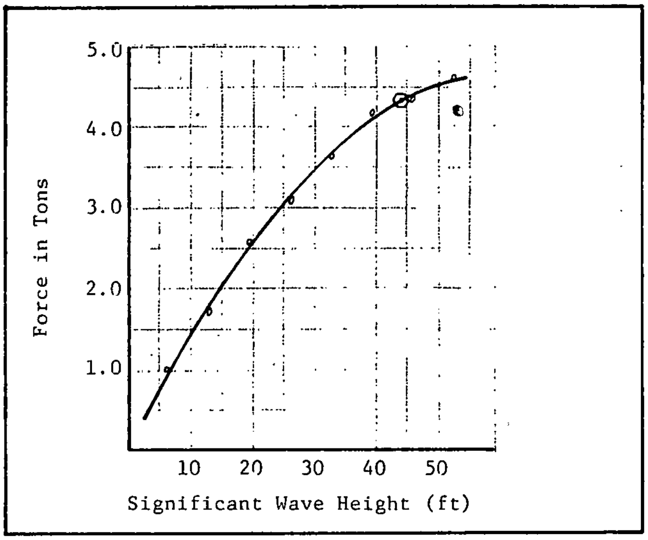

Figure 9. Extreme Values for Various Sea Severities.

Cyclic Loading History

Assuming that the long term probability density function has been determined for each demand source on the cable, the problem remains to combine the loads in a rational manner. At the present time, there is no generally accepted way to syntehsize all the individual p.d.f.'s, particularly considering the dimensions of the problem. Ideally, to develop a joint long-term p.d.f. from all the individual functions would require knowledge of the inter-dependency of all the random variables. This has not been attempted in lieu of a more feasible approach to the problem.

Since the platform motion is the predominent source of loading, its long term p.d.f. will be used as the "baseline". Figure 10 shows the long term p.d.f. for platform motions loading $\{L_2\}$. Since the total number of responses, n, can be obtained from Equation (7a), the number of cycles of various loadings necessary for evaluating possible fatigue failure of the cable can readily be evaluated from Figure 10. The cyclic loading curve which results appears in Figure 11.

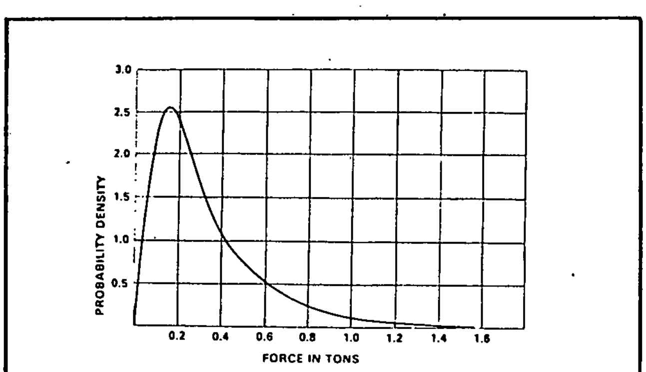

Figure 10. Long Term Probability Density Function of Platform Motion Loading $f(L_2)$.

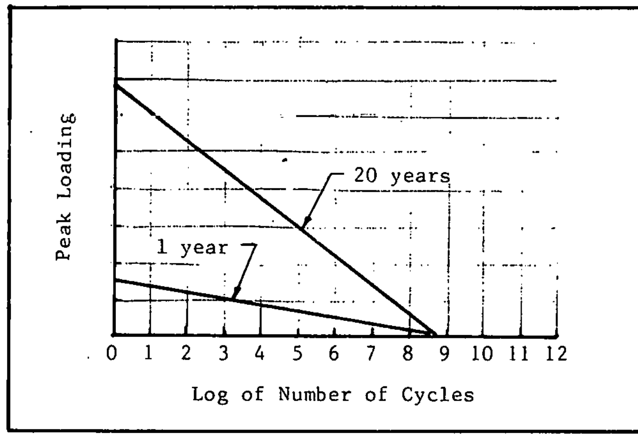

Figure 11. Cyclic Load History from Platform Motion Induced Loading.

The problem that remains is to determine by how much this curve should be moved vertically to account for the additional loads from other sources. One option is to use the design extreme value for each load source, determined using $\alpha = 0.001$ for example. The use of this value, however, would be unrealistically conservative, since it assumes that added to each and every response, over a lifetime, is the sum of the extreme values for all other loading sources. A more realistic approach would be to use the value for which half of all responses would be greater, and half would be less. In terms of the cumulative distributions, this value is associated with $F(X) = 0.5$. This approach would seem appropriate considering the number of responses in a 20-year platform life will be on the order of 10^7 - 10^8 cycles. With such a large sample, the combined $\{F(X) = 0.5\}$ load cycle curve would closely fit to an histogram of the actual load history. This procedure is depicted in Figure 12 on the following page.

This approach provides the cable designer with a cyclic-loading curve with which to carry out fatigue design procedures. Reference 17 outlines a methodology for riser cable fatigue design.

Summary and Conclusions

This paper presents a method to determine loading criterion for the cable designer. The procedure emphasizes the "rational" approach and proposes a methodology that is consistent with modern design practices for structures that operate in the marine environment. Conclusions derived from the results of the present study are summarized in the following:

1. Basic techniques are now available for making rational calculations in probability terms of most of the loads acting on the riser cable, including:

   - static tension
   - platform motion induced
   - current drag
   - wave forces
   - radial loads (hydrostatic pressure, thermal stresses)

   Further development is needed for the calculation of flow induced loads.

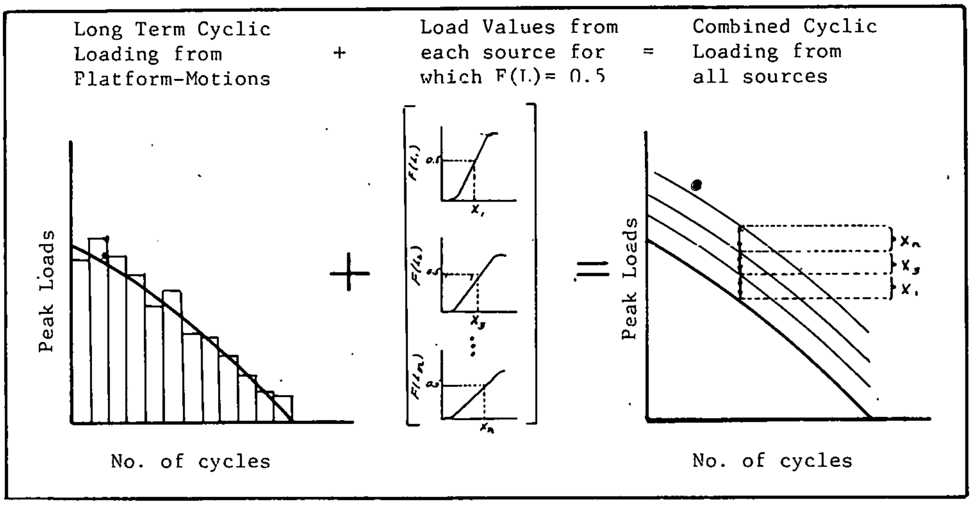

Figure 12. Scheme for Combining Loads for Long Term Cyclic Loading Curve

2. Input data for the calculation of loads on the riser cable is incomplete.

   a. Definition of the station keeping system is required for a rational determination of the p.d.f. for static tension loads.
   b. Definition of the type of cable termination device (i.e., fixed, swivel, etc.) and the location of the termination on the platform is required to determine platform motion induced loading.
   c. Definition of operational guidelines and restraints is required because it impacts on every source of loading, i.e., will the platform attempt to always maintain a certain heading in relation to seas, currents, subsurface buoy, etc.? Will the cable be disconnected in higher sea states?
   d. A definition of a set of design current profiles and their frequency of occurrence for the selected site is needed.
   e. The flow induced vibration problem must be confronted. If comprehensive analysis of the cable and environment prove that strumming is a significant problem, then a suppression device must be selected. These decisions impact on determination of current induced loading, and have a second order effect on platform motion loadings.
   f. Micro-analysis of the cable composite will determine if the radial compression forces from hydrostatic pressure, as well as thermally-induced stress variations are significant.

As the questions about station keeping systems, termination methods, etc., are answered, the data required for the application of the procedures proposed in this paper can be rapidly developed. Finally, attention must be given to the definition of an acceptably low limit for probability of failure.

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8. CLOSED-CYCLE OTEC POWER SYSTEMS

PRELIMINARY DESIGNS OF 10 MW_e and 50 MW_e POWER MODULES

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S. Cunninghis, *Gibbs and Hill*

Abstract

The results of preliminary design studies of a closed cycle 10 MWe modular experiment power system and the related commercial size 50 MWe power system module are presented. Indications are that the major equipment design goals can be met and that both the 10 MWe and 50 MWe modules can be built in existing facilities with the existing technology. Both heat exchangers and rotating equipment are scaled logically from 10 MWe modular experiment size to 50 MWe modules suitable for commercial plants in the 400 MWe range. Features include half size, double speed turbines for the 10 MWe modular experiment, modular tube bundles which are identical for all power levels, and a low loss turbine control system.

Each power module consists of an evaporator, a turbo-generator, a condenser and associated auxiliary equipment. They can be arranged in a variety of ways to fit alternative hull configurations. Heat exchanger shells can be arranged to provide a portion of platform bouyancy. Illustrative plant arrangements have been developed to minimize pressure losses in sea water and ammonia loops.

Capital cost has been optimized for the 50 MWe module size. Titanium was selected as the most economical material currently available for a 30 year plant life. If an aluminum alloy can be developed to qualify for equal service life in salt water, it will be the most cost effective material.

Introduction

The 10 MWe modular application OTEC power system and 50 MWe OTEC power system module preliminary designs discussed in this paper were developed by the Power Generation Divisions of the Westinghouse Electric Corporation under contract to the Division of Solar Technology, Department of Energy. Westinghouse was supported in its work by Carnegie-Mellon University in the areas of advanced heat exchanger technology and control system dynamic modeling; by Union Carbide Corporation also in the area of heat exchanger technology and with tube enhancement techniques; by Middle South Services, Inc. in the areas of electric utility practices and operating procedures; by Dr. A. E. Bergles with tube enhancement techniques; and by Gibbs and Hill Inc. with total power plant integration.

The 50 MWe power system module preliminary design is a development of conceptual designs made by Westinghouse of a 100 MWe demonstration OTEC power plant in a ship-type platform. Cost studies of the conceptual designs showed that minimum total system (power plant plus platform) cost would be achieved with 50 MWe power system modules. Each module is an independent system consisting of one evaporator, one turbine and generator, one condenser and associated piping and auxiliaries. [1]

The 10 MWe modular application OTEC power system is sized in accordance with terms of the DOE contract. Heat exchangers for the 10 MWe system exactly model those of a 50 MWe power system module because the tube bundles or modules are thermodynamically and structurally equivalent to those of the larger plant. The turbine and generator for the 10 MWe system are one-half size, quarter power, double speed versions of cost optimized 40 MWe units. A modest extrapolation from that size leads to units for cost optimized 50 MWe power system modules. [2]

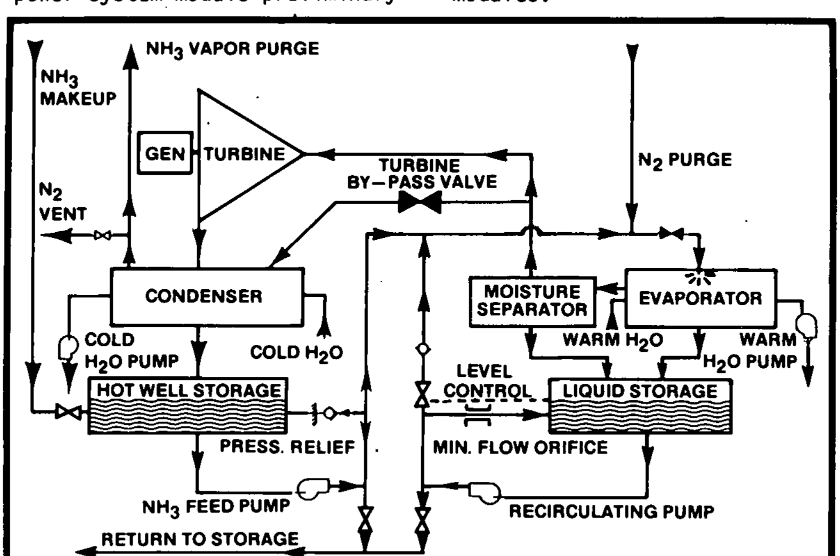


FIGURE 1
PRIMARY POWER SCHEMATIC

System Design

The OTEC power cycle schematic arrangement shown in Figure 1 has been basic to the study of system configurations since early in the conceptual design phase, with the exception of choice of control valves. The valve arrangement shown was selected finally for the preliminary design after detailed study of system control dynamics as discussed later in this paper. The evaporator tube bundles, moisture separators and liquid storage are all integrally contained in the evaporator shell; and the condenser tube bundles and hotwell storage are similarly contained in the condenser shell as discussed later.

Liquid ammonia is supplied to the evaporator as a mix of condensate from the condenser hotwell and recirculated liquid from the evaporator liquid storage. Ammonia vapor, which may contain as much as 10% liquid as it comes from the evaporator tube bundles, flows through moisture separators from which it passes to the turbine with less than 0.1% moisture. Exhaust vapor flows directly from the turbine discharge diffuser into the condenser. To minimize pressure losses there are no series control valves in the turbine inlet piping. Vapor flow to the turbine is controlled by diverting it through turbine bypass valves and by liquid feed and recycle flow valves.

Many design parameters affect the performance and cost of an OTEC plant. Some are strongly interdependent and must be optimized together. These include module size,temperature drop allocations between heat exchangers and turbine, water velocities in heat exchanger tubes, tube diameters and type of enhancement, heat exchanger dimensions, and platform dimensions. A computer-based optimization program was developed and exercised to determine optimum (least cost) combinations of the interacting parameters.

Figure 2 illustrates the logic employed in the economic optimization studies. The central cost optimization program, made up of sub-system design, thermal performance and cost models, used pattern search techniques to find values of continuous system variables which would result in the lowest cost power module for a given set of discontinuous variables. Specific optimum designs were further qualified by assessments of technology constraints, and power plant adaptibility to reasonable platform or hull configurations, producibility, and operability.

The continuous variables addressed were evaporator and condenser tube-side water velocity, seawater outlet temperature, and ammonia saturation temperature. Figure 3 illustrates a typical set of module cost data vs. continuous system variables.

Discontinuous variables addressed were evaporator and condenser tube diameter, material and enhancement, the number of evaporator and condenser shells per module, and net module power output.

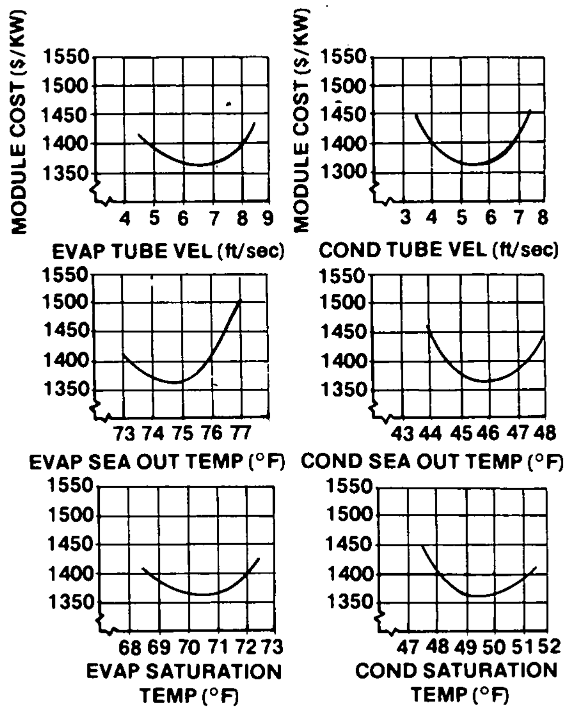


FIGURE 3
MODULE COST VS. SYSTEM VARIABLES

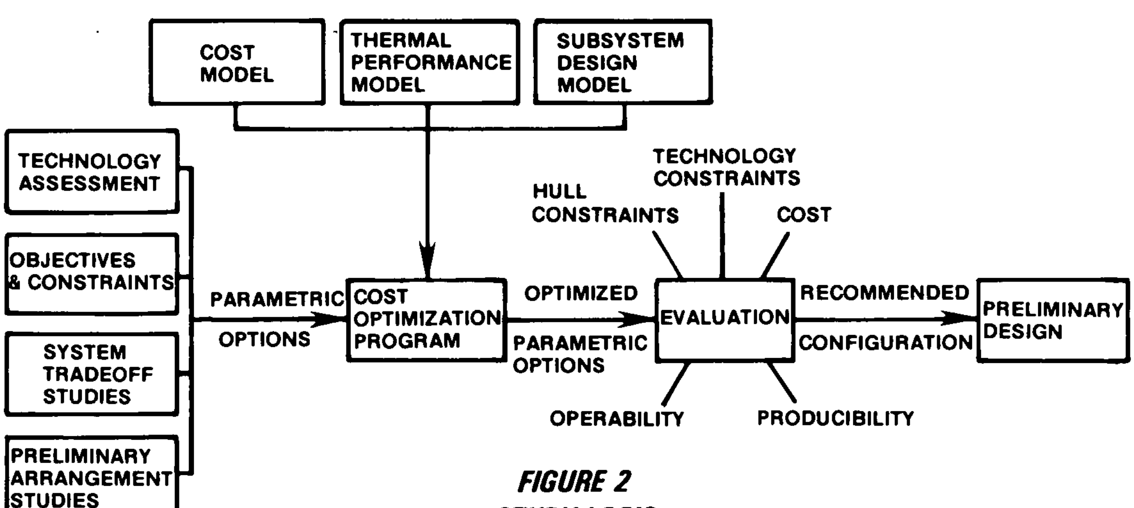


FIGURE 2
STUDY LOGIC

Seventeen types of proven tube enhancement techniques along the available range (1 to 2 inches) of tube diameters were studied. Linde High-Flux coating on the outside of the evaporator tubes with plain inside surfaces and plain-surfaced condenser tubes were shown to be the most cost effective combination. One evaporator and one condenser per module also proved to be the least cost approach. There was a slight decrease in power module cost per kw as module size increased from 50 MWe to 100 MWe. However, associated cost of a surface ship platform reached a minimum with 25 MWe modules. As shown in Figure 4 the resulting cost per kw for the total system appears to minimize at 50 MWe per module. Hence that size was selected as the baseline for subsequent studies. Table 1 lists some of the principal operating parameters derived from the optimization program.

	10 MW	40 MW	50 MW
Flows (10^6 lbm/hr)			
Ammonia	2.99	11.8	14.8
Cold Seawater	263	921	1300
Warm Seawater	305	1190	1500
Temperatures (°F)			
Warm Seawater Inlet	80.00	80.00	80.00
Warm Seawater Outlet	74.57	74.54	74.51
Evaporator Saturation	70.50	70.35	70.25
Turbine Inlet	70.20	70.05	69.95
Turbine Outlet	49.09	50.15	49.59
Condenser Saturation	49.00	50.06	49.50
Cold Seawater Outlet	46.13	46.89	46.17
Cold Seawater Inlet	40.00	40.00	40.00
Performance			
Turbine Efficiency	.790	.799	.800
Seawater Pump Efficiencies	.670	.680	.680
Ammonia Pump Efficiencies	.750	.750	.750
Condenser Effectiveness	.681	.685	.647
Evaporator Effectiveness	.572	.566	.564
Power (MWe)			
Turbine/Generator Gross	14.42	54.09	70.01
Cold Seawater Pump	2.30	6.74	10.05
Warm Seawater Pump	1.52	5.31	6.65
Ammonia Feed Pump	0.28	1.14	1.49
Ammonia Recycle Pump	0.05	0.27	0.11
Chlorination	0.17	0.63	1.21
Estimated Hotel Load	0.10	0.0	0.50
Net Power Output	10.00	40.00	50.00

Table 1
POWER MODULE COMPARISON

Fluid handling to minimize pressure losses was a major concern in developing system arrangements. This applied both to the ammonia working fluid and to the warm and cold sea water. For instance, it has been estimated that as little as a 1 psi increase in sea water pressure drop would cause a 3% increase in total plant cost. Therefore particular care was exercised in the design of sea water passages to ensure low flow velocities and to minimize changes in flow direction.

Figure 5 and Figure 6 illustrate the 10 MWe modular application OTEC power system installed in a barge-type surface platform. Principal features are:

. A linear arrangement of cold water discharge plenum, condenser, cold water entrance plenum, warm water entrance plenum, evaporator, and warm water discharge plenum.
. Turbine-generator mounted directly above the condenser for the shortest practical vapor discharge path commensurate with adequate exhaust diffusion.
. Relatively short direct runs for all ammonia vapor and liquid piping.
. Direct introduction of warm water into the evaporator and cold water into the condenser, and compact discharge plenums directly over the seawater pumps.
. Below hull location of the seawater circulation pumps, their diffusers, and the ammonia recirculation and feed pumps, provides sufficient suction head and minimizes volumetric requirements while meeting all spatial constraints.
. Provision of a removable bulkhead between the cold water and warm water entrance plenums so

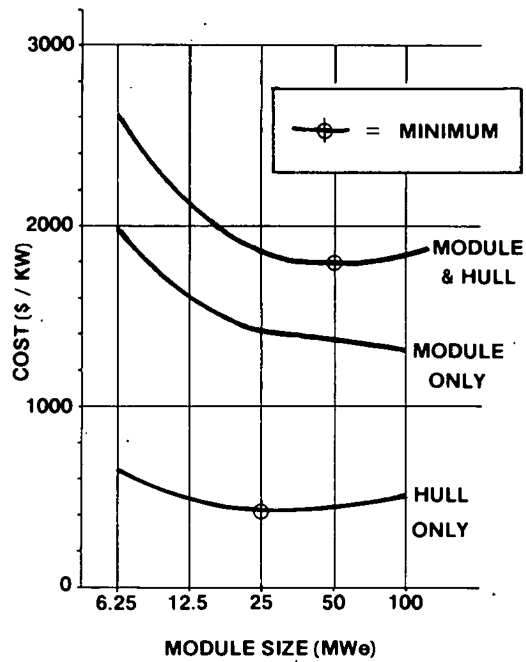


FIGURE 4
MODULE AND HULL COST

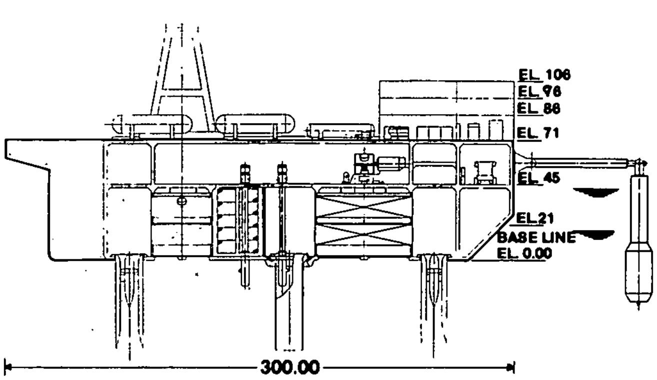


FIGURE 5
10 MWe POWER SYSTEM PROFILE

that their combined length provides adequate pull-out space for the longer condenser tube bundles without exacting an unnecessary volumetric penalty on the design.

. Provisions of unobstructed vertical access from the gantry crane to all major components and auxiliary elements which require lifting services during initial installation or during routine or emergency maintenance.

. Provision of gates or temporary closure capability at all hull openings.

It should be noted that the "ballast" or wing tanks are not essential to the volumetric requirements of the power plant. They are needed to provide adequate stability in a surface platform configured for a single power module.

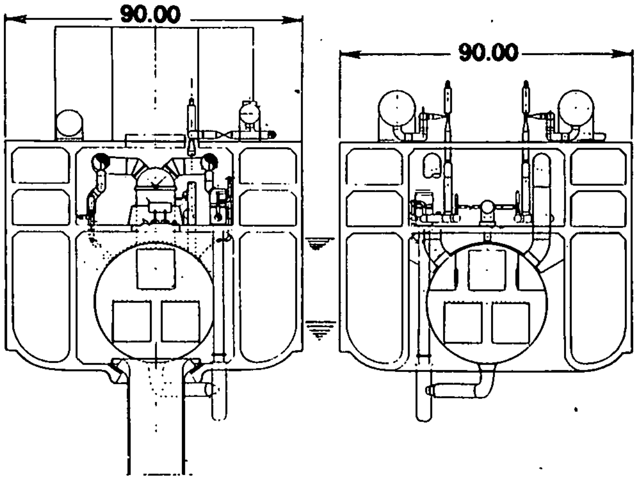


FIGURE 6
10 MWe POWER SYSTEM
SECTIONS 1-1 & 2-2

Figure 7 and Figure 8 illustrate two of several arrangement configurations of 50 MWe OTEC power modules which were developed to accomodate a variety of potential 400 MWe commercial OTEC plant platform concepts.

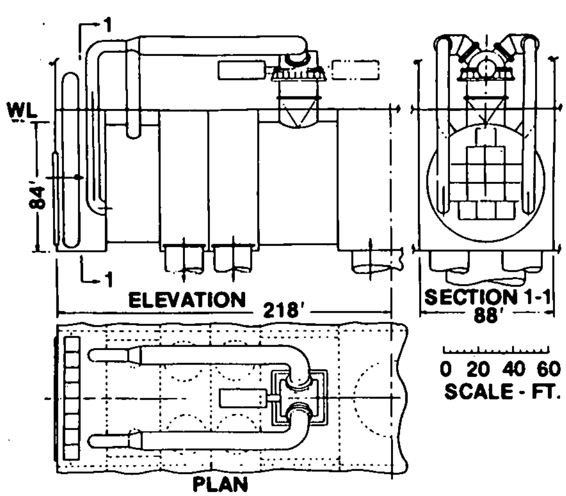


FIGURE 7
50 MWe OTEC MODULE
"BACK TO BACK" CONFIGURATION

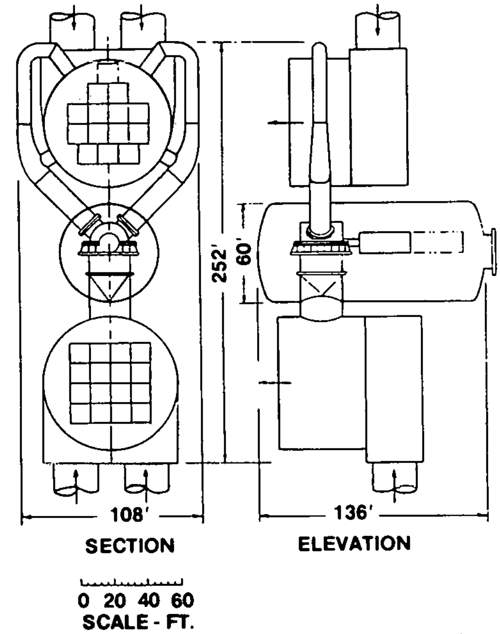


FIGURE 8
50 MWe OTEC MODULE,
STACKED CONFIGURATION

The "back to back" configuration of Figure 7 reverses the 10 MWe module arrangement. Sea water plenums are at the ends and discharge plenums are in the middle of the module. It was designed for surface ship-type platforms. Figure 9 shows two back-to-back modules in a quarter of a 400 MWe ship-type platform of approximately the size and configuration proposed in the conceptual designs developed by Gibbs and Cox.

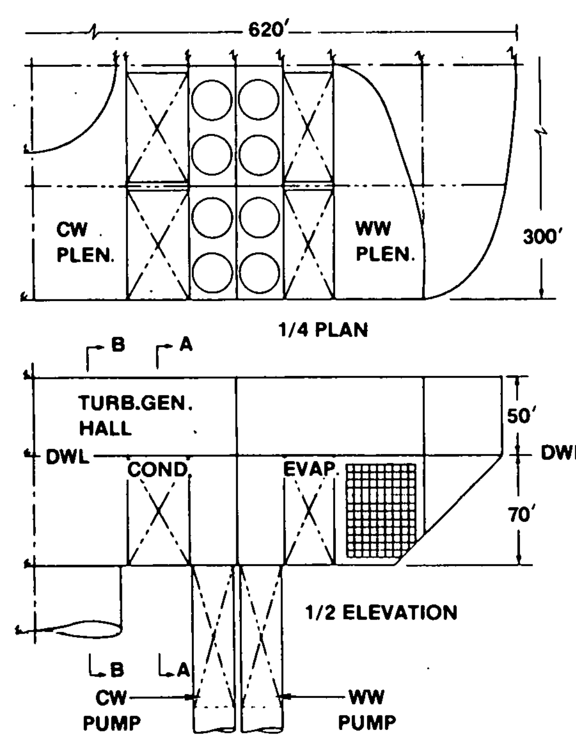


FIGURE 9
400 MWe SHIP-TYPE PLATFORM

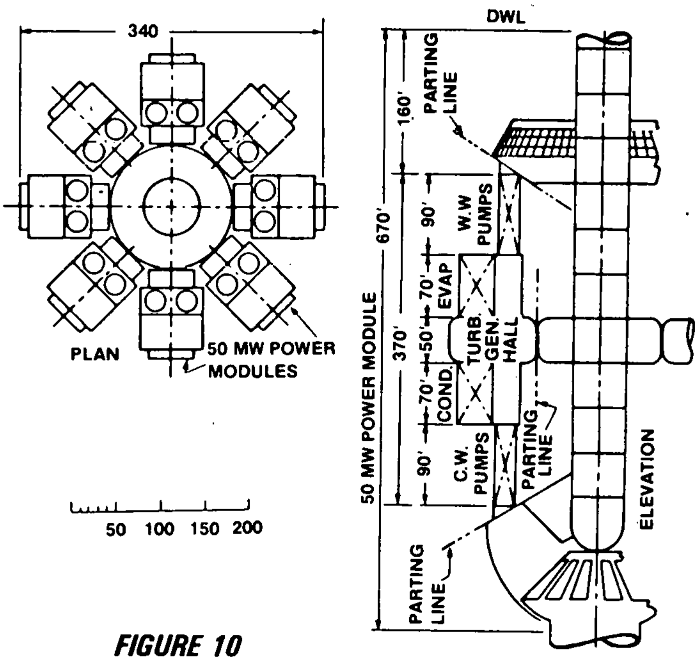


FIGURE 10
400 MWe SPAR PLATFORM

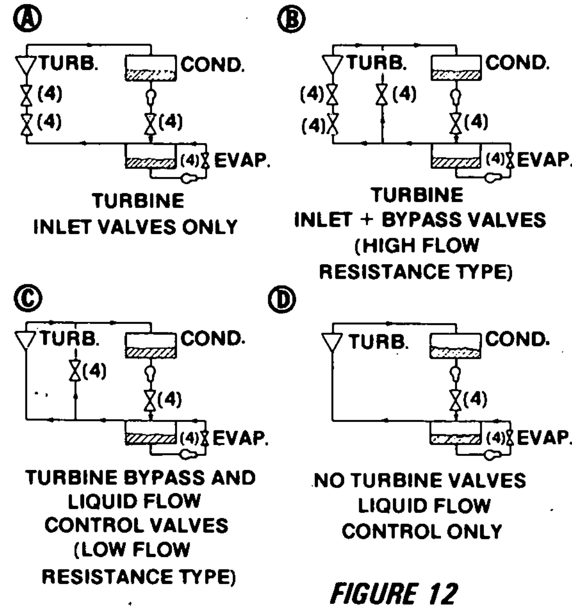


FIGURE 12
CANDIDATE CONTROL CONFIGURATIONS

The "stacked" configuration of Figure 8 was designed for semi-submersible and spar platforms. Figure 10 shows stacked modules on a 400 MWe spar platform similar to that proposed by Lockheed.

System Operation

The OTEC power modules are designed to operate between a warm seawater temperature of 80°F and a cold seawater temperature of 40°F. Biofouling is assumed to be controllable to a thermal resistance of 0.00025 BTU/Hr-Ft^2-°F. Studies of the effects of off-design characteristics indicate that the plant will experience a 30% drop in net power output for each 5°F drop in available ΔT. It will experience a 10% drop in net power output for each doubling of fouling resistance.

Dynamic response of the system to changes in load characteristics, and in valve settings were matters of major concern. Figure 11 indicates the response of the 10 MWe turbine to a load drop and the degree of overspeed control exercised by different combinations of valve operation. Design studies of four proposed valve configurations for controlling the plant were made. The candidate configurations are shown in Figure 12. Response characteristics of the plant under each control configuration were determined for a variety of operating situations through the exercise of a dynamic computer model. Operations evaluated included:

- Initial purging of air and N_2
- On-line purging of non-condensible gases
- System start-up
- Synchronization to an electrical grid
- Turbine speed and load control
- Condenser and evaporator hotwell liquid level control
- On-line valve testing
- Emergency trip response

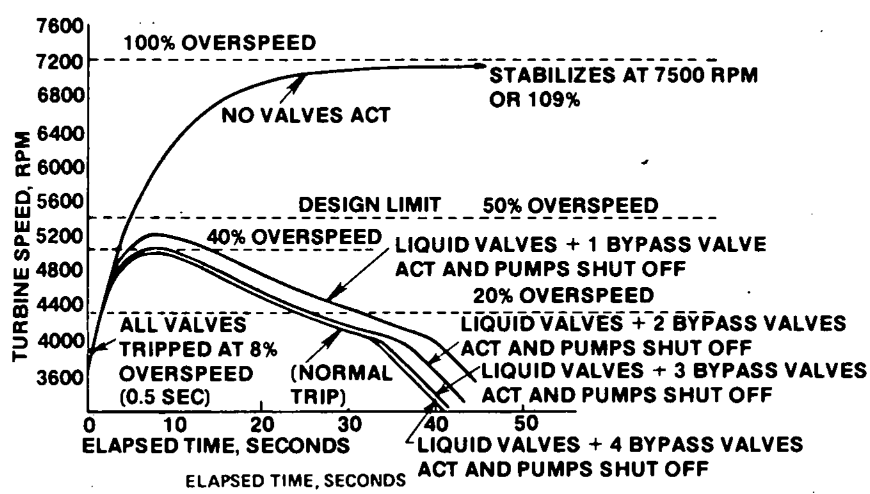


FIGURE 11
10 MWe TURBINE OVERSPEED PROTECTION

- System shut-down
- Purging NH_3 from the system

It was determined that system (C) using turbine by-pass valves and liquid flow control valves would give the best control over all normal and emergency operating conditions.

Ammonia storage is provided in four deck-mounted tanks of sufficient capacity to take the entire ammonia power cycle charge plus a reasonable reserve for make-up. During a shut-down mode or a maintenance cycle ammonia is pumped from the power cycle back into the storage tanks using both the ammonia feed pump and the ammonia recirculation pump. Vapor is removed and sent to storage by a compressor. During normal operation impurities are removed from the ammonia by a system consisting of a pressure tank, a compressor, and a distillate column.

Nitrogen is used to purge the ammonia power system prior to the introduction of ammonia, and again after ammonia has been removed prior to maintenance operations. A cryogenic storage system has been selected as the least expensive method of handling the nitrogen.

Primary fouling control will be achieved with a chlorination system. Back-up control will be with an AMERTAP mechanical cleaning system.

Major Components

Heat exchangers that are economical to build, efficient in operation and readily produceable are key to the successful development of OTEC power plants. The heat exchangers developed in these conceptual and preliminary design studies meet those three criteria. Building economy and produceability were obtained through the modular tube bundle approach. Maximum operating efficiency was ensured through detailed evaluation and trade-off studies of all known surface treatments for heat exchange enhancement over a range of tube diameters. Principal characteristics of 10 MWe, 40 MWe and 50 MWe condensers and evaporators are listed in Table 2.

Figure 13 illustrates the arrangement of a 50 MWe evaporator. Figure 15 shows a 50 MWe condenser. Each heat exchanger contains fourteen 12.5-foot x 12.5-foot modular tube bundles with approximately 1000 1-inch diameter titanium tubes each. Tube bundles for the evaporator are 36 feet long. Those for the condenser are 53 feet long. The 10 MWe heat exchangers contain three tube bundles each

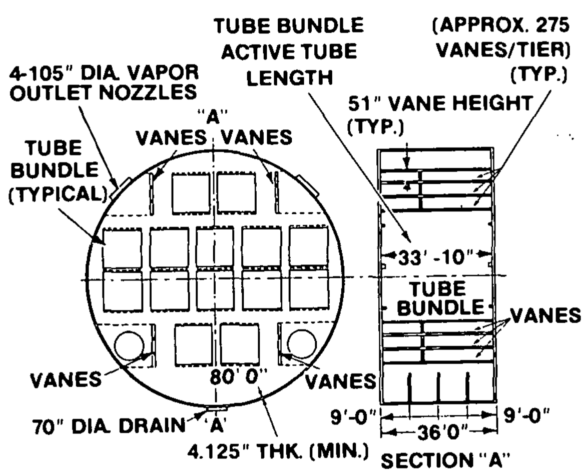


FIGURE 13
EVAPORATOR, 50 MWe

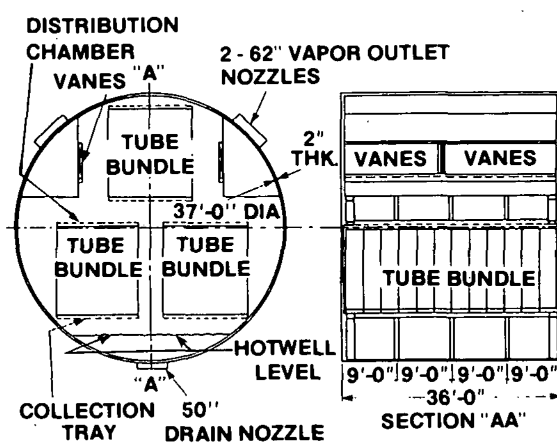


FIGURE 14
EVAPORATOR (10 MWe)

	CONDENSER			EVAPORATOR		
	10 MW	40 MW	50 MW	10 MW	40 MW	50 MW
Tube Velocity, ft/s	5.6	5.1	5.6	5.5	6.5	6.5
h (water), BTU/hr-ft^2-°F	854	804	862	1160	1160	1160
h (foul), BTU/hr-ft^2-°F	3780	3780	3780	3780	3780	3780
h (metal), BTU/hr-ft^2-°F	4820	4820	4820	4770	4770	4770
h (ammonia), BTU/hr-ft^2-°F	1870	1830	1920	4930	4980	5010
H.T Coeff., BTU/hr-ft^2-°F	459.3	441.6	464.2	648.2	649.0	649.5
L.M.T.D., °F	5.37	5.97	5.90	6.40	6.54	6.63
Surface Area, ft^2	623007	2294117	2787795	382786	1468648	1831138
Tube O.D. inches	1.0	1.0	1.0	1.0	1.0	1.0
Tubes	42150	159692	206453	42107	164613	206649
Bundles	3	12	14	3	12	14
Bundle Length, ft	56.5	54.9	51.6	34.7	34.1	33.8
Tubeside Press Drop, PSI	3.72	3.14	3.47	2.90	2.82	2.81
Amertap Press Drop, PSI	0.29	0	0	0.29	0	0

Table 2
HEAT EXCHANGER COMPARISON

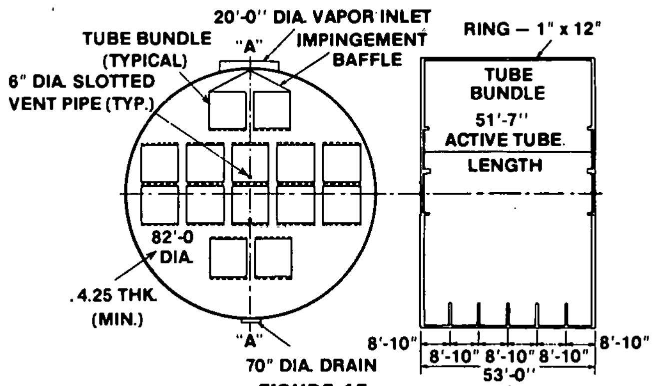


FIGURE 15
CONDENSER, 50 MWe

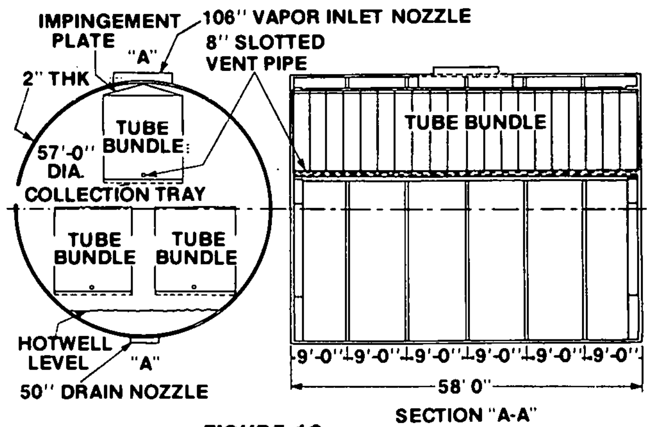


FIGURE 16
CONDENSER (10 MWe)

with identical cross sectional characteristics to those of the 50 MWe units. The 10 MWe is illustrated in Figure 14; the condenser is illustrated in Figure 16.

The relatively small size of each tube bundle permits the use of thinner tube sheets, and all production operations to take place in existing heat exchanger manufacturing facilities with their automated, ganged drilling and tube assembly machines. The large number of standardized parts (tube sheets, tube support plates, boxes, rails, tubes, stiffeners, etc.) will permit lot production even for a small number of heat exchangers with attendant manufacturing cost savings. Tube bundles will be rail-shippable to the power plant assembly facility. At that location the shells and supporting grids of I-beams and rails can be fabricated using standard ship construction techniques. Figure 17 is a conceptual illustration of assembly procedures which could be followed.

Each tube bundle is a completely independent unit. In the evaporator, liquid is distributed to a bundle by an integral covered and perforated tray. The cover permits pressurization to equalize flow to all perforations and ensures against sloshing and splash-over as a result of platform motion. Perforations are sized and located to ensure uniform liquid distribution to the evaporator tubes. Excess liquid is collected in drain trays attached to each tube bundle and piped to a liquid storage area in the lower portion of the shell where it is held to meet recirculation flow demand. Ammonia vapor flows from the tube bundles to moisture separator vanes also contained within the evaporator shell and thence through four outlet nozzles to the main ammonia vapor line and the turbine. Each condenser tube bundle is independently vented. They have attached drain trays, as do the evaporator bundles, which are piped to the condensate hot-well located in the lower portion of the condenser shell. There condensate is stored to meet ammonia feed demands.

Bundle positioning in the shells is designed to maintain low enough vapor flow velocities to minimize pressure loss, to prevent damage to heat exchanger internals, and to minimize re-entrainment and carry-over of liquid droplets in the evaporator. Tube support plates in the bundles prevent

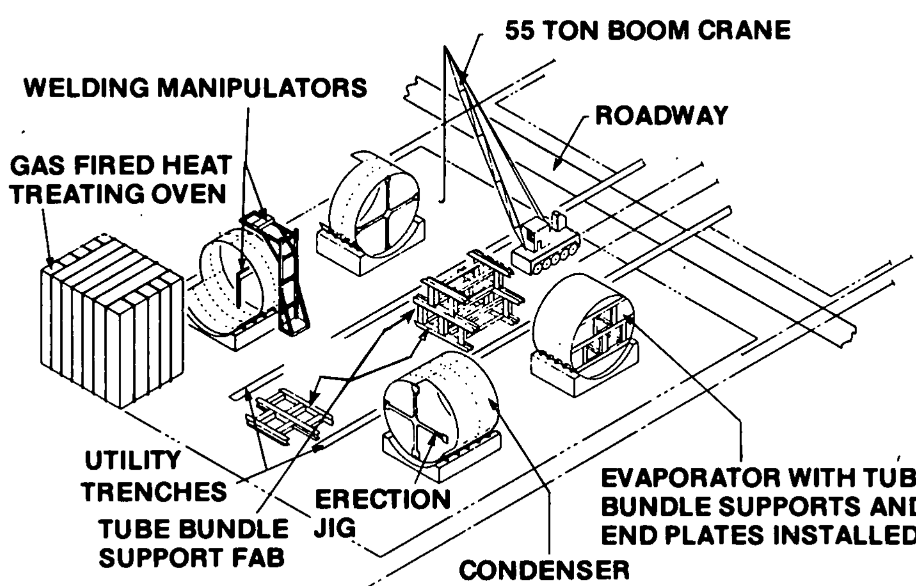


FIGURE 17
CONDENSER-EVAPORATOR ASSEMBLY

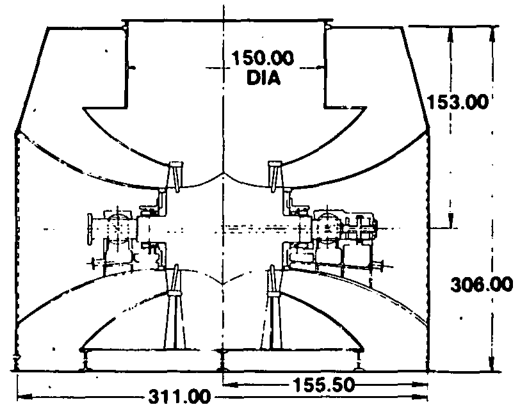
flow-induced tube vibration with its potential for tube damage.

Figure 18 shows a longitudinal section through the 50 MWe turbine. It has a double flow configuration with a single reaction stage at each end. The turbine for the 10 MWe modular application plant has a similar configuration. It's length, however, is 135 inches as compared with the 50 MWe turbine length of 311 inches. Figure 19 illustrates the 10 MWe (Net) generator.

Vapor enters the 50 MWe turbine through two 62-inch diameter pipes in the turbine cylinder cover. The vapor exhausts through a 135-inch by 140-inch rectangular opening in the base. The over-all cylinder and support configuration and rotor and blade structural and aerodynamic designs are very similar to large low pressure central station steam turbines. Therefore, design details have been based upon Westinghouse's vast experience in designing such units.

The rotor has a flexible shaft design with two integral blade discs. Critical speeds have been calculated, taking into account bearing oil film damping and bearing pedestal stiffness, to ensure that critical speeds are outside the range of 10% below to 10% above running speed.

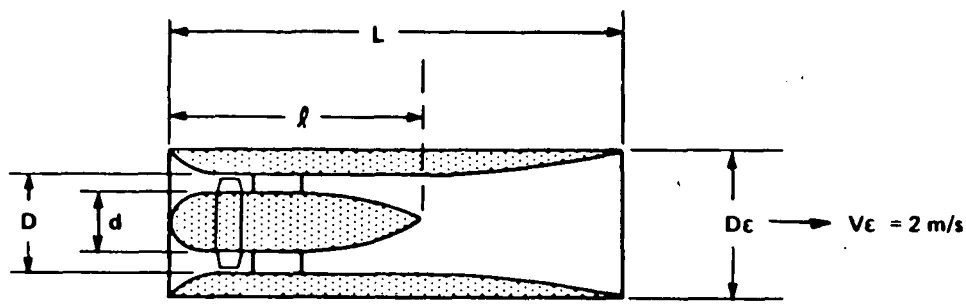
To ensure zero air leakage into the turbine or ammonia leakage to the atmosphere, hydraulic-type gland seals have been designed. Each gland seal ring contains two oil lubricated babbitted steel rings designed as air side and vapor side seals. The seals are supplied by hydraulic loops.



NOTE: INLET & EXHAUST SIZE BASED ON GAS VELOCITY OF 90 FT/SEC.

ESTIMATED WTS	LBS
CYLINDER COVER	73,500
CYLINDER BASE	80,000
ROTOR WTS COMPLETE	85,300
GLAND & JOURNAL BRG COVER	450
COUPLING COVER	300
TOTAL WTS	239,500

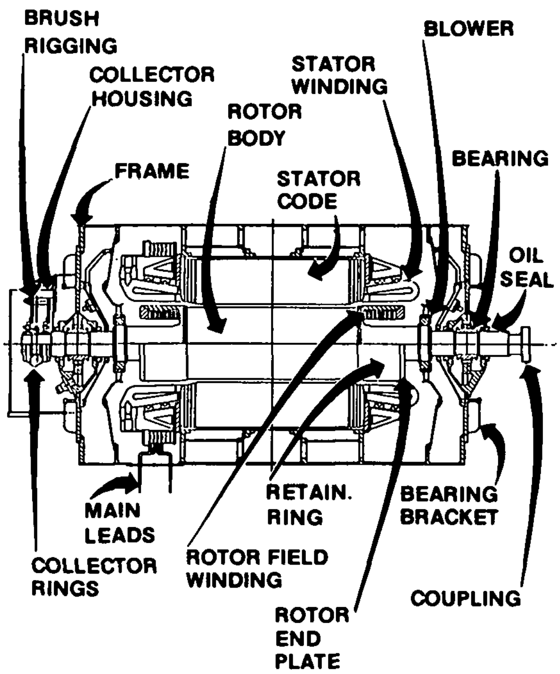
FIGURE 18
LONGITUDINAL SECTION
OTEC TURBINE 50 MWe NET



NUMBER OF ROTOR BLADES = 6
NUMBER OF STATOR BLADES = 11

APPLICATION	CAPACITY	HEAD	SPEED	POWER	EFFICIENCY	CASING DIA.	POD DIA.	LENGTH OVERALL	POD LENGTH	EXIT DIA.
	Q (m^3/s)	H (m)	N (rpm)	P (KW)	η' (%)	D (m)	d (m)	L (m)	ℓ (m)	D_ε (m)
50 MW_e MODULE	100	3.5	55	4650	76	4.90	2.91	26.3	11.6	8.00
10 MW_e MODULE WARM WATER	37.6	2.7	60	1267	80.6	3.43	2.30	19.6	17.1	4.89
10 MW_e MODULE COLD WATER	32.3	4.7	98	1911	80.6	2.76	1.85	18.9	14.6	4.52

Table 3
OTEC SEAWATER PUMP DIFFUSER
DESIGN CONFIGURATION



14.5 MW 0.80 PF 3PH 60 HZ 3000 RPM
TWO BEARING ENCLOSED AIR COOLED TURBINE GENERATOR USED WITH SEPARATE STATIC EXCITER

FIGURE 19
LONGITUDINAL SECTION
10.5 MWe (Net) GENERATOR

The air side loop is vented to atmosphere. The vapor side loop is pressurized and closed. Back-up static seals and static seal pistons have been designed to ensure against ammonia leakage during shut-down. The design is based upon rotating oil seals used for many years on large hydrogen-cooled generators.

Seawater pump configurations for both the 10 MWe and 50 MWe power plants combine an axial flow impeller direct coupled to a low speed synchronous motor in an integral pressurized pod design. Pump speed control will be provided by a remote solid-state variable speed control system. This system uses solid-state techniques to vary motor field frequency to control speed. Flow rate, pumping head and principal dimensions of warm and cold water pumps for both plants are shown in Table 3. The 37.6 m^3/sec at 2.7 m head warm water requirements, and 32.3 m^3/sec at 4.7 m head cold water requirements of the 10 MWe plant can be met by one pump in each loop. The 178 m^3/sec at 3.5 m head warm water, and 173 m^3/sec at approximately 3.5 m head cold water requirement of the 50 MWe plant will be met by four 100 m^3/sec pumps, two each for the warm and cold water loops respectively. Although such pumps are very large, propeller-type pumps of comparable size have been constructed. The pod configuration is becoming increasingly favored in European hydro-electric plant designs. To maintain optimum over-all pump efficiencies, long discharge diffusers are included. These limit discharge velocities to values less than 2 m/sec.

The ammonia feed and recirculation pumps for the 10 MWe modular application power plant are standard industrial vertical turbine type. Motor drive for the pumps will be located 45 feet above the impeller. The pumps will be located below the hull to provide adequate suction head. The line shaft is located in sections of flanged column pipe which support the shaft and serve as the pump discharge line. Pumps of this type with 50-foot or longer line shafts are common in industry. One manufacturer reports having fabricated several with line shafts on the order of 450 feet long. The ammonia feed pump for the 10 MWe plant will have a 600 hp, 3 phase, 60 cycle, 6900 V drive motor. The recirculation pump will be powered by a 70 hp, 3 phase, 60 cycle, 480 V motor.

Control valves for the ammonia power loops consist of vapor (turbine) by-pass valves, and liquid feed and recirculation valves. Requirements for size, flow capacity, and response time were determined by system steady-state and dynamic analysis. Plug-type valves were selected as best providing the required positive control response and fast action. They also have good vibration resistance and provide positive sealing against leakage of the process fluid to the atmosphere.

Dump or by-pass valves are particularly critical. To ensure against exceeding turbine-generator over-speed limits they must be fail-safe, and must have a full opening response time of less than 5 milli-seconds. This speed must be designed into 20-inch diameter valves for the 10 MWe plant and 32-inch diameter valves for the 50 MWe module.

Supporting auxiliary systems for which specifications have been developed for the 10 MWe modular application OTEC plant include a circulating sea water and component cooling water (CCW) system, a compressed air system, a start-up/standby diesel generator system, and an auxiliary or "ship's service" electrical distribution system. The circulation water system will supply approximately 9700 gpm of 40°F water to one of two sea water/fresh water heat exchangers as well as to the chlorination system and a screen wash booster pump. Approximately 5000 gpm of 80°F sea water is required to the second CCW heat exchanger. Demineralized (fresh) CCW will be supplied from the two heat exchangers at 45°F and 75°F respectively for ammonia storage tank cooling and for component cooling water. The compressed air system will supply start-up for the diesel generator, seal air for the warm and cold sea water circulation pumps, and service and instrument air throughout the plant.

A 4700 kw diesel generator will furnish stand-by and system start-up power. The set will be skid-mounted with principal engine auxiliaries. The engine will be a Vee-type turbo-charged, four stroke cycle diesel designed for automatic start on receipt of a remote signal, and can be fully loaded within 10 seconds. Fuel oil storage facilities sufficient for 170 hours operation will be supplied. The auxiliary electrical power conditioning and distribution system will take 6900 V, 3 phase, 60 Hz power and distribute it to various ship's service systems. These include a 480 V, 3 phase, 60 cycle distribution system; a 110 V, single phase, 60 cycle uninterruptible power supply system, and a 120/208 V, 3 phase, 4 wire lighting system.

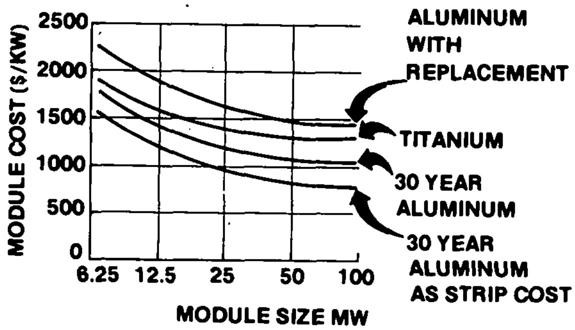


FIGURE 20
POWER MODULE COSTS

Cost Estimates

Figure 20 shows the estimated cost in $/kw of OTEC power modules as a function of module size (MWe). A major driving function is the cost of the heat exchangers which in turn is most strongly influenced by the cost of tube material. Available aluminum tube alloys would have to be replaced at least once (and possibly twice) in the assumed 30-year operational life of an OTEC plant, hence show the highest cost. If a 30-year life aluminum alloy were available and tube cost was based soley on the strip cost from the mill, e.g. no fabrication costs for forming the strip into tubes were included, the minimum power module costs shown would prevail. This in effect shows the floor below which it is unlikely that closed cycle OTEC power modules can be driven.

OTEC Plant Platform	FT^3/KW	Tons/MW	$/KW
100 MW Closed Cycle			
Concept. Des. (Steel)	172	3140	1109
10 MW Close Cycle			
Prel. Des. (Steel)	175	2550	1029
(Concrete)*	175	2550	393
50 MW Closed Cycle Module			
Prel. Des. (Steel)	74	1213	428
(Concrete)*	102	1695	225

Table 4
PLATFORM PACKING
FACTOR AND COST DATA SUMMARY

To obtain total power system costs, other factors such as platform, cold water pipe, mooring, and electrical transmission system costs to a grid must be added. Table 4 indicates packing factors and estimated costs for platforms which were developed in the course of these studies. Significant reductions were achieved from the conceptual design results. It also appears that reinforced and post-tensioned concrete construction will offer significant savings over steel for OTEC platforms.

Conclusions

The results of the preliminary design studies reported in this paper indicate that the power system modules for an OTEC power plant can be designed and constructed using proven technology and at a cost that appears competitive with other alternative energy plants.

References

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   Final Rcport, January 30, 1978
   Contract No. EG-77-C-03-1569
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   by - Westinghouse Electric Corporation
   Power Generation Divisions
   Lester, Pennsylvania

2. Phase 1 Preliminary Design
   Final Report, December 4, 1978
   Contract NO. EG-77-C-03-1569
   Department of Energy, Division of Solar Technology
   by - Westinghouse Electric Corporation
   Power Generation Divisions
   Lester, Pennsylvania

DISCUSSION

W. Howerton: What kind of a cleaning system are you proposing?

R. Miller: We are using chlorine as the major method; chlorine injection and Amertap as a backup, and I understand that there is a chlorine experiment going on now out at Pearl Harbor for the Navy by the principal manufacturer of chlorinators in the States, Eberhard. They have been injecting it into a submarine condenser since January (5 months). The injection rate is well under the requirements of the environmentalists, and apparently it has been holding the fouling to very low levels.

Question: Is any consideration given to taking the heat exchangers out for cleaning one at a time?

R. Miller: We would prefer not to, because of the down time and the loss to a grid. It may become necessary periodically, but I believe that the chlorine plus the Amertap system will control fouling between regular overhaul periods. There will, undoubtedly, be a down time for routine maintenance as there always is in a power plant.

R. Lyon, Oak Ridge: Since your fuel costs are essentially zero in this system, and the cost of operation is not a function of your power production, I wondered if, as a possible alternative to control of vapor flow, you had considered dumping power through resistors rather than controlling the vapor when your load drops off?

R. Miller: I suspect that the resistor banks that you would need to handle that emergency would be of significant size and cost themselves, and I think the valve solution is probably the more economical. However, we did not look at resistor banks.

K. Read: With regard to the first question asked by Bill Howerton, for the 10- and the 50-MW_e systems, the cleaning system is, indeed, chlorine backed up by Amertap, but to play it safe on OTEC-1, the test article will use Amertap backed up by chlorine. Another point with regard to that test article is that, although the design work of Westinghouse shows that titanium is the preferred material for the 10- and 50-MW_e systems (because it is, at this point, unlikely that aluminum will last thirty years, and therefore would have to be replaced), Westinghouse, in order to make a contribution to the overall program's evaluation of aluminum, has contracted to provide an aluminum test article for OTEC-1.

DESIGN OF A 10-MWe(NET) OTEC POWER MODULE USING VERTICAL, FALLING-FILM HEAT EXCHANGERS

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Abstract

The 10-MWe(net) OTEC power module preliminary design discussed ~~in this paper~~ was developed by a team led by TRW during Phase I of the Department of Energy's "OTEC Power System Development Utilizing Advanced, High Performance Heat Transfer Techniques" (PSD-I) program. The PSD-I power system employs vertical, falling film, shell-and-tube heat exchangers with titanium tubes with the ammonia working fluid on the shell side. The heat exchangers are externally mounted, immersed in seawater. The evaporator achieves an overall heat transfer coefficient (U) of about 900 Btu/hr-ft^2-°F, and the condenser a U of about 800 Btu/hr-ft^2-°F. The performance of these heat exchangers is based on experimental data taken at Carnegie-Mellon University and the Oak Ridge National Laboratory. The power system offers a compact arrangement integrated within surface-coupled (e.g., barge) and surface decoupled (e.g., spar type) hulls. No major technical problems were unsolved.

Introduction

The 10-MWe(net) OTEC power module preliminary design discussed in this paper was developed by a team led by TRW from August 1977 to October 1978 under Contract EG-77-C-03-1570 to DOE as part of DOE's PSD-I program. The heat exchanger concept selected exploited vertical-tube, falling-film-ammonia film on the shell side and longitudinal flutes that enhance heat transfer.[1,2] Figure 1 shows the arrangement of the main components including the externally mounted heat exchangers (immersed in seawater), the tube cleaning machine on top of the heat exchangers, the turbine/generator and ammonia piping, and pumps. The arrangement is compact, yet provides for space and access for maintenance and repair.

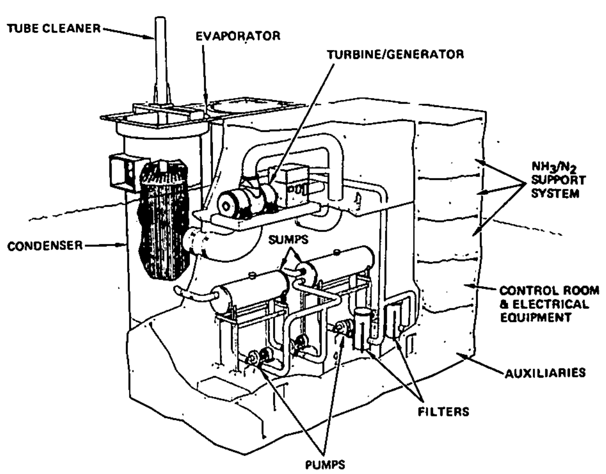

Figure 1. 10 MWe Power System Module

The evaporator and condenser achieve overall heat transfer coefficients of about 900 and 800 Btu/hr-ft^2-°F, respectively, based on experimental data taken at Carnegie-Mellon University (CMU) and the Oak Ridge National Laboratory. For minimum risk, titanium was selected for the tubes based upon its expected long life in a seawater environment. Fabrication of titanium into the desired flute configuration has been verified using an upsetting/rolling process. Aluminum tubes can easily be incorporated in the design if aluminum is qualified[3a] for this service.

The design includes a unique open top which simplifies the water box construction, reduces cost, and allows access to the waterside of the tubes for noninterruptive tube cleaning and repair. Per DOE direction, the heat exchangers are externally mounted and immersed in seawater. This results in a savings of containment hull volume at the expense of a more hostile environment as compared to internal mounting.

The piping is configured with long radius ells to minimize pressure drops and parasititc power losses and has welded joints to minimize ammonia leaks. The small ammonia and nitrogen makeup, letdown, and vent piping does not impact the system arrangement and has been omitted for clarity. Space has been provided for turbine removal for maintenance.

The heat exchangers are attached to the hull with coffer dams for at-sea makeup of through hull piping fittings. Flexible couplings are provided on both sides of the through-hull interfaces to provide for fabrication alignment and relief of differential movements between the hull and power system components. Open troughs are used for cold and warm water distribution to eliminate water hammer effects inherent in closed systems, decouple waterside hydraulics, minimize cavitation potential, and eliminate the requirement for expensive water-side valves and piping.

The warm and cold water pumps are shown in Fig. 2 for a 40-MWe reference hull. The cold water is pumped out of a common moon pool, and the warm water is pumped directly from the open sea (note that screens are not shown). The pumps maintain a water level in the open water boxes of approximately 10 ft above calm sea level. The water then flows through the heat exchangers by gravity. The steady-state water level in the moon pool will be 5 to 6 ft below calm sea level. This head elevation difference (drawdown) is sufficient to overcome density head (3 ft), friction, and other losses.

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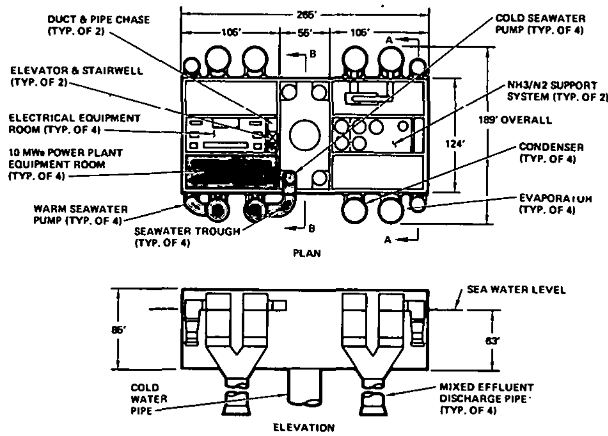

Figure 2. 40 MWe Platform

The turbine is a four-stage, double-flow, axial machine with an efficiency of 89.6% (versus 85% assumed for system sizing). The proposed control system includes variable nozzles (variable stator vanes) for the first stage to provide for increased power developed at off-design conditions compared to fixed nozzles. The ammonia subsystem provides three functions: storage, conditioning, and recovery. A unique feature of this subsystem is that warm seawater is used in the rectification column reboiler which recovers dry ammonia from the wet heel. This saves energy as compared to using electrical power. Ammonia is recovered from all major components of the power loop whenever practicable.

The following critical interfaces between the power system module and ship systems were identified and described by interface drawings: containment dimensions, hull penetrations and interfacing pipe materials, installed power system module weights, water trough duct connections, and immersed heat exchanger/hull attachments. Ship services to be provided by ship system auxiliaries include 50,000 CFM ventilation air to the power module equipment and NH_3/N_2 support systems rooms, 15,000 CFM ventilation air to electrical equipment room, air conditioning (2 tons of refrigeration) for the control room, lighting for the control room (100 ft candles; for the equipment areas, 80 ft candles), fire protection system, and 100-psig, dry, filtered shop air.

Major System Characteristics

Major physical and functional characteristics of the 10 MWe power module are shown in process flow diagrams, piping and instrumentation diagrams, arrangement drawings, and piping/support drawings prepared for DOE[2]. Major characteristics are summarized below in Tables 1 through 4 for

- Heat exchangers
- Water supply and return
- Ammonia power cycle
- Power budget.

Table 1. Heat Exchanger Subsystem

	Evaporator	Condenser
Tube Length - Total (ft)	29.83	29.83
Tube Outer Diameter (in)	1.00	1.00
Tube Wall Thickness (in)	0.031	0.026
Enhancement Ratio, Geometric (I/D)	1.46/1.50	1.46/1.50
Number of Tubes	42,667	43,883
Shell Diameter (ID) (ft)	26.25	27.08
Tubeside Water Velocity (ft/sec)	6.61	5.82
Tubeside Pressure Drop (PSI)	4.12	3.61
Water Inlet Temperature (°F)	80	40
Water Outlet Temperature (°F)	74.1	46.2
Ammonia Temperature (°F)	70	50
LMTD (°F)	6.62	6.41
Thermal Duty (MBthu/hr)	1543	1495
Overall Heat Transfer Coeff. (Btu/hr)	896	796

Table 2. Water (Hydraulic) Subsystem

	Warm	Cold
Temperature (°F)	80	40
Volumetric Flow Rate (gpm)	534,700	494,300
Mass Flow Rate (lb/sec)	75,970	70,500
Pipe Size (ID - Based on 6 ft/sec) (ft)	15.9	15.3
Total Pressure Drop (ft)	10.6	15.0
Pumping Power (MWe)	1.35	1.78
Shaft Horsepower (BHP) Assumptions: Motor/Gear Eff. 92.4% Pump Eff. 87%	1680	2210

Table 3. Ammonia Cycle

Ammonia Vapor Flow Rate (lb/sec)	807.2
Ammonia Volumetric Flow Rates Turbine Inlet (ACFS) Turbine Outlet (ACFS)	 1867 2594
Ammonia Recirculation Flow Rate (Evaporator Inlet Flow)(lb/sec)	1324
Feed Pump Head (psi)	41.8
Recirc. Pump Head (psi)	15.2
Ammonia Quality, Turbine Inlet (%)	100
Ammonia Quality, Turbine Outlet (%)	97.5
Evaporator Temperature (°F)	70
Condenser Temperature (°F)	50
Enthalpy Drop Across Turbine (Btu/lb) (at assumed eff. of 85%)	16.8

Table 4. Power Budget*

Gross Power (MWe)	14.03
Net Power (MWe)	10.00
Gross Power/Net Power	1.40
Warm Water Pumping Power (MWe)	1.35
Cold Water Pumping Power (MWe	1.78
Ammonia Feed Pumping Power (MWe)	0.24
Ammonia Recirculation Pumping Power (MWe)	0.14
Chlorination Power (MWe)	0.05
Mechanical Cleaning Power (MWe)	0.01
Control, Ammonia Support, Misc (MWe)	0.46

*Based on assumed efficiencies (85% turbine efficiency)

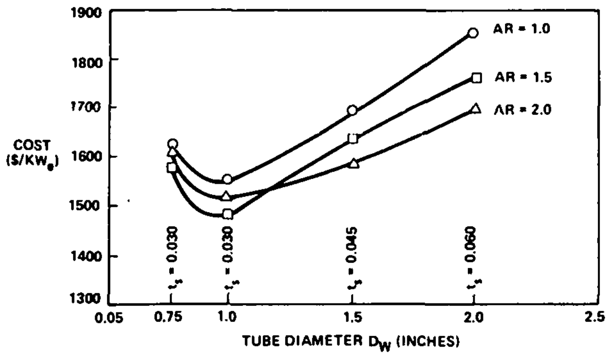

Figure 3. Direct Cost of 10 MWe Prototype Versus Tube Diameter and Water Side Area Enhancement Ratio, Ar.

Design Optimization

During the conceptual design phase, tradeoff studies were conducted to determine an optimum power module size. These tradeoff studies, described in the design report[1], involved considerations of heat exchanger manufacturability, containment hull size versus cost, and packaging. TRW's recommended configuration was a 12.5 MWe power module consisting of two 6.25 MWe submodules driving a common generator.

Subsequently, DOE directed TRW to change the rating of the power module to 10 MWe and to immerse the heat exchangers in seawater (external to the hull).

The major characteristics of the design, which evolved as a result of the reoptimization, are summarized above. Besides power module size and the general heat exchanger concept, the main tradeoff parameters include functional parameters, such as seawater flow rates and temperature allocations, physical parameters, such as heat exchanger tube geometry, and degree of water side enhancement (by longitudinal flutes). A computer program was developed for design optimization. Some results are described in the subsequent paragraphs.

Figure 3 shows the direct cost of a production power module versus tube diameter for three water-side enhancement ratios. The structurally required tube thickness is essentially proportional to tube diameter; however, for tube diameters less than 1 in. OD, the thickness was assumed constant dictated by forming constraints rather than strength. The figure demonstrates that approximately 1 in. OD is the optimum tube diameter, and that a water-side enhancement ratio of 1.5 is more cost effective than smooth tubes and the more extreme (and more difficult tube to clean) area ratio of 2.0. In all cases the ammonia-side enhancement ratio was kept equal to 1.5 as recommended by Dr. Rothfus of CMU.

The tube diameter and enhancement ratio have a strong impact on the optimum tube length. The tube length increases proportionally with tube diameter and decreases with increasing enhancement ratio. For tube diameters of approximately 1 in., the optimum tube length ranges from 25 to 50 ft, depending on the area ratio. For tube diameters of approximately 2 in., the optimum length ranges from 40 to 80 ft. Considering cost impact on the power system and other factors such as arrangement, draft, and heat exchanger performance versus risk (possible performance degradation due to ammonia-side Reynolds number effects), we selected a nominally 1 in. OD tube with 36 flutes with an actual enhancement ratio of 1.46 (area ratio).

The total length of the evaporator and condenser tubes was configured to be equal, approximately 30 ft long. This simplifies the topology of the design and does not penalize cost significantly.

System Operation and Control

Figure 4 shows a simplified schematic of the 10 MWe power module and the ammonia support subsystems. Note that redundant pumps and storage tanks are not shown. The power loop is a conventional Rankine cycle loop with ammonia recirculation in the evaporator.

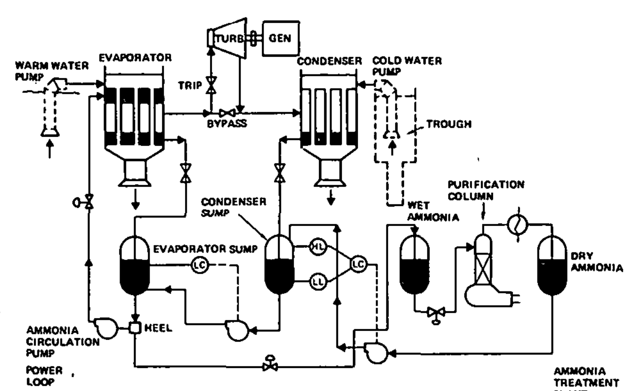

Figure 4. Power Module Schematic

The cold water trough is filled passively from the sea as water is pumped into the condenser. The cold-water and warm-water pumps run at constant speed, so motor-starters are satisfactory for control. No control is provided on water flow because of the large surge power requirements. The possibility of smoothing out heave-induced ammonia flow fluctuations by varying pump speed (though the power consumption is expected to be excessive) and of reducing external startup power requirements by varying pump speed remains to be investigated.

A fraction of the ammonia flow is continuously removed at the heel of the power loop and stored in a wet-ammonia tank. The ammonia is then fed to a purification column. Dry ammonia from the column is transferred to a storage tank from which it is pumped into the condenser sump. All controllers are located on the local control panel near the ammonia treatment plant. Four of them, required during normal plant operation, have remote set points in the PSD-1 control room:

- Feed rate of wet ammonia from the heel of the power loop
- Feed rate of wet ammonia to the purification column
- Reflux rate in the purification column
- Pumping rate to the condensate pump.

The ammonia treatment plant is monitored from the PSD-I control room and can be tripped by the operator, either from the local control panel or from the PSD-I control room. An ammonia plant trip isolates each piece of equipment from the others, without immediately affecting he power cycle. If the tripped state persists, the power cycle will eventually have to shut down.

The controls for the ammonia power cycle are shown in Fig. 4. The condensate feed pump's speed is controlled by the level in the evaporator sump, resulting in an average speed proportional to flow rate in the turbine. An on-off controller is not used because of the small volume of the tank relative to the flow rate and the consequent need for tight control. The ammonia recirculation pump operates at constant speed. Flow is controlled by a throttle valve that regulates coarse power output. It is operated either manually or from the digital computer.

A second liquid-side design option was considered, in which the condensate feed pump and the recirculation pump both feed the evaporator header in parallel. The evaporator sump level would be controlled by the recirculation valve, and the condenser sump pump would be controlled by the makeup valve. This second option may be somewhat less expensive than the chosen design because it requires a smaller evaporator sump and smaller recirculation pumps. However, it is subject to flow instabilities between the parallel pumps and requires a third valve to control gross power output. The chosen design is also simpler from a control viewpoint.

The turbine control actuator is a set of variable nozzles with a common controller. During startup, it is used for speed control; after synchronization, it is controlled for maximum net power output, either manually or by a hunting program in the central computer. When the generator is connected to an infinite grid, its speed is locked at 1800 rpm and a "wide-open valves" policy produces maximum power. If the generator were connected to an isolated grid, the variable nozzles would provide speed control, using inputs of shaft speed to the digital computer.

Discussions of full power versus part power operation, system dynamics, and off-design point performance appear in a companion paper presented at this conference by Dr. Kayton.[3b]

Materials

The materials and processes for the power module components and auxiliary equipment were selected with the objective of satisfying the 30-year design life through an optimum repair/replacement/maintenance program. In selecting specific materials, the following guidelines were used:

- All selected materials must have a reasonable delivery (availability) and be fabricable by methods currently available to the industry.
- Ammonia-side materials must conform to OSHA Standard 1910.111, "Storage and Handling of Anhydrous Ammonia," and the proposed amendments to the standard published in the Employment Safety and Health Guides, Number 259, 7 March 1976.
- The pressure-containing portions of the heat exchangers must conform to the applicable portions of the ASME pressure vessel codes (Sections II, V, and VII; Division 1 and IX).
- The selected materials and processes must meet the applicable structural and thermal requirements of the power system.
- Materials and fluids in contact must be compatible.

Recommended design materials for the heat exchanger system are shown in Table 5. Basically, steel or corrosion resistant steel (CRES) is used for ammonia system components. Steel or CRES, aluminum bronze, or glass fiber reinforced plastic are used for water system components.

Table 5. Recommended Design Materials

Heat Exchanger	
Shell	SA516 GR60 Steel
Tubesheets	SA516 GR60; SA285 GR B Steel
Tubesheet Cladding	SB265 GR1 or GR2 Titanium
Tube	SB338 GR2 Titanium
Internal Structural Members	SA516, SA283, SA285, ASTM A36 Steel
Water Boxes	SA516, SA283, SA285 Steel
Water Box, Shell or Pipe Coating	GACO N-200-1/N-11R Neoprene (or) NAPKO No. 5635 Coal Tar Epoxy
External Coating	Dimentcote 6/Amercote 83/84 Epoxy-Polyamide (or) GACO N-200-1/N-11R Neoprene
Designations are ASME except where noted.	

As expected for any marine system, periodic maintenance and repair is planned for the power module. For the heat exchanger, the prime areas of concern are the coated surfaces of the water box and shell. If the coating is damaged or fails by some mechanism, then corrosion of the steel substrate can occur. Periodic inspection and repair of the coatings will prevent unacceptable corrosive attack. If the cathodic protection system provided by the hull designer is inadequate, galvanic attack at the cofferdam, supports, and connections and/or blistering of protective coatings can occur. Again, periodic inspection and repair will be required to assure structural integrity. Any failure of the tubes will be detected by the ammonia leak detection system. Individual tubes will be plugged to prevent further leakage.

The ammonia and water system components will be maintained following standard shipboard practices. Replacement of seals, wear parts, and maintenance at coatings are routine.

Heat Exchanger Design

Our heat exchangers are characterized by high performance by geometric enhancement on both the shell (ammonia) and tube (water) side. Mechanically, these heat exchangers are of conventional shell and tube construction, and no fabrication problems are anticipated.

The design uses theoretical and experimental data available for vertical tube, falling film heat transfer, principally test data and analysis by Carnegie-Mellon University (CMU) and Oak Ridge National Laboratory (ORNL). The projected performance of evaporator and condenser was substantiated by test data from Argonne National Laboratory (ANL).

Key features of the design can be summarized as follows:

- Vertical orientation
  - Performance scale-up is simplified.
  - Provides an opportunity for a unique cleaning approach through the open head design.
  - Performance is relatively insensitive to platform motions.
- Tube surface enhancements (Fluting)
  - Ammonia-side heat transfer coefficient is significantly improved through falling film concept (h approaching 10,000 Btu/hr-ft^2-°F).
  - Water-side benefits are equal to the increased area ratio (i.e., enhanced area to equivalent smooth area).
  - Performance-to-cost of enhanced-to-smooth tubes favors enhanced tubes.
- Thermal/hydraulic performance
  - Insensitive to seawater maldistributions.
  - Insensitive to ammonia maldistributions.
  - Low sensitivity to platform roll, pitch, and heave.
- Long life and high corrosion resistance of titanium tubes in seawater.

Design Description

The design of the 10 MWe evaporator and condenser is illustrated in Figs. 5 and 6, while its major characteristics are shown in Table 1.

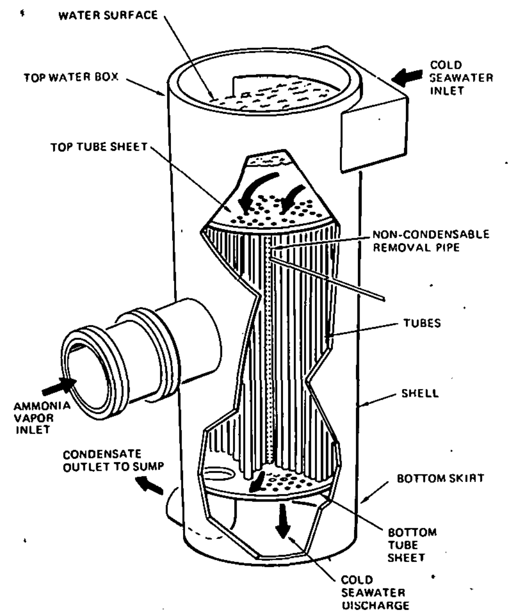

Figure 5. 10 MWe Condenser Features

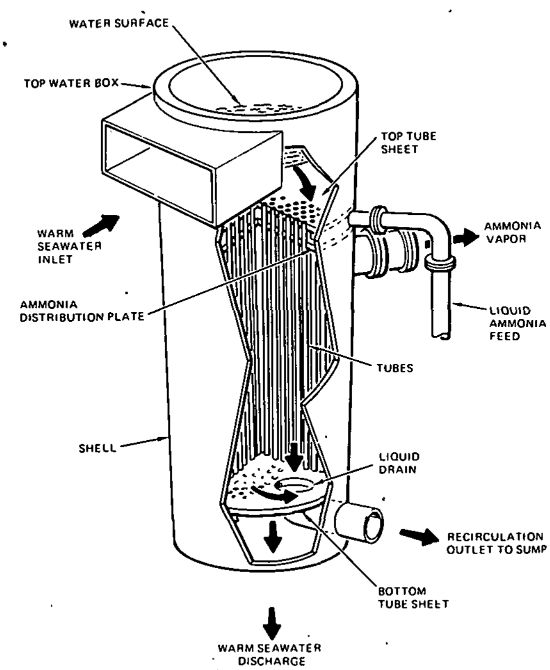

Figure 6. 10 MWe Evaporator Features

The heat exchangers incorporate fixed tubesheets and eccentric nonremovable tube bundles with the enhanced surface titanium tubes hydraulically expanded into carbon steel tubesheets with titanium cladding; the remainder of the units are all of carbon steel construction.

Protective nonmetallic coatings will be applied to the water box. The tubesheets will be titanium clad to inhibit corrosion and to permit tube-to-tubesheet welding. The inlet water box is an open channel type header to permit in situ mechanical cleaning. The water enters via a rectangular inlet nozzle to improve water-side distribution. Rod bundles, located at the vapor entrance in the condenser and at the liquid ammonia entrance in the evaporator distribution chamber, break and disperse the entering fluid and prevent impingement damage to the tubes.

The heat exchangers are immersed in seawater up to the level of the top tubesheet and are attached to the side of the floating platform by means of welded brackets. Design and construction includes the consideration of the requirements and limitations of transportation, barge loading for transportation to the platform, raising vertically for attachment, inservice loading due to operation, and Sea State 9 accelerations.

As illustrated in Figs. 5 and 6, the evaporator and condenser embody essentially identical bundles of vertical fluted tubes. The tubes are fluted to enhance heat transfer and offer significant performance/cost advantages. The tube bundle is constrained within the heat exchanger shell by two fixed tubesheets and by tube support baffle plates spaced at 4.5 ft intervals for the evaporator (4.8 ft for the condenser) between the fixed tubesheets. These tube support plates are designed to provide mechanical tube support without disrupting the falling film of ammonia on the tube exterior.

The design and performance projections of our heat exchanger design are predicated on the assumption that any one tube performs identically to any of the other 43,000 tubes in the bundle. Although single vertical fluted tube performance data are available, there are no similar comprehensive data on vertical bundles of significant size. However, this assumption regarding our heat exchanger design can be satisfied provided the following thermal hydraulic design considerations are met:

- The tube bundle shell-side pressure drop in minimal, so that the evaporation/condensation pressures (and temperatures) are uniform throughout the tube bundle.
- The liquid ammonia is uniformly metered over each tube in the evaporator to ensure uniform and optimal heat transfer.
- The ammonia vapor velocities over the tubes are limited so that the falling film on the tube exterior is not adversely affected, e.g., sheared-off or otherwise altered in configuration.
- Noncondensibles are readily extracted from the condenser.

Each of the above considerations is embodied in the design of our heat exchangers as discussed in the design report[2] and in a companion paper by Mssrs. Denton and Fukunaga,[3c] presented at this conference.

Biofouling Control

Biofouling control is accomplished by a combination of brush cleaning and chlorination. The importance of maintaining biofouling control is expressed by the sensitivity of net power (P_{net}) to the fouling factor R_f:

$$\frac{\Delta P_{Net}}{P^{\circ}_{Net}} = -0.070 \frac{\Delta R_f}{R^{\circ}_f}$$

The design goal is a fouling factor of $R_f{}^{\circ} = 0.0001$ hr-ft^2-°F/Btu. A fouling factor of 0.0005 results in a net power degradation of 28%. The design fouling factor will be achieved by a combination of continuous chlorine injection and periodic tube cleaning. Tradeoff studies remain to be performend to establish optimimum power output versus cleaning frequency and chlorine content. The levels assumed below for preliminary sizing of these subsystems should then be adjusted accordingly.

Based on an estimated tube cleaner subsystem cost of about $600,000, the resulting cost per kilowatt hour of approximately 1 to 2 mils makes this subsystem extremely cost effective. Parasitic power losses for tube cleaner operation (estimated at less than 10 horsepower) are negligible.

Tube Cleaner

The tube cleaner subsystem removes contamination from the inner surfaces of the evaporator and condenser tubes. Experience with commercial seawater exchangers indicates that the specified fouling factor can be maintained by brushing each tube approximately once per week with relatively light bristle pressure. To clean two exchangers in 1 week, allowing time for maintenance/repair or increased duty cycle and to minimize he positioning problems, a multiple brush (95 brushes) head is used. With the anticipated stroke times, a single exchanger can be cleaned in a 12-hour shift.

The tube cleaner can perform the cleaning function while the head exchangers are operating without noticeably reducing the number of active tubes. Only those tubes being cleaned are out-of-service during their cleaning cycle. The cleaning device cleans the entire inner surface of each tube and will not mar, scratch, or degrade the inner surface of the tubes, the tubesheets, or the tube-to-tubesheet joints. It consists of three major components:

1. A cleaning head with multiple cleaning devices (brushes)
2. A positioner which moves the cleaning head to progressively clean selected groups of tubes
3. A controller/recorder to command the positioner to its various positions and to operate the head through its cleaning cycle.

A perspective drawing of the tube cleaner is shown in Fig. 7. A positioner sits on a square bed called the X-Y positioner frame. This frame rests on top of the heat exchanger water box/tube cleaner support structure. The positioner moves the cleaning head in the X,Y, and Z directions by means of air motors through a rack and pinion drive.

The tube cleaner head is a 32-ft long cylinder with a hexagonal cross section. The cleaning head employs individual double action pistions to drive the brushes through the heat exchanger tubes. The pistons are fluid driven from a common manifold source of low pressure. This permits individual brushes to "dead head" against the tubesheet where there are no tubes or plugged tubes. After cleaning 95 tubes, the head is repositioned before cleaning the next set of 95 tubes. The pattern is indicated in Fig. 8 which shows the tubesheet overlayed with the cleaning sequence.

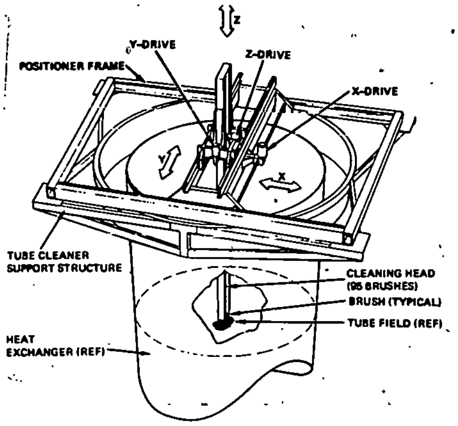

Figure 7. Tube Cleaner

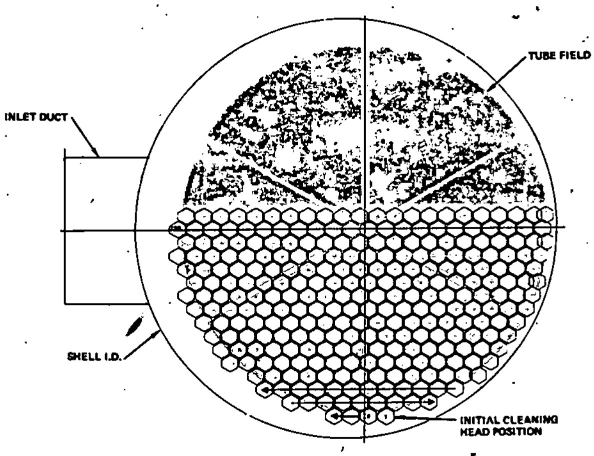

Figure 8. 10 MWe Evaporator Cleaning Sequence

Chlorination

The function of the chlorine is to deter microorganisms from adhering to heat exchanger surfaces and to inhibit their growth. Continuous low chlorination (0.02 to 0.05 ppm) well below acceptable seawater contamination levels has proven an effective deterrent for mussels in coastal and estuarine installations. A chlorination dose of 0.05 ppm was selected for this application.

The chlorinator consists of two separate units, a power supply and a chlorine generator. The power supply takes 480 VAC, 3-phase 60 Hz power and converts it to dc power as required by the chlorine generator.

The chlorine generator electrolyzes a seawater stream producing sodium hypochlorite (NaOCl) which remains in solution. The chlorine generator unit housing contains a seawater pump, electrolytic cells, a chlorine concentration sensor that regulates the power input, a means of safely disposing of the hydrogen generated in the electrolytic process, and the required piping and valves.

A commercially distributed chlorinator capable of supplying 30 lb/hr equivalent chlorine has been selected. For its operation, a unit of this size requires from 40 to 120 gpm of seawater. If this water is at about 80°F, the power consumed in generating chlorine ranges up to 2.5 kW hr/lb of chlorine.

Rotating Machinery

Rotating machinery includes a turbine generator and state-of-the-art ammonia and seawater pumps. The selected turbine design will be reviewed below.

Turbine Generator

The preliminary design efforts for the turbine generator concentrated on the turbine. No turbine (or expander) using ammonia as the motive fluid is known to exist, whereas the generator will be an off-the-shelf design. The main objectives of the preliminary turbine design work were:

- Select the optimum type of turbine — axial flow or radial inflow
- Select the optimum flow path — single or double flow
- Select the optimum number of stages — single or multiple
- Evaluate critical areas to a depth that provides technical and economic credibility
- Arrive at a cost estimate for detail design, development, and fabrication.

The bulk of the preliminary design work was done by the Elliott Company, a division of the Carrier Corporation, under subcontract with TRW. A separate paper by Mr. Vincent of TRW and Mr. Kostors of Elliott[3c], describes this work in detail. Only the conclusions are summarized here:

- Although an ammonia turbine has not been built, its design is well within the state of the art, and fabrication can be done with existing shop techinques and facilities.
- The optimum design for 10 MWe net power recovery is a four-state axial double flow turbine running at 1800 rpm connected to a four-pole synchronous generator.
- Variable nozzles for the first expander stage are justified for off-design conditions, for matching the expander to particular site conditions, and for high frequency control.
- Blade stress and frequencies are mechanically safe.
- Critical speeds are safely away from the operating speed.
- Stress corrosion cracking investigations should be authorized at the earliest possible date to ensure that material selections and certain design features are suitable before further turbine design work is done.

Plant Availability

The availability of the 10 MWe power module for a 30-year period was assessed by Monte Carlo simulation. The analysis took into account both scheduled outages for preventive maintenance and unscheduled outages. It also simulated buildup of a service queue when several parts wear out within a short time of each other. Figure 9 shows the reliability model of the power cycle. As a first approximation, treatment plant failures cannot cause power outages

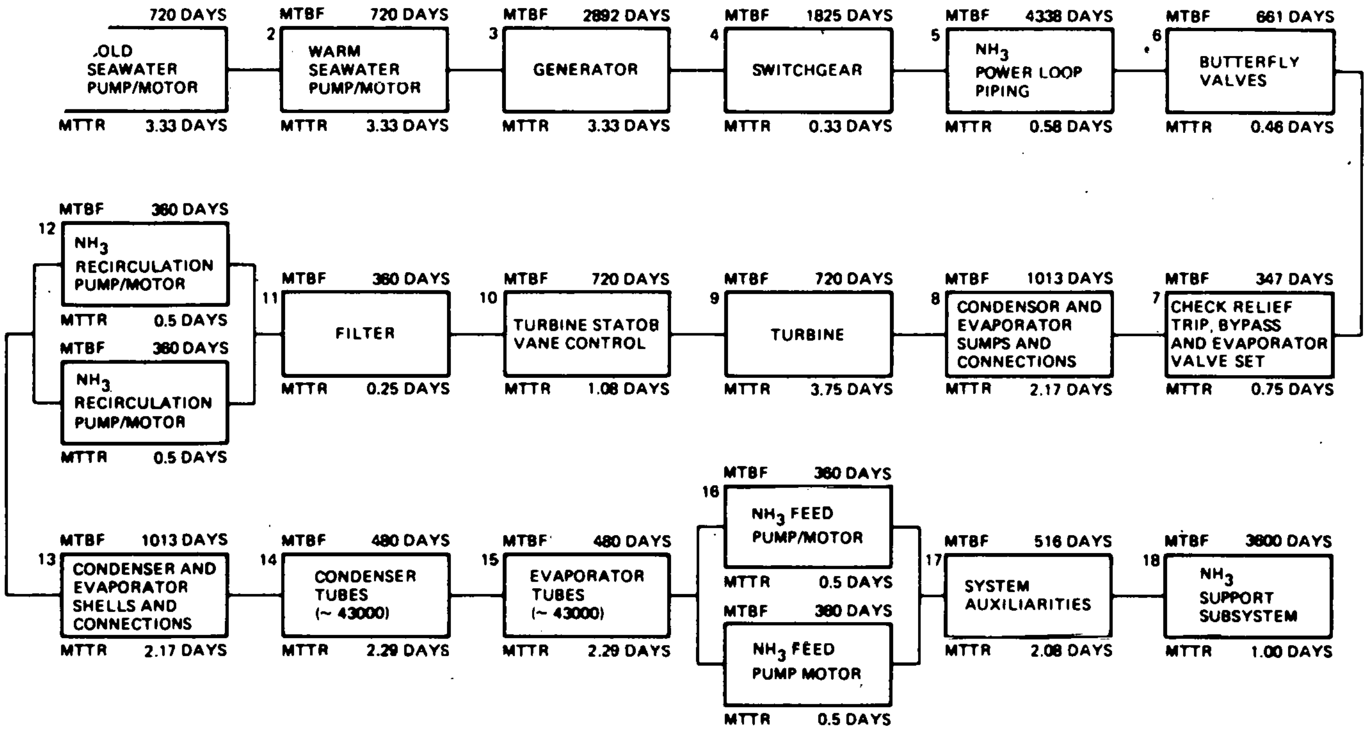

Figure 9. Operability Simulation Input Data Block Diagram
for Failure Repair Expectations

because the power cycle can operate for at least a day without the treatment plant. Thus, the figure shows the power cycle components whose failure disables the plant. Single-point-failure components are drawn in series; redundant components are drawn in parallel.

The results of the analysis are summarized in Table 6. The average availability is 0.906 for 30 years. Due to statistical variability, the annual average ranges from 0.88 to 0.93. Thirteen to 22 unscheduled outages are expected each year, causing unscheduled downtime of 13 to 30 days annually. Startup reliability ranges from 0.97 to 0.99. The availability and startup reliability predictions for the baseline design meet the DOE goals.

Table 6. Summary Results of Availability Analysis for 30 Years.

	Design Configuration	
	Redundant Ammonia Pumps	Single Ammonia Pumps
DOE-Required Availability	0.90	0.90
Computed Availability	0.906	0.886
Downtime, Scheduled and Unscheduled (days)	590	810
Power Plant Failures	468	503
Wearout Failures	142	130

Cost Estimates

Table 7 gives cost estimates for all the major subsystems and components of the power system for prototype and production systems. Installation costs and indirect costs (engineering services and home office services) are included. The major cost elements are the titanium tubed heat exchangers. No heat exchanger replacement is planned, corrosion allowances and provisions for in-service tube plugging are included in the design.

Conclusions

The results of the design study reviewed in this paper show that OTEC plants with heat exchangers employing vertical fluted titanium tubes in a conventional shell-and-tube configuration are technically feasible and require no extension of the state of the art. Plans for detailed design and construction are available. The heat exchangers are the major cost leverage item. Methods of reducing these costs are presently being pursued in the PSD-II projects employing plate type exchangers.

Acknowledgments

The Preliminary Design Phase of the PSD-I Project was conducted by a team consisting of:

- TRW Defense and Space Systems Group, the prime contractor, responsible for power systems design and analysis including: availability, materials technology, design integration, and project management.
- C. F. Braun and Company, subcontractor to TRW, responsible for mechanical design and fabricability of heat exchangers and design of ammonia support subsystems. They provided technical support in the areas of

Table 7. Power Plant Cost Breakdown Summary*

Item	Prototype ($/kWe) (10/MWe)	1st Production ($/kWe) (10/MWe)	8th Plant ($/kWe) (10 MWe)	8th Plant ($/kWe) (40 MWe)
Evaporator	890	763	565	505
Condenser	871	744	545	483
Turbine/Generator	307	226	168	153
Ammonia Feed Pumps	25	23	13	13
Ammonia Recirculation Pumps	28	26	15	15
Ammonia System Piping	119	108	75	72
Ammonia System Valves	111	100	67	63
Control Systems	64	59	46	44
Chlorination Systems	21	19	12	11
Cleaning System/Leak Detector	190	171	34	32
Ammonia Support Systems	208	112	79	72
Electrical Equipment	39	35	25	24
Totals	2873	2386	1645	1487

rotating machinery, thermal-fluid design of heat exchangers, system safety, and installation.

The following consultants:

- Dr. A. J. Impink, Jr., Dr. J. F. Osterle, and Dr. K. P. White, Jr., Carnegie-Mellon University. This team of consultants made major contributions in the areas of off-design and dynamic modeling of the ammonia power loop.
- Dr. R. R. Rothfus, Carnegie-Mellon University, provided basic design data and analyses in support of heat exchanger thermal design.
- Dr. L. L. Ambs, University of Massachusetts, provided turbine design and optimization support.
- Dr. R. T. McLaughlin (ENWATS), provided design and analysis support in the areas of seawater subsystem design including dynamic modeling.

Under a separate contract with TRW, Elliott Company performed an ammonia turbine/generator design and optimization study for OTEC.

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DESIGN OF A 0.2-MW_e (NET), PLATE-TYPE, OTEC HEAT EXCHANGER TEST ARTICLE AND A 10-MW_e (NET) POWER MODULE

*H T

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Abstract

This paper summarizes results on the preliminary design, projected performance, and costs of 10-MW_e (net) power modules, and on the preliminary design of 0.2-MW_e test articles made of full-scale panels of the novel, plate-type heat exchangers. ~~The work was completed in June 1979 under Phase I of the U.S. Department of Energy's Power System Development (PSD) II program.~~ The heat exchangers are made of either a low cost, Alclad aluminum alloy (Al 3004 clad with Al 7072 on the seawater side) or a super stainless steel comparable to Allegheny 6X. In either case a porous-type surface enhancement is used on the ammonia side. The heat exchanger plates are corrugated to form circular water flow passages, operationally accessible for cleaning, and ammonia passages with performance similar to conventional thermosiphon reboiler technology. The plates are assembled into 0.625-MW_e modules which are then assembled into 4 rows of 4 each, i.e., 4 are in series with respect to the seawater flow. This arrangement yields the lowest cost for 10-MW_e power modules. Present-values cost estimates for 30-yr life indicate that aluminum heat exchangers at \$367/kW_e with 15-yr life (i.e., replaced once) would cost slightly less than stainless steel exchangers at \$642/kW_e. An automated, brush-type cleaning system used once a week, along with continuous chlorination at 0.05 ppm concentration, is expected to keep the fouling resistance at 0.0003 hr-ft^2-^{o}F/Btu.

Introduction

Phase I of the U.S. Department of Energy's (DOE's) Power System Development II (PSD II) program, October 1978 through June 1979, had the following scope:

1) Conceptual and preliminary design of a 10-MW_e (net) power module using shell-less, aluminum heat exchangers.

2) Conceptual and preliminary design of 0.2-MW_e (net) evaporator and condenser test articles representative of the full-size exchangers.

3) Preparation of Phase II plan (proposal) for detail design, development, fabrication and checkout of the test heat exchangers and installation and operational support on an ocean-based, government-furnished test facility (OTEC 1). References 1 and 2 describe OTEC 1.

Relative to item (3), the Phase II plan as defined by DOE guidance of 3 April 1979 addressed the following areas.

a) Conceptual and preliminary design of a 10-MW_e (net) closed-cycle, ammonia power system module for a floating OTEC demonstration plant.

b) Design, fabrication, and delivery of test article heat exchangers representative of the power system design.

c) Design, fabrication, and delivery of any special ancillary equipment required for handling, delivery, field fabrication/installation, and operation of the test hardware on OTEC 1.

d) Participation and support of the OTEC 1 Systems Interface Control Working Group and Test Planning and Instrumentation Working Group.

e) Provision for manufacturer's representative support during three years of OTEC 1 operations.

This paper summarizes the results of TRW's Phase I work on PSD II. Further details are given in Ref. 2. The power system design is based on enhanced-plate-type heat exchangers (Figs. 1 and 2) to minimize the capital cost by coupling relatively efficient power conversion with existing mass-production techniques. It offers a competitively attractive power generation system for either a cruising OTEC plantship or an offshore OTEC plant connected to a utility grid.

Aluminum, due to its low initial cost, ease of fabrication and general availability, was the proposed material for this project. The Phase I design, therefore, has been specifically oriented toward a power system (Fig. 1) capable of being operated, maintained and repaired using aluminum heat exchangers. LaQue[1c] has projected the heat exchangers made from Alclad aluminum alloys (e.g., Al 3004 clad with Al 7072) will last

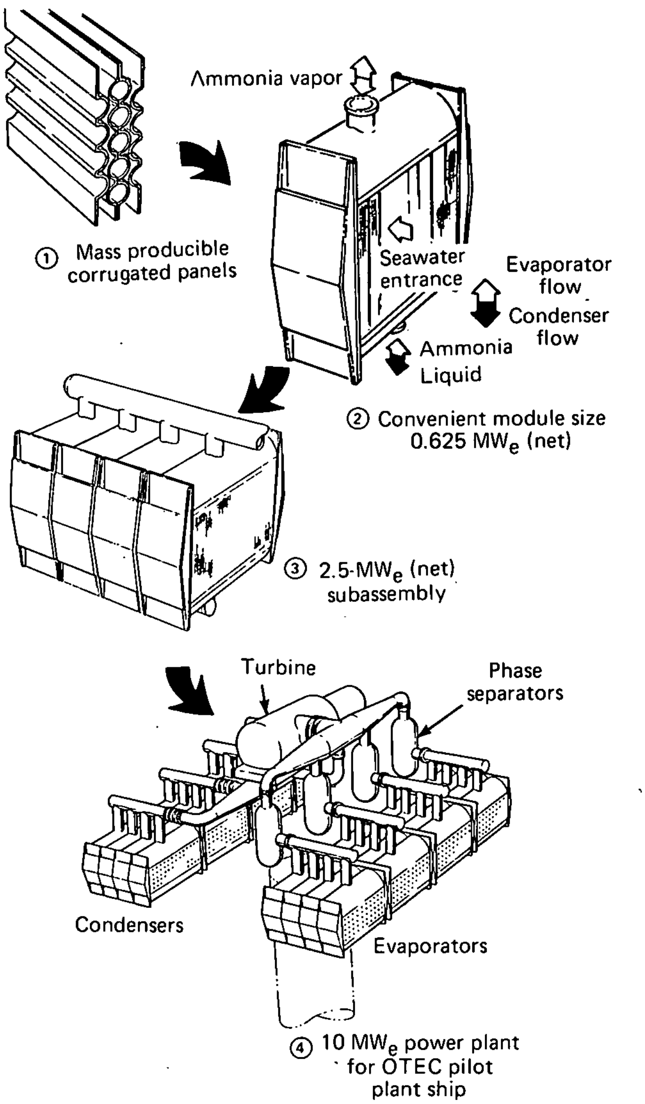


Fig. 1 Assembly and general arrangement concepts for OTEC power system employing the TRW/Linde concept for plate-type heat exchangers.

ten years or longer in a seawater environment if appropriate anti-corrosion protection is incorporated in the design. LaQue[1c] notes that aluminum industry representatives believe that Alclad aluminum should be credited with a longer life, and we believe that a 15-yr projection is warranted. However, the need for further qualification tests for aluminum led us to investigate alternative materials. The results presented herein indicate that equally viable, enhanced-plate-type heat exchangers can be fabricated from any one of several stainless steels typified by Allegheny 6X. Cost comparisons are presented on a present-value, 30-yr-life basis.

Several tasks for this project and those of TRW's PSD I project using vertical-tube, falling-film heat transfer techniques[1d] were similar, and the PSD I results have been employed wherever possible. In particular, the PSD I work was drawn upon to estimate the additional capital and operating costs

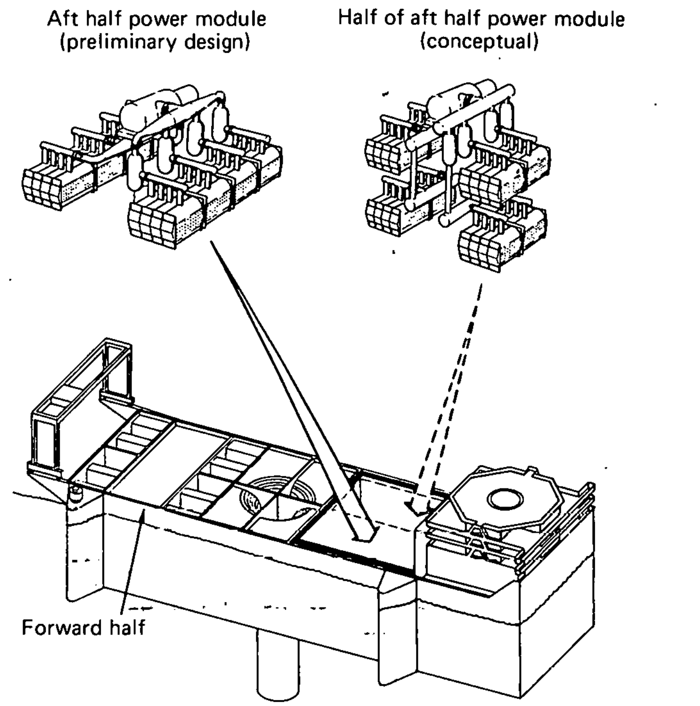


Fig. 2 The 20- and 40-MW_e (net) concepts for arrangements of the power modules in the OTEC pilot plantship hull.

for complete OTEC systems and hence to show the potential for attractive onboard power generation costs.

Technical Approach

The objective has been to design a power system which has low capital ($\$/kW_e$) and power generation (mills/kWh) costs. The 10-MW_e (net) power module design task primarily required matching the performance of the main power unit elements to the unique characteristics of the heat exchangers considering the need to package the power module efficiently into the candidate hull (Fig. 2). As our design is based on aluminum of 15-year life, emphasis was put on designing methods to repair and/or replace heat exchangers (the design requirement is for a 30-yr-life power system). Modularization of the heat exchangers to achieve maximum production cost benefits also allows flexibility for startup/shutdown and maintenance activities.

The 0.2-MW_e heat exchanger test articles are portions of the full-scale units (Fig. 3). The test articles would be built using prototype tooling and fixtures; the resulting hardware would confirm the fabricability and performance of the 10-MW_e module. Conceptual test plans and instrumentation for the tests on OTEC 1 have been completed.[2]

The 0.2-MW_e and 10-MW_e systems have been designed and analyzed with the aid of com-

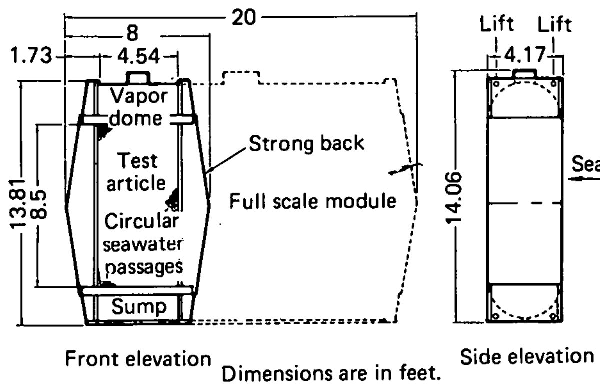


Fig. 3 Comparison of 0.2-MW_e test article with full-scale, 0.625-MW_e (net) modules.

puter programs which permit both design and off-design performance evaluations including effects of seawater temperature differences, seawater and ammonia flow and pressure drop variations, fouling and overall heat transfer, and hence availability. A gross power of 13.87 MW_e must be generated to give a 10-MW_e net output at the design point ΔT of 42^oF as discussed later.

The potential safety hazards of testing 0.2-MW_e heat exchangers and operating a 10-MW_e power module have been identified[2] for possible use by ship systems contractors to establish safety requirements for the PSD II systems. Preventive maintenance and scheduled downtimes are built into the design to assure system availability. Start-up considerations are included to assure starting reliability. The designs take into account the environmental considerations related to the mechanical cleaning of fouled heat exchanger tubes, chlorination, metallic elements, and release of working fluid.

Two significant technical alternatives affected the progress of the preliminary design and resulting conclusions:

1) The space allocation in the specified hull for the 10-MW_e (net) power module was initially understood to be the full aft half of the DOE pilot plantship as shown at top left in Fig. 2. The result is a power system consisting of four evaporator and four condenser module subassemblies on a single elevation. In December 1979 DOE requested that the PSD II contractors also address the packaging of 10 MW_e into one-half of the after half of the pilot plantship. This request has been responded to at a conceptual level as shown at top right in Fig. 2. This concept has two evaporator and two condenser module subassemblies on different elevations. A limited evaluation has verified the physical capability to install a power module with a plan area less than required by the basic design.

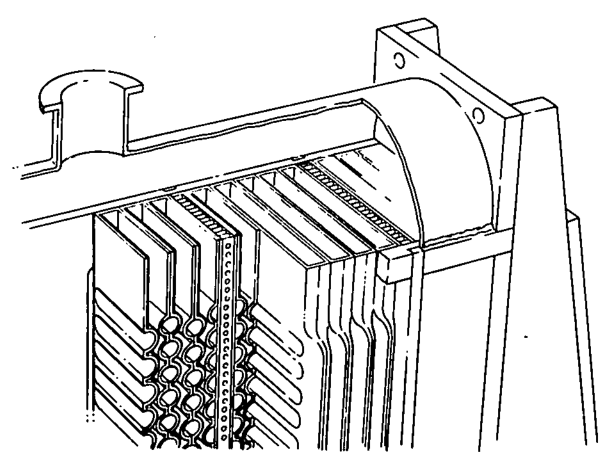
2) As noted earlier, the alternative use of AL-6X stainless steel for the heat exchangers has also been addressed.

10-MW_e Heat Exchanger Concepts

The major effort of this preliminary design project addressed the selection of heat exchanger designs which are thermally and hydraulically efficient and capable of mass production technology in order to meet the DOE cost goal of less than $700/$kW_e$.

The selected design is based on corrugated panels with porous, sintered-metal-type enhancements (UCC/Linde "HiFlux" proprietary process) on the ammonia side. These panels are assembled and joined into compact modules of 0.625 MW_e (net) each (Figs. 1 and 4) which are producible by current manufacturing techniques. The modules are further assembled with four in series with respect to the horizontal seawater flow (for near optimum thermal resource utilization) as shown by Figs. 1 and 5. This 2.5-MW_e subassembly is easily handled onboard the OTEC ship and can be isolated from the power loop when maintenance or repair is required (Fig. 5); it also can be disassembled to modules, if required, since mechanical sealing between modules is accomplished with a pressurized seal similar in concept to a bicycle tire (Fig. 6).

A significant feature of the evaporator module is the incorporation of an internal liquid return (see the downcomers in Fig. 4). The liquid accumulates from the primary phase separation provided by the semicylindrical dome structure (Figs. 4 and 5). Liquid flows internally to troughs which drain to the eleven downcomers in each module. (These downcomers are blocked structures in the condenser modules, so that the manufacturing pieces and processes remain essentially unchanged.) The physical parameters for the aluminum heat exchanger modules are listed in Table 1.



Panels of 3004-0 clad with 7072 aluminum or Allegheny 6X or equivalent stainless steel

Fig. 4 Cut-away view showing semi-cylindrical dome and downcomer for liquid ammonia that is separated from vapor by the dome.

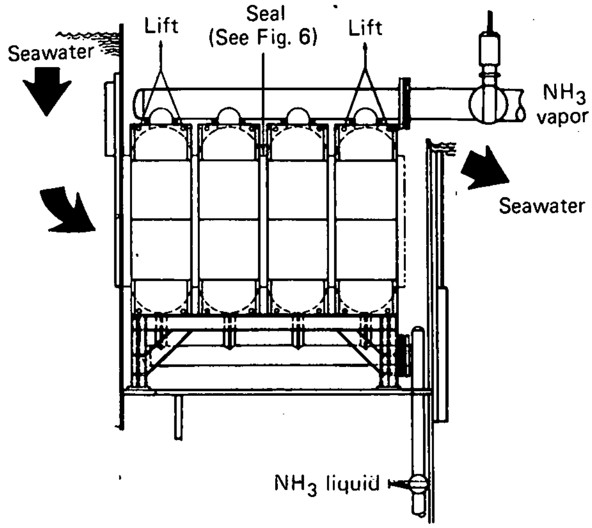


Fig. 5 Side view of 2.5 MW_e (net) module assembly, for evaporator, NH_3 flows upward; for condenser, downward.

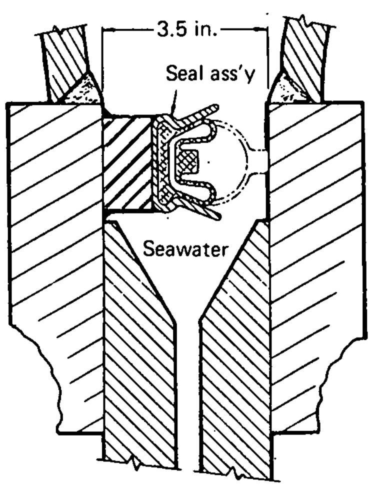


Fig. 6 Detail of inflatable inter-unit seal.

Table 1 Dimensions of Aluminum 0.625-MW_e (Net) Modules (Dimensions of 0.2-MW_e Test Articles That Differ Are Shown in Parentheses)

Active panels/module	204 (56)
Panel ht., heat transfer, ft	8.5
Panel structural ht., ft	9.0
Panel width, ft	4.0
Heat transfer area/panel, ft^2	75.3
Water passages/panel	100
Passage diameter, in.	0.75
Base metal (3004) thickness, in.	0.050
Cladding (7072), in.	0.01 (0.005)
Panel hole pitch, in.	1.019
Panel density, number/ft	14.4
Downcomers/Module	11 (3)
Upper and lower header diam, ft	4
Upper header nozzle (evap. outlet, cond. inlet) diam., ft	2 (1.17)
Velocity, ft/sec	32
Lower header nozzle (evap. outlet, cond. inlet) diam., ft	0.5 (0.33)
Velocity, ft/sec	6.0 (4.0)

Alternative Heat Exchanger Materials

As previously noted, assessment of possible alternative materials led to the conclusion that an alloy similar to AL-6X is an economically viable alternative to Al-clad aluminum. Any of the family of stainless steel alloys to the right of the curve in Fig. 7 is acceptable. These alloys have neither the environmental concerns of copper-nickel nor the high cost of titanium and are fully compatible with the ammonia working fluid. Our present judgment is that one of these alloys would offer lower risk than aluminum as the best heat-exchanger material for near-term, 10-MW_e power modules.

Heat Exchanger Performance

A close coupling of the mechanical design and thermal/hydraulic performance is required to assure optimum, cost-effective design. The basic structural design results in configurations which are thermally efficient and capable of analysis using vertical thermosiphon reboiler technology.

Ammonia-side enhancement (Linde's HiFlux) is used for both the evaporator and condenser, and the performance estimates are based on test data taken by Union Carbide for pool boiling on titanium tubes and

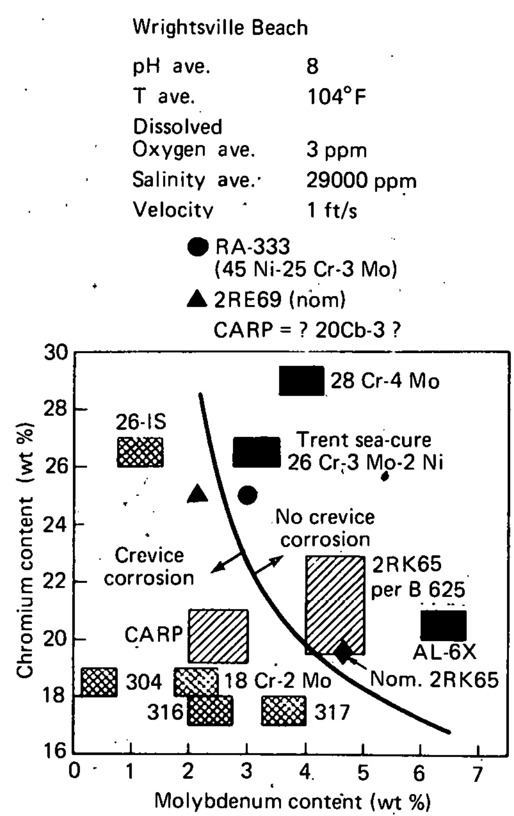


Fig. 7 Crevice-corrosion boundary between stainless alloys.[6] This figure differs from LaQue's Fig. 2[4] in that compositions have been added by Crucible Steel Corp. and UCC/Linde.

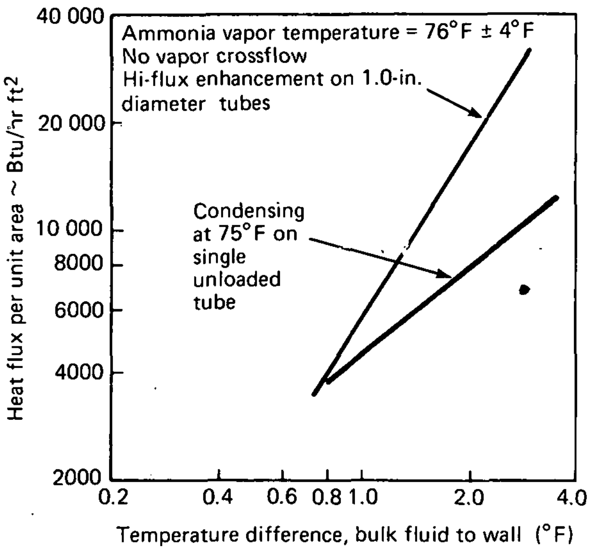


Fig. 8 Thin-film ammonia evaporator and condenser performance from UCC/Linde data.

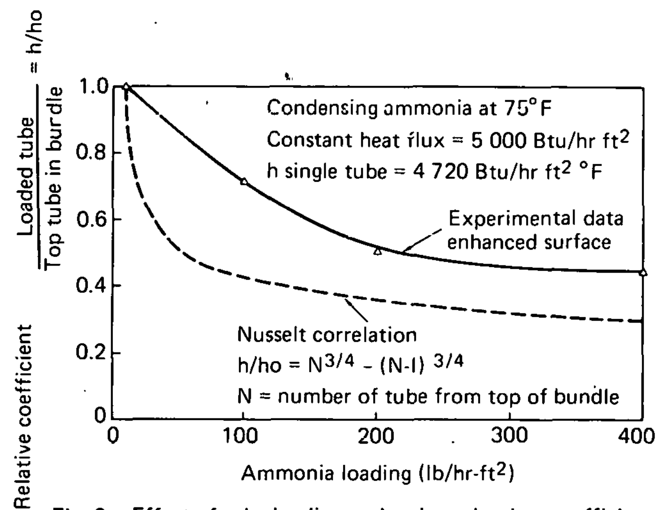


Fig. 9 Effect of tube loading on local condensing coefficients per UCC/Linde experiments.

spray boiling on aluminum tubes, which yielded the evaporator line shown in Fig. 8 and the condenser line for a single tube shown in Fig. 8. The experimental curve for loading in the condenser is compared with Nusselt's line in Fig. 9. Due to fabrication and cleaning considerations, waterside enhancement is not presently included.

Using the aforementioned data and evaluating the modules in series, the performance of module subassemblies has been projected. Table 2 presents the results for aluminum heat exchangers.

Biofouling Control

The specified requirement for biofouling control is to maintain the average fouling resistance R_f below 0.0005 (^oF-hr-ft^2/Btu). The solution is to employ mechanical brushes once a week along with a low dosage rate of chlorination (0.05 ppm) to maintain R_f at 0.0003 (average). The seawater passages are smooth and circular and fully accessible for in-situ cleaning. The design of the brush system (Fig. 10) is based upon conventional condenser tube cleaning concepts. The largely automated system indexes and locates a cleaning head containing 120 brushes which are driven horizontally through the tubes in push-pull strokes. In operation the cleaning head is submerged in the seawater inlet trough, and its use does not affect the remainder of the tubes in the module, hence it does not appreciably affect the module output. However, the length of the cleaning head requires the seawater inlet trough to be wider than the heat exchanger subassembly. The chlorination is through application of chloropac at an annualized cost of approximately 0.5 mill/kWh. The projected combined performance of these countermeasures is based primarily upon the DOE funded biofouling experiments at St. Croix and Keahole Point.[1e]

Heat Exchanger Costs

Detailed cost analyses based on an annual production rate of 100 MW_e (160 units each of the estimated installed costs for alumi-

Table 2 Characteristics for 4 Aluminum Modules in Series (Seawater Flow through the Subassembly)

Performance Parameters	Evaporator Modules No. 1	No. 2	No. 3	No. 4	Condenser Modules No. 1	No. 2	No. 3	No. 4
Water outlet temp., ^oF	80.75	79.67	78.73	77.92	41.22	42.32	43.31	44.20
Module LMTD, ^oF	8.78	7.60	6.58	5.70	10.79	9.65	8.62	7.70
Avg. quality, exit of panels	0.66	0.56	0.47	0.39	--	--	--	--
Ammonia ΔP in panels, psi	1.46	1.46	1.46	1.46	0.37	0.31	0.25	0.21
Avg. U, Btu/hr-ft^2-^oF*	819.4	818.5	817.5	816.6	613.6	617.9	622.3	626.5
h_a range, Btu/hr-ft^2-^oF	5847/ 5786	5787/ 5733	5741/ 5681	5683/ 5630	2947/ 1925	3040/ 1981	3136 2039	3236/ 2105
Heat duty/module, 10^6 Btu/hr	110.6	95.6	82.6	71.5	101.7	91.6	82.5	73.9
Relative performance/avg.	1.23	1.06	0.92	0.79	1.16	1.05	0.94	0.85

*Based on waterside area

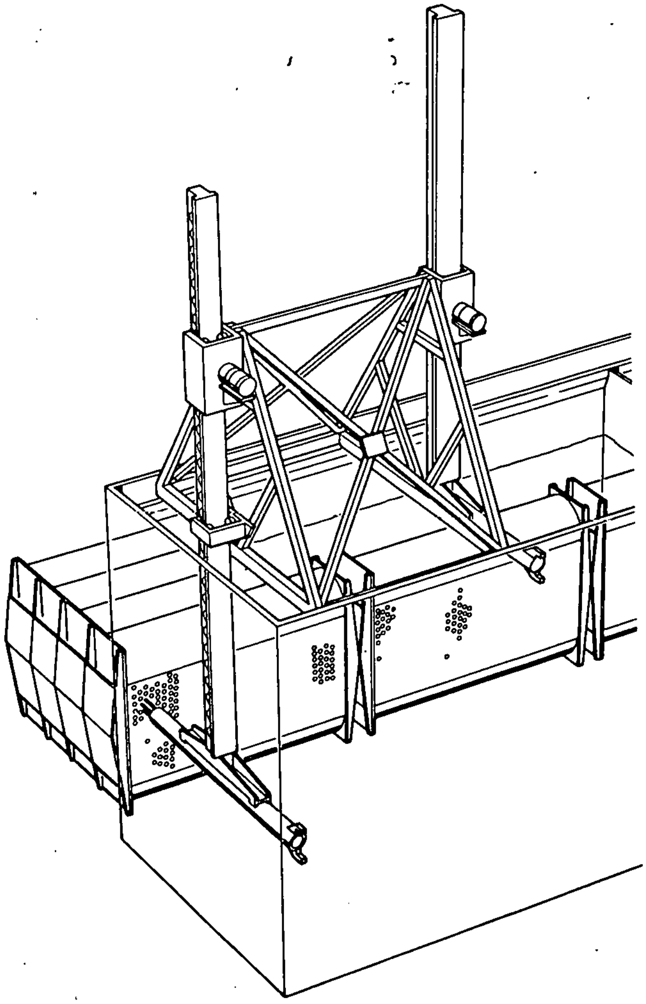


Fig. 10 Automated push-pull brush cleaning concept.

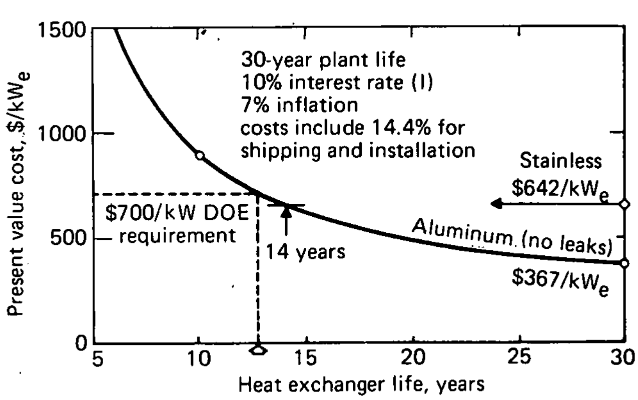


Fig. 11 Present value vs. life for 10-MW$_e$ heat exchangers.

num and stainless steel heat exchanger modules shown by Table 3. Evaluation of present-value costs for the aluminum heat exchangers as a function of life (see Table 4) results in the conclusion that either aluminum with a 15-year life (TRW's projection) or stainless steel with a 30-yr life will satisfy the DOE goal for the heat exchangers to cost less than $700 per kilowatt (Fig. 11).

Table 3 Specific Costs ($/kW$_e$ Net Power, March 1979 Dollars) of Aluminum and Stainless Steel Heat Exchangers Based on Annual Production of 160 Modules Each of Evaporators and Condensers (100-MW$_e$ Net Plant Requirement)

	Al	S.S.
Direct material	114.9	244.1
Direct labor	28.9	36.7
Indirect material	0.8	0.8
Indirect labor	11.4	13.2
Other services and expenses	19.2	19.8
Manufacturing staff	10.1	10.1
Sub-Total	184.3	324.7
Contingency 10%	18.4	65.0
Total	202.7	389.7
Warranty Expense at 2%	4.1	7.8
Product Overhead at 17%	34.4	66.2
Sub-Total	241.2	463.7
Depreciation Expense	4.7	6.4
Return on Investment at 20%	19.4	29.4
Cost	265.3	499.5
Enhancement Charges	55.0	61.2
Total Cost, $/kW$_e$	320.3	560.7
($/ft^2)	(6.52)	(10.25)
Freight & Instal'n at 14.4%	46.1	80.7
Installed cost, $/kW$_e$	366.4	641.4

Table 4 Present Value ($/kW$_e$) of Aluminum Heat Exchangers Having Lifetimes of 5-30 Years for a Power Plant Designed for 30 Years (March 1979 Dollars)

Yr	Present worth factor	Life cycle cost	Repl. Cost,[a] $/kW$_e$	Present value 10 yr	Present value 15 yr
0	1.000	367[b]	-	367	367
10	0.758	278	289	-	-
15	0.660	242	252	-	252
20	0.575	211	219	219	-
25	0.501	184	191	-	-
30	0.436	160	166	-	-
Total Present Value				875	619

[a] Replacement cost is 1.04 times first cost.

[b] First cost includes shipping to New Orleans and installation. This is also present value if H.E. lasts 30 yr.

10-MW$_e$ Power System Design and System Costs

The 10-MW$_e$ (net) power system designed for packaging within the aft half of the DOE specified plantship (Fig. 1 and top left in Fig. 2) consists of a basic power loop (heat exchangers, turbine-generator and pumps) and support subsystems. The optimization analysis, which evaluates heat exchanger performance characteristics and cost, options for installations (i.e., number of modules in series) and gross/net power outputs indicate that a gross power output of 13.87 MW_e will yield the 10 MW_e net required by DOE using aluminum heat exchangers. The power budget (Table 5) shows that the major system parasitic power use is for seawater pumps, allocated at 3.17 MW_e. Similar results occur for a power system using stainless steel heat exchangers except that the gross power output is increased slightly because of slightly lower heat transfer efficiency and a consequent increase in the number of heat exchanger tubes per panel.

The turbine and phase separator parameters and the efficiencies estimated for the major power system components are shown in Table 6. The resulting characteristics of the 10-MW_e (net) power module using the aluminum heat exchangers and the optimization program are shown in Table 7. The heat transfer coefficients and LMTD's in Table 7 are the average values for the total system that correspond to the individual module values shown in Table 2. The resulting power system cost estimates based on 100-MW_e's worth of production annually are shown in Table 8. It can be expected that when production levels reach several thousand MW_e's, use of somewhat larger turbines, generators and pumps, and learning-curve and technology improvements will reduce these costs sharply.

Figure 12 shows the effect of the number of aluminum HE modules in series on the power module life-cycle cost, and the reason a 4-module subassembly was chosen.

Table 5 Power Budget - 10-MW_e Power Module

Parameter		Power, MW_e
Gross power		13.87
Warm water pumps	-1.21	
Cold water pumps	-1.96	
Ammonia feed pump	-0.24	
Hotel load (per 10 MW_e)	-0.11	
Chlorination, mechanical cleaning, control & support subsystems	-0.35	
Parasitic power		-3.87
Net power		10.00

Table 6 Turbine and Phase Separator Parameters and Component Efficiencies, 10-MW_e (Net) Power Module

Turbine parameter	Value
Inlet T, °F/P, psia	72.0/133.4
Outlet T, °F/P, psia	51.6/92.0
Enthalpy drop, Btu/lb	18.6
Mass flow rate, lb/sec	764
Vol. flow in/out, ft^3/sec	1690/2357
Quality in/out, %	99.0/96.4
Phase separator inlet, °F/psia	72.2/133.5
Quality in/out, %	97.0/99.0
Efficiencies	
Turbine x generator	0.89 x 0.97 = 0.863
Seawater pumps x drive	0.87 x 0.924 = 0.804
Ammonia pump x drive	0.80 x 0.90 = 0.72
Plant, net power/evap. thermal duty	2.36%

Table 7 Characteristics of 10-MW_e (Net) Power Module

Parameter	Evaporator	Condenser
Water Side		
Velocity, fps	6.42	6.05
Inlet temperature, °F	82.0	40.0
Outlet temperature, °F	77.91	44.19
Avg. h_w, Btu/hr-ft^2-°F	1375	1138
Pressure drop, psid	2.56	2.44
Volumetric flow rate, gpm	721,800	679,600
Mass flow rate, lb/sec	102,500	96,920
Duct size (ID based on 6 fps), ft	18.5	17.9
Total pressure drop, ft	7.0	12.0
Pump power, MW_e	1.21	1.96
Shaft horsepower, BHP	1,500	2,420
Ammonia Side		
Inlet T, °F/P, psia	51.7/135.5	51.5/91.8
Outlet, T, °F/P, psia	72.2/134.0	51.3/91.4
Pressure drop, psid	1.46	0.39
Overall		
Heat transfer area, ft^2*	245,700	245,700
Thermal duty, MBTU/hr	1,444	1,395
LMTD, °F	7.17	9.12
U, Btu/hr-ft^2-°F*	819.7	622.6

*Referrred to water side

Table 8 Initial Costs ($/$kW_e$, March 1979 Dollars) for 10-MW_e (Net) Power Systems with Aluminum Heat Exchangers. (Costs That Differ with Stainless Steel Heat Exchangers Are Shown in Parentheses)

Item	Prototype 10 MW_e	First Prod'n 10 MW_e	Prod'n 100 MW_e per yr*
Heat exchangers	763 (1337)	455 (796)	367 (642)
Phase separators	9	8	5
Turbine generator	328	253	214
Ammonia feed pumps	25	23	13
Ammonia piping	140	127	85
Ammonia valves	222	200	126
Control systems	80	74	55
Chlorination sys.	21	19	11
Mech. cleaning sys.	92	81	19
Electrical equip.	39	35	24
Ammonia supp. sys.	208	112	72
Seawater pumps	340 (357)	310 (326)	217 (228)
Totals	2267 (2858)	1697 (2054)	1208 (1494)

*160 each of evaporators and condensers, which means that 320 H.E. modules are produced.

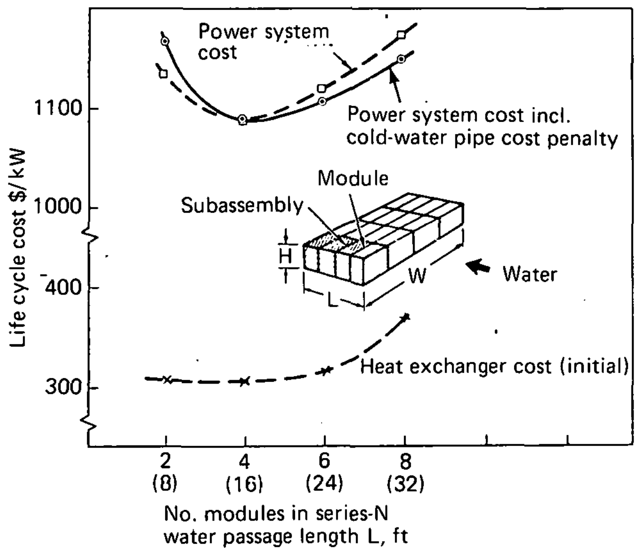


Fig. 12 **Cost versus number of modules in series (production articles).**

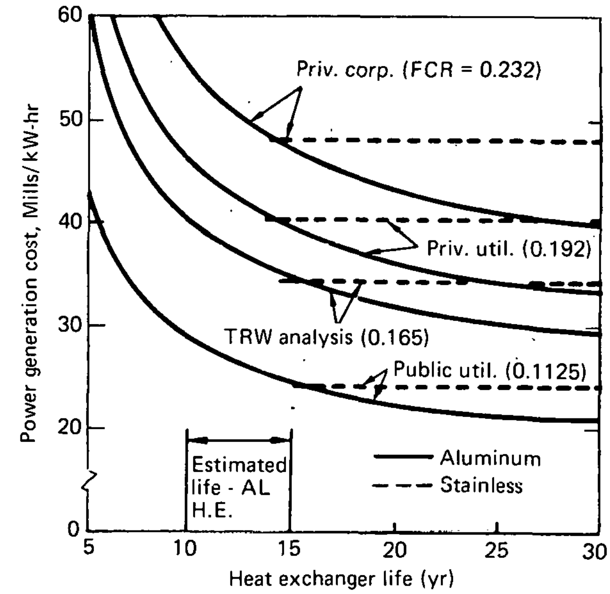


Fig. 13 **Power generation costs.**

Table 9 Fixed Charge Rate Percentages Based on Assumption of 7% Inflation Rate on Labor and Materials

	Public Util.	Priv. Util.	Priv. Corp.	TRW Anal.
Return	8.0	10.0	14.0	10.5
Income Tax	0.0	4.0	4.0	2.0
Depreciation	1.8	1.5	1.5	0.9
Ad Valorem Tax	0.0	2.25	2.25	1.5
G&A Expenses	1.25	1.25	1.25	1.0
Insurance	0.25	0.25	0.20	0.6
Total	11.25%	19.20%	23.2%	16.5%

Table 10 System Life Cycle Cost and Power Generation Cost (March 1979 Dollars)

H.E. Life, yr	Present value, H.E. Cost $/kW	Present value, total system $/kW	Power cost, mills/kWh Pub. Util.	Priv. Util.	Priv. Corp.
Aluminum					
10	875	1716	28.5	45.8	54.5
15	619	1460	24.8	39.6	47.0
30	367	1208	21.2	33.4	39.9
Stainless steel					
30	642	1494	25.3	40.4	48.0

For the Table 8 costs and the various fixed-charge rates shown in Table 9, the resulting power generation costs are shown in Table 10 and Fig. 13.

Finally, the variations of net power output and key loop parameters with the available ocean temperature difference, as it might vary with season in tropical ocean sites, are shown in Table 11.

Table 11 Variation of Key Power-Loop Performance Parameters with Season

Parameter	Winter	Nom.	Summer
Warm water temp., $^{o}F^{*}$	78	82	85
Cold water temp, $^{o}F^{*}$	40	40	40
Gross power, MW_e	11.3	13.9	15.9
Net power, MW_e	7.4	10.0	12.0
Ammonia system			
Flow rate, lb/sec	706	764	808
Evap. temp., ^{o}F	68.9	72.3	74.8
Cond. temp., ^{o}F	50.7	51.5	52.1
Evap. pres., psia	126.4	134.0	140.0
Cond. press., psia	90.4	91.8	92.9
Evap. U, Btu/hr-ft^{2}-$^{o}F^{\dagger}$	814	820	824
Cond. U, Btu/hr-ft^{2}-$^{o}F^{\dagger}$	623	623	623
Turbine efficiency, %	88.5	89.0	88.8

*Constant flow rate of warm and cold water

†Based on water side area

Alternative Configuration

A stacked heat exchanger packaging configuration (see top right part of Fig. 2) for the power system has also been evaluated to allow greater flexibility of power system-to-ship integration design. This option, although not optimum, indicates that the power system can, conceptually, be packaged in a space taking approximately 50 percent of the plan area currently used for the design (i.e., using only one-half of the aft half of the DOE plantship). The result would be approximately 25% reduction in platform cost per kW_e at 40-MW_e size, or 6-7% in overall pilot plantship cost.

Other System Factors

In addition to the aforementioned summarized results, the following performance and design factors, as specified by DOE, have been evaluated: reliability, availability, and maintainability; instrumentation and control design; system safety, regulatory requirements and environmental impact assessments; auxiliary equipment designs; manufacturability assessments; rotating machinery trade studies and net energy analyses.[2] The proposed 0.2-MW_e HE installations on OTEC-1 including water boxes and cleaning heads are sketched in Fig. 14. The parallel TRW work on turbine design and power system control considerations are presented in Refs. 1f and 1g, respectively.

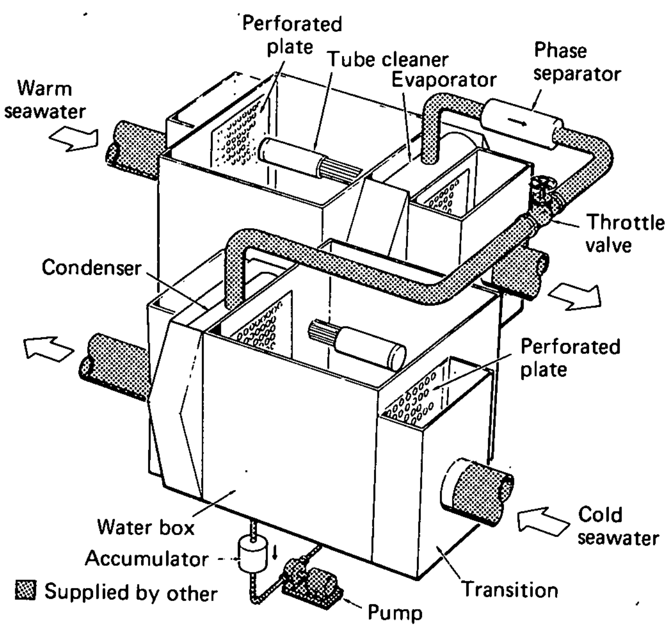


Fig. 14 General arrangement of 0.2-MW_e heat exchangers, waterboxes and support hardware.

Concluding Remarks

The TRW/Linde PSD II preliminary work that we have briefly summarized here resulted in a novel OTEC heat exchanger concept which offers potential to meet the requirement for a 30-year life-cycle cost below the desired limit of \$700/kW_e (in early 1979 dollars) set by DOE for these critical power system elements. Alclad aluminum can be expected to survive in a seawater environment for at least 10 years and, when adequately maintained and repaired, will have a useful life in excess of 15 years. The heat exchangers also can be fabricated in alternative materials; any of several of stainless steel alloys (similar to Allegheny 6X) will result in a 30-year cost comparable to that for aluminum. The high degree of modularization inherent in plate-type heat exchangers provides packaging options as to hull configurations and results in high availability/maintainability for the power system. Although overall OTEC plantship (or moored plant) design and cost estimates were beyond the scope of our PSD I work[1d] and this PSD II work, we are confident from our prior work, and our recent work on the cold water pipe[9] that no technical breakthroughs are required in order for OTEC to deliver an environmentally safe, inflation proof, renewable source of baseload energy and/or energy-intensive products[1h,4] at potentially competitive costs compared to conventional systems.

Acknowledgements

This material is the result of many hours of dedicated efforts by contributing TRW/Linde team members. These members include supporting personnel from TRW's Facilities, Mechanical and Power Integration Laboratory, led by M. K. Khandhar. At UCC/Linde, major contributions in materials assessments were made by R. Zawierucha and in structural design by Dr. J. H. Royal. The contributions of L. R. Niendorf (UCC/Linde), P. T. Fukunaga (TRW) and R. O. Pearson (TRW) are also significantly evident in the results of this project. As to the preparation of this paper, we acknowledge the help and guidance of Dr. G. L. Dugger of The Johns Hopkins University, Applied Physics Laboratory.

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DISCUSSION

J. Keirsey, JHU/APL: Would you please comment on possible problems associated with the ammonia loop in the TRW vertical stacking option. The difference in elevation would result in different ammonia saturation temperatures at the evaporator inlet.

J. Denton: We have not looked at that in detail, I have to admit. We have looked in some detail at the manifolding on the liquid ammonia side, because there is a potential for ammonia sloshing. The characteristic frequencies for sloshing of ammonia are considerably higher than the roll frequency of the ship, hence we are not concerned about ammonia sloshing. But the worst thing that could happen is that you would have to do some throttling and would have some penalty on the pumping power on the ammonia side. We are more concerned about the water side, but that, I think, is not unique to this concept. Water sloshing, or water hammer, or water level fluctuation in an open system is probably a concern for almost any kind of OTEC plant design. The report of our proposal for the next phase is now being evaluated, and the design part should be available to you soon.

Question: It is a resistance-weld, which has to penetrate through one plate.

D. Richards, JHU/APL: You had one curve that indicated the number of modules, from two up to eight modules. As you increase the number of modules, your cleaning brush mechanism, which had to have an equivalent or greater length, would get longer. Therefore, would it be practical to go beyond your present four-stack modules? Is that a major factor that could come into a plant packaging arrangement? Might it be even better to decrease it to three?

J. Denton: Yes, I think that is a valid point. The analysis shown did not take into account that it could be more difficult, or maybe impossible, to go beyond a certain length in the flow passage direction, in terms of the brush length. On PSD-1, the effective tube length is about 30 ft, and the brush length is about 33 or 34 ft. On the other hand, where you have a vertical design, you might not have the problem you have in the horizontal design. You might have some additional problems in the latter if you go more than about 16 or 17 ft.

DESIGN OF A 10-MW(e) POWER SYSTEM AND HEAT EXCHANGER TEST ARTICLES USING PLATE HEAT EXCHANGERS

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Abstract

A significant portion of the cost of an Ocean Thermal Energy Conversion (OTEC) plant is attributable to the power system, and the major portion of this cost is associated with the heat exchangers. The objectives of the Power System Development (PSD)-II program are to design a 10-MW(e) power system utilizing shell-less heat exchangers and capable of being incorporated into the grazing plantship proposed by The Johns Hopkins University's Applied Physics Laboratory, and to design, develop, fabricate, checkout and provide installation and operation support for a set of heat exchanger test articles for testing on the OTEC-1 platform. In the type of plate heat exchanger selected, titanium plates stamped into chevron patterns are gasketed and clamped face to face by bolts between a frame plate and a pressure plate. For the PSD-II 0.2- and 10-MW(e) systems, crossflow of the seawater would be used, with vertical flow of the ammonia (upward for the evaporators, downward for the condensers). Assembly and arrangement concepts for both an integrated approach and a more modular approach are described in some detail.

Introduction

The Power System Development (PSD)-I program was concerned with shell-and-tube heat exchangers since they were considered to be more state-of-the-art than shell-less heat exchangers which retain the working fluid within well-defined channels or ducts and, therefore, do not require an encompassing shell. This effect should reduce the required platform volume and reduce overall OTEC plant cost.

This paper presents the results of the Phase I portion of the Power System Development (PSD)-II program accomplished by Lockheed Missiles and Space Co. (LMSC) under DOE Contract ET-78-C-01-3407. The basic objectives of this portion of the program were to derive a conceptual and preliminary design of a 10-MW(e) power system compatible with a prescribed platform to provide a preliminary design of heat exchanger test articles sized to provide 1/50 the thermal rating of the power system heat exchanger. The conceptual design task was accomplished with the understanding that one half of the platform would be available for the 10-MW(e) power system installation.

The Applied Physics Laboratory of The Johns Hopkins University has proposed use of a grazing plantship which utilizes the power generated by the OTEC power system in an energy intensive manufacturing process. Ammonia and hydrogen are two potential products for such a plant. A 10/40 MW(e) platform has been proposed as a pilot plantship as shown in Fig. 1. The vessel could incorporate a 10-MW(e) Applied Physics Laboratory power system in the forward half and two alternative 10-MW(e) power systems in the after half. The space allocated for the power system proposed by LMSC is shown in Fig. 2. During the preliminary design task, direction was received to utilize one half of the after half of the platform for the power system installation.

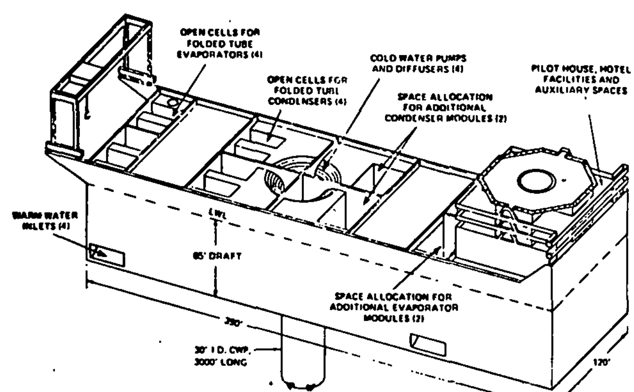


Fig. 1 Reference pilot plantship.

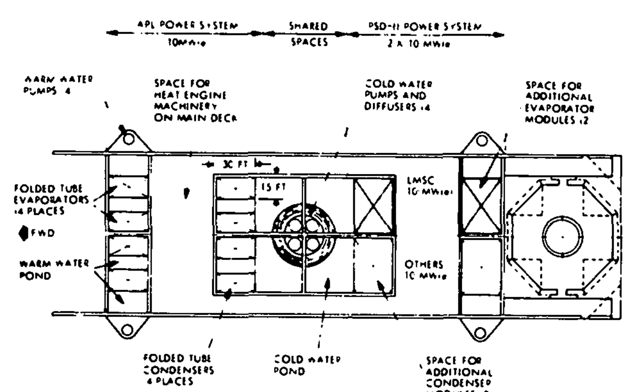


Fig. 2 Allocated spaces—OTEC reference.

Design Requirements

The proposed use of the reference surface platform is in a grazing mode in the Atlantic Ocean off the coast of Brazil. The design point thermal resources are 82°F surface water and 40°F deep ocean water. This 42°F ΔT is the annual median temperature over the anticipated grazing locale and is used to size the compact plate type heat exchangers. The design requirements are listed in Table 1.

Table 1 Design Requirements

Ocean ΔT -- 42°F (+3°F, -10°F)	
Power system containment -- Surface platform (Figs. 1 and 2)	
Heat exchangers -- "Shell-less" construction	
Design size -- 10-MW(e) net at bus bar after accounting for parasitic power and 112.5 kW hotel load	
Design life -- Major components, 30 years • 8,000 hr/yr operation • 1,000 start/stop cycles	
Ammonia vapor and seawater pressure drops -- Determined by most cost-effective power system configuration	
Working fluid -- To be selected by contractor	
Materials -- Compatible with ocean environment and working fluid	
Biofouling factor -- $\leq$ 0.0005 hr-ft^2-°F/Btu	

Table 2 Functional Requirements

- Control system -- Optimize power output for design and off-design operation
- Handling and installation -- Define features, special equipment
- Startup/shutdown -- Power requirements and sequences
- Safety -- Appropriate government and industrial codes
- Maintenance -- Provisions for efficient maintenance during downtime
- Availability -- Operability: 90% at 60% confidence; startup reliability: 95% at 60% confidence
- Power usage -- Part load, load shedding, startup
- Interfaces -- Compatible with OTEC-1 and OTEC-10

The working fluid and heat exchanger material were selected on the basis of trade studies which indicated that ammonia and titanium, respectively, are the best choices for the Alfa-Laval plate-type heat exchangers. Titanium also is appropriate for the initial application because of its proven effectiveness in seawater. As other materials are qualified for OTEC usage, they will be evaluated from a cost-effective standpoint.

Functional Requirements

The power system also must be designed to meet certain functional requirements. Previous power system studies indicated that the seawater and ammonia flow rates should be varied as functions of the available thermal resource. The control system should therefore be designed to optimize the power output at all operating conditions. The functional requirements to be considered in the design and operation of the 10-MW(e) power system and the heat exchanger test articles are presented in Table 2.

The Plate Heat Exchanger

The basic concept of the plate heat exchanger (PHE) was developed 40 years ago in response to the need for solutions to thermal problems not amenable to shell and tube application. The PHE comprises a pack of gasketed metal heat-transfer plates clamped face to face by bolts between a frame plate and a pressure plate. The heat transfer plates are suspended between two horizontal carrying bars and are compressed against the stationary frame plate by the movable pressure plate which is also suspended from the upper carrying bar (Fig. 3).

The configuration of gaskets and separately gasketed corner ports routes the two media through alternate interplate channels in cross-flow or in parallel groups (Fig. 4). The space between the port gasket and the body gasket is open to the surrounding atmosphere. The risk of intermixing of the two media is therefore considerably reduced, since leakage across one gasket will flow out to

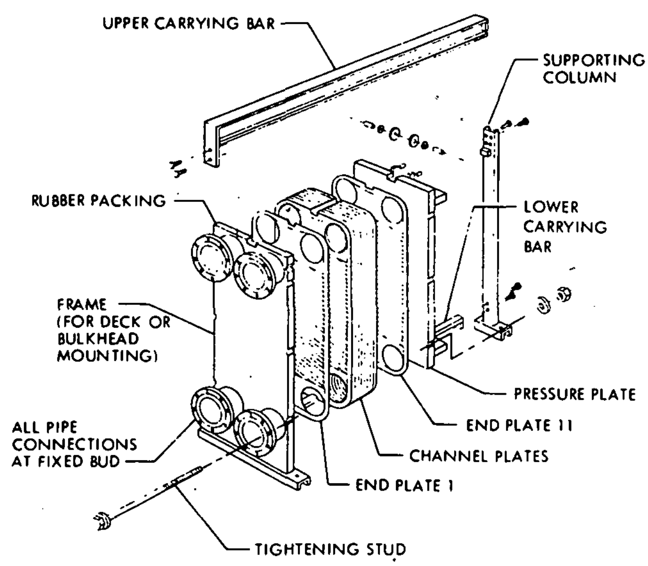


Fig. 3 Exploded view of basic heat exchanger.

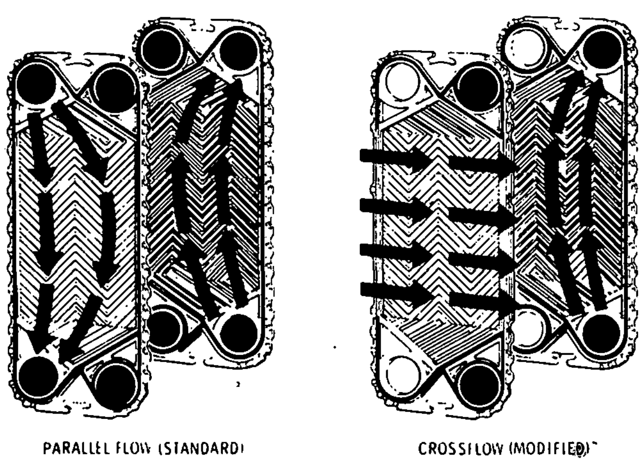


Fig. 4 Current production plate technology.

the surrounding atmosphere through the vent space. This feature permits easy location and repair of leaking gaskets. The principle of the PHE also allows combining various plate patterns for an intended duty by use of an effective thermal length, θ,

$$\theta = (t_i - t_o)/\Delta t_m$$

where: t_i and t_o are the inlet and outlet temperatures in a primary channel, and Δt_m is the logarithmic mean temperature difference between primary and secondary channels. High- and low-θ plates have been developed, and these are combined as required for the thermal duty.

Current and Future Production Plate Technology

The parallel-flow plates in Fig. 4 represent the current standard plates. The chevron pattern pressed into the plate determines the θ value of the plate. The pattern shown has an intermediate θ. The heat transfer characteristics are such that a low-θ plate, which has a large angle, is more applicable to the OTEC configuration.

The standard parallel flow arrangement has been used in a number of single-phase cooling and heating applications. For the two-phase flow required in the OTEC power cycle, the configuration shown represents the evaporation process. Liquid ammonia enters at the bottom, and ammonia vapor (at some quality less than 1) exits at the top. The seawater flow enters at the top. As indicated by the arrows, this is a counterflow arrangement. For condenser use, ammonia vapor would enter at the top with the condensate flowing out the bottom in a co-flow arrangement. This configuration is incorporated into the Mini-OTEC heat exchangers.[1a]

The current production plate can also be used in a crossflow configuration as shown in Fig. 4; the seawater flow is at right angles to the ammonia flow. The evaporator arrangement is as shown, whereas the ammonia flow path is reversed for the condenser.

Optimization of plate design for two-phase flow requires modification to the inlet and outlet ports to accommodate the large difference between liquid and vapor ammonia specific volume. Figure 5 indicates an approach that could be taken for the OTEC optimized plates. For parallel flow, a portion of the gasket is removed on each side, allowing seawater to enter. The gasket is removed from the base of the plate which serves as the exit. For crossflow, the gasket is removed from both sides as was done to provide crossflow in the current production plate configuration.

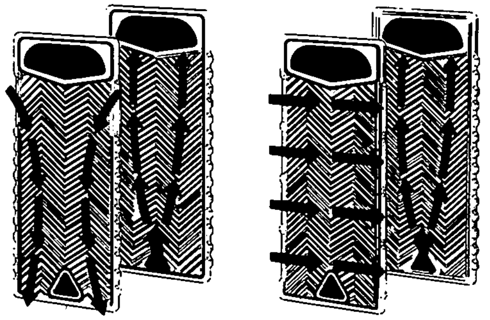


Fig. 5 Future production plate technology.

Recommended Plates for PSD-II Application

The PHE performance data indicate that the crossflow configuration is more cost effective than the parallel flow configuration. The gross power required is 0.56 MW(e) less, and the heat exchangers require 20 percent fewer plates. The recommended 10-MW(e) heat exchanger plate configuration is shown in the left hand portion of Fig. 6.

The heat exchanger test articles are required to provide data which are directly applicable to the 10-MW(e) power system. The anticipated schedule for fabrication and delivery of the test articles precludes the use of the advanced technology plates. Applicable data can be obtained by using the largest current production plates, model AL35, modified for

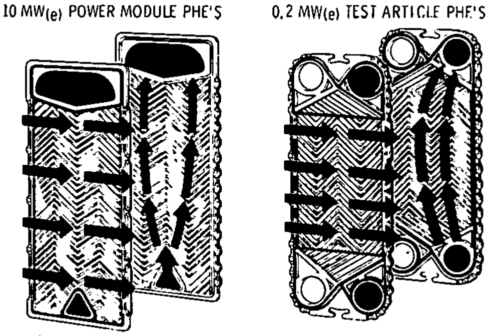


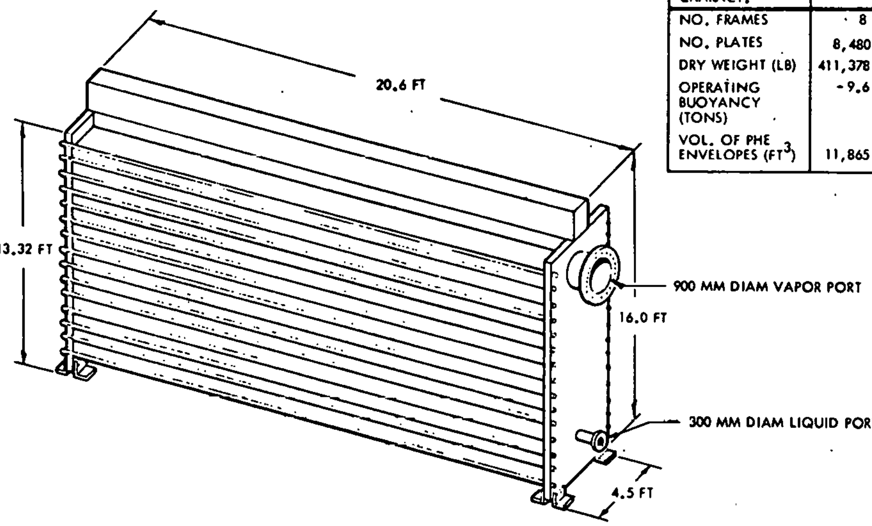
Fig. 6 Recommended plates for PSD-II OTEC application.

the crossflow configuration as shown in the right hand portion of Fig. 6. The gasket is modified to provide seawater crossflow with the added gasketing to prevent seawater flow into regions where stagnation and fouling could occur.

10-MW(e) Power System Heat Exchangers

Using the data synthesized for the advanced plate, the heat exchangers for the 10-MW(e) power system are as depicted in Fig. 7. Application of the Alfa-Laval design optimization program result in a configuration which required 4 percent more plates in the evaporator than in the condenser. Results from off-design performance computation indicated that the use of equal areas in the evaporator and condenser imposed a minor performance penalty. The system was therefore resized to provide 10-MW(e) utilizing identical heat exchanger modules in the evaporator and condenser subsystems.

A trade study conducted by Alfa-Laval indicated that the installed cost increased linearly with number of heat exchanger modules with a slope of 0.625 percent. This change is due to the increased number of end plates and increased manifolding required. The lower limit to the number of modules is defined by the maximum number of plates which can be incorporated into a single module. Alfa-Laval's experience indicates that the use of more than 1200 plates per module could result in maldistribution of the working fluid. Since 8480 plates are required, the use of seven modules would be marginal and eight would be more desirable. The selected preliminary design configuration shown in Fig. 7 consists of eight modules in each heat exchanger subsystem, with each module incorporating 1060 plates. The number of plates is five percent greater than the theoretical value to allow for variations in plate wall



SUMMARY OF 10-MW(e) PHE ASSEMBLY		
PHYSICAL CHARACT.	EVAP.	COND.
NO. FRAMES	8	8
NO. PLATES	8,480	8,480
DRY WEIGHT (LB)	411,378	411,378
OPERATING BUOYANCY (TONS)	-9.6	-10.1
VOL. OF PHE ENVELOPES (FT^3)	11,865	11,865

Fig. 7 10-MW (e) heat exchanger plate pack.

thickness and to provide preheating, if required. The corresponding 0.2-MW(e) design is shown in Fig. 8.

10-MW(e) Power System Design

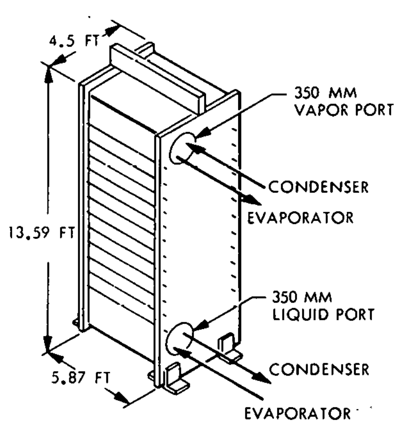
As noted above, the Conceptual Design portion of the Phase I program was accomplished with the premise that the reference platform would be designed for 20-MW(e) net power output. In this arrangement, the after half of the vessel was available for the PSD-II power system. Four heat exchanger modules were installed in each well with vapor and liquid manifolding located on the centerline of the vessel. This arrangement was not volumetrically efficient since it utilized only approximately one-third of the available well volume.

The Preliminary Design portion of the program was therefore accomplished utilizing one half of the after half of the platform. All eight evaporator heat exchanger modules are installed in one well using vertically stacked pairs. Each pair is located in a separate compartment to permit cleaning and periodic refurbishment without requiring shutdown of the remainder of the heat exchanger subsystem. A similar arrangement is utilized for the condenser subsystem and the only difference is the orientation of the modules. The evaporator modules are oriented with the inlet and exit faces athwartship while the condenser faces are parallel to the vessel centerline.

Two approaches were taken for the design of the 10-MW(e) power system. The Integrated Arrangement assumes that the power system design and the platform design are integrated through a working group to ensure incorporation of all interface requirements. The Modular Arrangement assumes that the vessel is designed as a separate entity with the proviso that well dimensions are adequate for power system installation.

Integrated Configuration

The general arrangement of the 10-MW(e) integrated configuration is shown in Fig. 9. The installed power system occupies a space envelope on the reference platform of 113.5 ft fore and aft and 60.5 ft from centerline to outboard of the torque box. The condenser well dimensions are 30



SUMMARY OF 0.2-MW(e) PHE TEST ARTICLES		
PHYSICAL CHARACTERIS.	EVAPORATOR	CONDENSER
NUMBER OF PLATES	210	210
DRY WEIGHT (LB)	27469	27469
VOLUME OF PHE ENVELOPE (FT^3)	359	359

Fig. 8 Heat exchanger test article design.

ft long by 38.5 ft across. The evaporator well is similar to the condenser but rotated 90 deg to allow for direct seawater inlet from the warm water pump. A machinery space is located between the well and platform centerline bulkhead.

Eight condensers and eight evaporators are arranged in vertically stacked pairs and divided into compartments by water-tight partitions to allow isolation from the rest of the heat exchanger subsystems for cleaning, maintenance or removal. Piping to and from the heat exchangers penetrates the water-tight well bulkheads to pumps inside the machinery space. Flexible pipe joints perform two functions: (1) reduce stresses in the short, coupled pipes that may occur as a result of relative PHE-bulkhead motion, and (2) compensate for any misalignment that may occur after installation. The heat exchangers are side-mounted on the partition water box, to allow the hydrostatic load, due to the pressure drop across the PHE, to bear directly on the waterbox.

The ammonia vapor piping is sized to meet the requirement that "the ammonia vapor pressure drop within each heat exchanger power module (inclusive of manifolding) shall not exceed 2 percent of the vapor pressure difference at 82°F in the evaporator and at 40°F in the condenser." On this basis, the allowable system pressure drop is 1.7 psi. The design-point performance indicates that the ammonia vapor pressure loss in the evaporator is 0.6 psi and the pressure gain in the condenser is 0.2 psi. The separator specification limits the pressure drop to 0.5 psi at the design flow rate. In order to meet this design requirement, the vapor piping pressure loss must, therefore, be equal to or less than 0.8 psi. The piping pressure loss is a function of length-to-diameter ratio and velocity which, in turn, is also a function of diameter. In addition, bends and fittings can be equated to equivalent L/D. An iterative procedure was used to determine the diameter which results in a line loss of less than 0.7 psi. This approach was used to allow for the incorporation of standard diameter pipe sizes in the installation.

The turbine-generator is oriented in the fore and aft direction to minimize piping complexity and reduce gyroscopic loads on the bearings. The condensate, evaporator feed, and CIP pumps are located in the pump room below the 32 ft elevation to ensure adequate pump suction head. An access hatch of 9 by 9 ft is provided in each deck to allow installation of equipment in the machinery space. An individual access hatch, 18 by 35 ft, is provided for the turbine-generator installation. A fume-tight, positively ventilated stairwell extends from the turbine-generator deck to the pump room to provide maintenance personnel access, and to act as an ammonia-vapor-free escape route in the event of any accidental spills or leaks. Decks will be required at the 32.13-, 45- and 61-ft levels, in addition to partial decks in between, for auxiliary system equipment mounting. Because of the potentially hazardous location, all motors in the machinery space are explosion proof and all electrical power conditioning

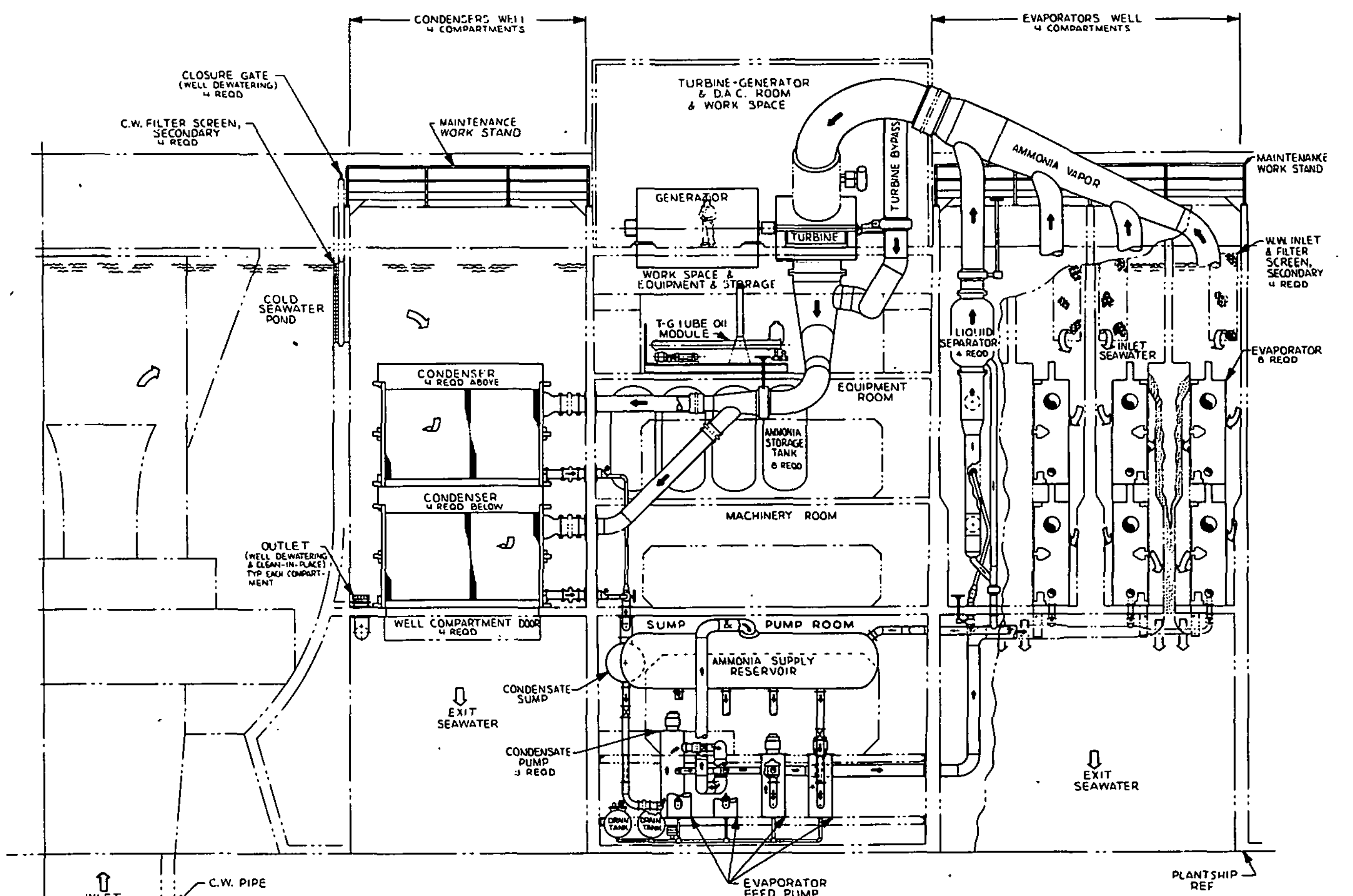


Fig. 9 Integrated arrangement.

equipment such as switchgear, transformers, etc., is located in the torque box structure above the main deck in a positively ventilated room. This permits isolation from any explosion hazard without the necessity of providing inordinately long electrical cable runs.

Modular Configuration

An alternate approach, the modular configuration, was based on the assumption that minimum integration coordination would be available during the platform design. Criteria used in the design of the modular configuration were that all heat exchanger waterboxes, supports and compartmentation structure would be supported from the main deck and there would be no piping pentrations in the vertical well bulkheads.

The heat exchanger and waterbox arrangement within the wells, the auxiliary systems and the electrical power conditioning equipment arrangement external to the wells, are similar to those of the integrated configuration. The ammonia vapor and liquid system, and the heat exchanger clean-in-place (CIP) system require vertical risers and downcomers within the wells. The impact of these criteria led to the necessity of having:

- Well support structure attachment interfaces at the main deck bordering each well, a support edge along the torque box at the main deck, and indentation of the torque box structure
- Pads embedded in the concrete well to absorb well structure bearing loads
- A change in well dimensions from 40 ft by 30 ft to:

	Evaporator Well	Condenser Well
Athwartship	35.3 ft	36 ft
Longitudinal	36 ft	33.3 ft

  The machinery space length was kept at 50 ft, but the width was reduced to 36 ft to conform to the well dimensions
- Deep well ammonia condensate and CIP pumps in the wells, plus separator drain pumps to return the separator liquid drains and the liquid runback from the vertical runs of the evaporator vapor piping
- Additional corrosion protection for the heat exchanger support structure

The condenser and evaporator well arrangements are similar to the integrated configuration in which the heat exchangers are side mounted to the well structure water boxes, and the heat exchangers are compartmented in pairs. The seawater enters through an opening in the structure, travels vertically downward to the heat exchangers, across the heat exchangers, and exits through openings in the bottom. Each compartment can be isolated from the others by an inlet sluice gate and by a hydraulically actuated closure gate located at the compartment bottom.

The heat exchanger support structure is a welded steel structure bolted to a steel bearing pad around the perimeter of the well. Each structure, weighing approximately 500 LT empty of heat exchangers, piping, and pumps, can be installed in one of three ways:

- Continuously fabricated while being lowered from the main deck
- Preassembled in modules and welded together as each module is lowered from the well
- Preassembled as a completely fabricated structure - then lowered either by a crane or a jack-down system

The first two methods allow the well structure to be installed either dockside or on station where material and/or modules can be transported to the site and the available platform handling systems can lower the structure into place. Because the 500 LT structure weight would require handling equipment beyond the capacity of that on the platform, the third method could be accomplished either dockside, using land-based crane facilities or on site, using high-capacity floating crane barges, such as GOLIATH II or SUN 800 or others.

The ammonia condensate pumps are deep well pumps in 45-in.-OD pressure-resistant cans that serve also as sumps. The pump cans are an integral part of the well structure at the juncture of the well partitions and well frame. The cans are evacuated by means of a jet pump at the base of the can. Any residual ammonia that cannot be removed by the jet pump is removed by the ammonia unloading system vacuum pumps. The can volume is then purged prior to pump removal. One CIP pump is located in each well. Manifolding allows the pumps to dewater individual compartments, fill and circulate CIP solution through the heat exchangers in an opposite direction to normal seawater flow, and then return the CIP solution to holding tanks. The residual CIP solution, approximately 6 percent of compartment seawater volume, is first diluted with seawater by opening ports in the inlet sluice gate prior to releasing solution into the ocean. If removing all residual solution is required for maintenance, a small portable pump can be lowered into the well and all residual water pumped overboard.

To minimize vapor piping losses and make available more space in the machinery area, the liquid separators are installed in the evaporator well. This results in a small increase in well length to accommodate the heat exchanger installation and removal envelope. The drainage from the liquid separators and any liquid run back from the vertical evaporator vapor pipe is pumped via deep well separator drain pumps to the ammonia supply reservoir.

To balance heat-exchanger seawater flows, the well structure on the inlet side of the heat exchangers is shaped to provide a constant seawater velocity. This arrangement also permits heavier structure at the base of the compartments. Nonstructural fairings are installed on the discharge side of the waterbox to create an equivalent back pressure and balance the pressure drop across the upper and lower heat exchangers.

Heat Exchanger Test Articles

The 0.2-MW(e) heat exchangers are designed to simulate the 10-MW(e) power system arrangement from thermodynamic and fluid dynamic standpoints. Since the modification of the Model AL35 plates (Fig. 6) reduces the effective plate heat transfer area, the number of plates required is greater than 1/50 the number of plates in the 10-MW(e) power system. The resultant configuration is shown in Fig. 8.

To provide the seawater side crossflow on board the OTEC-1 test facility, the heat exchangers must be enclosed in waterboxes. An exploded view of the waterbox arrangement is shown in Fig. 10. The waterbox is designed to mate with the 2-ft-dia. seawater piping on board OTEC-1 and with the clean-in-place system that is used periodically to supplement the use of chlorine as a biofoul countermeasure. An artist's rendition of the test hardware installed in OTEC-1 is shown in Fig. 11. The heat exchangers are installed on the machinery flat at the 21 foot level. The ammonia pumps, sumps and preheater, and the pumps and storage tank for the clean-in-place solution are mounted on a foundation floor above the double bottom.

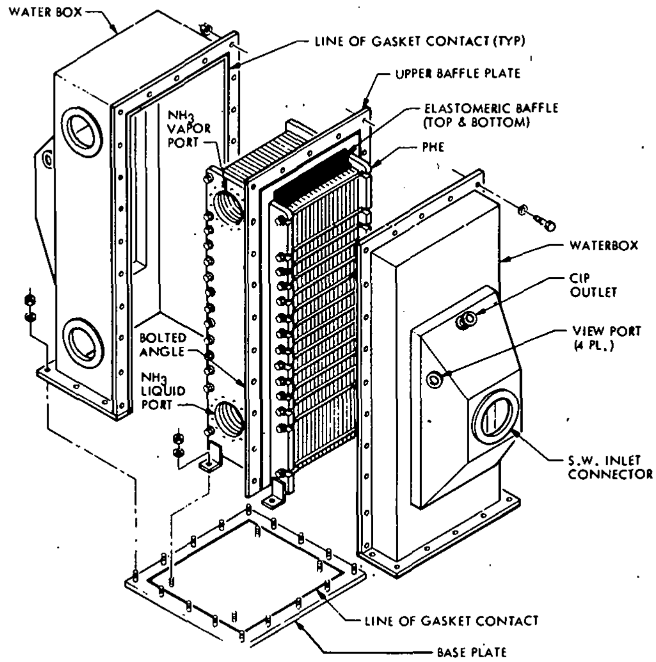


Fig. 10 Exploded view of waterbox arrangement.

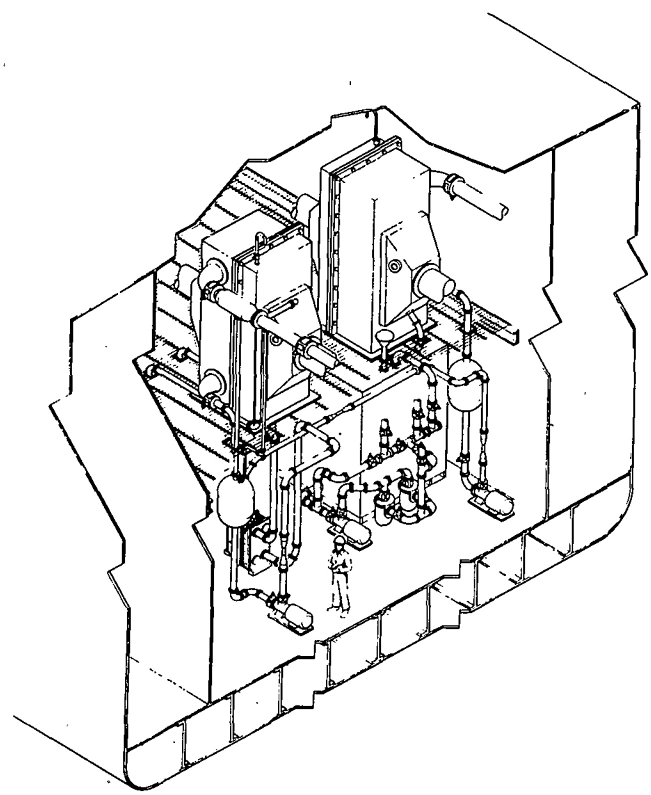


Fig. 11 OTEC-1 hardware installation.

The objectives of the test program are (1) to evaluate the thermodynamic and fluid dynamic performance of the heat exchangers; (2) to determine the biofouling characteristics and evaluate the effectiveness of the biofouling countermeasures; and (3) to evaluate the long-term performance from materials and maintenance standpoints. A part of the Phase II effort is related to the finalization of the test plan.

Concluding Remarks

We have summarized the results of our Phase I effort on the Power System Development (PSD)-II program. The 10-MW(e) power system is designed to be accommodated in one half of the after half of the reference platform. The compact plate heat exchangers are based on advanced production plates proposed by Alfa-Laval. Use of this generic type of heat exchanger is presently being demonstrated on Mini-OTEC. The heat exchanger test articles are sized to provide 1/50th the thermal flux of the 10-MW(e) power system heat exchanger. The test articles and related hardware are designed to be compatible with the OTEC-1 test facility. A companion paper by Owens addresses system response and control.[1b]

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OPTIMIZING PLANT DESIGN FOR MINIMUM COST PER KILOWATT WITH REFRIGERANT-22 WORKING FLUID

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Abstract

A rationale for the selection and use of Refrigerant-22 as the Ocean Thermal Energy Conversion (OTEC) working fluid and the basic logic and results of a power system design optimization and off-design analysis for a 0.2-MW_e, plate-type, OTEC heat exchanger and a 10-MW_e power module are presented. The results based on thermal performance indicate that a system using R-22 will cost only 10% more than one using ammonia when both systems have enhanced surfaces on the fluid side of the heat exchangers. However, even this small difference may disappear when operational systems are built.

Introduction

The design of an OTEC power module with flate-plate, cross-flow heat exchangers must address a very large number of variables. A computer program for power system (module) design optimization and off-design analysis was developed by General Electric and Sea Solar Power during Phase I of the Department of Energy's OTEC Power System Development (PSD) II program. Sea Solar Power, founded by the Andersons, has long advocated the use of a hydrocarbon or a Freon-type refrigerant as an OTEC working fluid,[1,2] whereas others have advocated ammonia.[3a-c] The computer program is designed to find minimum power system cost. It incorporates existing Freon and ammonia property programs, power system performance programs, heat exchanger design programs, and cost minimization programs.

Selection of Working Fluid

Initially, some 17 fluids were evaluated and then reduced to seven prime contenders: R-31/114, R-12/31, R-500, R-22, isobutane, ammonia, and propane. The specialty freons are azeotropic mixtures manufactured to obtain specific performance characteristics. Efforts were directed to development of a holistic approach to working fluid selection that would take into account various properties in addition to efficiency of heat transfer. Although certain supporting system costs and requirements can be developed to compare relative costs of ammonia versus freon, it is difficult if not impossible to quantify health risks or material handling problems. Table 1 compares various properties of Freon-22 (or R-22) and ammonia.

Fifteen fluid properties were identified and then prioritized according to OTEC program impact. Weighting factor points were assigned to these characteristics as shown in Table 2. In this quasi-objective comparison ammonia did not finish near the top, because it has some serious shortcomings: tip speed is too high for a single-stage turbine; its corrosive characteristics limit materials available for the heat exchangers; it is not soluble in oil, thus leaving open the possibility of the oil fouling heat transfer surfaces; but it is soluble in water leaving no opportunity to separate the working fluid from any seawater that leaks in. Leakage of either seawater into ammonia or ammonia into seawater leads to calcareous scale deposits. There is an unresolved concern about corrosion product film development related to oil. Ammonia is toxic, both to humans and aquatic life, and it is flammable. In summary, ammonia offers excellent heat transfer capabilities, but this advantage is offset by several secondary concerns.

Table 1

Comparison of ammonia and freon characteristics.

FLUID PROPERTIES	FREON-22	AMMONIA
HEAT TRANSFER COEFFICIENTS		
BOILING	317 BTU/HR·FT^{2}·°F	1702 BTU/HR·FT^{2}·°F
CONDENSING	340 BTU/HR·FT^{2}·°F	1430 BTU/HR·FT^{2}·°F
CYCLE EFFICIENCY	3.15%	3.21%
VAPOR PRESSURE AT 52°F	103 PSIG	92.7 PSIG
VAPOR PRESSURE AT 72°F	142 PSIG	133.4 PSIG
TURBINE SIZE (25 MW)		
WHEEL DIAMETER	158 IN	98.3 IN
TIP SPEED	275 FPS	698 FPS
SOLUBILITY IN WATER	NO	YES
SOLUBILITY IN OIL	YES	NO
CORROSIVITY	NO	YES
FLAMMABILITY	NO	YES
TOXICITY	NO	YES
ENVIRONMENTAL EFFECTS	NO	YES
SAFETY ASPECTS	NO	YES

From the ratings in Table 2, R-22 emerged as a leading contender. Although R-22's theoretical heat transfer characteristics are much lower than ammonia's, Ganic & Wu found that with enhanced heat transfer surfaces, the coefficients become very close.[4] R-22 permits a much lower turbine tip speed, is not corrosive or flammable, and requires no particular handling precautions. It is soluble in oil, avoiding hydrocarbon deposition, and can be readily separated from seawater. Although some fluorcarbons have been of concern in ozone degradation, R-22 represents no environmental problems in this regard.

System Description

The system used for the preliminary design development consists of a basic Rankine cycle which uses warm (surface) seawater to boil the working fluid, expands the vapor through a turbine, and then condenses it using cold seawater from a 3000 ft depth. The design uses auxiliary turbines to drive the warm and cold seawater pumps and the boiler feed pump(s) as indicated in Fig. 1. Turbine driven pumps improve the overall plant efficiency and provide better control during off-design conditions. A comparison of the turbine-driven pump system with a motor-driven pump system demonstrated that the former would lower the plant cost by 7%.

Flat-plate, cross-flow, "shell-less" heat exchangers are used.[2] Seawater will flow horizon-

Table 2.

Working fluid evaluations.

NO.	CHARACTERISTICS	FACTOR POINTS	AMMONIA POINTS	R–31/114 POINTS	R–12/31 POINTS	R–500 POINTS	R–22 POINTS	ISOBUTANE POINTS	PROPANE POINTS
1.	HEAT TRANSFER COEFFICIENTS FOR ● BOILING UNIT ● CONDENSING UNIT	200	200	70	55	60	80	90	110
2.	TURBINE SIZE WHEEL DIAMETER WHEEL TIP SPEED	120	85	65	95	90	120	65	85
3.	VAPOR PRESSURE AT 52°F FOR CONDENSER DEPTH 72°F FOR BOILER DEPTH	100	70	90	85	75	60	100	65
4.	PUMPING POWER	80	50	80	80	80	80	40	20
5.	CYCLE EFFICIENCY (ACTUAL)	80	80	65	65	65	60	70	75
6.	ENVIRONMENTAL EFFECTS	75	20	70	70	70	75	25	25
7.	CORROSION ACTION ON MATERIALS	75	20	65	65	65	75	65	65
8.	SAFETY ASPECTS	70	20	70	70	70	70	25	25
9.	CORROSIVENESS IN WATER	65	20	65	65	65	65	65	65
10.	TOXICITY	50	10	50	50	50	50	50	50
11.	FLAMMABILITY	35	7	35	35	35	35	1	1
12.	SOLUBILITY IN WATER	35	9	35	35	35	35	35	35
13.	SOLUBILITY IN OIL	35	5	25	25	25	14	25	25
14.	LIQUID FLOW VOLUME	30	30	25	17	18	20	15	10
15.	FLUID COST	30	25	NA	NA	20	20	25	30
16.	TOTAL (POINTS)	1080	651	810	812	823	859	696	686

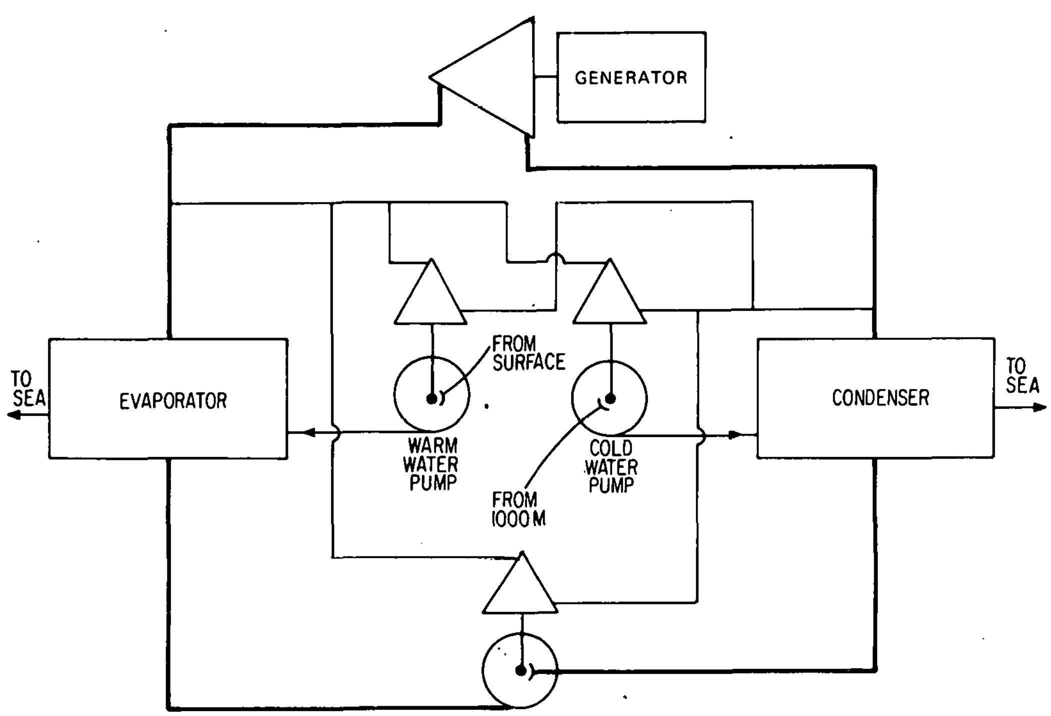

Fig. 1 Power plant system concept.

tally through channels making a single pass from one end to the other. Alternate vertical, 2-ft-high channels separated from the water by the "flat plates" contain the working fluid. In the evaporator, the working fluid floods the vertical channels. Nucleate "pool" boiling occurs on the fluid side plate walls. The vapor bubbles rise to the tops of the channels where a low-velocity separator or demister separates the vapor and liquid. The vapor is then directed to the main and/or auxiliary turbines for expansion. In the condenser, the working fluid condenses on the vertical channel walls and drains to the bottom where it is collected and taken away by the condensate feed pump.

Subroutines in the Computer Program

The heat exchanger subroutines calculate the water-side heat transfer coefficients based on one of several correlations (Diessler, Colburn, Popov, etc.). Alternate water-side fouling factors or correlations can also be used. Enhancement of the fluid-side coefficients is not extensively documented for freon and is therefore the subject of a separate test program and paper. After the coefficients have been calculated, the subroutine calculates the heat transfer area requried, the length and the width of the units, and water-side pressure drops. An input unit cost ($\$/ft^2$) then yields a total cost of the heat exchanger units.

The performance subroutine is based on a program presented by Abelson.[5] Changes to the logic were required to incorporate the turbine-driven pump features, to delete the separate demister, and to account for piping pressure drops. The subroutine uses component unit costs, turbine efficiencies, evaporator outlet temperature, and the relative elevations of equipment as input data and then determines fluid properties at each state point, load for each component, minimum pipe sizes, piping pressure drops, Carnot, Rankine and plant efficiencies, and the cost of each major component.

There are two subroutines which can be used to calculate the fluid properties: one based on curve fits for freon and ammonia, and the University of Utah "Freon Properties Program."[6] Another program based on ASHRAE calculation procedures (used by Sea Solar Power) is available but is not presently incorporated into the optimization program.

Two minimization programs are incorporated into the program, one based on a heuristic direct search approach,[7] and one called "MAXOPT" which is on the General Electric TIPO time share computer. Use of both programs permits an independent check of validity.

Dr. Mosteller's program[7] uses a direct-search minimization algorithm which consists of global and local search strategies. Direct-search algorithms do not require derivatives; only function values.

In general, these methods are successful in locating a small neighborhood containing the minimum and are attractive since the exact minimum is not needed and computer time and storage can be saved by eliminating derivative computations.

The "MAXOPT" Fortran program maximizes an unconstrained multivariate function. Derivatives are not required. This program consists of two simple maximizing codes, LOGIC1 and LOGIC2, both written as subroutines, and an analysis program which calculates the value of the response function and issues a call to one of the subroutines. Both strategies are simple hill-climbers: LOGIC1 implements an optimum gradient technique when used in MODE = 1 or a steepest ascent method when MODE = 0. The program is dimensioned for 10 adjustable parameters. For the present use it was necessary to modify the "MAXOPT" program slightly to have it find the minimum ($/kW) value in lieu of the maximum.

The logic of the program is shown in Fig. 2. Typical input data and results are shown in Table 3. The input data and initial data and initial conditions for each of the variables are used by the heat exchanger, fluid properties and system performance subroutines to establish the initial design conditions and component sizing data. Input unit costs are used with the calculated sizes to provide the component costs, which with the support system costs, are totaled to determine total plant cost and $/kW for the plant. This procedure is repeated with new sets of variable numbers until the minimum cost design is found, then the off-design subroutine can be activated.

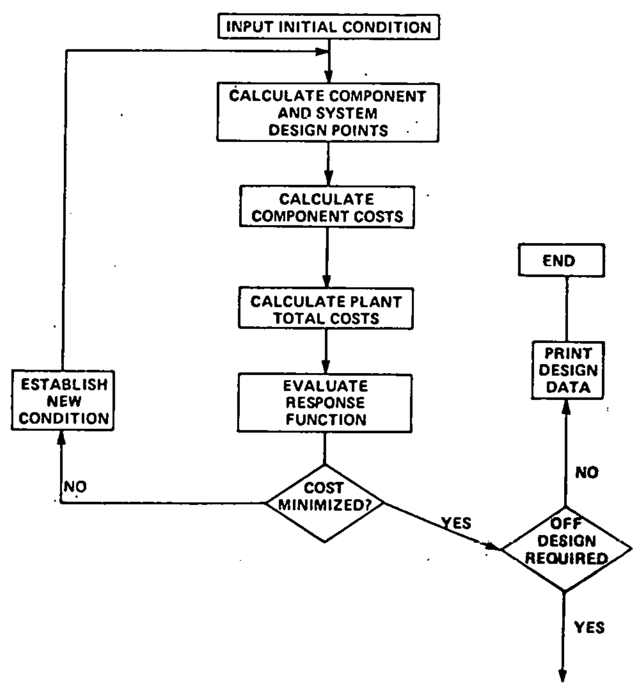

Fig. 2 Optimization program logic.

Parametric Analysis

Typical sensitivity analysis is shown in Figs. 3-6.

Figure 3 shows water velocity heat exchanger versus fluid side coefficients (enhancement) for the evaporator and condenser with R-22 and ammonia.

Table 3

Typical input parameters and results.

	Evap.	Cond.
Input Parameters, HX System		
Core material	Cu-Ni	Cu-NI
Conductivity of core mt., Btu/hr-ft^2-^oF/ft	35	35
Heat transfer coeff. function (W) (Dysler)	1.00	1.00
Heat transfer coeff. function (F)	2500	3500
Fouling factor (water side), hr-ft^2-^oF/Btu	0.0003	0.0003
Fin factor	3.500	4.251
HX core plate thickness, in.	0.050	0.050
Water passage shape	Rect.	Rect.
Passage height (water side), in.	0.500	0.500
Passage width (water side), in.	0.375	0.375
Passage width (fluid side), in.	0.188	0.188
End spacer thickness (W pass), in.	0.500	0.500
Spacer thickness (W pass), in.	0.050	0.050
Water temp. (entrance), ^oF	82.0	40.0
Cost of heating area, \$/ft^2	15	15
Cost of aux. turbine, \$/kW	350	350
Results, HX System		
Fluid temperature (4 & 6), ^oF	71.0	50.26
Seawater velocity, ft/sec	4	3.715
Temperature diff. across HX, ^oF	3.255	5.241
Log mean temp. diff, ^oF	8.937	7.329
h_w, water side, Btu/hr-ft^2-^oF	1209	819.5
h_f, fluid side, Btu/hr-ft^2-^oF	2500	3500
h_fp, preheating section, Btu/hr-ft^2-^oF	1000	
Preheating area, ft^2	11432	
Boiling area, ft^2	195528	
Total HX area, ft^2	206960	291074
Heating area/net kW, ft^2/kW_e	20.696	29.107
U, overall coeff.	607.5	519.5
Number of water passages	5167	3181
Length of HX, ft	7.96	17.3
Width of HX, ft	285	175.6
ΔP across HX (water), ft	2.12	6.34
Heating area cost, \$K	2155	2884
HX aux. equip. cost, \$K	348	2229
Total cost of HX, \$K	2504	5113
Heating area cost, \$/kW_e	215.5	288.4

	Main Turbine	Aux. Turbine
Cycle Analysis		
Turbine efficiency	8.90	0.80
Genrator efficiency	0.97	
Cost/turbine kW	250	350
Pump efficiency	0.85	0.85

Results, Cycle Analysis

State no.	Press. psia	Temp. ^oF	Spec. vol. ft^3/lb_m	Enthalpy Btu/lb_m	Entropy $\frac{Btu}{lb_m-^oR}$	Quality
1	99.3	50.2	0.0128	24.08	0.0514	0
2	146.7	50.7	0.0128	24.21	0.0514	0
3	138.9	71.1	0.0132	30.21	0.0630	0
4	138.8	71.1	0.3766	106.15	0.2059	0.959
5	138.2	70.8	0.3785	106.19	0.2061	0.951
6	99.4	50.3	0.5181	103.31	0.2066	0.937
7	99.4	50.3	0.5201	103.63	0.2073	0.940

		Flow rate, 10^6 lb_m/hr
Main turbine load	35.58 MBtu/hr	11.975
Cond. turbine load	2.21 MBtu/hr	0.8387
Boil. turbine load	1.23 MBtu/hr	0.467
Pump turbine load	185 MBtu/hr	0.699
Aux. turbine load	5.29 MBtu/hr	2.004
Condenser load	1108.3 MBtu/hr	13.979
Boiler load	1145.4	13.979
Feedwater pump HP	726 H.P.	
Warm water	729,200 G.P.M.	374.32
Cold water	438,200 G.P.M.	224.95
Carnot cycle eff.	3.87 %	
Rankine cycle eff.	3.11 %	
Plant efficiency	2.98 %	
Cost of main TB	\$261/kW	
Cost of aux. TB	\$54.3/kW	
Total cost output	\$1041/kW	

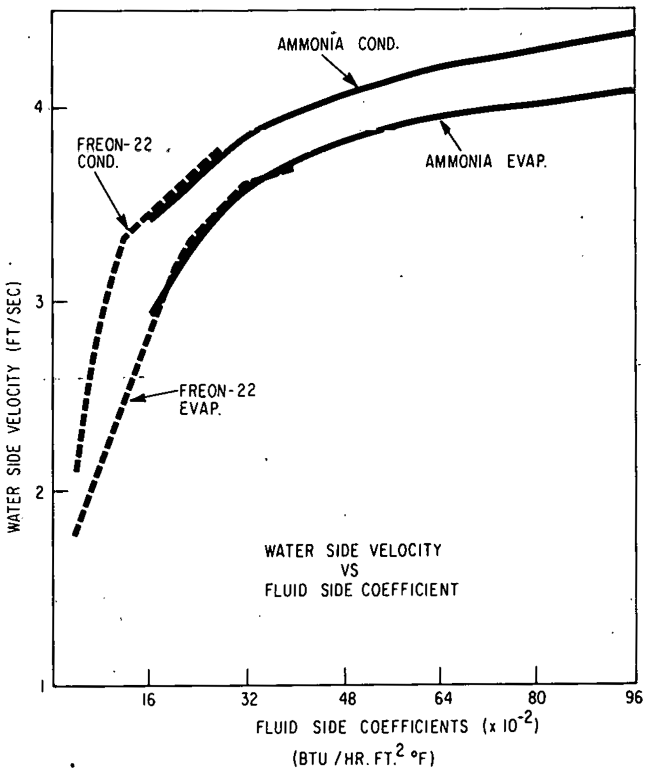

Fig. 3 Relationship between water velocity and fluid-side heat transfer coefficients.

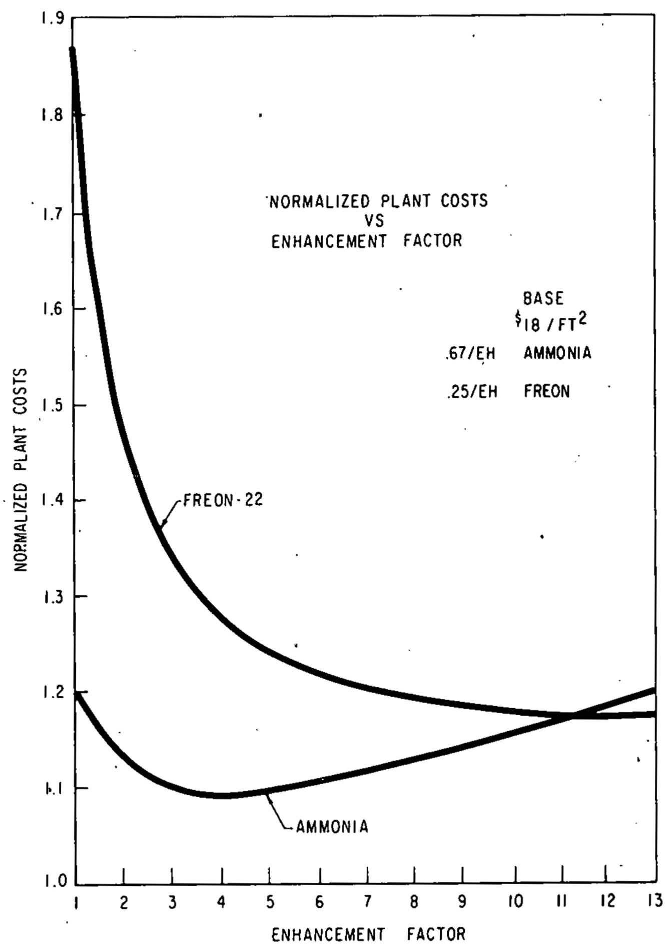

Fig. 4 Effect of fluid-side surface enhancement on plant cost.

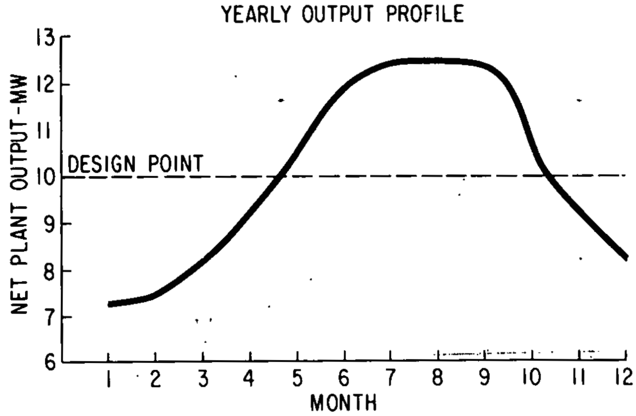

Fig. 5 Monthly variation of plant output.

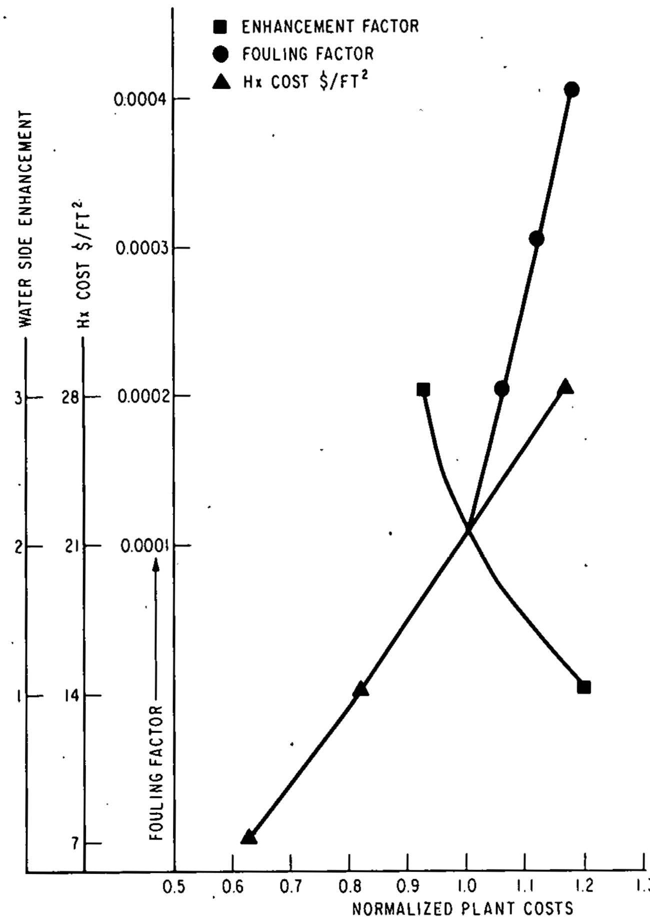

Fig. 6 Effects of enchancement and fouling factor on heat exchanger and plant costs.

This figure does not necessarily represent a real plant since the condenser and evaporator coefficients were set equal to each other. The difference between the condenser and evaporator velocities can be attributed to the fact that a cost function for the cold water pipe is included in the program. An increased velocity in the condenser will result in a smaller diameter (i.e., cheaper) cold water pipe than would result if equivalent velocities were used. It can be concluded from this evaluation that the velocities in the heat exchangers should be between 2 and 4 ft/sec for either R-22 or ammonia assuming that there is no water side enhancement factor.

Figure 4 shows normalized plant cost vs enhancement factor for the fluid side of the heat exchangers. G.E. and S.S.P. consider that the flammability, toxic, corrosive, etc. concerns with ammonia vs R-22 will ultimately result in more expensive auxiliary systems and turbines for an ammonia plant. This figure does not reflect this difference, but rather assumes that all systems and equipment will have the same base unit costs. As can be seen, enhancing surfaces for an ammonia system results in less than a 12% reduction in plant costs, whereas for the R-22 system a 70% reduction could be achieved. There is less than a 10% difference between the minimum cost points.

Figures 6 and 7 show additional sensitivity results.

Off-Design Performance

The off-design subroutine uses the system design specified by the optimization program and applies monthly temperature difference data. A typical result is shown in Fig. 5.

Conclusions

The following general conclusions can be drawn based on the data presented.

1) Refrigerant 22 should continue to be considered as a viable alternate to ammonia as an OTEC working fluid.

2) An OTEC power system design which uses enhanced fluid-side surfaces on the heat exchangers and ammonia as the working fluid could result in only a 12% reduction in plant cost when compared with the unenhanced case, whereas a 70% reduction in plant costs could result for R-22.

3) The performance results indicate that the optimized R-22 power system will cost 10% more than the optimized ammonia power system, but this small differential may disappear when a detailed design is done which takes into account the health, safety, corrosion and environmental concerns of using ammonia.

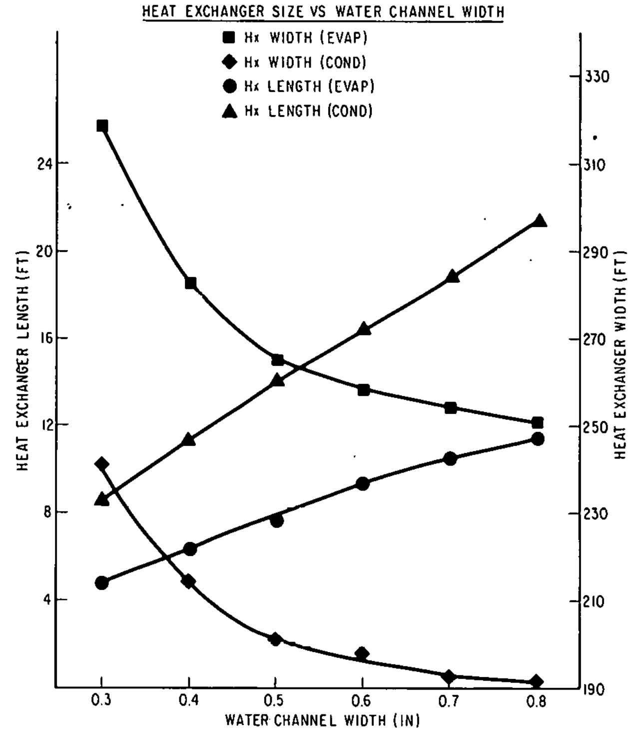

Fig. 7 Relationship between water channel width and heat exchanger length and width.

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3. Dugger, G. L., editor, Proceedings, Third Workshop on Ocean Thermal Energy Conversion (OTEC), Houston, Texas, May 8-10, 1975, APL/JHU 75-2, August 1975:

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   b) Trimble, L. C., Messinger, B. L., and Ulbrich, H. G., "Ocean Thermal Energy Conversion System Study Report," pp. 3-21.

   c) Dugger, G. L., Olsen, H. L., Shippen, W. B., Francis, E. J., and Avery, W. H., "Tropical Ocean Thermal Power Plants Producing Ammonia or Other Products," pp. 106-115.

DISCUSSION

Paper 4A-5, Olmsted

M. Olmsted: The enhancement factor is the ratio of the coefficient that we expect divided by the smooth plate coefficient. In some of the testing we have measured factors of 6 to 7 over a plate for freon. I think Union Carbide's surface yielded a factor of 3 or 4 for ammonia in some of the ANL testing.

Question: In one figure you showed the effect of enhancement factor on relative costs. How did you arrive at the values of cost per enhancement factor or ratio per enhancement factor being 0.67 for ammonia and 0.25 for freon? What are the differences that cause that?

M. Olmsted: That was based on a bid that we received from a commercial applier of enhancements, who said that the cost would be a certain number of dollars, and we would get a certain performance, and the same costs would apply whether it was an ammonia or a freon unit. I agree that this may not be the best way to use that parameter, but it was the best we had to use at the time.

Question: Do you assume that the waterside coefficient is constant while enhancing the ammonia?

M. Olmsted: The water-side coefficient varies with the velocity, which is the free floating variable in the program. As the enhancement goes up (and the coefficients go up), the optimum velocity on the water side tends to go up.

Question: So you are trying to optimize both sides, not just the working fluid side?

M. Olmsted: Yes.

R. Lyon, ORNL: In your comparison of the environmental effects of the freon as opposed to the ammonia, did you take into account the possible effects on the ozone layer of the freon?

M. Olmsted: The freon that we selected, R-22, has a relatively small effect on the ozone layer compared to the freons that are going to be taken off the market, such as R-12 or R-114. We can not put in a cost for the latter in our comparison, because they are going to be taken off the market because of the ozone layer.

R. Lyon: You showed a large environmental impact factor for ammonia. I assume that effect is because it is released into the ocean. Is that correct? What is the actual effect?

M. Olmsted: I'm not sure. Mike Mann...

M. Mann: Basically, ammonia is toxic in the acute or the short-term type of release. When you look at the long term, ammonia will add a nutrient factor to the ocean, and there will not be much degradation of the local area. But we are looking mostly at the short term, catastrophic-release effects.

OCEAN THERMAL ENERGY CONVERSION PLANT WITH FREON 22

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and
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Abstract

~~In this paper is treated~~ the performance of Ocean Thermal Energy Conversion plants using Freon 22 as the working fluid is studied. The new plate type heat exchanger is used as the evaporator and the condenser in this plant. The overall heat transfer coefficient of this plate type evaporator is about 3000 kcal/m^{2}h°C for Freon 22 for the case that the velocity of warm water is 1.2 m/s, and the inlet temperature of warm water is 28°C. The overall heat transfer coefficient of this plate type condenser is about 1500 kcal/m^{2}h°C with Freon 22 for the case that the velocity of warm water is 1.0 m/s and the inlet temperature of cold water is 8.5°C. (a)

On the basis of these results, the design for a 25 MW OTEC plant module with Freon 22 is conducted and is compared with that of Ammonia.

Nomenclature

U	Overall heat transfer coefficient	(kcal/m^{2}h°C)
G	Mass flow rate	(t/h)
C_p	Specific heat	(kcal/kg°C)
T	Temperature	(°C)
Δtm	Logarithmic mean temperature difference	(°C)
A	Heat transfer area	(m^2)
P	Pressure	(ata)
λ	Heat conductively	(kcal/mh°C)
ρ	Density	(kg/m^3)
μ	Viscosity	(kg/ms)
ν	Kinematic viscocity	(m^2/s)
σ	Surface tension	(N/m)
L	Latent heat	(kcal/kg)

Subscript

I	Inlet
O	Outlet
H	Warm Water
C	Cold Water
L	Liquid Water
V	Vapor

* Associate Professor
** Professor
*** Associate Professor
**** Research Assistant
† Graduate Student

Introduction

It is important to develop the high efficient heat exchanger and choose the working fluid well suited for OTEC plant in order to operate OTEC plant in practice.

Working fluids for OTEC plant have already been proposed. Zener[1], Lavi[2], Dauglous[3], Trimble et al[4], Dugger et al[5], and Japanese OTEC committee[6] recommend Ammonia as the most suitable working fluid. Anderson[7] recommendes Freon 21/31. Goss et al[8] recommend Propane. It is presumably thought that Ammonia is superior to other working fluids in heat transfer characteristics as well as thermo-dynamic properties and that dimensions of heat exchanger and turbine for Ammonia cycle is smaller than those of other fluids. But Ammonia has the following weak points;

[1] attack copper
[2] explosive
[3] poisonous

Considering that Freon 22 has not the above-mentioned weak points, we choose Freon 22 as one of working fluids suitable for OTEC plant.

The various heat exchangers for OTEC have been proposed and tests of these exchangers are run under the conditions of small temperature differences. Heat transfer coefficients of the double fluted tubes recommended by Rothfus and Neuman[9] and Rothfus and Lavi[10] are measured for each of Freon 11 and Ammonia. One of authors[11] measures condensation heat transfer coefficients on vertical fluted tubes for Freon 11 and 113. Combs and Murphy[12] measure condensation heat transfer coefficients on vertical fluted tubes for Ammonia.

Lorenz et al[13], Owens[14], Conti[15], Czikk et al[16] and Sabin and Poppendick[17] experiment on horizontal thin film evaporators. Sather et al[18] and snyder[19] test on 1 MW_t OTEC heat exchanger with Ammonia. Kajikawa et al[20] and authors[21] test on 100 KW_t heat exchanger with Freon 114. Authors[22] propose the concept of the plate type condenser and evaporator for OTEC plant and report the experimental results of the heat transfer coefficients on the small style of heat exchangers.

In this paper, we report the results of the performance of plate type heat exchanger for $100KW_t$ OTEC plant and of the conceptual design for 25MW Module using Freon 22 as working fluid.

This work is sponsored by Sunshine Project in AIST, MITI and GRANT in Aid for Scientific Research (B 246081) in MESC.

Experimental Facility

Layout of System

The loops of circulating Freon 22 used as the working fluid, warm water and cold water are shown schematically in Fig. 1. Figure 2 shows a picture of the Shiranui V.

The vapor is generated in the plate type evaporator and then the vapor flowing out of a main condenser and an auxiliary condenser is condensed there and is returned to a Freon tank. The liquid in the Freon tank is pumped to evaporator by a Freon pump. A boiler is used to keep warm water at a constant temperature (25 to 35 °C, ± 0.1 °C). Cold water is kept at a constant temperature (5 to 10 °C ± 0.1 °C) by means of a refregirator. The auxiliary condenser made of stainless steel is vertical shell and tube type.

The respective temperature of vapor, condensate, inlet and outlet is measured by the thermocouples connected with a digital voltmeter in order to measure the voltage difference of them.

Flow rates of cold and warm water are measured by the rotameter. Flow rates of condensate and Freon fed by the Freon pump are measured by the magnetic flow meters.

The pressures at the mark Ⓟ in Fig. 1 are measured by the pressure gauge.

The temperatures at the mark $\boxed{T}$ in Fig.1 are measured by the thermocouples.

Evaporator

Shell and plate type evaporator used in this plant is made by Hisaka Work Co. in Japan. Figure 3 shows a picture of a surface of the plate type evaporator. The surface made of Titanium is coated with Aluminium powders to increase heat transfer rate.

The dimensions of the plate of this evaporator are 1255 mm in length, 415 mm in width and 0.8 mm in thickness.

Figure 4 shows a sketch of shell and plate type evaporator.

Figure 5 shows flow passages of warm water and Freon 22.

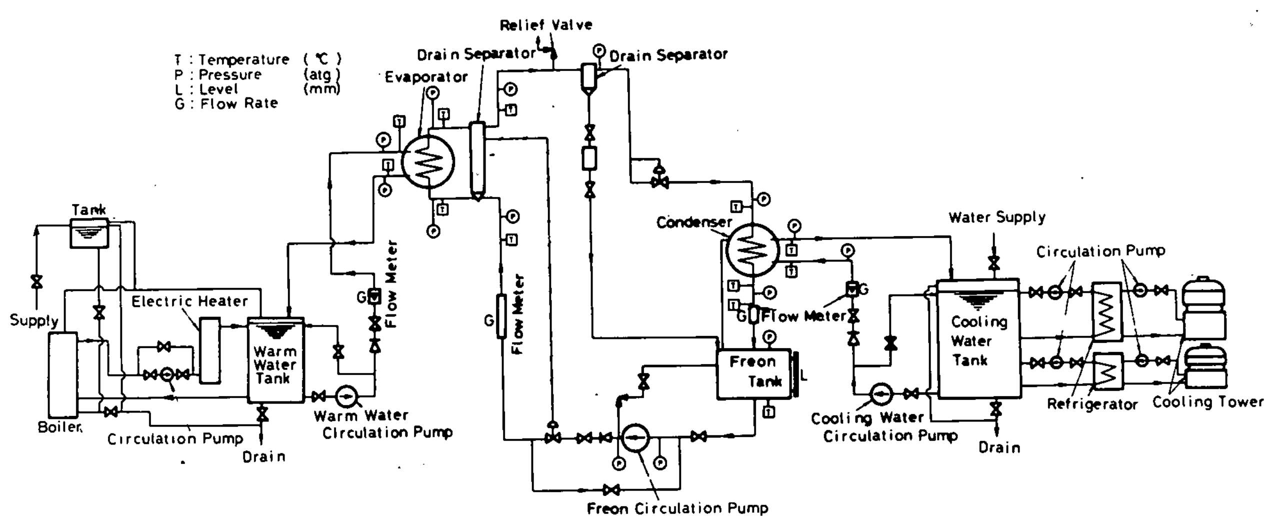

Fig. 1 Layout of system.

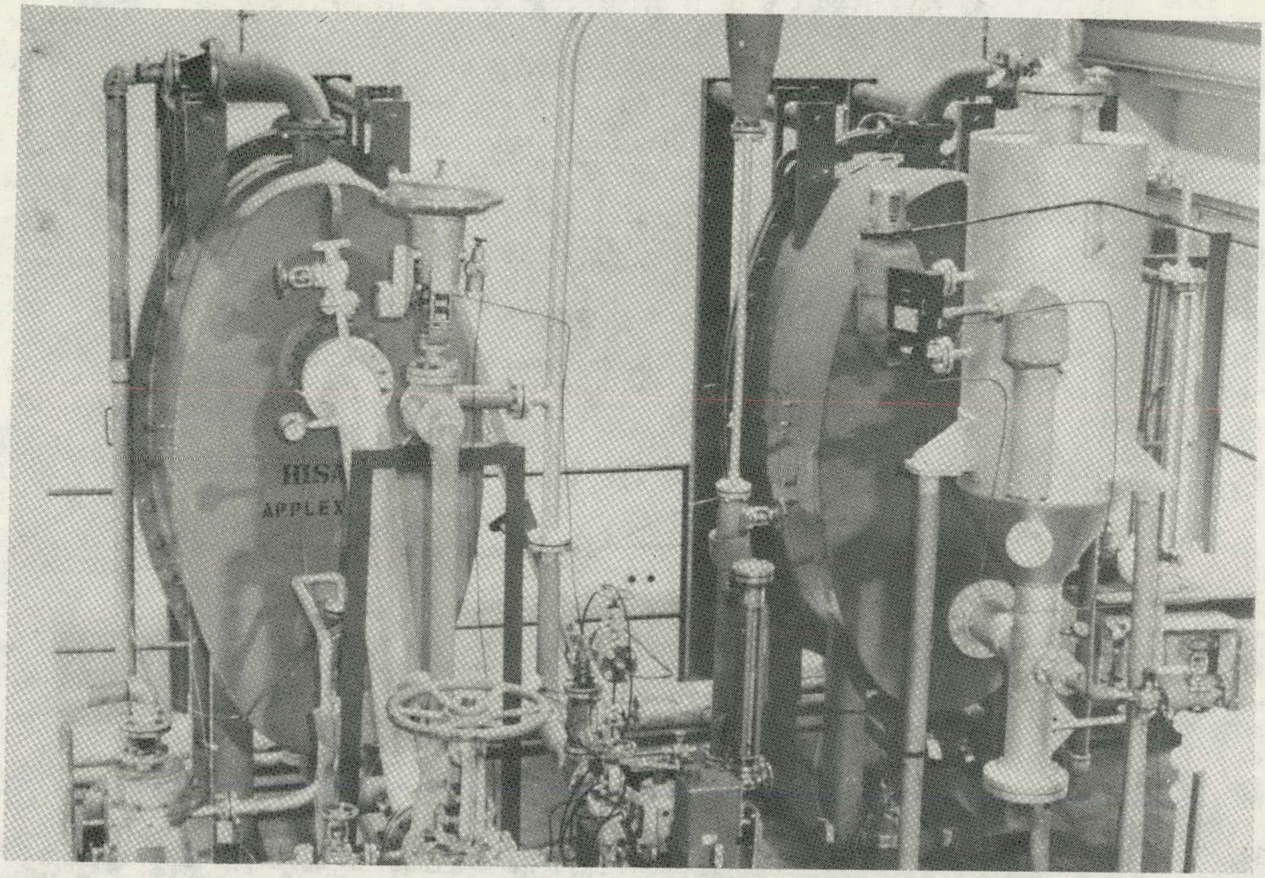

Fig. 2 Picture of Shiranui V.

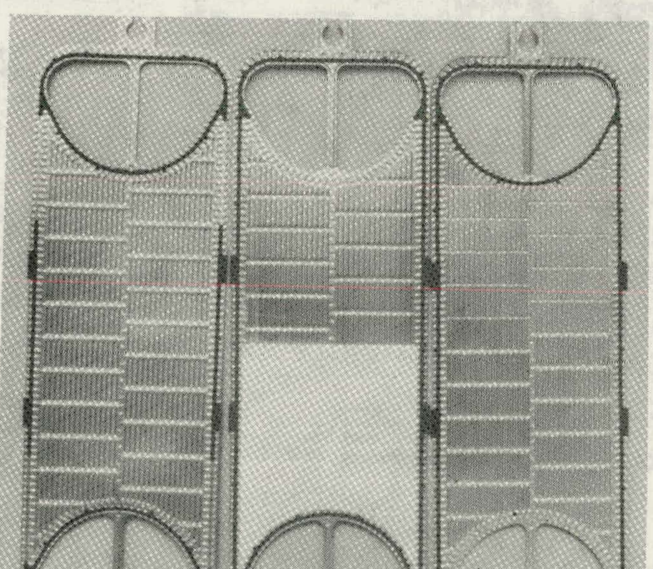

Fig. 3 Picture of heat transfer plates for evaporator.

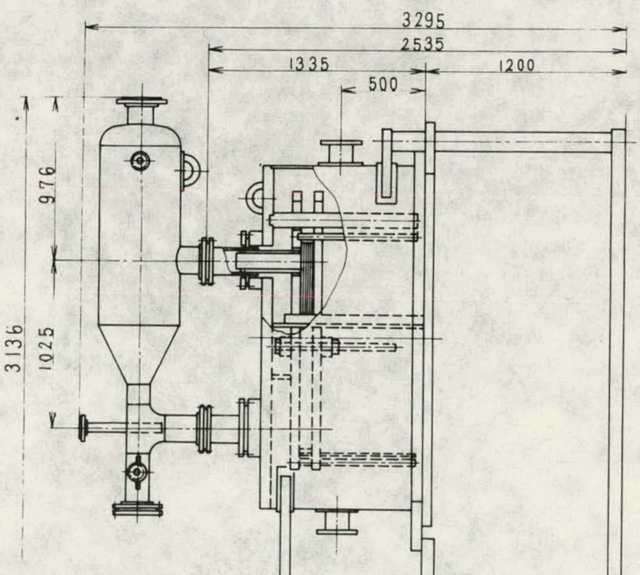

Fig. 4 Sketch of evaporator.

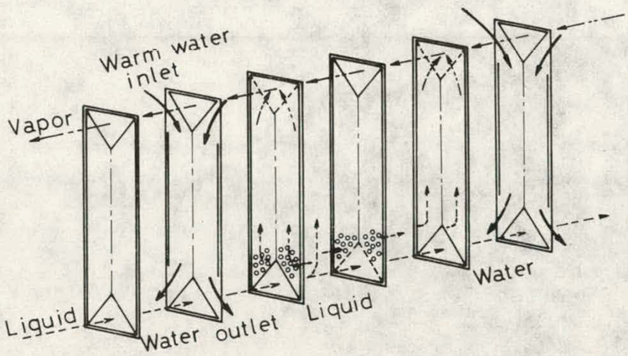

Fig. 5 Flow passage of warm water and Freon 22.

Condenser

Shell and plate type condenser used in this plant is made by Hisaka Work Co. Ltd. in Japan. Figure 6 shows a picture of plates. The flute as well as draninage on the surface of plate enhances the condensation heat transfer coefficient.

The dimensions of the plate of this condenser are 1255 mm in length, 415 mm in width and 0.8 mm in thickness.

Figure 7 shows a sketch of shell and tube condenser.

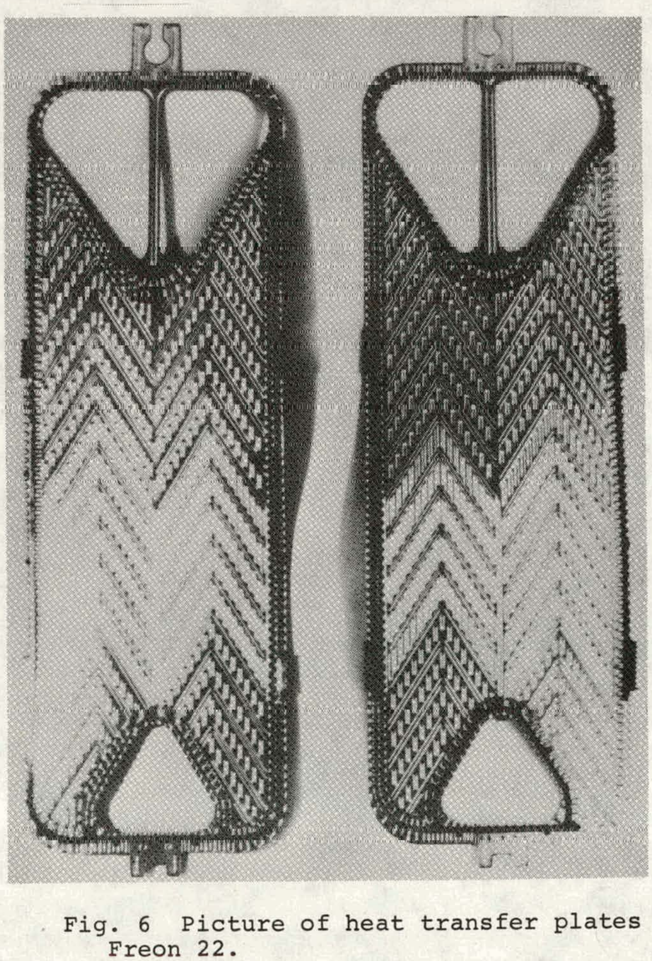

Fig. 6 Picture of heat transfer plates Freon 22.

Experimental Result

Experiments are run for the following conditions;
inlet temperature of cold water 6 to 9 °C
inlet temperature of warm water 25 to 30 °C
velocity of cold water 0.5 to 1.3 m/s
velocity of warm water 0.5 to 1.3 m/s.

Evaporator

The overall heat transfer coefficients of evaporator are defined as follows;

$$U_H = \frac{G_H C_{PH}(T_{HI} - T_{HO})}{A_H} \frac{1}{\Delta t_m}$$

$$\Delta t_m = \frac{(T_{HI} - T_{VO}) - (T_{HO} - T_{LI})}{\ln \frac{(T_{HI} - T_{VO})}{(T_{HO} - T_{LI})}}$$

Table 1 shows warm water flow rate, warm water inlet temperature, liquid inlet temperature, log-mean temperature difference, warm water velocity, heat transfer rate, and overall heat transfer coefficient. For the case that the velocity of warm water is 1.2 m/s and the inlet temperature of warm water is 28°C (82.4°F) the overall heat transfer coefficient is about 3000 kcal/m^{2}h°C (600 Btu/ft^{2}h°F).

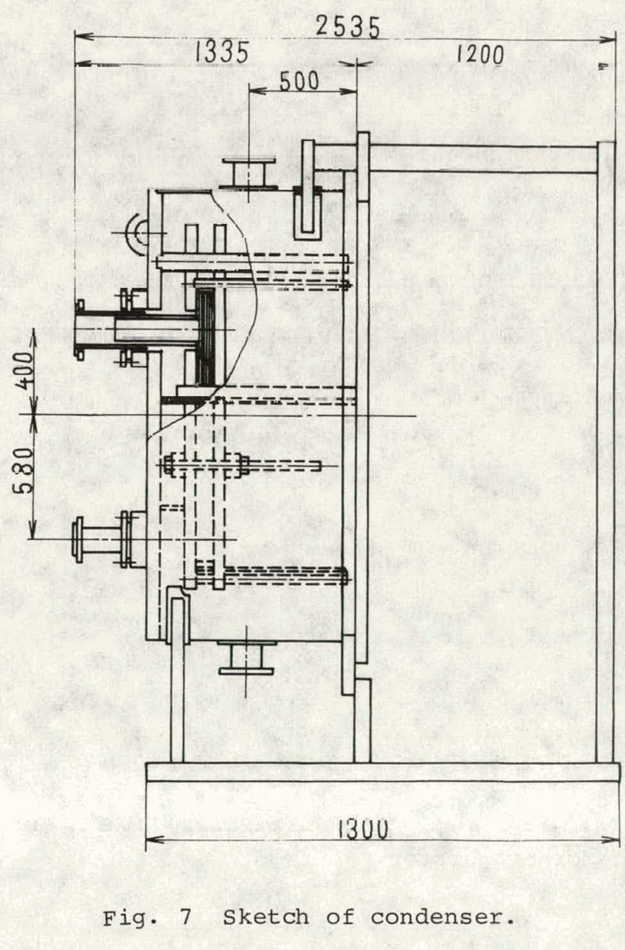

Fig. 7 Sketch of condenser.

Condenser

The overall heat transfer coefficients of condenser are defined as follows;

$$U_C = \frac{G_C C_{PC}}{A_C} \frac{(T_{CO}-T_{CI})}{\Delta tm}$$

$$\Delta tm = \frac{(T_{VI}-T_{CO}) - (T_{LO}-T_{CI})}{\ln\left(\frac{T_{VI}-T_{CO}}{T_{LO}-T_{CI}}\right)} = \frac{\Delta 1-\Delta 2}{\ln\left(\frac{\Delta 1}{\Delta 2}\right)}$$

Table 2 shows cold water flow rate, cold water inlet temperature, vapor inlet temperature, log-mean temperature difference, cold water velocity, heat transfer rate, overall heat transfer coefficient.

For the case that the velocity of cold water is 1.0 m/s and the inlet temperature of cold water is 8.5°C, overall heat transfer coefficient is 1500 kcal/m²h°C (300 Btu /ft²h°F).

Conceptual Design of 25 MW OTEC Plant

Figure 8 shows the Rankine efficiency of OTEC plant using Freon 22 as the working fluid and also shows the Rankine efficiency using Ammonia in order to compare with that of Freon 22. As shown in Fig. 8 Rankine efficiency of Freon 22 cycle is almost the same as that of Ammonia cycle. The cycle efficiency is about 3.5 persent for the case electric generator efficiency is 96 persent, the inlet and the outlet vapor temperature of turbine are 22.5 °C and 12.2 °C, respectively.

Characteristics of Freon 22

Table 3 shows characteristics of the two working fluids of Freon 22 and Ammonia. It is clear that Ammonia is superior to Freon 22 in heat transfer as well as thermo-dynamic properties. But Freon 22 is selected as the working fluid in this plant because wet vapor of Ammonia is not only incompatible with copper but also is apt to explode.

Table 1 Performance of Evaporator

Warm Water Flow Rate (t/h)	Warm Water Inlet Temperature (°C)	Liquid Inlet Temperature (°C)	Log-Mean Temperature Difference (°C)	Warm Water Velocity (m/s)	Heat Transfer Rate 10^4(kcal/h)	Overall Heat Transfer Coefficient (kcal/m²h°C)
43.0	26.89	22.21	3.28	1.19	8.90	3326
43.0	28.00	23.51	3.16	1.19	8.99	3485
43.0	27.90	23.44	3.23	1.19	8.94	3393
43.0	27.76	22.80	3.55	1.19	8.69	2998
43.0	27.76	22.29	3.84	1.19	8.56	2731
43.0	27.81	22.60	3.69	1.19	8.69	2885
43.0	28.00	22.78	3.69	1.19	9.56	2842
43.0	28.03	23.49	3.31	1.19	8.47	3136

Table 2 Performance of Condenser

Cold Water Flow Rate (t/h)	Cold Water Inlet Temperature (°C)	Vapor Inlet Temperature (°C)	Log-Mean Temperature Difference (°C)	Cold Water Velocity (m/s)	Heat Transfer Rate 10^4(kcal/h)	Overall Heat Transfer Coefficient (kcal/m²h°C)
45.0	7.51	16.89	6.57	1.43	9.81	1330
45.0	8.11	16.48	6.11	1.43	10.80	1578
45.0	8.36	16.89	6.31	1.43	10.53	1489
39.5	8.71	16.48	5.61	1.25	10.30	1639
39.5	8.73	16.48	5.57	1.25	10.43	1671
39.5	8.74	16.89	5.77	1.25	10.46	1618
30.0	8.42	17.30	6.29	0.95	10.32	1464
30.0	8.40	17.30	6.32	0.95	10.23	1445

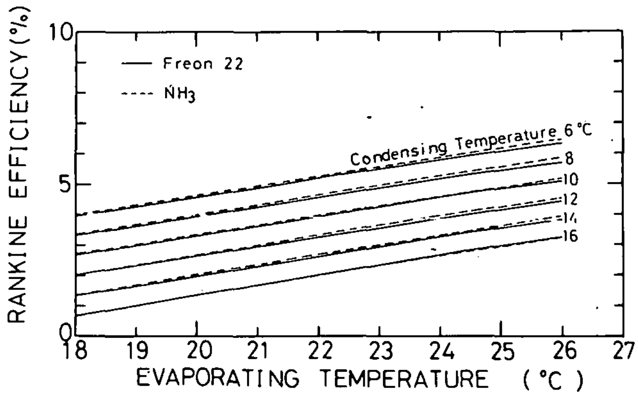

Fig. 8 Rankine efficiency.

Table 3 Comparison of Thermo-dynamic Properties

Item	Unit	Ammonia	Freon 22
T	°C	15	15
P_v	ata	$0.743x10^1$	0.805x10
λ_L	kcal/mh°C	0.453	0.079
ρ_L	kg/m^3	$0.618x10^3$	$0.123x10^4$
μ_L	kg/ms	$0.225x10^{-3}$	$0.195x10^{-3}$
ν_L	m^2/s	$0.364x10^{-6}$	$0.159x10^{-6}$
C_{PL}	kcal/kg°C	$0.114x10^1$	0.291
ρ_V	kg/m^3	$0.570x10^1$	$0.33x10^2$
μ_V	kg/ms	$0.976x10^{-5}$	$0.109x10^{-4}$
ν_V	m^2/s	$0.171x10^{-5}$	$0.331x10^{-6}$
σ	N/m	$0.227x10^{-1}$	$0.91x10^{-2}$
L	kcal/kg	$0.288x10^3$	$0.459x10^2$
R	-	$0.449x10^2$	$0.258x10^2$
P_{rL}	-	$0.203x10^1$	0.259x10

Dimensions of Turbine

Table 4 shows dimensions of Ammonia turbine and turbine of Freon 22 for OTEC plant.[5] Turbine of Freon 22 is larger than Ammonia turbine.

Table 4 Comparison of Dimensions of Turbine

Item	Unit	Freon22	Ammonia
Flow Rate	kg/h	$1.25X10^7$	$2.00X10^6$
Blade Height with Double Flow	m	370	140
P. C. D.	mm	750	1850
Rotational Speed	rpm	1800	1800
Inlet Temperature	°C	24	24
Outlet Temperature	°C	9	9
Output Power	MW	25	25

Charactristics of Plate Type Heat Exchanger for 25 MW Output Module

Table 5 shows an estimation of the characteristics of the plate-type evaporator for the 25 MW module using Freon 22. The table also shows an estimation of the characteristics of Ammonia to compare with Freon 22. As for the material of the heat exchanger plate, Aluminum-brass is used for Freon 22, and Titanium is used for Ammonia.

Table 5 Comparison of Specifications of Evaporator for 25MW Output Module

Item	Unit	Freon 22	Ammonia
Warm Water Temperature at Inlet	°C	28	28
Warm Water Temperature at Outlet	°C	24.62	24.5
Vapor Temperature at Outlet	°C	22.50	22.9
Material of Surface Plate		Aluminum Brass	Titanium
Logarithmic Mean Temperature Difference	°C	3.56	3.02
Liquid Flow Rate	kg/h	$1.39x10^7$	$1.39x10^6$
Heat Transfer Rate	kcal/h	$6.58x10^8$	$4.1x10^8$
Overall Heat Transfer Coefficient	kcal/m^{2}h°C	2500	3475
Heat Taansfer Area	m^2	$7.39x10^4$	$3.9x10^4$
Warm Water Flow Rate	kg/h	$1.95x10^8$	$1.21x10^8$
Warm Water Velocity	m/s	1.1	0.79
Water Head Loss	mAq	5.2	2.8

Table 6 Comparison of Specifications of Condenser for 25 MW Output Module

Item	Unit	Freon 22	Ammonia
Cold Water Temperature at Inlet	°C	7.0	7.0
Cold Water Temperature at Outlet	°C	11.28	11.1
Vapor Temperature at Inlet	°C	12.20	12.5
Material of Surface Plate		Aluminum Brass	Titanium
Logarithmic Mean Temperature Difference	°C	2.47	3.0
Vapor Flow Rate	kg/h	1.39×10^7	1.39×10^6
Heat Transfer Rate	kcal/h	6.36×10^8	3.96×10^8
Overall Heat Transfer Coefficient	kcal/m^{2}h°C	1480	3015
Heat Transfer Area	m^2	1.74×10^5	4.39×10^4
Cold Water Flow Rate	kg/h	1.50×10^8	1.01×10^8
Cold Water Velocity	m/s	0.6	0.87
Water Head Loss	mAq	3.36	4.1

The surface area of Freon 22 evaporator is two times as large as that of Ammonia evaporator. Table 6 shows the estimation of the characteristics of condenser for the same module. The surface area of Freon 22 condenser is four times as large as that of Ammonia condenser. Table 7 shows the total cost of the plate used in the evaporator. Table 8 shows the total cost of the plate used in condenser. It is found that the total cost of plate adopted Freon 22 cycle is lower than that for Ammonia cycle. But now we can not estimate the total cost of Freon 22 OTEC system.

Conclusions

1. Tests are run to grasp the characteristics of 100 KWt OTEC plant using Freon 22 as the working fluid.

2. The new plate type heat exchanger is used as the evaporator and condenser in this plant.

3. The overall heat transfer coefficient of this plate type evaporator is about 3000 kcal/m^{2}h°C for Freon 22 for the case that the velocity of warm water is 1.2 m/s, and the inlet temperature of warm water is 28°C.

4. The overall heat transfer coefficient of this plate type condenser is about 1500 kcal/m^{2}h°C for Freon 22 ,for the case that the velocity of warm water is 1.0 m/s and the inlet temperature of cold water is 8.5°C.

5. On the basis of these results, the design for 25 MW OTEC plant module with Freon 22 is conducted and is compared with that of Ammonia.

Table 7 Comparison of Material Cost of Plate of Evaporator for 25MW Output Module

Item	Unit	Freon22	Ammonia
Surface Area	m^2	7.4×10^4	3.9×10^4
Material		Aluminum blass	Ti
Density	kg/m^3	8410	4540
Thickness	mm	0.8	0.8
Material Weight (Plate Only)	kg	4.98×10^5	1.4×10^5
Unit Price	yen/kg	650	4000
Total Cost (Plate Only)	yen	3.24×10^8	5.7×10^8

Table 8 Comparison of Material Cost of Condenser for 25MW Output Module

Item	Unit	Freon22	Ammonia
Surface Area	m^2	12.8×10^4	4.4×10^4
Material		Aluminum brass	Ti
Density	kg/m^3	8410	4540
Thickness	mm	0.8	0.8
Material Weight (Plate Only)	kg	8.61×10^5	1.54×10^5
Unit Price (of Material)	yen/kg	650	4000
Total Cost (Plate Only)	yen	5.6×10^8	6.2×10^8

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DISCUSSION

D. Lyon, ORNL: In the comparison of your costs of the heat exchanger, you chose 0.8 mm for the thickness in both cases. Are the mechanical properties different enough between titanium and brass so that perhaps the optimum thicknesses would have been different?

H. Uehara: The optimum thickenesses of plates for titanium and aluminum will be different, but we have not performed that calculation yet.

M. Leitner, Lockheed: Your last figure also showed that the cost for the aluminum-brass heat exchanger was considerably less than that of titanium, but did you consider that the titanium heat exchanger will last for 30 years? How long did you consider that the aluminum-brass would last, from a corrosion or erosion standpoint?

H. Uehara: About 16 years.

PERFORMANCE OPTIMIZATION OF AN OTEC TURBINE

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Abstract

The need for an ammonia turbine with maximized efficiency over the expected range of operation was determined during OTEC 10 MWe net power system optimization studies. It was determined that for every extra KW produced it was worth expending $1,000 in plant cost. Specific application of state-of-the-art hardware designs were used to assure that the efficiencies determined were realizable. The turbine aerodynamic design considered axial flow and radial inflow turbines, single and double flow designs, and variations in the number of stages and base diameters. Consideration was given to the turbine-generator control scheme and specific areas of the turbine mechanical design such as the blades, seals and bearings. The optimum design is a four-stage axial flow double flow turbine directly connected to a four pole 60 hertz synchronous generator. The efficiency of the double-flow, four stage design is more than two percent higher than a one-stage, single flow design with diffuser. This is due to the latter's inherent very high exit velocity. Additional optimally designed stages can more efficiently recover this energy than a diffuser. Variable nozzles for the first stage are utilized for power maximization at off-nominal conditions.

Introduction

The OTEC ammonia turbine generator is one of the critical elements of the OTEC power system. Its energy conversion efficiency and the efficacy of design, considering the unique use of ammonia as a working fluid will have a major effect on the overall system performance. Also, a complete ammonia turbine, including the mechanical aspects, has never been designed for OTEC service. To arrive at a manufacturable aerodynamic design requires specific hardware design studies to examine system efficiencies and the feasibility of implementation. The key question to be addressed is

- What efficiency can be obtained utilizing realistic blade geometries and other mechanical design features?

This paper reports on the results of a study to examine the feasibility of developing an OTEC ammonia turbine-generator that will operate efficiently, economically and safely, using state-of-the-art hardware designs and materials. Particular emphasis is on the optimization of the turbine aerodynamic efficiency and off-nominal operating conditions. A summary of other results are given, including:

- Control design
- Key realistic mechanical design features
- Applicability of scaling over the potential OTEC turbine generator power range.

The study was focused on specific solutions to identified nominal conditions. The turbine generator requirements used are shown in Table 1; Table 2 gives the design point criteria. A nominal 14.0 MWe gross power output from a synchronous 60 Hz electric generator was chosen for the study. The nominal, minimum, and maximum ammonia operating conditions (flow, temperature, pressure, and quality) were specified for an OTEC power system corresponding to this power level.

Table 1. Requirements

Output, 14.0 MWe* (gross) at 60 Hz (synchronous speed)
Nominal input, 807.2 lb/sec saturated ammonia vapor at 70°F, 50°F outlet state
Turbine flange to flange efficiency greater than 85 percent
Generator efficiency greater than 97 percent
Provision for power peaking control during input variation
Use of existing technology
30-year life
Base load duty cycle
*Based upon the assumed efficiencies of 85 percent for the turbine and 97 percent for the generator and parasitic loads.

Table 2. Design Point Criteria

	Minimum Flow (Winter)	Nominal Flow	Maximum Flow (Summer)	Variation (Approx.) (±)
Inlet Conditions (Total)				
Ammonia mass flow, lb/sec	727	807.2	888	10
Ammonia temperature, °F	67.0	70.0	73.0	5
Ammonia pressure, psia	122.1	128.7	135.7	6
Ammonia quality, percent*	>99.0	>99.0	>99.0	
Ammonia volume flow, acfs**	1,801	1,865	1,951	5
Outlet Conditions (Total)				
Ammonia temperature, °F	49.0	50.0	51.0	2
Ammonia pressure, psia	87.5	89.2	90.9	2
Gross Generator Power				
MWe***	11.4	14.03	16.8	20

*Saturated conditions
**Estimated assuming 100 percent vapor
***Estimated assuming constant efficiency with variations

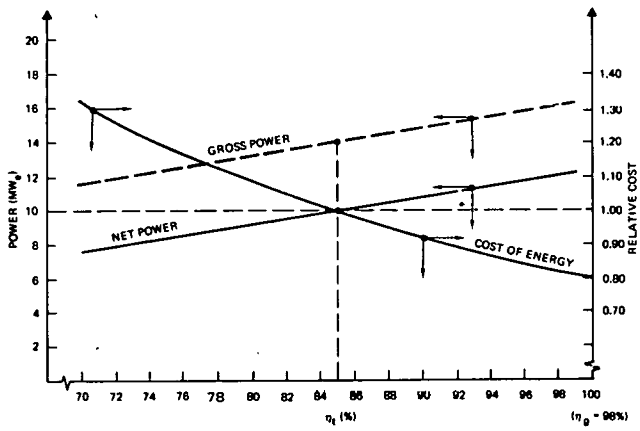
Plant/Turbine Optimization

The requirements for the OTEC turbine were developed during a preliminary design study which had as its objective the overall optimization of the OTEC plant. The early system design assumed a turbine efficiency of 85 percent at design conditions. Since gross output of a given plant is directly proportional to turbine efficiency, simple calculations are sufficient to determine the sensitivity of net power and cost of delivered energy to actually achieved turbine efficiency. Figure 1 shows the inherent assumptions and results of the calculations. A 3 point change in turbine efficiency from 85 to 88%

*Assistant Project Manager, OTEC-1
†Senior Development Engineer

results in a 3.6 percent increase in gross power, and a five percent increase in net power developed.

Figure 1. Sensitivity of Net Power and Energy Cost to Turbine Efficiency



A cost analysis, using an OTEC computer model of a 10MWe net plant to optimize designs for three different turbine efficiencies (80, 85 and 90 percent), provides additional insight into the importance of turbine efficiency. The model was run with a fixed best exchanger design, optimized with respect to number of tubes and seawater flow rates. The results are indicated in Table 3. Using a turbine efficiency of 85 percent as a base, it is demonstrated that each efficiency point change in turbine performance is worth approximately 17 dollars per kilowatt of net plant output. Therefore, it is cost effective to pay up to $170,000 for a 10MWe net plant to increase the turbine efficiency one percentage point. One KW increase in gross turbine output is worth approximately $1,000 in additional capital expenditure;

$$\frac{\$170{,}000/\% \times 3\%}{5\% \times 10{,}000 \text{ KW}} \cong \$1{,}000/\text{KW}$$

Table 3. Cost Versus Turbine Efficiency

Turbine Eff., Percent	Net Power, MWe	Cost Model Prototype, $/kW	Cost Sens. $/kW
90	10	1362	-81
85	10	1443	0
80	10	1536	+ 93

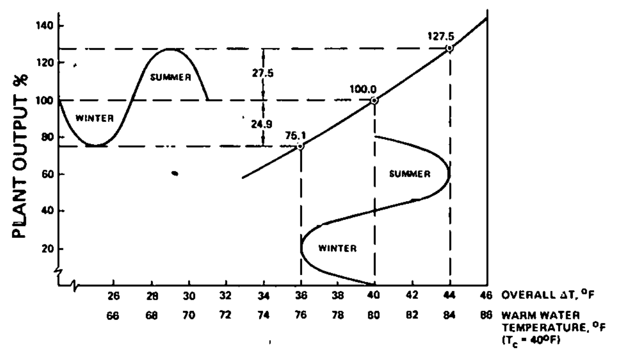
$$\text{Slope at } 85\% = \frac{93 - (-81)}{10} = 17.4 \text{ \$/kW/\%}$$

Off design behavior is also an important consideration. Because the plant and turbine-generator are optimized at the nominal conditions, fixed flow turbine nozzles would not be able to pass all of the flow for the off-design "summer" conditions (see Figure 2). This is due to the unique characteristic of OTEC whereby the higher warm seawater temperature causes an increase in unit energy level and also generates more ammonia vapor mass flow. Similarly, "winter" operation, at the minimum seawater temperature, causes the mass flow rate and evaporator pressure to drop below the level where fixed flow turbine nozzles can maintain design efficiency levels. Therefore, consideration of variable turbine inlet geometry, which would accommodate the changes in mass flow rate and available energy, is most desirable.

There are three options available for reducing the power output of the plant; turbine throttle (or variable inlet nozzles), turbine bypass, and evaporator recirculation rate control.

A turbine with variable inlet nozzles can be used both to modulate and optimize power and would have a high frequency response. Therefore, a plant control mechanization is suggested in which large changes in power could be accomplished with an evaporator recirculation flow control and small perturbations controlled by the turbine inlet nozzles.

Figure 2. Plant Output Variation With Available ΔT



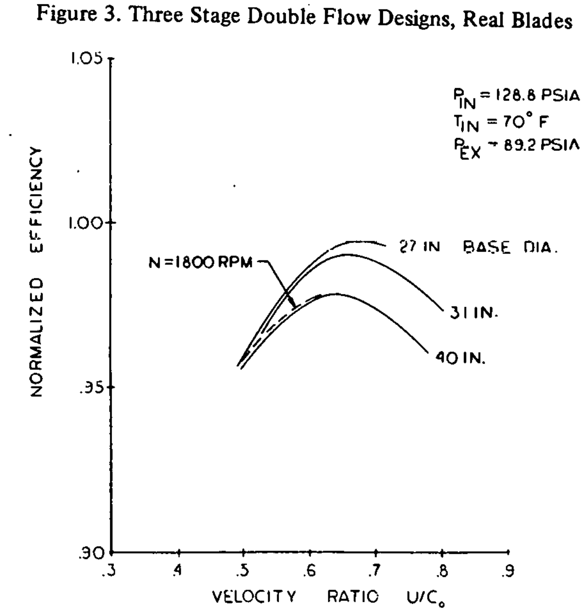
The operation of an OTEC plant at reduced power for a given set of pressure-temperature conditions does not result in a saving of consumable resources in contrast to conventional power plants. For this reason, the interest in part-power operation is primarily that of control and how it may be obtained; efficiency at reduced output is not important.

Turbine Aerodynamic Design

The primary consideration in the aerodynamic design of the turbine is to achieve an optimum performance level with manufacturable state-of-the-art technology. As indicated above, the turbine efficiency is of vital importance in the overall cycle performance, consequently extreme care must be taken to obtain the optimum performance, consistent with mechanical design limitations. The elimination of a speed decreasing/increasing gear with about 2 percent losses would appear to have definite advantages; and indeed this is true as will be seen later.

Figure 3 shows turbine designs with three stages of expansion using real blades. The three base diameters are 27 inches, 31 inches, and 40 inches. The latter base diameter maximizes the three stage design efficiency at 1800 RPM. All efficiency

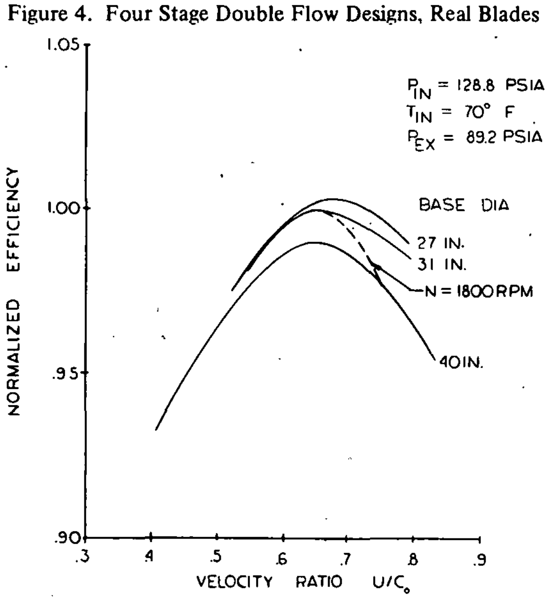
Figure 3. Three Stage Double Flow Designs, Real Blades



curves have been normalized to the efficiency of a double flow four stages axial turbine operating at 1800 RPM. Higher peak efficiencies can be obtained by going to higher shaft speeds and smaller base diameters. In this event, a gear would have to be used and a two percent gear loss would be incurred. Since the improvement is not two percent, the best performance can be obtained from the 40 inch base diameter design. This particular design has much lower efficiency compared to the four and five stage designs because of decreasing L/D ratios, where L is the blade length and D is the mean diameter of the blade annulus. This peak normalized efficiency is .978, which means this design will produce 2.2 percent less gross power than the four stage design. This works out to be approximately 330 kilowatts.* The 330 KW, valued at $330,000, can be obtained by adding a fourth stage at the cost of approximately $100,000. For this reason, the three stage design is not cost optimized.

Figure 4 shows four stage, double flow designs on three different base diameters. The smallest base diameter peaks at the highest efficiency, but at a speed other than 1800 RPM or 3600 RPM. The 31 inch base diameter design is at the peak efficiency when operating at 1800 RPM. No other synchronous speed is in the proper range to appear on these velocity ratio curves. Figure 5 presents similar information for five stage designs, with the 27 inch base diameter giving the highest efficiency when operating at 1800 RPM, although it exceeds the L/D limits. Figure 6 is a plot of normalized efficiency versus velocity ratio for all of the 31 inch base diameter double flow turbine designs.

Figure 4. Four Stage Double Flow Designs, Real Blades



A five stage, double flow design does offer better performance than a four stage double flow design, but caution must be exercised about the value of this small improvement. The 27 inch base diameter design peaks at nearly one percent better performance, at the same shaft speed as the four stage design. But from the L/D curve, Figure 7, we can see that this design has blades that violate L/D limits, making this design unfeasible. The 31 inch base diameter design is also better than the four stage design, but there are two problems. First, the maximum L/D ratio for the five stage design on a 31 inch base diameter is slightly higher than the accepted limit for this blade. This would necessitate a new, untested blade to be used. The second problem concerns practicality. The five stage design is about three-tenths of one percent better in efficiency, equivalent to about 50 KW, but the cost of adding a fifth stage to both ends of the double flow design would be approximately $100,000. This figure, compared to the $50,000 worth of additional power produced makes the five stage design cost inefficient. Also, using a four stage design having a shorter bearing span than the five stage design, helps avoid critical speed problems.

*Approximately 15 MWe is the nominal power output for specified conditions at optimized efficiency.

Figure 5. Five Stage Double Flow Designs, Real Blades

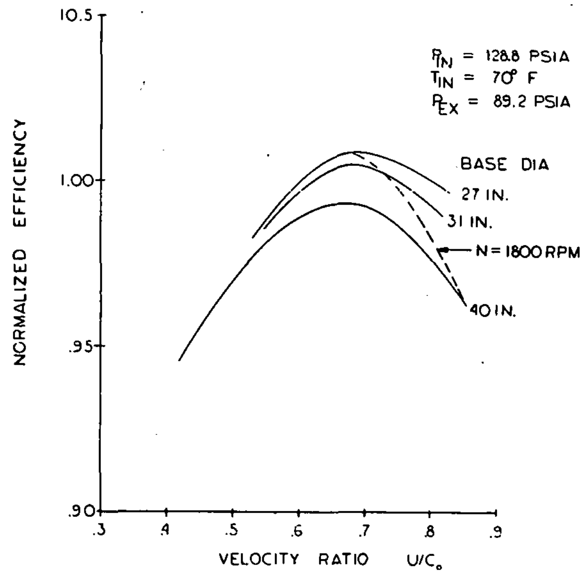
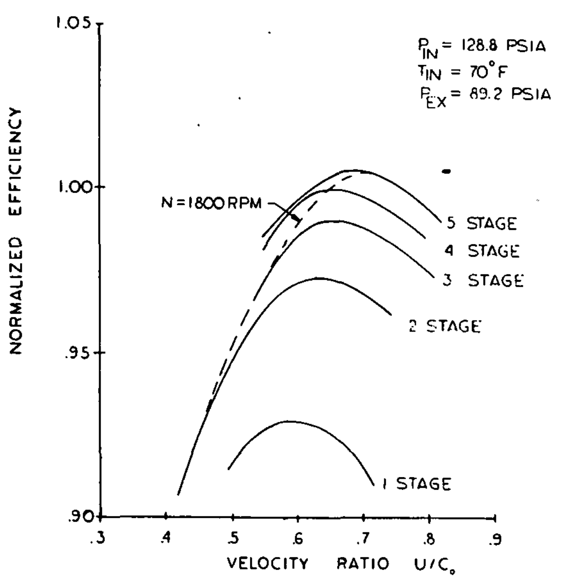


Figure 6. Double Flow Designs, 31 Inch Base Diameter, Real Blades



From the data presented in the foregoing figures, several conclusions can be reached. 1) Four stages are the minimum economic number of stages to provide peak efficiency. 2) A 31 inch base diameter provides the best efficiency for a four stage, double flow design when operating at 1800 RPM and when limited to the blade profile L/D limit shown in Figure 7. 3) A five stage design yields better efficiency than the four stage design, but when the constraints of this system are applied (L/D ratios, 1800 RPM speed, using a gear for non-synchronous operation) the gain is so small that the cost of adding the fifth stage would be higher than the benefit of the additional power.

So far we have only considered double flow designs since they eliminate any high thrust bearing losses. Bearings for single flow designs can be quite large since it is accepted practice not to load thrust bearings over 250 psi, and preferably to keep the loading below 200 psi.

Figure 7. Maximum L/D Vs. Base Diameter, Real Blades

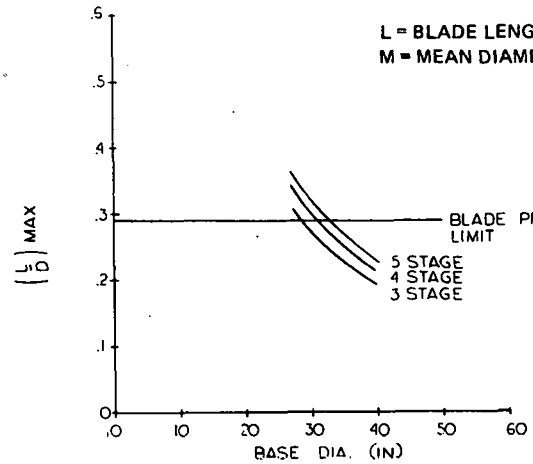
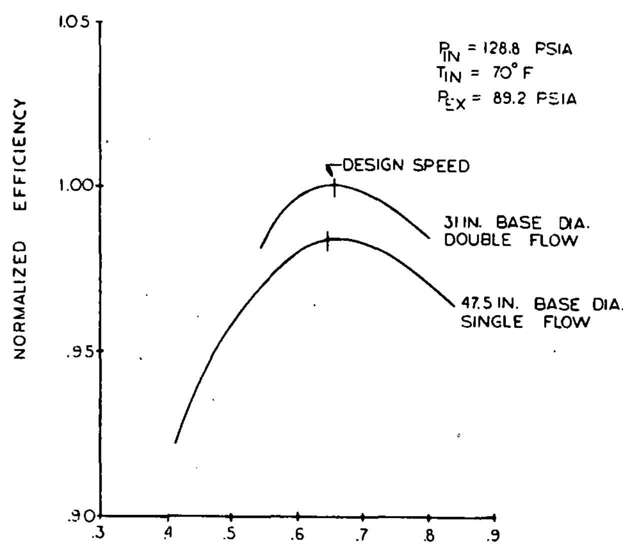


Figure 8 compares a double flow and a single flow design. The single flow design is under some constraints discussed below which limit its efficiency to 1.6 percent less than the double flow. The thrust bearing produces a large loss, but another loss is introduced by having to run at a synchronous speed of 1200 RPM. In peaking the velocity ratio of the single flow design for 1200 RPM operation, the base diameter has been increased and L/D ratios have gone down. This also contributes to the lower efficiency. Measures could be taken to reduce the thrust loading, such as balance holes, but an added performance penalty would result.

Figure 8. Four Stage Designs, Real Blades



Also considered is a single stage, single flow design shown in Figure 9. An important part of this design is the diffuser, which is the only means of recovering some of the substantial amount of exhaust kinetic energy that would otherwise be wasted. For this reason, efficiency curves for a range of diffuser pressure recovery factors have been included in this figure. The highest value for the pressure recovery factor, 0.7 is based on the maximum area ratio possible, that is, the diffuser exit area identical to the condenser inlet flange area. This area ratio produces a diffuser length well over 40 feet for the optimum recovery factor. With this type of diffuser, an overhung design would be nessary, with the bearing compartment being designed into the inlet side of the turbine. The probability of obtaining a pressure recovery factor of 0.7 is considered to be very small, even with such a design.

As noted on Figure 9, the base diameter is 31 inches, which produces an L/D ratio of 0.33. Since it is considered impossible to successfully use a shroud with the L/D ratio being so high, these efficiency curves reflect the difference between four-stage, double flow and the one-stage, single flow designs. The normalized efficiency of the one-stage, single flow design is more than two percent lower than the preferred design, indicating that additional stages do a better job of converting energy to power than a diffuser. Other disadvantages of the one-stage, single-flow design include the difficult inlet case design and the requirement of more exotic blade materials to handle the higher stresses.

The final topic under consideration in the optimization study is that of using radial inflow turbines. Many different types of systems for radial inflow turbines are possible, but state-of-the-art technology limits the choice. No wheel diameter over 52 inches was considered, since this is approximately the largest radial inflow wheel size design now being considered. Since radial inflow turbines normally have a high energy transfer per stage, only single stage designs were studied for this low heat drop application. For the nominal conditions given, a synchronous speed of 3600 RPM gives the best range of velocity ratios for the wheel diameters studied (see Figure 10). The only

Figure 9. Single Stage, Single Flow Designs, Idealized Blades, Base Diameters = 31 In.

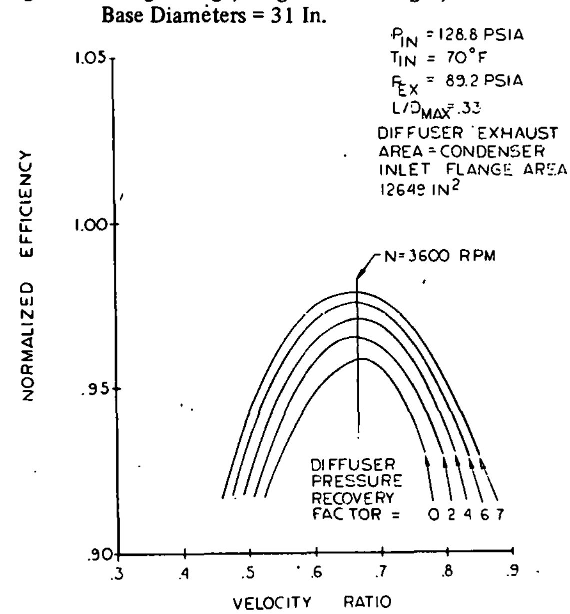
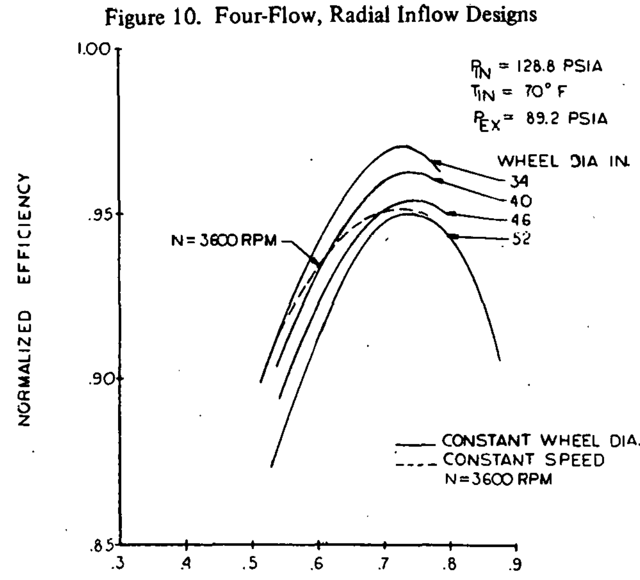


Figure 10. Four-Flow, Radial Inflow Designs



design choice open is the number of ways the flow must be divided. For one-half the total mass flow through a single wheel, the inlet blade heights get large, and the exducer outside diameter approaches the wheel tip diameter. This results in an impractical design for high efficiency.

Considering these facts, four-flow (flow divided between four parallel wheels), single stage radial inflow turbines of various wheel diameters and running at various speeds were analyzed. Minimum angles were assumed at stator and rotor exits to maximize efficiency. The results are shown in Figure 10. Notice that the 46 inch wheel produces the peak efficiency for synchronous operation, and that this point is less than 2 percent lower than the peak efficiency for nonsynchronous operation, making a gear-driven generator unattractive. All performance levels are considerably lower than those obtained with the optimum axial flow designs. Therefore, a radial-inflow design is undesirable for this application.

Considering the above factors, the double flow axial flow turbine with four stages of expansion when operating at 1800 RPM is the optimum aerodynamic design.

Turbine-Generator Control Scheme

Control of the turbine-generator covers the three basic phases of start-up/shutdown, steady state, and emergency trip. This system is rather unique due to the characteristics of this process, the requirement to limit the generator output, and the requirement to maximize turbine output at any steady state condition. A simplified block schematic of the proposed control system is shown in Figure 11.

The requirement to maximize power at any steady state condition dictates a power maximizing type of control, which is only possible if variable nozzles are provided and the generator is base loaded. The control would be as follows:

The generator is first locked into synchronous speed with the speed control. The load control then actuates the variable nozzles a predetermined amount at pre set time intervals. Initially, there is no movement of the variable nozzles until the dead band of the control is exceeded. When the variable nozzle position changes, the load control will sense the increase (or decrease) in power output and will continue to move the nozzles or reverse its motion to drive the power in an increasing direction. This is a slow response activity compared to the speed control.

The dead band, proportional band and reset on the speed setting must be adjusted and tuned on site to match the response of the system.

The above control scheme appears to be practical, although there are no known applications of this kind.

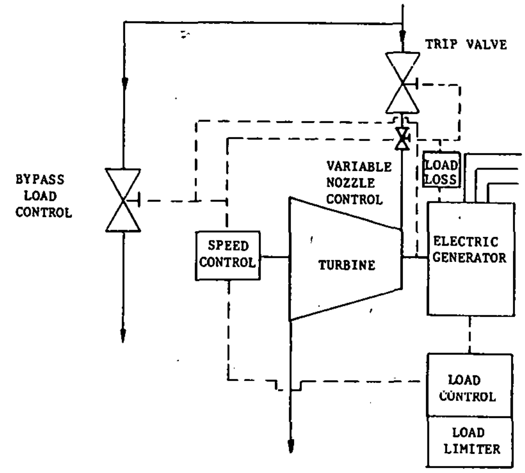
If the generator output exceeds its rating, a load limiter will decrease the variable nozzle setting, thereby reducing the mass flow to the expander based on the heat source characteristics. Thus, the power input to the generator will be reduced. The load limiter signal could be a part of the load control and will override the maximization of power signal to the variable nozzles.

Turbine Mechanical Design

The rotating element is the critical item in any piece of rotating equipment. Consequently, the designer is compelled to insure that the design of the blading is both adequate and practical from a mechanical viewpoint and at the same time keep a watchful eye on optimizing the aerodynamic performance. The proposed expander design has attained these objectives.

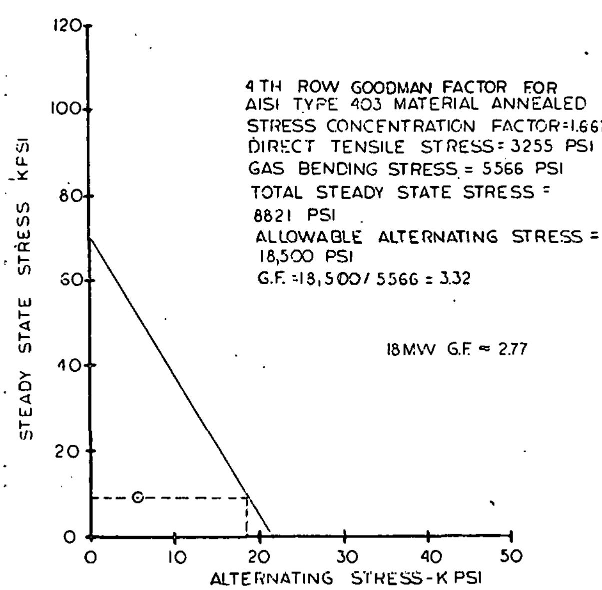
All four rotor blades were analyzed, but the fourth row has the highest stresses and lowest frequency.

Figure 11. Proposed Control Schematic



Using the calculated blade stresses and material properties, a Modified Goodman can be constructed as shown in Figure 12. For this particular blade the Goodman Factor is 3.3. From historical data, it has been found that blades with Goodman Factors of 2.5 and larger will not experience failures due to these stresses for this type of staging.

Figure 12. Ammonia Turbine Goodman Diagram



The rotating blades should be designed so that their natural frequency exceed the low multiples of the operating speed and do not coincide with the nozzle passing frequency. Since the fundamental natural frequencies of the rotating blades range from 8 to 11 times the operating speed, no blade vibratory problems are anticipated.

The operating speed is between the first and second lateral critical speeds resulting in a flexible shaft design, which is common practice in machine design. This approach increases performance and provides increased reliability with improved maintenance accessibility compared to a stiff shaft design where the operating speed is below the first critical speed.

Spherically seated 10-1/2 inch diameter by 7 inch wide cylindrical journal bearings provide the required stiffness and a 55 square inch thrust bearing absorbs the axial forces of the rotor.

A highly refined contact type face seal with shutdown features and a proven record of performance in process gas compressor installations should be used for this application, where leakage of toxic ammonia to the atmosphere must be eliminated.

Scaleability

The optimized turbine can be scaled upwards. One way of increasing the power output of the train is simply to double the capacity of the generator, double end it, and put duplicate turbines at each end of the generator. The controls for such a train become more complicated.

Table 4 shows various scaled arrangements of the optimized 15 MWe turbine-generator.

Table 4. Scaled Turbine-Generators

Gross Output (MWe)	Speed (RPM)	Base Diameter (Inches)	Relative Throughput	Remarks
15	1800	31.0	1.0	Base Design
30	1800	31.0	2.0	Two 15 MWe Turbines
33.75	1200	46.5	2.25	
60	900	62.0	4.0	
67.5	1200	46.5	4.5	Two 33.75 MWe Turbines
120	900	62.0	8.0	Two 60 MWe Turbines
25	1394	40.0	1.67	Gear Required
75	805	69.3	5.0	Gear Required

Another approach is to scale the optimized turbine designed for 1800 RPM to other synchronous speeds, 1200 RPM and 900 RPM. This is done by holding the velocity ratio (U/C_0, mean blade speed divided by isentropic velocity) constant, holding the blade L/D ratio constant, and varying the base diameter. The base diameter increases by the inverse ratio of the synchronous speeds (1800/1200, for example). The capacity and the gross power output increase by the square of this ratio. Once again, scaled duplicate turbines could be coupled to a double-ended generator to double the capacity of the train.

By accepting a nonsynchronous speed turbine, the optimized turbine can be scaled to any desired gross power output. However, gearing will have to be used to drive the generator at synchronous speed. This is undesirable because gears reduce the power at the generator coupling by about 2 percent, increase the size of the oil facilities, and are a frequent source of maintenance problems.

Achnowledgements

The authors gratefully acknowledge the invaluable contributions of Paal Bakstad of TRW for his work in system optimization and Arthur J. Miller, John E. Claydon, and Thomas J. Elliott of the Elliott Company for the design of the ammonia turbine.

Summary

An ammonia turbine has been designed using high performance state-of-the-art hardware. It was found that four stages of blading is the minimum economic number to provide peak efficiency for the OTEC design conditions. The performance level of the double-flow axial-flow four-stage turbine exceeded the performance level of other single-flow axial and radial inflow turbines for the same operating conditions.

A unique control scheme has been devised to continuously optimize the power generated by the generator.

The mechanical design is conservative for this service. Special provisions to insure positive sealing of the toxic ammonia vapor have been incorporated into the design. Materials of construction, blade stresses, blade frequencies, rotor lateral critical speeds, bearings and seals have all been investigated.

DISCUSSION

Question: In your definition of mean blade speed, do you use the arithmetic mean between tip and hub, or the mean-area diameter, or what?

C. Kostors: We use the arithmetic mean diameter between the hub and the tip, and since it is a multistage unit, we use then the square root of the sum of the squares to get effective value for the machine. Something else that I do not think is in the report is that if you look at this machine design on a basis of a different parameter than loss velocity ratio, that being specific speed, these stages are running at a specific speed of approximately 120, 125.

Question: Did you take secondary losses into account?

C. Kostors: Yes, we considered profile, leakage, disk, inlet, and exhaust losses using the pressure loss correlations.

OTEC SYSTEM RESPONSE AND CONTROL ANALYSIS

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ABSTRACT

An analysis is presented for OTEC system response and control. Two basic operational modes are considered consisting of constant and variable flow seawater pumps. Variable flow seawater pumps allow optimization of the OTEC thermal cycle state points for maximization of net generated power. Constant flow pumps are cheaper and simpler, but do not permit direct control over the evaporator and condenser operating temperatures. A system of non-linear differential equations representing the basic elements of a constant seawater flow OTEC plant has been formulated for computer solution. Typical response curves are presented for pressures, temperatures, mass flow rates, and generator speed for the Mini-OTEC plant scheduled for operation later this year.

NOMENCLATURE

A	Area
a	Constant
b	Turbine-generator loss coefficient
c_p	Specific heat
g_c	Gravitational constant
H	General transfer function or pump head
J	Moment of Inertia
K	Valve gain
L	Torque
M	Mass
$\dot{m}$	Mass rate
N	Turbine-generator speed ratio
p	Pressure
p'	Set pressure
q	Heat rate
R	Gas constant
s	Laplace transformation variable
T	Temperature
t	Time
UA	Thermal conductance
$\mathcal{U}$	Step function
V	Velocity
X	Valve displacement or vapor quality
Z	Compressibility factor

GREEK

α	Condensation coefficient
Δh_i	Isentropic enthalpy difference
ϵ	Turbine thermal efficiency
η	Turbine speed ratio correction factor
λ	Heat of vaporization
τ	Time at which $\omega = \omega_{os}$
ω	Generator speed
ω_s	Generator set speed

SUBSCRIPTS

1	Bypass valve
2	Turbine overspeed valve
3	Turbine control valve
B	Vapor line
BP	Bypass
c	Condenser
e	Evaporator
G	Generator
L	Liquid film or turbine-generator losses
os	Overspeed
P	Plate
R	Vapor line
RC	R to C
S	Separator or saturation
SB	S to B
SW	Seawater
T	Turbine or tube

INTRODUCTION

The design of an OTEC system is generally based on steady-state operating conditions which do not provide an insight into the dynamic characteristics of the system. Analysis of system stability along with response to changes in operating conditions, such as electrical load, requires the development of a mathematical model which will provide an adequate simulation of transient response. The principal elements to be considered are generally the heat exchangers and the turbine-generator although this depends on the particular OTEC system under consideration. The basic procedure followed in the present solution is to assume lumped conditions for line and heat exchanger vapor volumes and heat exchanger temperatures. Using this assumption, the appropriate time derivatives are written for the various system elements and solved simultaneously to provide the interaction between the various components when subjected to changes in the important forcing functions such as generator load and seawater flow or temperature variations.

THERMAL CYCLE CONTROL

The basic requirement of the OTEC thermal cycle control scheme is to provide stable regulation of flow rates, pressures, and turbine speed or load throughout the full range of operating conditions including instantaneous load loss. This entails regulation of vapor and liquid working fluid flow rates and, in some cases, seawater flow rates for various seawater temperatures and generator loads.

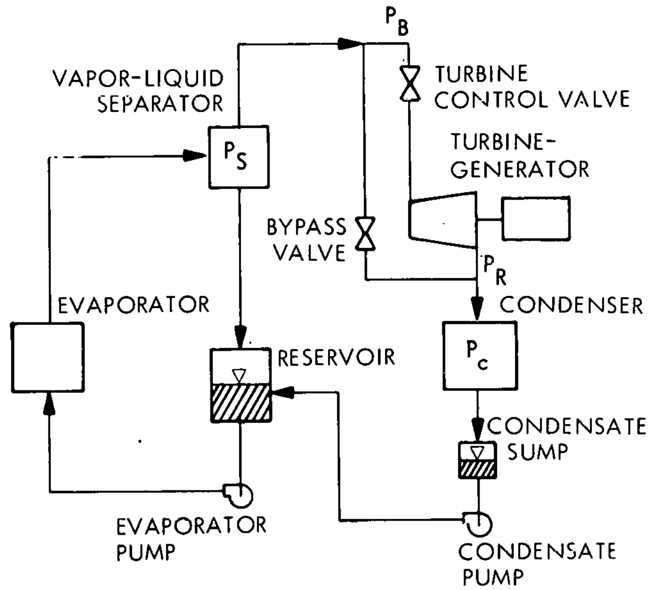


Fig. 1 Flow Schematic

Figure 1 shows a flow diagram for a plate heat exchanger OTEC cycle. With this arrangement, it is possible to use independent control for the vapor and liquid flow rates allowing a simpler and more stable flow control technique. Liquid flow control is limited to the condensate and evaporator pumps. The condensate pump is regulated to maintain the liquid level in the condensate sump between prescribed levels to ensure the required NPSH and non-flooding of the condenser. The plate evaporator is designed to operate at maximum rated load with any excess vapor being diverted through the bypass line shown in Fig. 1. Provision for constant liquid flow to the evaporator can best be maintained with a positive displacement pump which will be insensitive to pressure fluctuations. This will eliminate any need for feedback control of the evaporator pump and possible liquid flow stability problems. This type of evaporator feed is made possible by sizing the reservoir, shown in Fig. 1, to permit sufficient storage to accommodate system flow variations from the condensate pump and vapor-liquid separator.

Decoupling of the working fluid liquid and vapor flow control provides greater stability with a simpler control system. The basic requirement of the vapor control system is to provide stable electric power generation at cycle state points which have been selected to optimize net power generation or provide plant ship power requirements. For the system of Fig. 1, this is provided by appropriate regulation of the turbine and vapor bypass valves.

Two applications currently envisaged for OTEC are the plant ship and the base-loaded system tied in with a shore-side power grid. The choice of seawater flow control for either application has been shown to be dependent on the type of heat exchangers and the yearly seawater ΔT variation.[1,2] The choice of seawater flow control for a particular plant and location generally involves a cost tradeoff between increased pump costs versus the increase in net power for off-design operation. Current results for horizontal shell and tube heat exchangers indicate that seawater flow control should be considered for a sea-

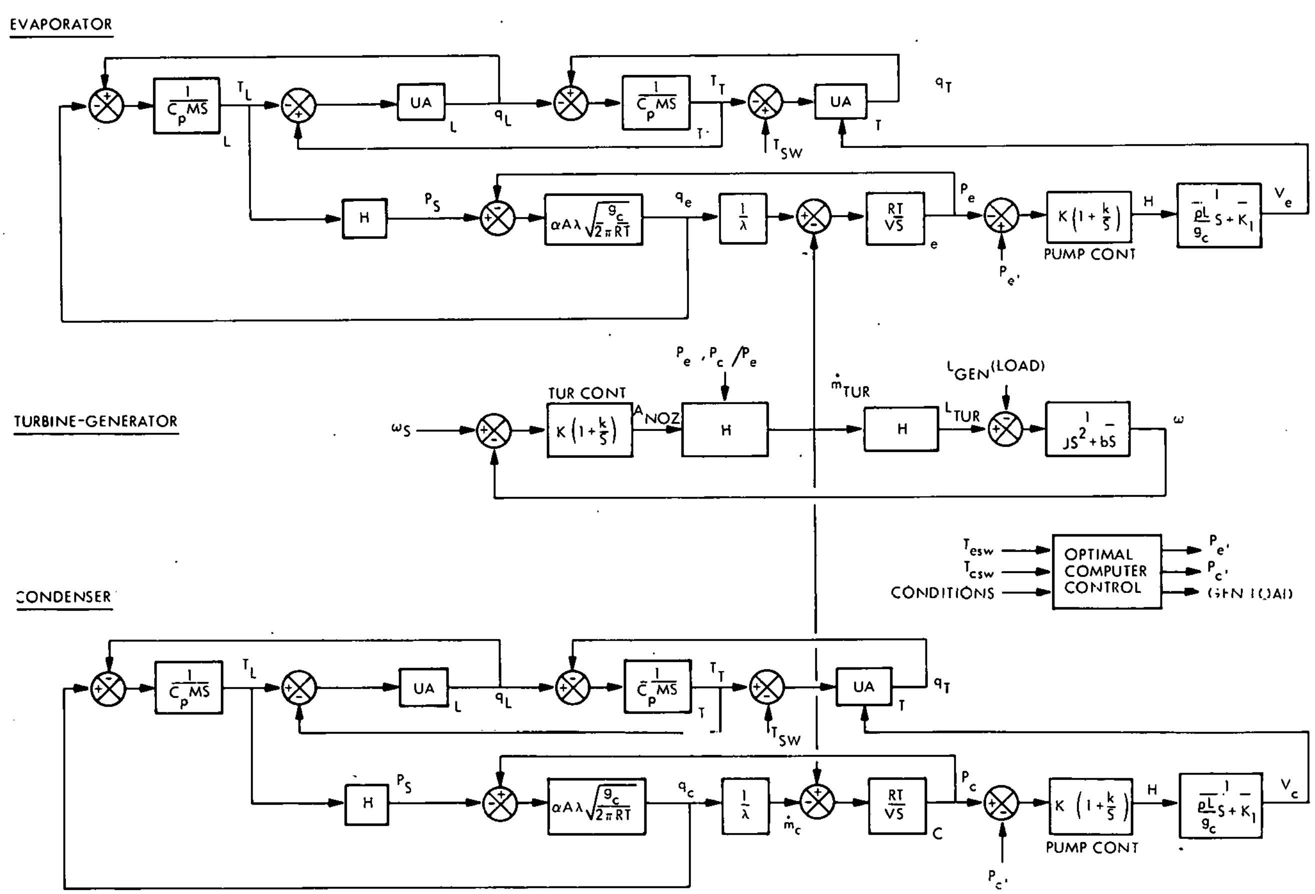


Fig. 2 OTEC System Analogue with Seawater Pump Control

water ΔT variation in excess of approximately 4^o or 5^oF.[1] Preliminary results for a plate heat exchanger OTEC system indicate an insignificant advantage for seawater flow control over a fairly wide range of ΔT (32^o to 46^oF).[2]

A system analogue for a base loaded plant with seawater flow control is shown in Fig. 2. Such an arrangement allows optimum net power generation for all conditions of seawater temperatures, fouling, and electrical load. The analogue is typical of a shell and tube heat exchanger OTEC plant where the various lumped heat exchanger temperatures are obtained from an energy summation while the various system pressures are obtained from a summation of mass rates. The set pressures for the evaporator and condenser are obtained from an in-line computer program which uses cold and warm seawater temperatures along with other operating conditions to generate the optimum pressures and generator load. The evaporation and condensation rates are controlled through the effect of seawater velocity on the seawater heat transfer coefficient. A reduction in evaporator pressure below the optimum set pressure would result in an increase in the warm seawater pump speed or blade pitch to produce an increase in the seawater flow and higher evaporator tube-side velocities. The response of the seawater velocity would depend on the pump controller gain along with the inertial effects of the seawater flow passages. The effect of the higher seawater velocity is to increase the working fluid liquid film temperature on the evaporator tubes and thereby increase the saturation pressure. The actual evaporation rate is obtained from Eq. (11) which requires a knowledge of the working fluid evaporation coefficient and vapor-liquid surface area. The same general observations apply to the condenser portion of the analogue shown in Fig. 2.

Figure 3 shows a control and system response analogue for a plate heat exchanger OTEC plant. In this system, with unregulated (constant flow) seawater pumps, vapor flow control is provided by controlling the vapor bypass valve to maintain the desired turbine inlet pressure, P_B. A simplified evaporator analogue has been shown as the dynamic characteristics of the plate type evaporator are not completely understood at the present time. The plate type condenser has been assumed to be similar to the shell and tube in terms of lumped plate and liquid film temperatures as well as vapor volume.

Turbine control is effected by sensing generator speed and regulating the turbine flow rate to match the generator torque requirement. Turbine-generator overspeed through load loss or other causes is accommodated by a fast response valve arranged in series with the turbine control valve. Exceeding a preset level, ω_{OS}, causes actuation of the overspeed valve, which interrupts vapor flow to the turbine.

Many variations of the two analogues shown in Figs. 2 and 3 are possible depending on the particular conditions. The suitability of the various assumptions can, ultimately, only be assessed by

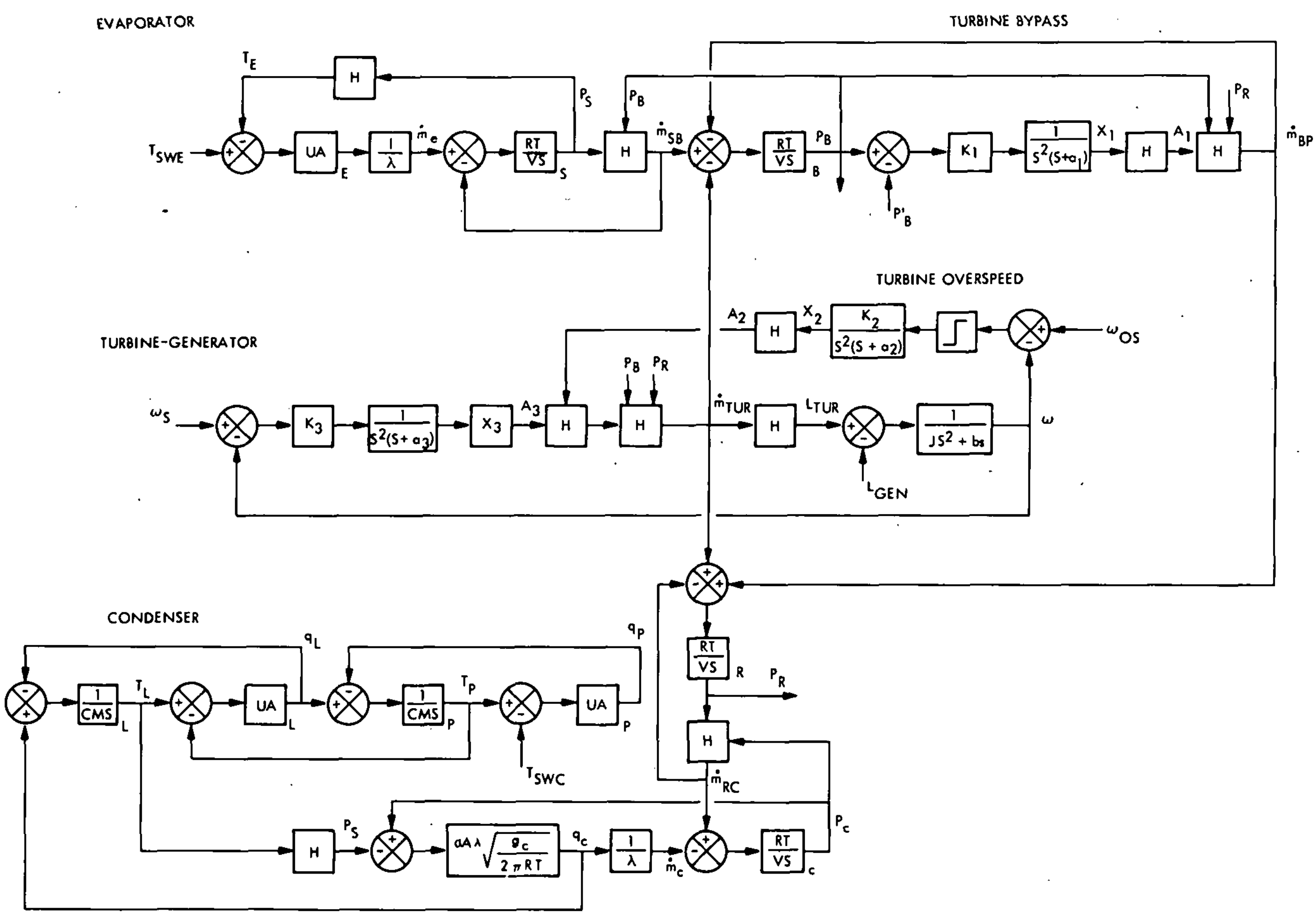


Fig. 3 OTEC System Analogue for Plate Heat Exchangers (Mini-OTEC)

comparison of predicted system response and control stability with actual OTEC plant data. Such an opportunity exists with Mini-OTEC which is scheduled to begin operation during, the summer of 1979. The analogue of Fig. 3 is based on the Mini-OTEC design and the following mathematical treatment and computer results represent an attempt to assess the Mini-OTEC system response.

SYSTEM OF EQUATIONS

The system of nonlinear differential equations used to simulate the Mini-OTEC plant are basically those indicated in the analogue of Fig. 3. The four system pressures are obtained from the continuity equation while the energy equation is used to calculate the lumped condenser plate and liquid film temperatures. The turbine bypass and control valve positions are obtained from second order differential equations with the error signals as forcing functions. The turbine-generator overspeed valve position is also obtained from a second order differential equation with a step function which is triggered by the turbine speed exceeding some predetermined value. The differential equation of motion for the turbine-generator is obtained from a summation of torques.

Pressure

$$\frac{dP_S}{dt} = \left(\frac{ZRT}{V}\right)_S (\dot{m}_E - \dot{m}_{SB}) \qquad (1)$$

$$\frac{dP_B}{dt} = \left(\frac{ZRT}{V}\right)_B (\dot{m}_{SB} - \dot{m}_T - \dot{m}_{BP}) \qquad (2)$$

$$\frac{dP_R}{dt} = \left(\frac{ZRT}{V}\right)_R (x\dot{m}_T + \dot{m}_{BP} - \dot{m}_{RC}) \qquad (3)$$

$$\frac{dP_C}{dt} = \left(\frac{ZRT}{V}\right)_C (\dot{m}_{RC} - \dot{m}_C) \qquad (4)$$

Temperature

$$\frac{dT_P}{dt} = \frac{1}{(C_pM)_P} (q_L - q_P) \qquad (5)$$

$$\frac{dT_L}{dt} = \frac{1}{(C_pM)_L} (q_C - q_L) \qquad (6)$$

Control Valve Position

$$\frac{d^2X_1}{dt^2} + a_1 \frac{dX_1}{dt} = K_1 (P_B - P_B') \qquad (7)$$

$$\frac{d^2X_2}{dt^2} + a_2 \frac{dX_2}{dt} = K_2 \,\mathcal{U}(t - \tau) \qquad (8)$$

$$\frac{d^2X_3}{dt^2} + a_3 \frac{dX_3}{dt} = K3 (\omega - \omega_s) \qquad (9)$$

Turbine-Generator Speed

$$\frac{d\omega}{dt} = (L_T - L_G - L_L) / (J_T N^2 + J_G) \qquad (10)$$

Compressible flow was assumed for the line and turbine mass rates for Eqs. (1) through (4). The condenser mass rate was calculated using the Knudsen equation as indicated schematically in Fig. 3.

$$\dot{m} = \alpha A_c \cdot \sqrt{\frac{g_c}{2\pi RT}} (P_C - P_L) \qquad (11)$$

The various heat rates used in Eq. (5) and (6) were calculated in the manner shown in Fig. 3. The film thickness used to calculate the working fluid mass in Eq. (6) was assumed to be constant as was the condensation heat transfer coefficient.

Turbine torque was calculated from the following expression where the isentropic enthalpy difference was based on the pressures P_B and P_R.

$$L_T = a\, \epsilon \eta\, \dot{m}_T \Delta h_i / \omega \qquad (12)$$

Turbine thermal efficiency, ϵ, is a function of flow rate and η is a correction factor dependent on the turbine speed ratio. The difference between turbine torque and the torques due to generator electrical load and windage plus mechanical losses determines the rate of change of turbine-generator speed. The electrical load is arbitrarily specified as a time dependent input in the present solution and generator torque is proportional to load divided by speed. The torque resulting from windage and mechanical losses has been assumed to be proportional to the square of the speed.

The time dependent flow areas for the turbine control, bypass, and overspeed valves have been assumed to be functions of controller position as indicated in Fig. 3. The turbine control and overspeed valves for Mini-OTEC have a linear variation while the bypass valve area has an "S" shaped variation with controller position.

RESULTS

The preceding equations were programmed for solution using design values from Mini-OTEC, which is a small OTEC plant scheduled for operation off Keahole Pt., Hawaii.[3] The design consists of a closed-loop Rankine cycle using ammonia for a working fluid with plate-type heat exchangers. The system has been sized to produce approximately 50 kW gross for the design and operating conditions shown in Table 1.

Figure 4a illustrates speed, mass rate, and pressure response for a two-stage ramp generator loading. The first ten seconds represent a zero electrical load although the turbine load due to windage and mechanical losses, is 20 kW. The initial fluctuations shown result from initial conditions such as condenser pressure and turbine and bypass valve openings which did not correspond to steady-state values. Steady-state has been approximately established after 10 seconds at which time the electrical load was increased, linearly, for 10 seconds to 50 percent of the

Table 1 MINI-OTEC DATA AND CONDITIONS

HEAT TRANSFER	EVAP	COND
Q_{sw} (GPM)	2700	2700
h_{sw} (BTU/HR-FT^2-°F)	988	833
h_{NH_3} (BTU/HR-FT^2-°F)	573	852
R_F (BTU/HR-FT^2-°F)$^{-1}$	3×10^{-4}	0
R_W(BTU/HR-FT^2-°F)$^{-1}$	2×10^{-4}	2×10^{-4}
U (BTU/HR-FT^2-°F)	307	388
A (FT^2)	4390	4390
T_{sw} (°F)	80	39

COMPONENT VOLUME	LENGTH (FT)	DIA (FT)	VAP. VOL. (FT^3)
EVAPORATOR	–	–	20
SEPARATOR	12	2	20
LINE-B	11	0.5	3.2
	13	0.33	
CONDENSER	–	–	20

rated value. This produced a sag in generator speed and an increase in the turbine valve opening and mass rate along with a corresponding decrease in bypass line mass rate. The turbine and bypass control valves have a saturated controller response time of approximately one second for full travel. The generator load is further increased between 30 and 40 seconds to 100 percent of the rated load which is 50 kW. This produces a further increase in turbine valve opening and mass rate with an attendant decrease in bypass line flow. The second load increase appears to have induced some instability in the system and the generator speed fluctuations are on the order of 6 to 7 percent. During one of these oscillations, the turbine mass rate exceeds the evaporation rate and at 42 seconds the bypass valve closes completely while attempting to sustain a separator pressure of 130 psia.

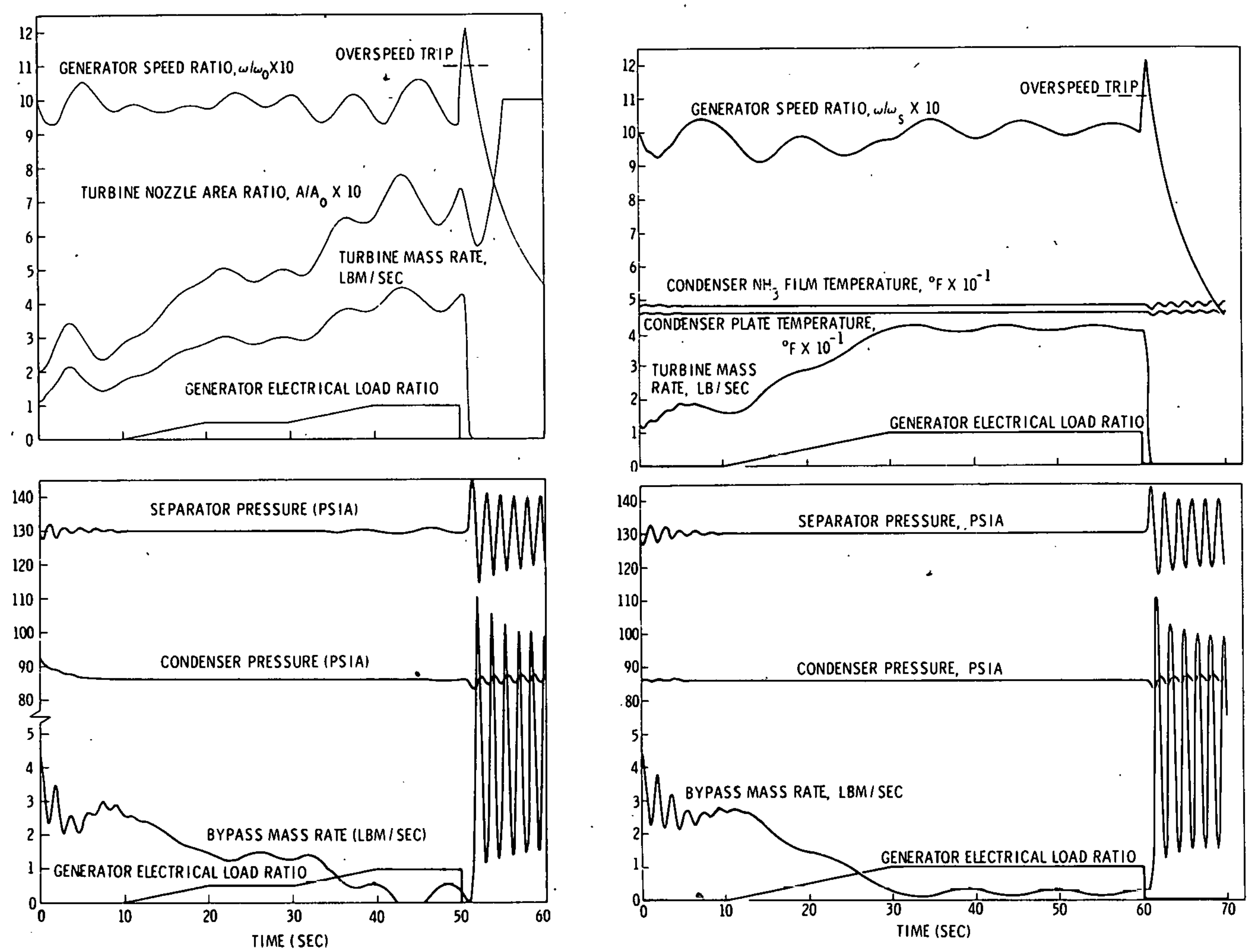
At 50 seconds the electrical load is instantaneously dumped simulating the opening of a circuit breaker. This results in an immediate increase in generator speed beyond the turbine controller's capability to limit the maximum speed to a safe value. The overspeed trip has been set to begin shutting the overspeed valve when the speed exceeds 10 percent, or 3960 rpm. The overspeed valve has a saturated closure time of 0.5 seconds which results in the maximum speed being limited to 21 percent of operating speed. The effect of the overspeed valve closure is also shown in the turbine mass rate which is essentially zero, 1.5 seconds after the load was dumped. The turbine controller initially reacted to the overspeed by starting to close until the speed dropped below the set point at which time it began to open to counter the diminishing generator speed.

Sudden interruption of flow to the turbine causes the separator pressure to increase since the bypass valve cannot react fast enough to maintain the set pressure of 130 psia. This results in fairly large fluctuations of bypass line flow rate and separator pressure which damp out with time. The oscillations could be eliminated by an override on the bypass valve controller triggered by the overspeed trip, or by using the pressure downstream from the turbine overspeed valve for the bypass valve controller.

The condenser pressure, for the present control scheme, is not actively controlled but floats at the value which results in a vapor flow rate equal to the thermal potential for condensation. The response of the pressure to flow perturbations is fairly rapid due to the small time constants (~ 1 second) associated with the thermal response of the plate and ammonia liquid film. Data for the condensation coefficient for ammonia in Eq. (11) could not be found and a value of 0.04 for water was used. The pressure drop in Eq. (11) was less than 0.1 psi for the present conditions, and the possible error associated with the lack of knowledge of α is felt to be small. The relatively small pressure drop results, primarily, from the large plate condensing area of 4390 ft^2. There is a slight decrease in the condenser pressure as load is applied due to the approximately 2 percent reduction in heat load. The slight increase in average pressure after 50 seconds results from the increase in heat load as well as a small change in the average evaporation rate resulting from the separator pressure oscillations. Inclusion of a thermal model for the evaporator would probably reduce the amount of separator pressure fluctuation and increase the damping, although modeling of the evaporator flow under these conditions would be extremely difficult.

The generator speed shown in Fig. 4a exhibited fairly large fluctuations between 30 and 50 seconds and appeared to be marginally stable. An indication of the influence of turbine controller gain was obtained with a second run with half the gain of the first. The results are shown in Fig. 4b for a continuous ramp generator loading between 10 and 30 seconds. The sag in speed at the beginning of the load increase is slightly larger than that for Fig. 4a as a result of the lower gain setting. In general, the generator speed oscillations have been significantly reduced along with a more stable turbine mass rate.

Figure 5 shows the general relationship between turbine controller gain, stability, and turbine-generator period of oscillation for the present conditions. Reference to Figs. 4a and 4b indicates a bypass mass rate period of oscillation on the order of 2 sec. This also appears to be approximately the limiting value of turbine-generator period with increasing turbine controller gain shown in Fig. 5. A possible conclusion might be that turbine-generator stability increases with increasing ratio of turbine-generator to bypass flow oscillation period. Figure 5 also indicates that, for the present conditions, this ratio must be larger than approximately two to produce stable operation. Final choice of the turbine-controller gain setting must be based on a consideration of generator speed stability as well as the maximum speed reduction which can be tolerated with a prescribed generator loading rate.



a. Turbine Controller Gain 0.0088 Ft/rad-sec

b. Turbine Controller Gain 0.0044 Ft/rad-sec

Fig. 4 Computer Solutions for Mini-OTEC Ramp Loading and Instantaneous Load Loss for Two Values of Turbine Controller Gain

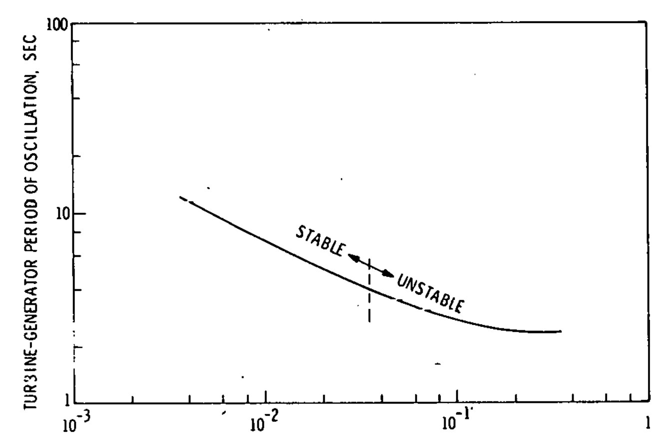


Fig. 5 Relationship Between Turbine-Controller Gain, Turbine-Generator Period, and Stability for Mini-OTEC

The present two examples apply to a single generator with speed control, which is representative of the Mini-OTEC system. Synchronous operation or parallel DC operation would require a different control scheme and consideration of the dynamic characteristics of the electrical circuit. The basic intent of the present effort is to provide a dynamic model for the thermal-fluid-mechanical aspects of an OTEC plant. Based on the computer solutions for Mini-OTEC shown in Figs. 4a and 4b, it is felt that the present method offers a fairly simple and effective means of simulating OTEC plant response.

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2. "Ocean Thermal Energy Conversion Power System Development-II, Phase I - Final Report," HCP/3407-2, LMSC-D637832, 31 July 1979.

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DISCUSSION

D. Richards, JHU/APL: Do you anticipate any dynamic problems from wave and heave-induced seawater flow variation and variation of ammonia flow at wave periods corresponding to system time constants?

W. Owens, Lockheed: The influence of waves and heave can be important through their effort on seawater flow variation as noted by Mr. Richards. Variable seawater flow manifests itself as a variation in working fluid condensation and evaporation rate; the relative magnitude of which is dependent on the influence of the seawater heat-transfer coefficient on the overall heat-transfer coefficient. The analogue shown in Fig. 2 of the paper would require the input of variable head in the pump-control heat-exchanger seawater velocity portions to assess the influence of waves and heave. This could be an important consideration in the overall control system stability determination depending on the relative values of the various system constants and amount and period of pump-head variation.

DYNAMIC AND OFF-DESIGN ANALYSIS OF OTEC CLOSED CYCLE POWER SYSTEMS

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ABSTRACT

An advanced OTEC closed Rankine cycle was formulated and analyzed via power system analytical simulation programs developed by Westinghouse and Carnegie Mellon University, ~~on D.O.E. contract number EG-77-C-03-1569.~~ Since cycle uniqueness precluded selections based on experience, least cost optimization algorithms involving pattern search techniques were used to verify that anhydrous ammonia is the most suitable working fluid and that the particular configuration selected provided maximum power cycle efficiency. A highly favorable net energy balance was achieved and individual components (Evaporator, Moisture Separator, Turbine, Generator, Exciter, Condenser and Seawater Pumps) with a minimum loss, novel control system possessing the requisite flexibility, stability and availability were developed. A unique power plant configuration was developed in response to contract specifications. Dynamic and off-design conditions (start-up, shutdown, part load, overload, varying seawater temperatures, and varying levels of biofouling) were simulated on the OTEC power plant and were extensively analyzed with the aforementioned computer programs to demonstrate system transient response, working fluid inventory transfers, and to determine the unique specifications of the balance of plant. Thus, requirements for heat exchanger hotwell capacities, liquid ammonia pump characteristics, connecting pipe sizes, valve specifications, storage capacities, condenser venting loops, etc. were all analytically determined. Finally, requirements for auxiliary power for start-up, purging, emergencies, shut-down, etc. were determined for this complete power cycle-especially with a view towards it being in a ship type hull.

This paper is organized to the following sections:

1. Introduction
2. OTEC computer models
3. Application of program results to component design
4. Conclusions and Recommendations

1. INTRODUCTION

Background

The extraction of useful electrical power from the tropical ocean thermal gradient between the 80°F surface and the 40°F depths requires a solution of the technical problem of exploiting this very small thermal difference.

OTEC closed Rankine cycles have a uniqueness deriving principally from the low energy availability of the ocean which requires the handling of very large volumetric working fluid and seawater flows. This novel cycle precluded optimization decisions based on experience and led to the application of a system approach to the design of a complete power cycle wherein system optimization studies resulted in the selection of the ideal closed cycle design parameters.

* Net Energy Balance results when more energy is produced by the plant than is used in its construction.

The closed cycle operates by using warm surface seawater to vaporize ammonia in an evaporator. This vapor is then expanded through a turbine and condensed in a condenser which is cooled with cold water pumped from the ocean depths of approximately 3,000 feet. The direct coupled turbine generator delivers electric power. A simple schematic diagram of this power system is shown in Figure 1. While this historical solution[1,2] was available, it remained to determine optimum cycle and individual equipment design parameters to make the concept economically attractive with a highly favorable* net energy balance.

The mass flow rates of working fluid as well as seawater in both the warm and cold water loops are quite large for the electrical power output as compared to a conventional fossil or nuclear plant. The OTEC plant ammonia flow rate is approximately 23 times the steam flow rate of a nuclear plant and 45 times the steam flow rate of a fossil plant. These flow rates pose a considerable challenge for the efficient design of the major equipment components.

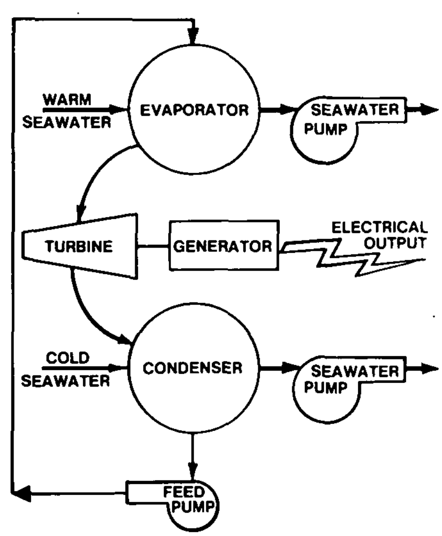

Fig. 1 Schematic diagram of elementary OTEC power system.

Control System Requirements

The inherently low efficiency of the closed cycle is a direct result of the aforementioned low enthalpy drop of the working fluid as it is expanded through the turbine and the large quantity of energy required to power seawater pumps. Efforts to minimize the power cycle pressure losses resulted in elimination of the conventional control valve preceeding the turbine in favor of a valved parallel by-pass loop. A schematic diagram of this arrangement is shown in Figure 2. Thus the conventional turbine-generator rotor overspeed control problem has been further aggravated by the indirect arrangement of control valves as well as a more severe ratio of working fluid mass flow rates to turbine-generator rotor mass. An additional complication arises from the selection of a porous heat transfer enhancement on the outer surface of the evaporator tubes which results in more trapped liquid and more entrained energy in the evaporator. This larger entrained energy with its relatively long decay time presents a particular problem for system safety at time of load rejection. The turbine by-pass valves are capable of being full closed to full opened in a quarter of a second to divert vapor flow around the turbine directly to the condenser, while the liquid ammonia feed valves to the evaporator are capable of being quickly closed to interrupt vapor production. These valves, acting together, limit turbine-generator overspeed. Sudden interruption of vapor production by electrical shutoff of the liquid ammonia feed pumps provides redundant overspeed protection.

Westinghouse - Carnegie Mellon Computer Programs

The solution to this transient control problem was achieved by Westinghouse and Carnegie Mellon University through the development and application of a dynamic simulation. This is one of three comprehensive analytic system simulation programs developed to investigate and cost optimize the OTEC closed cycle power system and its equipment components. The various computer programs which are the subject of this paper are briefly described as:

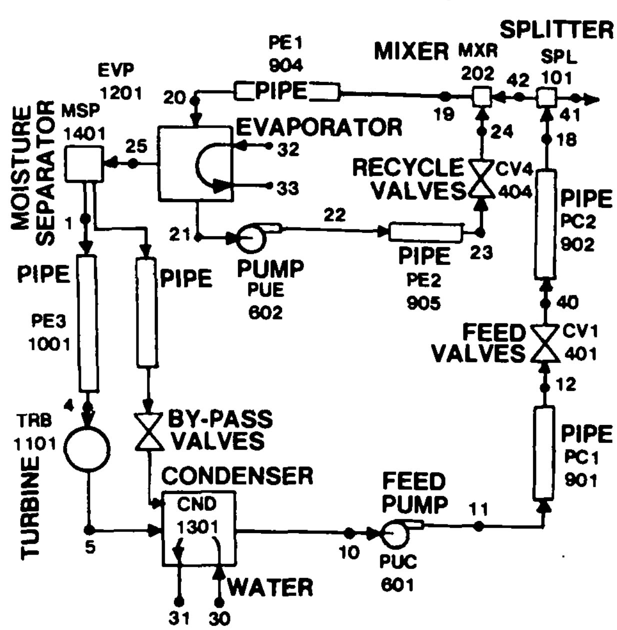

Fig. 2 Schematic diagram of OTEC system with turbine by-pass valves.

1. Design Optimization Program, OTOPT, designs the lowest cost OTEC Power System within the constraints of specified warm and cold seawater inlet temperatures, biofouling resistance, and module net power output.

2. The Steady State Flowsheeting Program, OSAP-1, calculates the steady state thermal performance and turbine mechanical performance of a specified power system at design and off-design conditions, such as would occur with variations in the warm and/or cold seawater inlet temperatures.

3. The Dynamic Simulation Program, ODSP-3, calculates the transient thermal response and turbine mechanical response of a specified power system, initially at design load, to changes in thermal conditions. Such changes would be caused by seawater temperature variation or during load lowering or shedding, resulting in transient flows and/or emergency trips.

This set of computer programs proved to be an important design tool which determined the size of system components to meet control response requirements and accommodated system flow transients, performed design tradeoff studies of the system variables, verified the static and transients response characteristics of the system, and insured that individual components interacted and functioned as an integrated control system.

Baseline Power Module & D.O.E. Contract Requirements

The OTOPT program, in addition to other design decisions discussed, indicated power plant cost per kilowatt, for the surface ship type platform, continued to decrease with increasing module size beyond 100 MWe per module. However, platform cost per kilowatt reached a minimum value at 25 MWe per module size because of the significant increase in hull depth required to accommodate the optimum heat exchanger diameters of larger modules. Consequently, total system (power plant plus platform) cost showed a minimum value with 50 MWe modules. Subsequently, emphasis was shifted to a 10 MWe Modular Application Power System under Contract Number EG-77-C-03-1569 for the Division of Solar Energy of the Department of Energy. The work presented herein was performed under this contract. The control system and concepts described are for the 10 MWe system, but are component-for-component directly scalable to the 50 MWe cost optimized commercial application power system.

Turbine Design Considerations

The turbine design has a strong influence on the overall system. Turbine efficiency directly affects the net power output and therefore has considerable influence on the specifications of all other components in the system, and thus strongly affects cost per unit of power output.

The turbine pressure ratio is suited to a single stage of axial flow reaction blading. The blading design selected in response to a specification for a base-loaded turbine is one that results in peak efficiency at the base-loaded point and somewhat diminished efficiency at (near) off-design operation.

Control System Design

The design of the control system for both the 10 MWe Modular Application and the 50 MWe Demonstration Plant under both steady-state and transient operating conditions was preceded by analysis of steady-state performance at part-load, off-design seawater temperature conditions, variations in biofouling levels, transient performance at overspeed, start-up, shut-down, synchronization to an electrical grid, and loading changes. Use was made of these various computer programs in supporting design and analytical studies. Results of these studies were used to produce the control hardware and software designs and related product design specifications. System operation at part load, system operation at overload and effect of biofouling resistance on electrical output are discussed in detail.

2. OTEC COMPUTER MODELS - OTOPT, ODSP-3 and OSAP

Cost Optimization Program OTOPT

Program OTOPT determines the lowest capital cost OTEC power system. The main constraints of this Westinghouse program are the warm and cold seawater inlet temperatures and a shell-and-tube configuration for both the evaporator and condenser. All other thermodynamic and mechanical parameters are manipulated by the program to arrive at a parametric combination yielding the lowest cost system. This modular computer model contains these sections - Main optimization program, plant heat balance subroutine, design subroutines for each component, and pricing subroutines for major components. The modular approach was chosen to facilitate algorithm improvements.

Optimization Method

The cost and performance of the power system are functions of independent system variables. The number of system variables is determined by the equations used in the heat balance and design calculations. If the mathematical model consists of E equations and U unknowns, there are (U-E) system variables. While the number of system variables can be calculated using this method, the identity of the system variables is still open to judgment. For example, the seawater pressure drop may be calculated from the tube velocity, or the velocity may be calculated from the pressure drop. The optimization would yield the same result regardless of which system variables were chosen, so the choice depends on the order in which the equations are solved.

The system variables used in the optimization of the OTEC power module are of two types: continuous and discontinuous. Continuous variables are those such as temperatures and velocities which can vary infinitesimally over the range to be studied. Discontinuous variables are variables such as tube diameter, tube material and the type of tube enhancement which exist only as discrete alternatives. The optimization technique was used to optimize the continuous variables for each combination of discontinuous variables.

Calculation Sequence

The calculation sequence for the heat balance and design equations begins with the thermodynamic properties of the condensate leaving the condenser. The pressure drop between the condenser and the ammonia feed pump is calculated to determine the entrance pressure to the pump. The next set of calculations determine the thermal specifications for the evaporator, after which the evaporator is designed. The remainder of the heat balance equations are solved as needed to provide information for the design of the separator, the ammonia drain tank, the ammonia feed pump, and the ammonia recycle pump. The last three items of the ammonia cycle to be designed are the turbine, the condenser, and the large cold water pipe which brings the deep, cold water up to the condenser.

After the two heat exchangers have been designed, the seawater pressure drops through the tubes of the heat exchangers are calculated.

The seawater pressure drop in the cold water pipe is added to the tubeside pressure drops in the condenser, after which the cold and warm seawater pumps are designed. When the components have been designed and their associated parasitic losses have been calculated, the net power output of the power module is determined by subtracting the power requirements of the ammonia pumps, seawater pumps, and chlorination system from the gross power output of the turbine. If the net power output differs from the

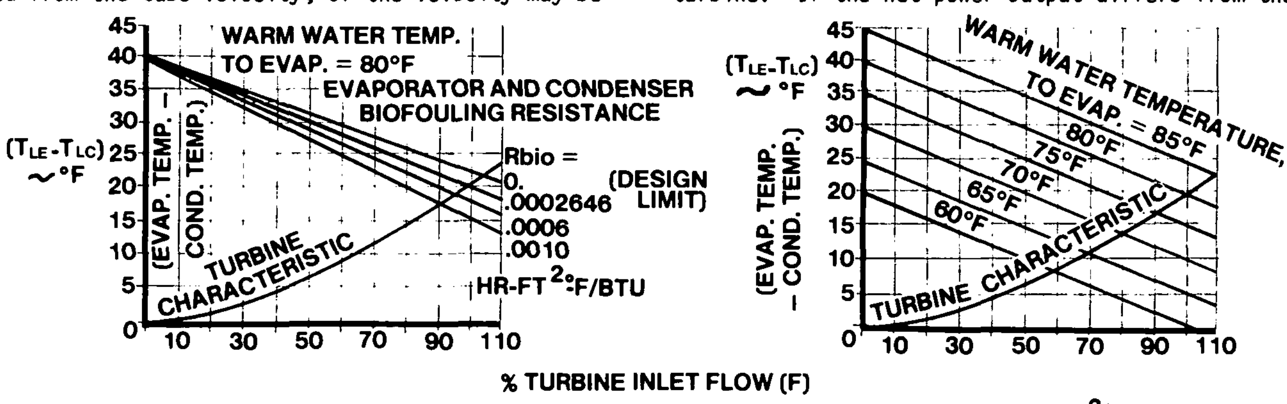

Fig. 3 10 MWe modular application temperature differentials, evaporator and condenser.

specified output power, the cycle mass flow rate is adjusted and the entire calculation sequence is repeated.

Various types of cost estimates were used for the components depending on the availability of data. These included a single estimate, a factored estimate, a factored estimate with the capacity exponent, a summation of labor and material costs with an administrative cost multiplier, a curve fit through a range of supplier cost estimates, and price list data.

OTEC Modeling Aspects of ODSP-3 and OSAP

In this section the characteristics of the models developed at Carnegie-Mellon University to describe the transient and steady state behavior of the components within the OTEC plant are described. Two programming systems, ODSP-3 and OSAP, which use these models to effect a dynamic simulation of an OTEC plant and an off-design steady-state calculation, respectively are discussed.

Figure 2 illustrates a particular OTEC plant configuration as an aid in following this description.

Component Modeling Aspects

Figure 2 shows that this OTEC plant configuration can be modeled by considering models for the following components: mixing unit, stream splitting unit, pipe segment (pipeline), pump, valve, turbine-generator, evaporator, condenser and moisture separator. The general characteristics of each of these units as modeled for use in the simulation system ODSP-3 and OSAP, are as follows:

The model of the splitter is the simplest of all the components in the system. A mass balance is required among the incoming and outgoing flows. The thermodynamic state of each stream is identical. For the mixer, an enthalpy balance is required in addition to the mass balance.

The model of a pipe segment considers flow resistance and transport lag in the pipe. The fluid in the pipeline is assumed to be isothermal and incompressible. The pressure drop is calculated using Darcy's Law for fluid friction with a correction for head change due to an elevation difference between its inlet and exit. Equivalent length is included to account for the frictional loss due to bends and fittings. The flow delay time for a pipe segment in a transient simulation is calculated from an integral equation which equates the integrated flow during the delay time to the total volume of the specific pipeline.

The valves are modeled utilizing a valve co-efficient. The moisture separator is also considered as a flow resistance but with a volume which cannot be neglected. The pump models include simple mass balances and energy balances. The adiabatic temperature rise in the pump considers a frictional heating component and a flow work component. The pump characteristic curve is supplied by the user and gives fluid pressure increase by the pump versus fluid flow rate.

The model for the turbine generator includes calculation of the input energy or potential for the speed change initiated by a load dump from the design point. First, the energy balance at steady state (design) conditions is obtained according to the requirement for the rate of energy converted by the turbine to electrical power generated plus loss (generally friction). In the steady state, with the turbine generator locked into the utility system at synchronous speed, the load change is calculated corresponding to a given control change in the turbine flow and pressure ratio. After a load dump, the turbine energy balance is modified to change the kinetic energy involved in altering the speed of the turbine generator. The dependence of turbine and generator losses on rotational speed and the performance equations of the turbine were included in the model. The turbine mass flow rate was modeled by use of the Stodola Ellipse Law and went into the model for turbine thermodynamic power. This turbine power model consisted of expressions in terms of speed, pressure ratio and inlet temperature.

The models for the condensers and evaporators are complicated. A lumped system is considered in their dynamic modeling. Only one horizontal tube is considered in the modeling. Transient heat balances are formulated for the water, the tube, the ammonia liquid on the tube and the vapor in the vessel, and finally for the liquid in the hotwell.

The transient water heat balance considers the heat transfer to the tube and the incoming and outgoing enthalpies of water. To approximate the dynamic behavior, the transit times for the water are introduced as a time delay. The transient enthalpy balance for the ammonia liquid on the tube and the ammonia vapor in the vessel considers the heat transfer from tube to vapor, liquid feed onto the tube and draining to the hotwell, evaporation from the hotwell, and the vapor leaving the vessel.

Mass balances are also formulated separately to describe the holdup for the vapor, the liquid ammonia on the tube, and liquid in the hotwell. Additionally, the state equation for the vapor temperature is written with the assumption that the vapor in the condenser is in the saturated state and that it follows the ideal gas law.

In general, this model can be applied to both the condenser and the evaporator. Specific considerations are that the condenser does not have liquid feed onto the tube and no vapor leaves the vessel. On the other hand, no vapor comes into the evaporator. Under normal operating conditions the evaporator will have ammonia transport from the liquid film to vapor in the vessel; this transport will be a negative quantity for the condenser.

The liquid in the hotwell will flash when the vessel pressure becomes lower than the saturation pressure of the hotwell liquid. Also, at reduced load conditions, the evaporator may operate at a low recirculation rate and the tube may experience partial or complete dry-out as well as during transients. Since the exact relationship between the heat transfer and film flow during partial dryout is not available in the open literature, only complete dry-out is modeled.

The present model is limited by the assumption that the tube bundle in the thin film heat exchanger is considered as a single lumped tube. In practice, this lumped approach is good for a transient where the ammonia liquid feed rate varies slowly with time. For this case the characteristic time of the feeding transient is longer than the time of the liquid flowing from the top of the bundle to its bottom. On the other hand, when a severe liquid feed rate transient occurs, the reliability of the lumped approach will be reduced unless an appropriate liquid flow delay time is included. (This delay time is being included in a modified model for the dynamic simulation of the OTEC system currently being developed at Carnegie-Mellon University).

The ODSP-3 Program

This section outlines the basic features of the dynamic simulation system developed at Carnegie-Mellon University, (C.M.U.) to simulate the performance of the proposed Westinghouse OTEC module.

The model developed is applicable to a number of particular cases of interest, predicting system response to the following types of control action:

- Opening the bypass valve in response to a severe reduction in electrical load on the generator. (rapid response)
- Closing off of the liquid flowing from the condenser back to the evaporator, pumping it instead out of the system to a liquid ammonia storage facility. (inventory control)
- A combination of these actions.
- Control of the turbine speed by manipulation of the bypass valve (high frequency response to hunting).
- Control of the turbine by controlling the inlet flow of ammonia onto the evaporator tubes. Here the flow would be less than the evaporator is capable of evaporating so there is no overflow of excess ammonia off the tubes and thus no recirculation of the excess ammonia. This mode is for startup and part load situations.
- Sinusoidal manipulation of the valves to provide frequency response information.

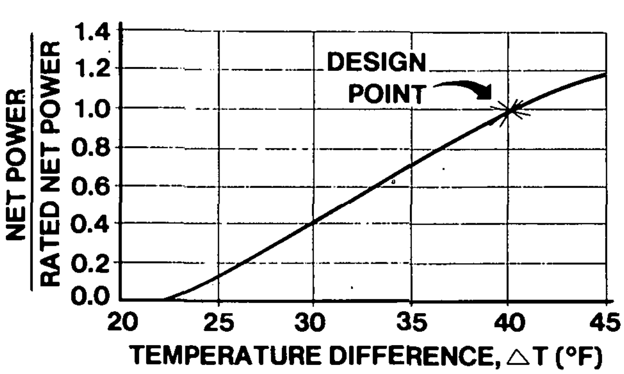

Fig. 4 Temperature effect on performance.

Because of the very short time allowed to develop ODSP-3, it was written specifically to solve the OTEC plant shown in Figure 2. Full advantage was therefore taken of the particular structure of the algebraic equations which are solved in two steps. total plant model is a collection of dynamic and algebraic equations which are solved in two steps. First an initial steady-state for the plant is established to obtain the design pressures and temperatures of the ammonia and the areas of the ammonia control valves consistent with the design point. This involves simultaneously satisfying the dynamic equations with all time derivatives set to zero, the algebraic equations and the required user specifications. Once an initial steady-state was established the dynamic simulation began. Needed here were specifications for the input "independent" variables versus time. These variables permitted the perturbation of the plant.

The structure of ODSP-3 is rigid. The user has virtually no alternatives in which variables he must specify to effect a simulation. This rigidity is obviously a disadvantage, but it permitted ODSP-3,a very efficient program in terms of computer time,to be developed quickly. Solution algorithms were determined for the 150 or so simultaneous nonlinear algebraic equations modeling the steady-state, which required guessing and converging simultaneously only three variables in one large iteration loop. The 140 or so algebraic constraints were solved during the dynamic simulation, by guessing and converging only 5 variables in two non-overlapping iteration loops, one loop with two iteration variables and the other with three. Special algorithms available in the literature on "tearing" methods were used for the solution of the simultaneous nonlinear algebraic equations to find these solution procedures; then a Harwell library subroutine (VAØ5AD) was employed to effect the convergence of these iteration loops.

VAØ5AD is a sophisticated numerical routine which can solve the problem

$$\min_{x_1, x_2, \cdots x_n} \sum_{j=1}^{m} f_j^2 (x_1, x_2, \cdots x_n)$$

When n=m the routine will adjust n x_i variable values to drive n error functions f_j to zero (as that would be the minimum of

$$\sum_{j=1}^{n} f_j^2(x_1, \cdots x_n)$$

To integrate the set of ordinary differential equations, the IBM Scientific Subroutine library program DHPCG was used. It is a Hamming-Predictor/Corrector integration package for numerically integrating sets of first order ordinary differential equations (ODE's) from an initial starting condition. The equations are of the form

$$\frac{dx_1}{dt} = f_i(x_1, x_2, \cdots x_n, t)$$

$$x_i\ (0) = x_i^0\ ,\ \text{given}$$

$$i=1,2,\cdots_n$$

A user written routine is called by DHPCG, repeatedly, to evaluate the $f_i(x_1,x_2,\cdots x_n,t)$ for values of $x_1,x_2,\cdots x_n$ and t supplied by DHPCG.

Because the model is really a mixture of fourteen ODE's and about 140 nonlinear algebraic equations, the actual form of the model is

$$\frac{dx_i}{dt} = f_i(x_1,x_2,\cdots x_n,z_1,z_2,\cdots z_m,u_1,u_2,\cdots u_r,t)$$

subject to $g_j(x_1,x_2,\cdots x_n,z_1,\cdots z_m,u_1,\cdots u_r) = 0$

where $i=1,2,\cdots n, \quad j=1,2,\cdots m$

The variables z_1 to z_m are algebraic variables whose values are obtained by solving the m nonlinear algebraic equations $g_j=0$, $j=1,2,\cdots m$. Thus they are functions of the state variables x_i and the independent variables u_k. Time dependent values for the variables u_k supplied by the user are the mechanism by which a disturbance is described for the process. An example is the electrical load on the turbine which may be decreased abruptly. Thus the variables u_k act only as time varying parameters to the restated problem in the desired form for DHPCG:

$$\begin{aligned}\frac{dx_i}{dt} &= f_i(x_1,x_2,\cdots x_n,z_j(x_1,x_2,\cdots x_n,u_1(t)\\ &\quad u_2(t),\cdots u_r(t),\ j=1,2,\cdots m)\\ &= f_i(x_1,x_2,\cdots x_n,t)\end{aligned}$$

It is necessary to solve a set of about 140 simultaneous nonlinear algebraic equations each time DHPCG requests values for the f_i function as it is integrating the equations. This is accomplished efficiently using the tearing concepts to find the correct iteration variable together with the Harwell library routine VAØ5AD to converge the set of 140 simultaneous nonlinear algebraic equations.

The OSAP Program

OSAP is a flowsheeting program prepared by CMU for analyzing the steady state behavior of an OTEC plant such as that shown in Figure 2. OSAP is a modular flowsheeting program based on the same modeling equations as used in the ODSP-3 program. Any OTEC plant can be analyzed which can be modeled by interconnecting the component modules listed in Table 1.

OSAP is an unusual flowsheeting program because, provided the choice of input variables is legitimate, the user can select any set of variables to provide prespecified values. OSAP will then set up and execute automatically a program which will calculate values for all the remaining variables. In an abstract sense, OSAP considers the problem to be one of solving n nonlinear algebraic equations in m ($m>n$) variables. Thus m-n variables must have their values prespecified by the user, leaving OSAP to solve the n nonlinear algebraic equations for the remaining n variable values. These n equations must not be "singular" in the remaining n variables left for solution, but this is virtually the only restriction placed on the user's choice of variables to specify.

This means the user could employ the following plan to analyze the steady state behavior of an OTEC plant. (Refer to Figure 2).

TABLE I

OSAP-1 Unit Module List

1. Simple Stream Splitter
2. Mixer Unit, liquid streams
   Mixer Unit, vapor streams
3. Valve, liquid stream
   Valve, vapor streams
   Moisture Separator (Modeled as a Valve)
4. Pump
5. Pipe, laminar flow, liquid stream
   Pipe, laminar flow, vapor stream
   Pipe, turbulent flow, liquid stream
   Pipe, turbulent flow, vapor stream
6. Turbine
7. Evaporator and Condenser

1. User establishes the plant structure to analyze and develops an OSAP model for it.
2. To establish the equipment sizes needed for given ammonia flows and given turbine design characteristics, select the flows as given, select the turbine design characteristics as given and let the system calculate some of the sizes for various pieces of equipment (such as number of tubes in the evaporator and condenser units, valve throat areas, etc.). The user specifies elevation, diameter, length, etc., for all pipelines; valve constants for all valves; tube lengths and diameters, etc., for this run. OSAP is used to solve the model.
3. To determine the effect of closing off a fraction of the tubes in the evaporator on the steady state behavior of the plant, the user specifies and fixes the number of tubes in the evaporator (a quantity calculated in the first analysis, but is now to be given). A variable previously designated as an input (i.e., given) must be released to allow OSAP to calculate a value for it. Select the ammonia flow rate through the turbine and run OSAP to analyze the plant.
4. To calculate the valve constants which result from specification of the valve throat areas, (fixed as inputs), release for each valve the valve constants to be calculated.
5. Etc.

Thus, with relative ease, a whole host of calculations are possible, allowing the user to do "design" calculations (fix the flows and calculate equipment sizes) or "performance" calculations (fix the equipment sizes and calculate flows, T's and P's, etc.). Also, if a baseline is first calculated, then parametric sensitivity analysis is easily accomplished. Thus a user can usually set up a sequence of calculations which will not cause the resulting problems to be "singular."

OSAP is based on the Newton-Raphson iteration technique to solve the simultaneous nonlinear algebraic equations which model a plant. The Newton-Raphson technique establishes at each iteration a linear model which approximates all the equations around the current guessed solution. The linear model is solved for the values of the variables of the linear equations to make them simultaneously have zero error. These variable values become the next guess, and, if the initial values are close enough to the solution, convergence to very high accuracy is accomplished in 7 to 12 iterations.

To set up the linearized models for an arbitrarily configured flowsheet requires a modular program design in which the equation coefficients and errors are generated separately for each component. The executive routine collects these and removes the contributions from the prespecified variables when it sets up the complete problem. It then submits the linearized equations for each iteration to a sparse matrix package (Harwell's MA28 package) for solution.

The program aids the user by having the following features:

1. Only a small subset of the total of problem variables need be given initial guesses. The rest are guessed automatically. However, any variable may be given an initial guess by the user which overrides that determined by the program.

2. Internal variable scaling and equation scaling are automatically performed to reflect appropriate engineering accuracies. This scaling aids the convergence characteristics of the program without the user's knowledge of it.

3. Intermediate printouts can be requested to allow the user to track convergence.

4. A rerun may be performed with initial variable values for the rerun being based on those from the previous run. A significant reduction in the number of iterations frequently occurs.

5. A sledge hammer convergence method based on a variation of so-called "continuation methods" is incorporated and usually can force convergence, even if slowly, if the Newton-Raphson method is failing.

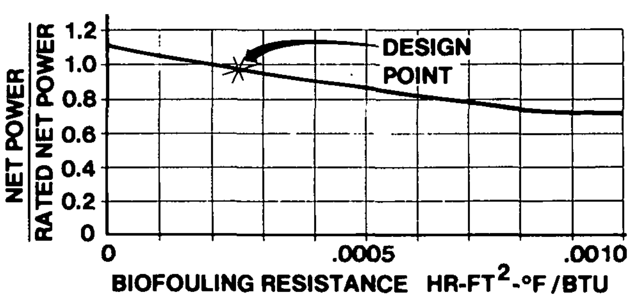

Fig. 5 Biofouling effect on performance.

The model for the plant in Figure 1 generated 157 nonlinear algebraic equations in 215 variables. Fifty-eight variables were then specified (mostly equipment dimensions), reducing the problem to 157 equations in 157 unknowns. OSAP converged these equations in 17 iterations (10^{-7} accuracy) from an initial guess which did not reflect "knowing the solution" before starting. A rerun in which a slightly different set of 58 variable values were fixed converged in 4 iterations. Most other runs gave convergence usually with 7 to 12 iterations. Reruns were typically much faster, as illustrated by the above example.

3. APPLICATION OF PROGRAM RESULTS TO COMPONENT DESIGN

Control System Design

Following the selection of the preferred configuration (Fig.2) with turbine bypass valves for plant efficiency reasons, use was made of the various computer programs to effect the design phase of the OTEC Closed Cycle Power System Controls. The objective was to optimize the system as a whole, rather than to assemble a set of components, each of which was designed on an individual conventional basis, as would typically be supplied by a vendor. The control hardware and software designs and related product design specifications were obtained as a result of optimization and steady-state and transient synthesis and analysis. Use was made of these various programs in supporting computer studies.

Heat Exchanger Modular Design

The evaporator and condenser were designed for optimized power system performance at the design point by considering such requirements and tradeoffs as

- Type of working fluid
- Type of tube material, diameter and wall thickness for evaporator and condenser
- Consideration of tube corrosion and erosion rates
- Effective tube surface area, along with possible use of several different types of enhancement on ID or OD in the evaporator and condenser
- Tube spacing geometry for efficiency of vapor paths in the evaporator and condenser for optimum heat transfer
- Integral hotwells for the evaporator and for the condenser
- Evaporator and condenser seawater flow velocity for controlled pressure drop
- Permissible levels of biofouling on the evaporator and on the condenser heat transfer surfaces
- Choice of seawater on tube side and working fluid on shell side and heat transfer modes (nucleate boiling in evaporator)
- Evaporator recirculation arrangements for optimum liquid loading on tubes
- Condenser venting procedures
- Turbine exhaust flows and bypass flows to condenser (baffle arrangement)
- Heat exchanger design, spacing of tube supports, etc. to preclude fatigue loading of tubes by flow excitation

Steady-State Analyses

Steady-state analyses were performed (1) to map the part load and overload characteristics of the OTEC closed power cycle, (2) to predict the effect on power output of variations in seawater temperature difference, and (3) to calculate the effect of varying amounts of biofouling upon the power output.

Transient Analyses

Transient analyses were performed to size and otherwise design a number of closed power cycle components. The evaporator and condenser hotwells were sized for working fluid inventory transfers with emptying and filling during severe transients, such as emergency trips of the turbine, following full load rejection. The requirements for turbine and generator to withstand overspeed conditions were derived from the same emergency trip studies. Control of overspeeds under these conditions was investigated by transient performance studies of responses to trips with different bypass valve areas, including a fault tree analysis. This fault tree analysis consisted of calculations of maximum overspeeds after turbine trip following full load rejection, for various combinations of failures to act by feed valves and bypass valves. In the OTEC schematic diagram of Figure 2, there are actually four valves in parallel for the turbine bypass, for the feed flow to the evaporator, and for the evaporator recycle flow. This fault tree analysis shows that the maximum overspeed reached does not significantly change until three of the four bypass valve fail to act (see section, Backup Overspeed Protection). Further increases in flow area have negligible effect on turbine flow rate. For this configuration, the bypass valve sizes were selected to provide this additional overspeed protection characteristic. Total wide open flow area of valves was sized to bypass 70%-75% of the total vapor flow from the evaporator at 110% of rated flow. In Figure 10, turbine flow and bypass flow rates are shown as a function of total wide open flow area of the bypass valves. Flows are plotted as a percentage of total vapor flow rate from the evaporator.

Liquid pumps and valves were sized to provide margins for feed flow and evaporator recycle flow of the working fluid. Liquid pipes were oversized to provide low velocities of the working fluid (5 ft/sec at design conditions) to provide exceptionally low flow losses. Liquid valves were of the same size as their associated pipes. With turbine inlet valves discarded to minimize flow losses, vapor pipes from the evaporator to the turbine were greatly oversized to provide low velocities of the vapor (40 ft/sec at design conditions) and hence to create only exceptionally low flow losses. Also, great care was taken to design the seawater passages for low velocity and few changes in direction to minimize flow losses.

Valve Testing

To insure the reliability of the control system valves to respond quickly when called upon, periodic testing of each of these flow control valves should be done while the system is in operation. The number of control valves in parallel depends upon the valve testing requirements, e.g., if four liquid feed valves are used to pass flow through the turbine and one of the valves is closed during testing, approximately 25% of rated flow would be interrupted. It is estimated that there would still be sufficient electrical output to supply auxiliary power requirements without reducing the electrical load to zero. If four bypass valves are used, about 60% of rated turbine flow is maintained when one bypass valve is opened for testing (See Figure 10). It is estimated that there would still be sufficient electrical output to supply auxiliary power requirements, without reducing the electrical load to zero.

Off-Design Performance

Significant impacts on performance would result from variations in seawater temperature and fouling of heat exchangers. Lesser effects would result from part-load and load-cycling operations.

The effect of non-condensible content in the ammonia can be overpowering at a very low level. This sensitive item will be controlled with our condenser venting approach, thereby lessening the effects of non-condensibles in the ammonia upon power cycle performance.

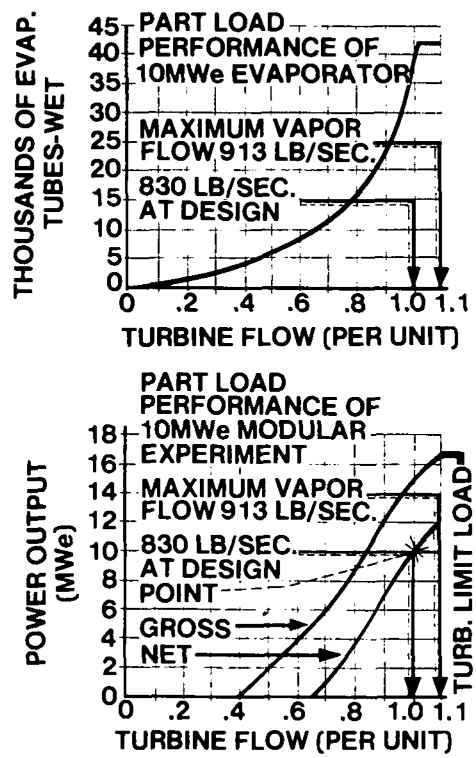

Fig. 6 Part load performance of 10 MWe evaporator, 10 MWe modular equipment.

As warm seawater temperature varies, saturation temperature and pressure change appreciably in the evaporator, but only to a slight extent in the condenser. The temperature differential which may be obtained between the evaporator and the condenser is shown for various water conditions in Figure 3. The effect of biofouling resistance of the evaporator and of the condenser, when the warm temperature is at the design point of 80°F, and the effect of warm water temperature when the biofouling is at the design amount of 0.0002646 hr-ft^2-°F/BTU are given in these figures.

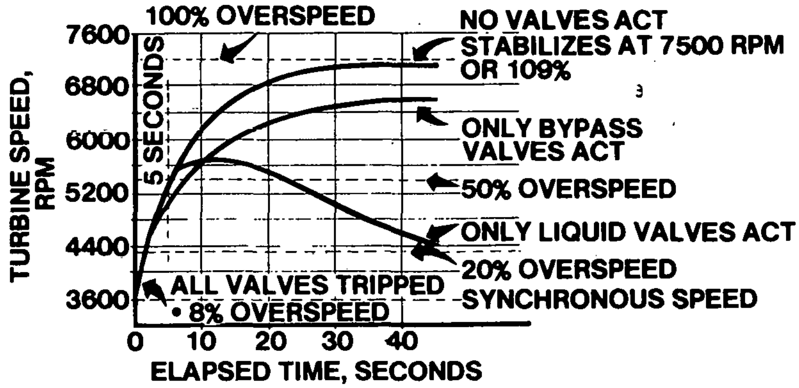

Fig. 7 10 MWe turbine speed characteristics.

This temperature differential is plotted in these two figures against the percent turbine inlet flow for various values of biofouling resistance at design warm seawater temperature, and for various values of warm seawater temperature at constant design biofouling resistance. Superimposed on these curves is the turbine characteristic; the intersection of the evaporator-condenser curves and the turbine characteristic yields the flow obtainable versus biofouling resistance and against seawater temperature, respectively. The changes in feed flow to the turbine are controlled with the feed flow control valve until the feed pump head corresponds to the required flow.

Temperature Differences

Variations of the temperature difference between warm surface seawater and cold deep seawater impact heavily on performance. Figure 4 shows an approximate 4% variation of net power per 1°F variation near the design temperature difference of 40°F.

The number of evaporator tubes that should be wet depends upon the flow to be evaporated to the turbine, the difference between the warm water temperature and the evaporator saturation temperature, and upon the amount of biofouling resistance. The changes in number of tubes to be wet means a corresponding change in recirculation flow to the evaporator. The recirculation flow control valve is therefore modulated until the recirculation pump head corresponds to the recirculation flow required to keep just enough tubes wet to evaporate the necessary vapor flow to the turbine under the specific conditions of warm seawater temperature and of biofouling.

From the flow obtainable to the turbine under these conditions, the turbine power output may be read from Figure 6.

The net power output is then plotted against warm seawater temperature at design biofouling conditions of the evaporator and condenser in Figure 4.

* 2 mils = 0.002 inches

** Turbine pressure ratio = inlet pressure/exhaust pressure

Fouling

The effect of fouling on power output is expected to be less severe than temperature difference variations, since fouling control methods will be utilized.

At the initial or clean condition, incremental fouling has a larger percentage effect than at the design point of 0.0002646 hr-ft^2-°F/BTU; there is a tendency for the effect of fouling to level off (Figure 5). This model considers only the increase in heat transfer resistance due to fouling (which is the major parameter in thin slime films of 2 mils* or less), and it does not consider the effect of increased fluid frictional resistance or wall diameter decreases. When the latter parameters are considered, the slope of the curve will become more negative beyond the design point than the present model estimates.

System Operating at Part Load and Overload

System operation at part load is accomplished by reduction of feed flow to the evaporator. This is accomplished by closing down the feed flow control valve (Figure 2) until the feed pump head corresponds to the required flow. In Figure 6 the flow needed with turbine bypass closed is given versus the gross and net output power for the module. A gross power of zero is reached at about 40% flow, and a net power of zero is reached at about 65% flow with the single stage, opposed flow reaction turbine. In this application, gross power is electric power at the generator terminals and net power is the power to the bus after deducting power for driving auxiliaries.

Load greater than full load may be accomplished, if the seawater temperature difference exceeds 40°F. For the off-design case of elevated temperature of warm seawater, turbine power may be increased above design as evaporator output is increased by increasing feed flow and/or recycle ratio. A power limit is imposed by the feed pump characteristic of 110% flow with the feed flow control valve wide open. When above 105% feed flow, the turbine power increases only with flow because of the limit load effect, where increased turbine pressure ratio** no longer contributes to any increase in turbine power. This is because turbine blading geometry has been optimized at the design point (40°F to condenser and 80°F to evaporator). These nozzles (passages between adjacent blades) are flow limiting at mass flows in excess of the design point, but are not a step function.

Dynamic Response

Analyses of dynamic responses of the OTEC power system to fast actions of the control valves were conducted. One dynamic response was the power-loop-to-valve motion for turbine-generator overspeed protection, following an electrical load dump at rated turbine flow and load conditions. Another dynamic response was the power loop response to a small step in turbine bypass valve area from fully closed to slightly open at rated turbine flow conditions.

Turbine Overspeed Characteristics

When a turbine is used to drive an electrical generator, turbine overspeed protection becomes an important design consideration. With an electrical load, most malfunctions of the electrical system will quickly disconnect the generator to protect the system. This sudden interruption of turbine load results in turbine overspeed.

The following overspeed characteristics are based on computer system simulations.

Figure 7 shows turbine speed versus elapsed time following an electrical load dump at maximum rated turbine flow conditions. Before the trip, the turbine bypass valves are closed and the liquid ammonia feed and recirculating flow valves into the evaporator are open. Three responses to assumed actions taken following the loss of electrical load are shown as follows:

a. No Valves Act

Turbine speed quickly accelerates to a level just below 100% overspeed in approximately 30 seconds -- eventually stabilizing at 109% overspeed. The limiting 50% overspeed level is reached in approximately 5 seconds. Therefore, rapid signal response and fast-acting valves are required to provide adequate turbine overspeed protection.

b. Only Turbine Bypass Valves Act

When the turbine reaches approximately 8-10% overspeed (0.5 second), only the bypass* valves are tripped to their full-open positions. Turbine speed accelerates to approximately 90% overspeed before stabilizing.

c. Only Liquid Valves Act

When the turbine reaches approximately 8-10% overspeed (0.5 second), only the liquid feed* and recirculating* flow valves are tripped closed. Turbine speed accelerates to approximately 60% overspeed. With vapor production stopped, pressure differential between evaporator and condenser decays, reducing turbine speed.

Turbine Overspeed Protection

Figure 8 shows curves of turbine speed versus elapsed time for the normal trip operation and for the redundant trip operations.

From the previous overspeed characteristics, it was shown that neither the bypass valves acting alone nor the liquid valves acting alone were capable of preventing excess overspeed. However, by tripping all valves together, this design limitation can be met.

Normal Overspeed Trip

In the normal overspeed trip situation, the trip signal is simultaneously given to the four bypass valves to open, and to the liquid feed and recirculating valves to close. Redundancy in interrupting the liquid flow into the evaporator is provided by also electrically shutting off the evaporator liquid feed and recirculation pumps.

The bottom curve of Figure 8 shows this simulated operation. All valves are tripped when the turbine reaches 8 to 10% overspeed (0.5 sec. elapsed time). This trip operation limits turbine speed to approximately 40% overspeed, well below the design limitation of 50%.

*Westinghouse designed bypass valve trip from full closed to full open in a quarter of a second after an additional quarter second to sense trip condition and dispatch command function. Likewise the liquid feed and recirculating flow valves trip from full open to full closed in the same quarter second.

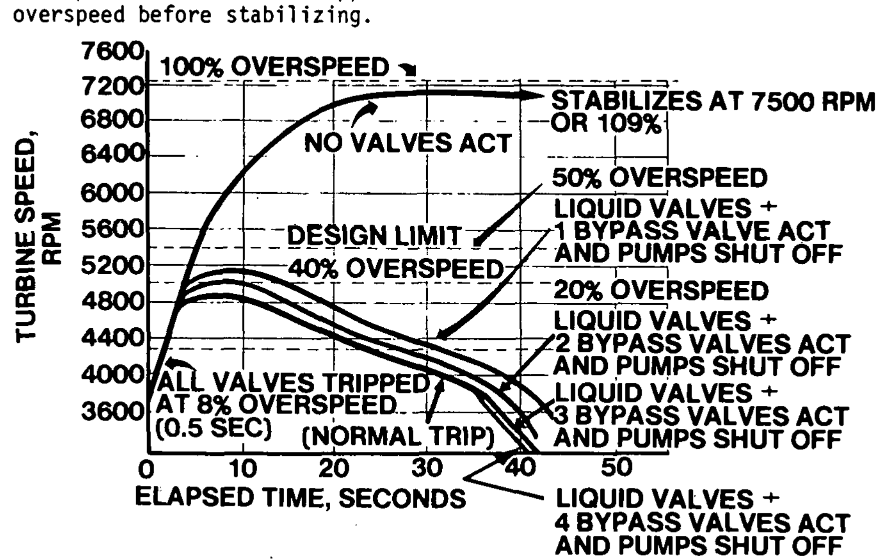

Fig. 8 10 MWe turbine overspeed protection.

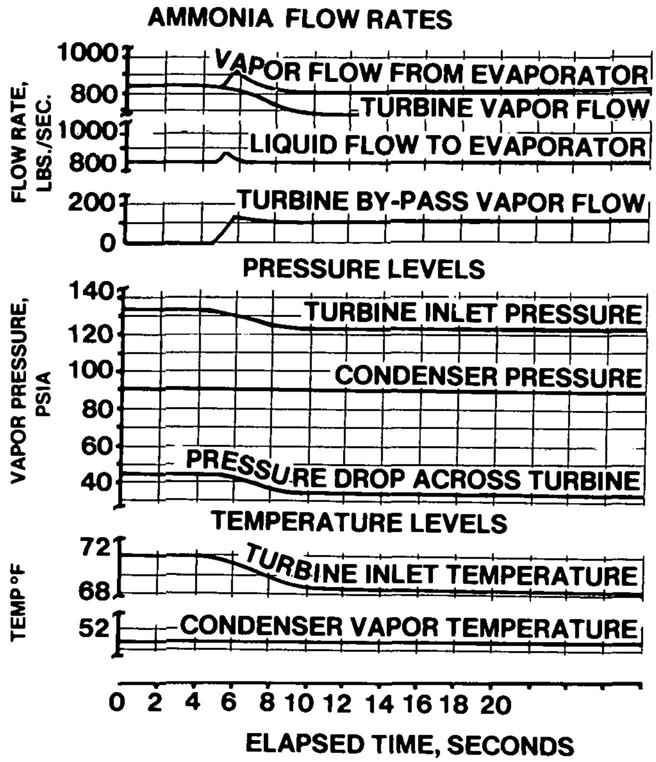

Fig. 9 10 MWe transient system response to opening step function of bypass valves flow area from 0 to 88.9 square inches.

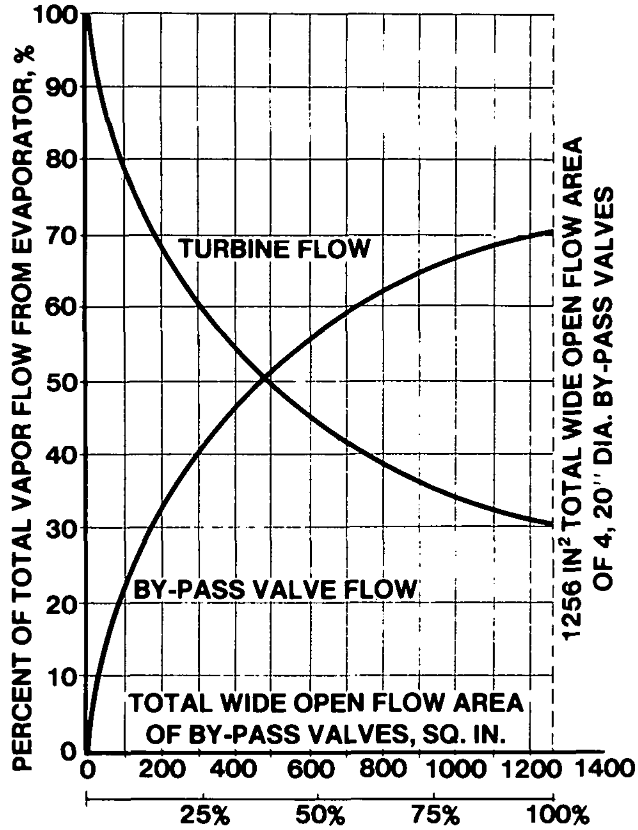

Fig. 10 10 MWe turbine flow characteristics vs total wide open flow area of bypass valves.

Backup Overspeed Protection

The results of a fault tree analysis to determine the effect on overspeed margins, if combinations of bypass valves fail to respond to the trip signal, are also shown in Figure 8.

If three of the four bypass valves fail to act, turbine speed would reach a maximum value of 45% overspeed, still below the design limit of 50%. In figure 7 it was shown that if all bypass valves fail to act (only liquid valves act), turbine speed would reach a maximum overspeed of 60%. This is above the design limitation of 50% overspeed, but still within the mechanical safety margins used to establish this design limitation.

This fault tree analysis shows that the maximum overspeed reached does not significantly change until three of the four bypass valves fail to act. This is characteristic of the parallel flow paths formed by the turbine and bypass valves. As the total wide open flow area of the bypass valves is increased, the effect on total flow resistance approaches a constant minimum value. Further increase in flow area has negligible effect on turbine flow rate. For this configuration, the bypass valve sizes were selected to provide this additional overspeed protection characteristic.

Transient System Response Characteristics

Figure 9 illustrates the transient system response characteristics to an opening step function of the bypass valves. With the vapor flow rate out of the evaporator just below the maximum vapor production capability of the evaporator, the closed bypass valves are stepped open from a total flow area of 0 to 88 square inches.

The step function was arbitrarily chosen to illustrate system response. Actual time response is a function of both the magnitude and rate of change in valve position.

From Figure 9 it is apparent that the following significant effects occur when the system stabilizes following the transient step function:

- Turbine inlet pressure and temperature levels decrease
- Total mass flow rate of vapor out of the evaporator returns to its initial value, but the volumetric flow increases and energy flow rate (BTU/sec) decreases
- Pressure drop across turbine decreases
- Mass flow rate through the turbine decreases

Turbine load, which is a function of both mass flow rate and inlet pressure and temperature conditions, decays.

In Figure 10 is a plot of turbine flow characteristics versus bypass valve area used in the simulations.

Evaporator and Condenser Hotwell Storage Capacities

At stabilized design flow conditions, the maximum liquid ammonia feed flow rate into the evaporator is equal to the design vapor flow rating for the turbine. The change in liquid inventories in the evaporator and condenser hotwells during transients is a function of the differences between the liquid and vapor flow rates in these vessels and the flow response capabilities of the controlling valves. Therefore, sizing criteria for the liquid storage capacities of the hotwells is referenced to minutes of duration at rated turbine flow.

During transient conditions, transfer of liquid hotwell inventories between the evaporator and condenser hotwells will occur in response to varying pressure, temperature, and flow conditions. In general, liquid storage capacities are sized to accomodate these transients conditions and are based on the capability of the liquid control valves to respond to these transient conditions and maintain stable hotwell levels.

Auxiliary Power Requirements

Once having designed the power system components to requirements established from calculation results, consideration was given to auxiliary power requirements for the various operations which are necessary while the turbine-generator is not supplying power. Full and sufficient auxiliary power must be available for necessary operations when turbine flow is less than 65% of rated, which is the point where system power is expected to become zero. This power may be supplied either by an auxiliary diesel-driven electrical generator, or, if available, from the electrical grid to which the OTEC power plant is tied.

These essential operations requiring auxiliary power include (1) the initial purging of air from the cycle by nitrogen and then expelling the nitrogen by the ammonia before start-up and (2) the initial acceleration of seawater pumps and then the pumping of cold and warm seawater and of the working fluid and of lubricating oil during the start-up of the turbine, and also during synchronization to the electrical grid. Also included are (3) the same pumping functions during normal or emergency shutdown of the turbine-generator and (4) purging of ammonia from the system with nitrogen after shutdown of the plant.

4. CONCLUSIONS AND RECOMMENDATIONS

The use of OTOPT to arrive at a cost optimized closed cycle power system was a highly successful effort. As more precise analytic expressions for the performance characteristics of the major cycle components are developed from the recommended additional analytic and/or experimental OTEC related research, additional exercise of the OTOPT program will promote improvements in the baseline optimized closed cycle.

Likewise, the use of the off-design and dynamic simulations revealed sensitive design information on the OTEC Closed Cycle configuration selected (Refer Figure 2). The net power output from the generator was 120% of design at maximum turbine flow capability of 110% of design with 85°F warm seawater. This result was obtained from a sensitivity analysis which indicated a 4% net power variation for 1°F variation of warm seawater.

These studies also revealed that with design levels of fouling and cold seawater temperature constant at 40°F the power system would reach zero net electrical output as the warm seawater temperature drops from the design point of 80°F to a value between 62°F and 63°F. At that point a decision would be required as to the advisability of continuing to operate. This also says something as to the geographic positioning of these plants.

Variations in biofouling analyzed near the design point indicated a 4% variation of net power for 0.0001 HR-FT^2-°F / BTU variation of fouling. These sensitivity analysis results provide the logical base for directing additional OTEC analysis to the most cost effective areas.

The ODSP-3 dynamic simulation provided working fluid inventory considerations to size hotwells, storage tanks, pump characteristics and pipe areas. This allowed another level of precision in further dynamic simulation of the entire Closed Cycle.

Overspeed transient studies were used to discern a specification for the by-pass control valves. This was of strategic importance to the cycle cost and design manpower studies because it revealed a valve design specification for such a rapid responding valve as to preclude commercial availability. In other words, large, custom designed rapid acting valves with a 30 year life are required. The dynamic simulation also determined the number and size of valves in a parallel series network to provide the ability to test (exercise) control valves on-line for RAM (Reliability, Availability, Maintainability)contract requirements.

The dynamic simulation also provided design information for start-up, synchronization, load-change, load dump and shut-down modes of the system.

The successful deployment of closed cycle ammonia plants is directly dependent on design information derived from the dynamic (transient) and static (off-design) calculation programs. Therefore, the prime recommendation is for additional work in the areas of realistic modeling of system and subsystem component performance. Improvements in the functions describing turbine performance, generator losses, evaporator and condenser heat transfer characteristics as a function of film thickness, enhancement and liquid velocity are all desirable inputs. Evaporator performance related to the outside tube surface enhancement requirements for liquid loading is an additional area requiring verification.

Developments in the future could substantially lower OTEC Closed Cycle plant costs through innovative design and the qualification of less costly materials than are currently specified.

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STEADY-STATE AND DYNAMIC PERFORMANCE OF AN OTEC PLANT

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Abstract

The objectives of the computer simulations of TRW's designs for the OTEC PSD-I and PSD-II plants were: 1) determine the steady-state performance of the plants at nominal and off-nominal conditions; 2) design the steady-state control system; and 3) determine the dynamic power response caused by ship's heave. The simulations were constructed of modules that modelled the heat exchangers, turbine, ammonia pumps, water pumps, pipes, valves, and fittings in the steady-state, using a relaxation method. A simpler dynamic program was written, considering only the oscillatory heave motion of the ship and those plant variables affected by ship motion, using numerical integration. The results led to a quantitative understanding of the off-nominal system performance. Control mechanizations were developed for each plant that permit optimum full-power operation and economical part-power throttling of the plant. After being validated by small-scale test data and being updated to reflect new design information, the simulation models will be used to predict the performance of larger plants.

Introduction

Steady-state mathematical models were written for the OTEC PSD-I and PSD-II plants designed by TRW.[1, 2, 3] Both simulations analyze the power output as a function of the states of the water loops and the ammonia loop. The principal difference between the PSD-I and PSD-II plants is that the former uses parallel-flow, falling-film heat exchangers whereas the latter uses cross-flow, plate-type heat exchangers. A simple dynamic model was written that analyzes the transient response of the PSD-I plant.

The objectives of the steady-state simulations were as follows:

1. Determine the steady-state performance of the plant at nominal and off-nominal conditions.

2. Design the steady-state control system. The model calculated the response of the plant to actuator manipulations (valves and pumps).

3. Determine the dynamic power response caused by ship heave.

The simulation results were also used to determine operating policy and to make design decisions, such as the need for variable speed water and ammonia pumps, the economic frequency of tube cleaning, and the static characteristics of the control valves. The plant response to actuator manipulations determines the steady-state control law. Steady-state control is needed to: 1) maintain maximum power as ocean temperatures and fouling factors change and 2) maintain reduced power during start-up and programmed load reductions. There is no need to optimize anything at reduced power since there would be no saving in consumable resources.

*Member of the Senior Staff.

Model Description

The steady-state model includes simulations of all major system elements, organized into subroutines that can be individually modified. A relaxation algorithm determines the steady-state flows and temperatures given the dimensions and other variables. The simulation was written for time-shared usage on remote terminals of TRW's Cyber 174.

The inputs are power-plant dimensions (such as diameters, heights of pipe center lines, the number of pipe bends in each run), pump speeds, valve settings, and inlet water temperatures and water flow rates. There are two output options:

1. The production format is used for routine printout and shows key power outputs, temperatures, pressures, flow rates, and speeds.

2. The debug format prints all system states and is used during the program checkout.

The ammonia pumps and water pumps are modelled by polynomial fits to their head/flow-rate/efficiency/speed characteristics. Because specific vendor choices had not yet been made, typical pumps were used. The ammonia turbine model was taken from one created at the University of Massachusetts with funding from the National Science Foundation.[4] It models the power output, exit enthalpy, exit temperature, and exit pressure as a function of mass flow rate and inlet temperature. The turbine models for PSD-I and PSD-II were sized to deliver maximum efficiency under the nominal flow conditions for each plant. The heat-exchanger models were different in PSD-I and PSD-II. The PSD-I heat exchangers are falling-film, parallel-flow units in which the ammonia flows down the exterior of the fluted tubes and evaporates or condenses. Each tube is modelled as ten axial segments, each one having uniform properties. For each segment, the film thickness, evaporation rate and water temperature are computed. The thermal and mass-flow continuity equations are written among adjacent segments. All tubes are assumed to be identical and the shell space is assumed to be at uniform pressure, thereby neglecting cross-flow effects within the heat exchanger. The water-side heat transfer coefficient is proportional to $(\text{mass-flow-rate})^{0.8}$, $(\text{diameter})^{-1.8}$ and to a polynomial in $(\text{temperature})^2$ that represents the water properties. The ammonia-side heat-transfer coefficient is modelled as a polynomial in (temperature) and as a function of (Reynolds number). The heat exchanger model was developed at Carnegie-Mellon University and is consistent with data taken on falling film heat exchangers at Carnegie-Mellon by Professor R. Rothfus.[5] The PSD-II plant uses cross-flow heat exchangers in which water flows horizontally through tubes while vertically-flowing ammonia condenses or evaporates. The PSD-I evaporator and condenser were single units whereas the PSD-II evaporator and condenser are each divided into four removable subassemblies for ease of repair. The PSD-II model used a log-mean-temperature-difference fit based on a segmented model run at Union Carbide's Linde Division based on their test data with similar heat exchangers.

The pressure drops in all pipes were computed, including losses in bends, valves, fittings, and orifice plates. The PSD-II piping is considerably more complicated than is the PSD-I piping. Height differences among all the elements of the system were modelled in the piping. Heat transfer from the near-room temperature pipes to the environment was neglected. The steady-state model achieves 1% accuracy in power, 0.3% accuracy in flow rate, 0.3% accuracy in pressure, and 0.1°F accuracy in temperature.

An approximate dynamic analysis, whose error is approximately 10%, estimates the response of the PSD-I system to ship heave. The dynamic model consists of a hydraulic model, which calculates the water-flow velocities in the tubes, and an ammonia-loop model which computes the power output versus time. There is no feedback from the ammonia-loop model to the water-loop model. Only gross power output is computed because the pump power fluctuations were not modelled. The hydraulic model includes flow rate and inertia of the water in cold water, warm water, and discharge pipes, and in the heat exchanger tubes. It models the changing water height in the troughs and the resulting head on the heat exchanger tubes. It includes the head-flow characteristics of the water pumps. Its output is velocity versus time in the heat exchanger tube, which is input to the ammonia-loop model. In that model, the heat exchangers are modelled as thousands of thermodynamically identical tubes. Each tube is divided into 10 axial segments, as in the steady-state model, except that heat storage in the metal is included. Heat transfer and mass-flow differential equations are written for each segment that include evaporation/condensation in each segment and transfers to other segments. The transit time of water flow in the tubes is included. On the liquid ammonia side, the model includes sump volumes and pump head-flow-rate curves. The model includes the volume of the shell-space and turbine inlet piping, which attenuates power surges. Other plant time-constants were analyzed separately and found to be less than 5 seconds. The heave-induced oscillations are far slower than the power-line frequency; therefore, the turbine was assumed to be locked to the power grid during heave-induced oscillations and not to have any dynamic behavior.

The model was verified for several hand-calculated cases and was checked against a design optimization model that only considered nominal performance.

Results

The nominal system states for the two OTEC power plants are shown in Table 1: the output power, sea-water temperatures, water flow rates, ammonia temperatures, and ammonia flow rates. Nominal turbine efficiency is 89% in both cases. The PSD-II plant has higher water flow rates and lower heat-transfer coefficients, partly because it has smooth tubes, whereas the PSD-I tubes are fluted on both sides. The nominal fouling factor of PSD-II was higher because its heat exchangers are made of aluminum and are harder to clean than the titanium heat exchangers of PSD-I. The difference between the evaporator and condenser temperature is 20°F for both plants. The PSD-I ammonia flow rate is somewhat higher, and that plant delivers 5% higher power.

Table 1 Nominal conditions for OTEC power systems.

PARAMETER	UNITS	PSD-I	PSD-II
• POWER OUTPUT			
GROSS	MWe	14.6	13.9
NET	MWe	10.5	10.0
• SEA-WATER TEMPERATURE			
WARM	°F	80	82
COLD	°F	40	40
• WATER FLOW RATE			
EVAPORATOR	LB/SEC	76,400	102,500
CONDENSER	LB/SEC	70,500	96,900
• HEAT EXCHANGER			
MATERIAL	—	TITANIUM	ALUMINUM
AREA, EVAPORATOR	FT^2	260,000	246,000
AREA, CONDENSER	FT^2	295,000	246,000
U, EVAPORATOR	BTU/HR-FT^2-°F	896	820
U, CONDENSER	BTU/HR-FT^2-°F	796	623
• DESIGN FOULING FACTOR	FT^2-HR-°F-BTU^{-1}	0.0001	0.0003
• AMMONIA TEMPERATURE			
EVAPORATOR	°F	70.0	72.3
CONDENSER	°F	50.0	51.5
• HEAT ADDED IN EVAPORATOR	BTU/SEC	430,000	410,000
• AMMONIA FLOW RATE THROUGH TURBINE	LB/SEC	807	764

Figure 1 shows the gross and net power output for both plants as a function of cold-water inlet temperature and as a function of warm-water inlet temperature. Both curves are linear over the temperature ranges of interest, specified by the Department of Energy (DOE). The PSD-I curves lie above the PSD-II curves because of the lower fouling factor and higher heat transfer rate. Figure 2 shows the seasonal power profile for the PSD-II plant, based on the DOE-furnished warm-water temperature profile from January through December at a typical site. In all cases, the bottom-water temperature is assumed to be 40°F. Because of the linearity of power output with temperature, the annual mean output power (10 MWe) is the same as the instantaneous output power at the mean temperature (82°F).

Figure 3 shows the power output versus fouling factor for the two plants. For any given fouling factor, the PSD-II curves lie above the PSD-I curves since PSD-II was scaled to deliver 10 MWe at a larger fouling factor than that for PSD-I. The fouling factor curves are virtually linear as a function of warm-water and cold-water fouling. The following equations relate net power output (P_N), warm water temperature (T_W), and fouling factor (R_{FW} and R_{FC}) to an accuracy of 0.02 MWe.

▶ PSD-I

$$P_N = 10.52 + 0.694\,(T_W-80) - 0.693\,(T_C-40) - [0.37 + 0.0088\,(T_W-80)][10^4 R_{FW}-1] - [0.33 + 0.0088\,(T_W-80)][10^4 R_{FC}-1] \text{ MWe}$$

▶ PSD-II

$$P_N = 10.00 + 0.645\,(T_W-82) - 0.645\,(T_C-40) - [0.37 + 0.0086\,(T_W-82)][10^4 R_{FW}-3] - [0.37 + 0.010\,(T_W-82)][10^4 R_{FC}-3] \text{ MWe}$$

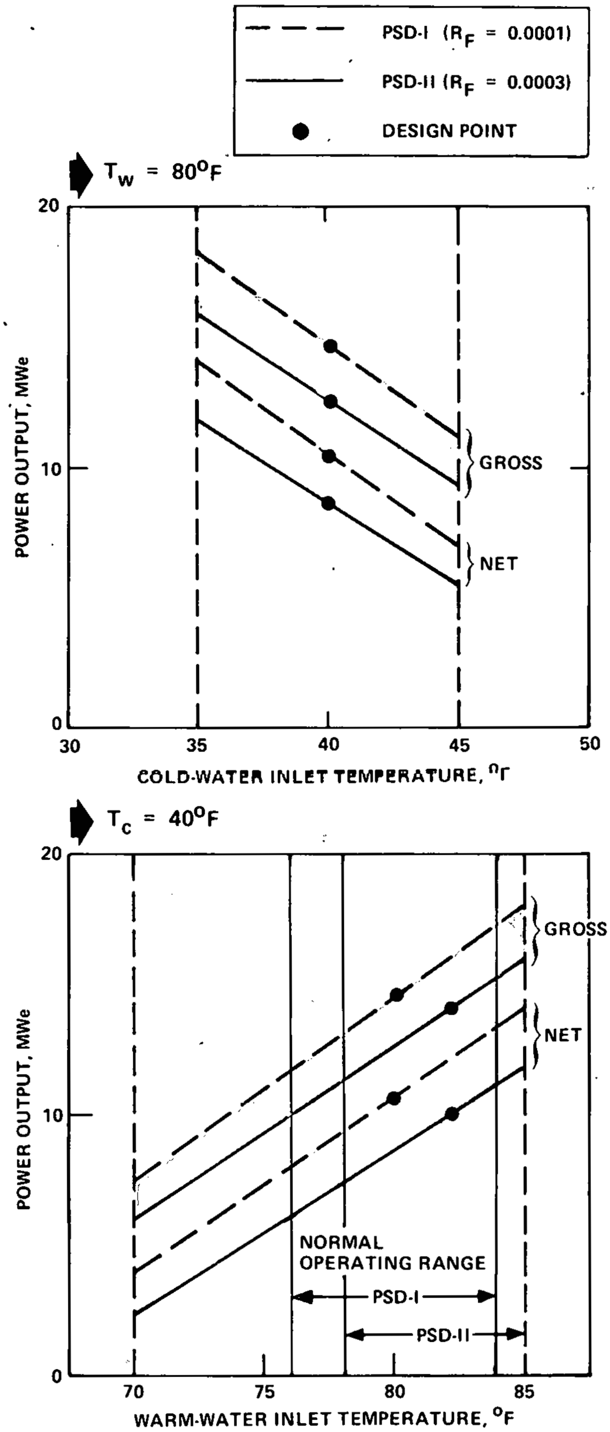

Fig. 1 Power output versus water temperature.

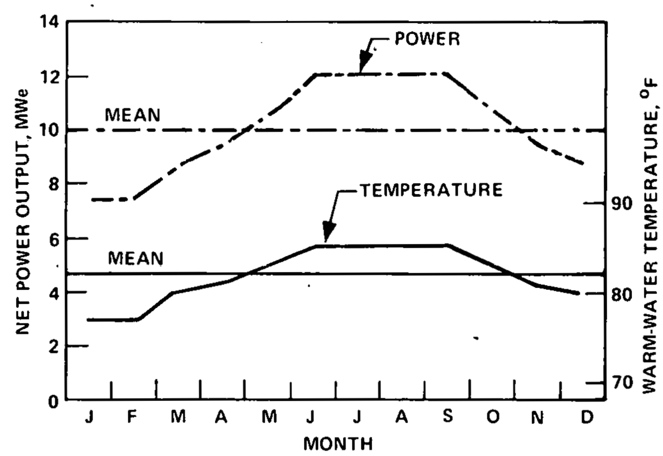

Fig. 2 Seasonal power profile for PSD-II.

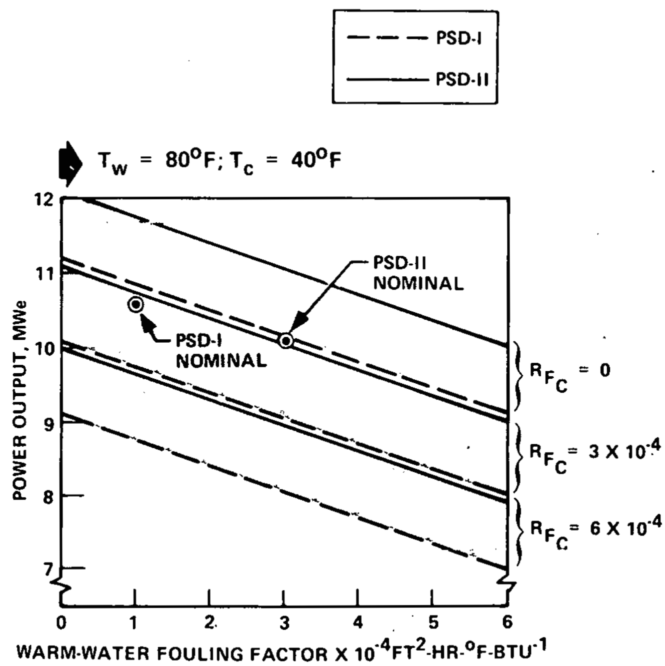

Fig. 3 Power output versus fouling.

The PSD-II nominal fouling factor, 0.0003 ft^2-hr-°F-BTU^{-1}, is three times that of PSD-I because the aluminum tubing is harder to clean, because of the dead spaces between the individual heat exchangers, and because of the larger degree of conservatism adopted in the PSD-II design.

Figure 4 shows the effect on PSD-II of varying warm-and cold-water flow rates. The curves were used to determine the best water flow rates and to decide if variable-speed water pumps are economical. Notice that, at a warm-water temperature of 85°F, the cold-water flow rate is correctly chosen at the peak power point. At 82°F, a slight power increase would be achieved if the flow rate were lowered 5,000 lb/sec. At 78°F, a further increase would be obtained by lowering the water flow rate 10,000 lb/sec. On an annual basis, an average increase of 0.1 MWe would result if the cold-water flow were lowered 5,000 lb/sec. As for the warm-water flow rates, represented by the 21 parabolas in Figure 4, peak power output is reached at the nominal warm-water flow rate, within 0.01 MWe, no matter what the water temperature or cold-water flow rate.

Table 2 shows the effect of removing one of the four heat exchanger subassemblies. When a subassembly is removed, the gap is blocked-off, the ammonia flow rate to it is shut-off, and the water pumps continue to operate at constant speed. The water trough levels rise until a new equilibrium is established at which the higher level in the troughs forces water through the tubes at a higher rate than before. Table 2 shows that with one failure in

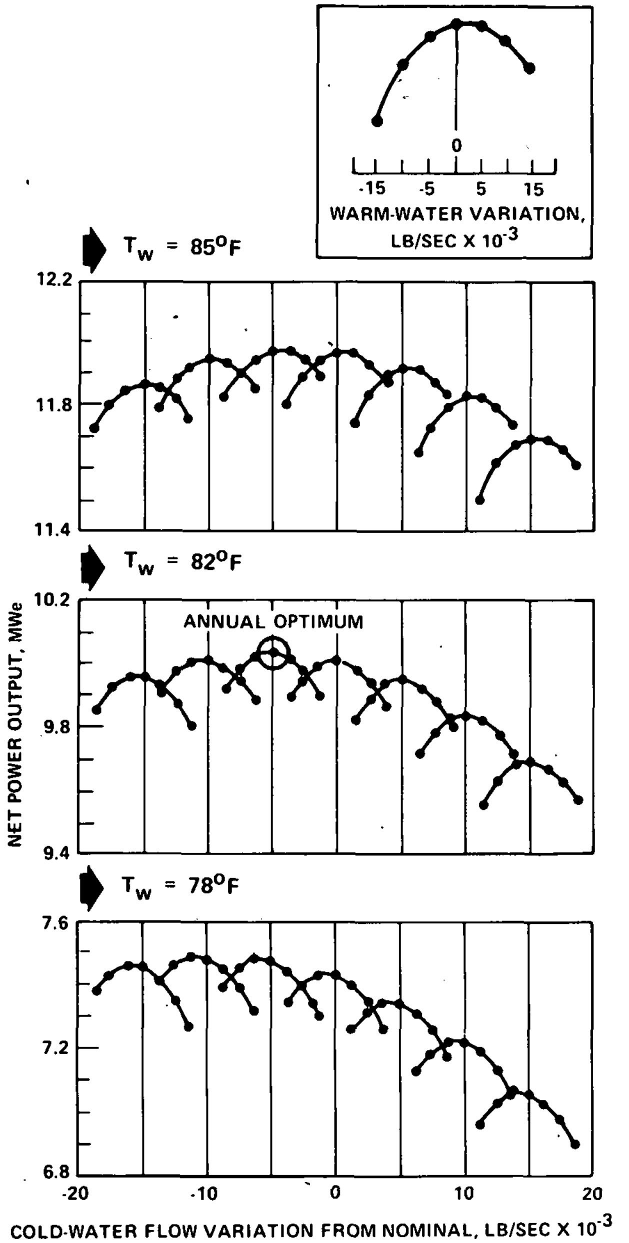

Fig. 4 Water flow effects on power ($T_C = 40^oF$).

Table 2 Plant conditions with failed heat exchanger.

PARAMETER	NO FAILURES (NOMINAL)	REMOVAL OF SUBASSEMBLY: EVAPORATOR REMOVED	CONDENSER REMOVED	BOTH REMOVED
• WATER FLOW RATE				
WARM, LB/SEC	102,500	89,900	102,500	89,900
COLD, LB/SEC	96,900	96,900	88,900	88,900
• AMMONIA TEMPERATURE				
EVAPORATOR, °F	72.3	70.6	72.4	70.8
CONDENSER, °F	51.5	51.1	53.7	53.2
• AMMONIA FLOW RATE, LB/SEC	764	735	753	725
• TURBINE EFFICIENCY, %	89	89	89	89
• WATER PUMPING POWER				
WARM, MWe	1.21	1.34	1.21	1.34
COLD, MWe	1.96	1.96	2.13	2.13
• NET POWER OUTPUT, MWe	10.0	8.60	8.26	7.00

either the evaporator or condenser (removing 25% of the evaporator or condenser area), the power output drops between 14% and 17%. If one evaporator and one condenser are removed simultaneously, the power output drops 30%. The effects are nearly additive.

Figure 5 shows the results of the dynamic analysis. The lower curve on the right shows the water height in the cold water trough as the ship heaves. The height in a calm sea is 5 feet below sea level, because the cold-water pump continually removes water from the trough into the condenser header. When the ship heaves (±10.7 ft at 16.2 second period in sea state 6) the water height oscillates ±1.2 ft at the heave frequency. The upper curves on the right of Figure 5 show the water levels in the evaporator header and condenser. In the absence of heave, the water level in the headers is 10 feet above sea level, driving the water downward through the vertical water tubes and discharge pipe. As the ship heaves, the header levels in the evaporator and condenser vary ±1.1 foot. It was found that the water in the 3000-foot cold-water pipe has a nearly constant flow rate in inertial space so the relative velocity between water and tubes varies ±4 ft/sec as the ship moves up and down on the water column. However, the flowing water in the 300-foot discharge pipe is substantially accelerated by the changing hydraulic head so its velocity relative to the tubes varies only ±1.4 ft/sec. These velocity variations with time cause a varying heat transfer rate.

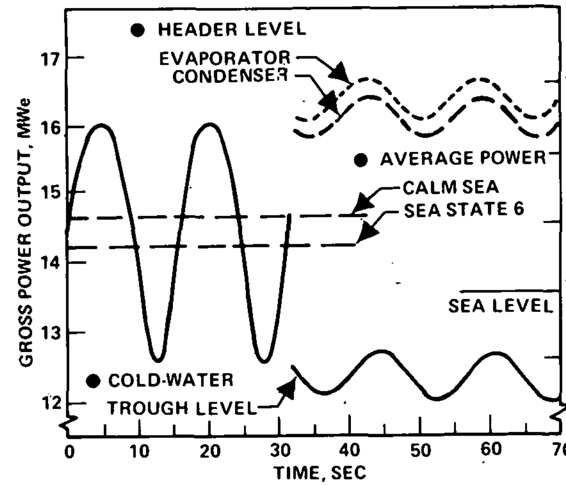

Fig. 5 Asymmetric power oscillations of PSD-I caused by heave.

The gross power output is shown at the left of Figure 5. The power output in a calm sea is 14.6 MWe. As the ship heaves, the power output oscillates, reaching a peak of 16 MWe and a minimum of 12.6 MWe in sea state 6. Because the oscillation is asymmetric, the mean power output is reduced from 14.6 MWe to 14.2 MWe. Of course, in calmer seas, the ship's heave would be much smaller and the variations would drop. if the OTEC plant were connected to an "infinite grid", that has a large amount of generating capacity, the plant owner would probably accept the time variations in power (utilities are learning to accept many unsteady power generating sources, such as windmills, tidal plants and solar plants). On the other hand, if the OTEC plant were an isolated plant (for example one of several small plants on an island), a control system would be designed to attenuate the oscillations to a level acceptable by the owner. The oscillations could be attenuated by controlling to their minimums, which would result in a further loss of average power output of approximately 1.6 MWe in sea state 6. The oscillations could also be attenuated with less loss of average power output by increasing the volume of the heat exchangers, thereby storing the oscillatory energy. That solution was deemed expensive and impractical in the confined space of the OTEC ship.

Oscillations of power output in response to ship heave are believed to be characteristic of all OTEC designs. Their prediction requires more analysis and experiment.

Several steady-state control laws were analyzed for the PSD-I and PSD-II plants. Since no consumable fuels are used, the control strategy is to generate as much power as possible during full-power operation and, when part-power operation is requested, to control the reduced power inexpensively. The selected control actuators and set-point control laws are shown in Table 3. Of

Table 3 OTEC control law.

ACTUATOR	PSD-I	PSD-II
• RECIRCULATION PUMP	FIXED-SPEED	NONE
• RECIRCULATION THROTTLE	$f(T_w-T_c$, DESIRED POWER)	NONE
• FEED PUMP	EVAPORATOR SUMP LEVEL	$f(T_w-T_c$, DESIRED POWER)
• BYPASS VALVE	TRIP	$f(T_w-T_c$, DESIRED POWER)
• TURBINE THROTTLE	TRIP	TRIP
• TURBINE NOZZLES	MAXIMUM POWER	MAXIMUM POWER
• WATER PUMP SPEED	NO CONTROL	NO CONTROL

the seven actuators available for control, four will be actively manipulated to set the power level and turbine speed. This simulation analyzed their sensitivity and interactions in the steady-state. The dynamic control law was not considered in this early phase of design. It is expected that frequency-sensitive filters will be designed analytically and will be adjusted experimentally during plant installation and checkout.

The power output of both plants is controlled by the ammonia feed rate, at and near full-power, and by the turbine bypass during start-up. The PSD-I ammonia feed is controlled with a valve at the output of a constant-speed recirculation pump. The PSD-II feed is controlled by a variable-speed feed pump. Those devices were selected because of simulations of constant-speed pumps and valves on one hand versus variable-speed pumps on the other. Figure 6 shows the net power output of PSD-II as

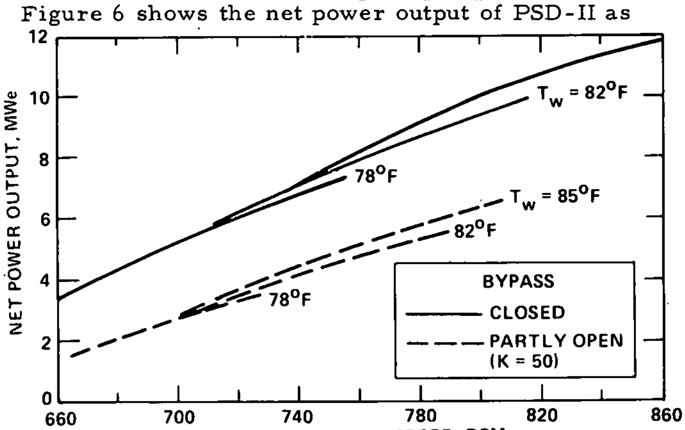

Fig. 6 Variation of net power output with pump speed and bypass valve, PSD-II.

the feed-pump's speed changes. Pump speeds of 600 rpm to 880 rpm cover the full operating range. However, the sensitivity of power output to pump speed is 50 kW/rpm. Thus, if start-up is required at several kilowatts because of limited power available to the water pumps, the bypass control shown in Figure 6 will also be needed. Bypass control is expected to result in more rapid response than is feed-rate control.

When the plants are connected to an infinite grid, the operator is expected to set all valves for maximum power; the grid will maintain the turbine in synchronism and the turbine nozzles will be varied slowly, by computer, to hunt for the condition of instantaneous maximum power. On the other hand, when either OTEC plant is connected to an isolated grid, the variable turbine nozzles will be used to synchronize the turbine and to maintain its speed. To reduce power, the operator or the control computer selects a suitable combination of ammonia feed-rate and turbine bypass.

A turbine trip valve, close to the turbine inlet, is used to reduce turbine-speed overshoot following a load disconnect. The power loss caused by the pressure drop across a wide-open trip valve was found to be less than 0.05 MWe. The added losses (if any) that would result from a series turbine throttle valve (instead of a trip valve) are avoided by using feed rate and bypass for control.

Future Work

Modelling activities will continue as the plant designs improve. The following desired improvements have already been identified for the steady-state model.

1. Improve the heat-exchanger model for PSD-II, modelling each of the 16 subassemblies separately.

2. Improve the heat exchanger models to include cross-flow and to model off-nominal heat transfer more exactly.

3. Improve the integrated simulation to operate at large bypass.

4. Add water pump efficiency as a function of pump flow conditions and speed.

5. Replace the single stage NSF turbine model with a multi-stage model.

6. Revise the subsystem numerical data based on OTEC-1 tests, laboratory tests, and sea-going tests, rather than on analytical predictions.

The dynamic heave model should also be improved. It should include the water pumping power as a function of flow conditions and additional ammonia system lags that may have been omitted from the present model. The model should be validated with test data as soon as possible by observing heaving troughs and long pipes at sea.

A trip overshoot model was not included at this time because trip overshoots are common to all power plants and are not unique to OTEC. The use of a turbine trip valve close to the turbine inlet reduces the importance of such an analysis. At later stages of the PSD-I and PSD-II designs, when details are firmer, an analytical model could be constructed that predicts turbine trip overshoot in the presence of a load disconnect.

Event-timing models and small-perturbation control-system models may also be constructed in later phases. The event-timing model will be designed to estimate the time between events during startup and shutdown, times that could alternatively be determined during the installation and start-up procedures. The small-perturbation model will analytically determine the characteristics of the filters that are needed in the control loops. Such filters could alternatively be designed empirically during plant start-up. The de-coupling of multiple measurements into the multiple actuators for set-point control is determined from the steady-state dynamic model.

Models are regarded as the principal tool to extrapolate OTEC designs from small plants to large plants.

Acknowledgments

The writer wishes to acknowledge the help of R. L. Barkley, Jr. and K. H. Moh of TRW, who integrated the simulation programs and analyzed the data; G. Gibson of TRW, who coded the hydraulic model; P. J. Bakstad of TRW, who provided guidance on the OTEC design; Professors A. Impink and K. Preston White, Jr. of Carnegie-Mellon University, who wrote the falling-film heat exchanger model; Professor L. Ambs of the University of Massachusetts, who wrote the turbine model; and R. T. McLaughlin of ENWATS, who wrote the hydraulic model.

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DISCUSSION

Question: Did you include biofouling in the calculations of the variable speed pumps where you varied the water flow rates?

M. Kayton: No. We looked at those curves at fixed biofouling numbers of 0.0001 to 0.0003 hr-ft^2-^{o}F/Btu. We could have run another whole picture like that for each of several biofouling coefficients and run an economic average over biofouling knowing or at least guessing at what rate the biofouling would build up, and at what biofouling number we would clean it and start all over again. We did not do that. We just ran the curves for the two averages, 0.0001 and 0.0003.

D. Richards, APL: The response ± 10% would indicate that your system is reacting directly to the heave. I suppose this was with this specific platform, which would indicate that the platform was heaving very significantly.

M. Kayton: Yes. We used the DOE-provided heave amplitude and period for the platform. Not being Naval architects, we went through a lot of soul searching on what the heave motion of the platform should be, and, in the end, we used the numbers furnished by DOE: ± 11 ft roughly, in sea state 6.

D. Richards: Well, in relating to heave, I am very interested, because we are saying the maximum operating heave of the APL platform is ± 2 ft. The heave effect on the cold water pumps will be reduced by the suction pool arrangement.

G. Dugger, APL: The point that Dennis is making is that on the APL platform, which is the one prescribed for 5- or 10-MW_e, PSD-II power module analyses, there probably will be considerably less heave effect, and hence smaller power oscillations than you indicate.

Question, Lockheed: In respect to the sensitivity to flow rates of warm and cold water — we compared our cases with your PSD-I studies on the sensitivity, and we found generally that we were in agreement, I think you looked at a much larger range of sea state variation and warm water temperature than we did. Would you comment further on your analysis to see whether a variable-speed pump would be cost effective?

M. Kayton: For the purpose of the economic analysis, we looked at the cost of the control system for the motor driving the pump, and somebody just got a standard control system out of the catalog and a variable-frequency control scheme of some sort to vary the motor speed. He said, how much will that cost? From these numbers, I calculated for him how much average power output over the year would be if he controlled the speed to follow the optimum function of water temperature, and the answer is you don't save enough power to pay for the capitalized cost of the pump and the variable pump speed controls, either cold-water pump control, warm-water pump control, or both. None of those three options pays for itself in our opinion. However, one time at which you might need variable speed is at start-up, depending on how you work the plant, and that is the subject of a whole chapter on in our write-up. You might need a variable-speed water pump just to get the plant started; you might not. You do not need it in the steady state, and I do not think it is economically attractive.

OFF-DESIGN PERFORMANCE AND CONTROL CONSIDERATIONS FOR AN OTEC PLANTSHIP

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Abstract

An Ocean Thermal Energy Conversion (OTEC) plant will have to operate over a range of temperature differences due to seasonal variations. A computer simulation program has been developed for the closed Rankine cycle system including a detailed simulation of the two-phase-flow, folded-tube heat exchangers being developed by APL. For a specified heat exchanger geometry, and with algorithms to represent turbine performance and the various fluid dynamic losses and elevation effects in the system, the program can be used to determine the ammonia and seawater flow rates which yield maximum net power for design-point seawater temperatures. Having found a geometry which satisfies other constraints related to integration with the plantship hull to achieve minimum system cost while including reasonable physical arrangements for handling the heat exchangers, etc., the program can be used to compute off-design performance. It is also desired to determine the best method of controlling the system to achieve the best performance at the off-design conditions, again without making choices that lead to unnecessary or excessive cost for the turbine, the seawater pumps, or the ancillary equipment. The limited work to date has been directed to the requirements for a cruising 10/40-MW_e pilot plantship. It indicates that a variable-admission turbine and an evaporator-inlet-flow control will give a suitable basic system. In the concept described ~~here~~ for demonstrating only the basic power plant's performance in the cruising mode, switched resistor banks would be used to dissipate the generated power. ~~Ongoing work will address requirements for onboard ammonia production (from seawater and air as the H_2 and N_2 sources) or electricity transmission to shore by undersea cable.~~

Introduction

The development of a 10/40 MW_e pilot plantship has as a primary objective the demonstration of OTEC power generation. Important supplemental objectives include the attainment of specific data on equipment sensitivities, off-design performance, and related dynamic effects and control requirements. To develop some insight into these issues an analytical performance program has been developed which computes the heat transfer and flow characteristics of the APL folded-tube heat exchangers and power system that has been designed for use on cruising OTEC plantships. While the development of the computer program was intended to optimize the system design by varying the system geometries and attendant losses including auxiliary power requirements, it has also permitted computation and assessment of off-design performance and has provided an insight into overall control requirements.

This paper describes the design of the power system and its sensitivities to variations in operating parameters.

It must be noted that the power system performance reported herein was based upon a pilot plant conceived as a 10/20 MW_e(net) power demonstration plant utilizing resistor banks for load dissipation, and that preliminary evaporator test data on the folded-tube heat exchanger[1a] indicate significantly better performance than predicted here and in Refs. 1b, 2 and 3.

Power and Control Systems

Power System Design

The APL OTEC power system utilizes a closed Rankine thermodynamic cycle with anhydrous ammonia as the working fluid. The thermal driving potential is developed from warm surface water and cold deep water pumped up via the cold water pipe. System design is modular, based upon a 5-MW_e power module consisting of two evaporators, two condensers, a turbogenerator, a cold water pump, a warm water pump, and ancillary equipment. The heat exchanger design is the APL folded-tube shell-less concept.

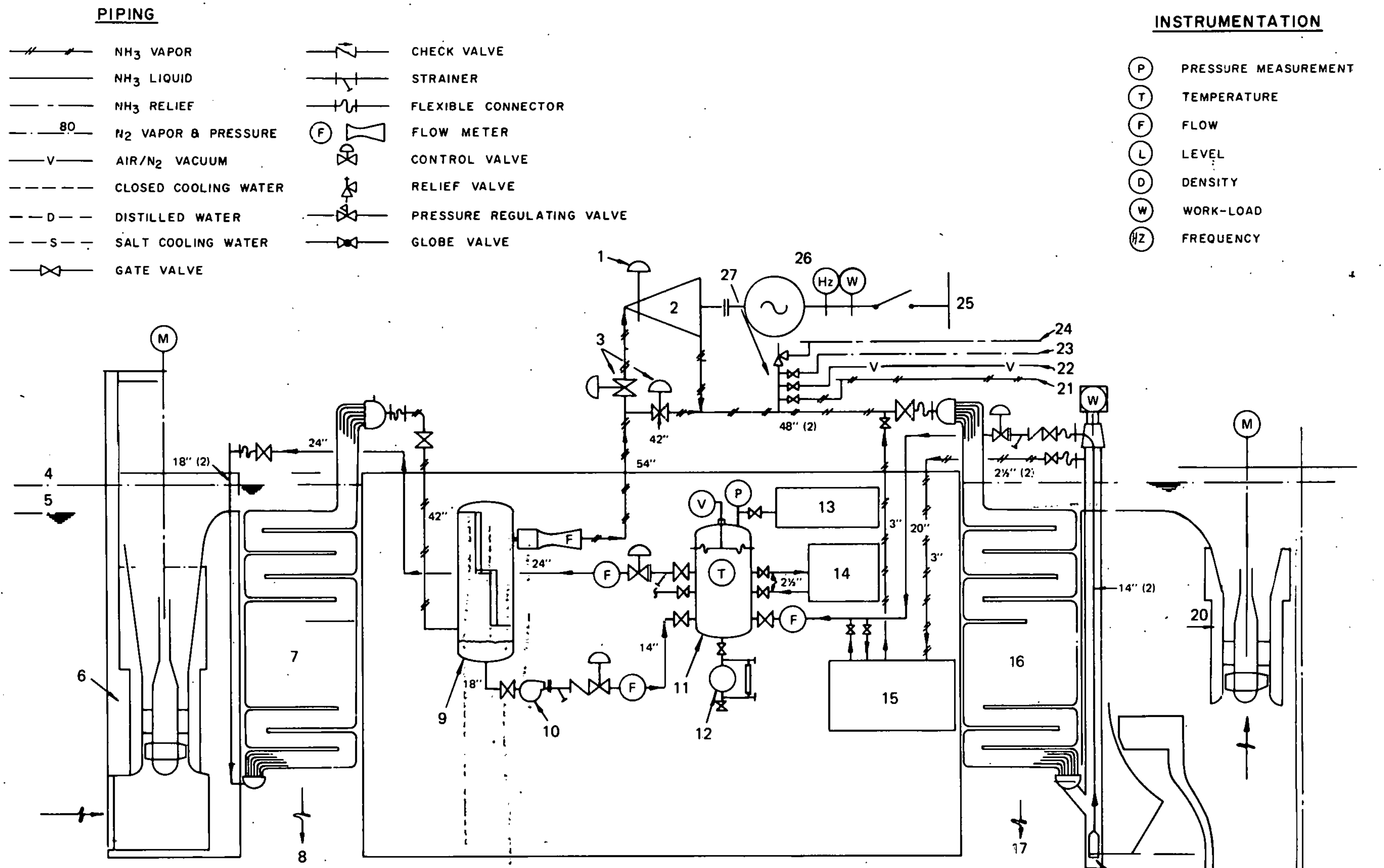
The power cycle is depicted in Fig. 1 with the essential instrumentation and controls required for pilot plant evaluation. For design-point operation, warm water at 82.3°F is pumped to head ponds over the evaporators and then flows by gravity through the evaporators. Cold water at 39.3°F is pumped (by pumps mounted above the cold water pipe) to head ponds above the condensers and flows through them by gravity. Liquid ammonia is admitted to the evaporator inlet manifolds at the bottom of the evaporators at 57°F and is distributed to the multiple tubes comprising the heat exchanger. Available test data indicate that the flow will be stable with equal distribution among tubes.[1a]

From computed heat transfer and ammonia pressures, evaporation will begin in the first or second horizontal pass and with each subsequent pass the ammonia temperature and percentage vapor (quality) increases until at exit from the twenty-fifth pass the vapor quality is approximately 70 percent at 69.8°F.

From the evaporator, the wet vapor flows to a separator where the liquid is removed. The separated liquid ammonia is recycled to the system via a recycle pump, mixing with the condensed ammonia in the pressure control tank.

The ammonia turbine is s synchronous machine with nozzle inlet vanes controlling frequency, and with generator load controlled to match the available turbine power. The turbine trip and bypass valves shown are necessary in that admission control design is not required to have shut-off capability, and the bypass can incorporate a step load change and startup/shutdown control function as well as permitting cycle tests and checkout prior to power generation load calibration and rotating equipment checkout.

After expansion in the turbine the ammonia vapor is exhausted to the condenser inlet manifolds at



1 Turbine admission control
2 Turbine
3 Turbine trip and bypass valves
4 Ponds level
5 Sea level
6 Warm water pump
7 Evaporator (2)
8 Warm water out
9 Moisture separator
10 Recycle pump
11 Pressure control tank
12 Settling drains tank
13 N_2 pressure control
14 Volume control
15 Non-condensables purge
16 Condenser (2)
17 Cold water out
18 Condensate pumps (2)
19 30 foot diameter CWP
20 Cold water pump
21 NH_3 evacuation
22 Air/N_2 vaccum
23 N_2 purge
24 Relief main
25 Bust
26 Generator
27 Typical between all valved connections

Fig. 1 Basic 5 MW$_e$ (net) power module cycle.

50.2°F. The condensate flows from the manifolds to sumps containing "well" type pumps which pump it to the pressure control tank where it mixes with the recycle liquid before passing to the evaporators to repeat the cycle. The liquid level in the condenser manifolds is controlled to maintain a positive pump suction head to prevent cavitation and to provide for removal of non-condensable gases.

The pressure control tank, besides acting as a mixing tank for the condensate and recycle liquids also acts as a surge tank to absorb slight liquid volume changes with changing system operating conditions. Excepting step load changes or equipment shutdown, system operating conditions will change so slowly that a small accumulator volume will suffice, and will provide a signal to transfer ammonia from the system to storage or admit ammonia from storage. The pressure control tank's nitrogen charge will be determined from the system operating conditions and will provide the coarse control on evaporator inlet pressure. The evaporator inlet control valve will provide the fine control of evaporator inlet pressure and flow. In the event of load or equipment failure, a system shutdown would occur and system pressures would equilibrate.

The seawater pumps required are of large capacities and low heads, leading to low-speed, axial-flow designs. Water flow variation can occur from dynamic inlet head variations at the pumps due to wave action and platform motion. For the warm water pumps, the primary factors are the relative sea pressures and evaporator pond level change with water flow rate at the ships station location, while the cold water pumps are primarily affected by platform heave relative to the cold water pipe contents. The sea state transfer function calculated for the pilot plant indicates that platform motion will not be significant except above sea state 5; however, operating load oscillations may require variable pitch vane controls and severe sea states may necessitate blade feathering with shaft locking.

The OTEC electrical system has full controls and instrumentation at switchgear, switchboards and motor controllers to accomplish the normal operating and equipment protection functions required. All unit controls are connected to the central control-room console to permit one operator to monitor and control all system startup, shutdown and loading operations.

OTEC startup is accomplished via the tie feeders from the platform thruster bus, with initially one cold water pump, one warm water pump, and the ammonia condensate and recycle pumps started in series. The maximum (5-MW module) OTEC-startup load is approximately 2.8 MW, assuming seawater pump inrush current is limited to <2.3 times full-load current; well within the platform's total diesel generator capabilities. After ammonia vapor flow is established by the turbine bypass, the turbine generator can be started and when operating speed and synchronization are attained, the generator can be connected to the bus and can assume the loads via programmed computer control.

This preliminary design of the OTEC pilot plant does not include a product demonstration plant or any useful power utilization from the OTEC generators other than platform and thruster load and the OTEC auxiliary (parasitic) loads. To attain the pilot-plant objective of demonstrating power generation and control capability, a water-cooled resistor-bank system is provided which can absorb loads up to 10 MW. When the power system generates up to 12 MW_e(net) in February (warmest surface water at ATL 1) the thrusters at 1/2 knot and ships service power will absorb the excess. Additional resistor banks can be added to increase the dissipation capability, if additional power systems are added.

Power Cycle Performance Program

A performance program has been developed to analyze the performance of the platform-integrated OTEC power plant system with the APL heat exchanger. The current program is based largely on the code described in Ref. 4 with a major change in the solution algorithm: the prior code developed the required geometry of the heat exchangers to attain a specified net power output, whereas the current code determines the net power output based on a specified geometry. This new approach also is better suited to the determination of off-design performance of the OTEC power plant and to the investigation of related control system requirements.

In developing the computer code, the primary goal was to have the program applicable to investigate geometric changes in the OTEC plant. Toward this end a high degree of generality has been incorporated and an attempt has been made to account for all the identifiable losses. Some of the features of the program are:

- Provision for losses in piping, external to the evaporator and condenser modules, via dynamic head loss coefficients.
- Propulsion power estimates have been made a function of drag coefficients and areas of the ship and pipe with an input propulsion plant efficiency.
- The head loss for a specified pipe depth has been made a function of a typical density profile.
- Seawater and ammonia properties have been revised to currently accepted values.
- Provision for machinery (e.g., turbine, separator, manifolds, etc.) elevation changes on the cycle performance have been incorporated.

Figure 2 shows the 23 stations that have been designated around the cycle where the state variables are defined. The program requires some 125 input variables that describe the power plant geometry and key operating conditions. The turbine is entered as a fixed-speed, constant-area device that controls mass flow in the cycle based on continuity considerations, and the solution algorithm proceeds as follows: the pressures upstream and downstream of the turbine are estimated and the mass flow through the turbine is then computed and the turbine exhaust is adjusted to yield a fully condensed state within the last 2% of the length of the condenser tubes. The ammonia pump pressure ratio is estimated as is the state of the recirculated fluid. The evaporator inlet pressure is then adjusted to yield the desired evaporator exit quality and turbine mass flow. The algorithm proceeds around the cycle until the pressures upstream and downstream of the turbine and the mass flow converge. The hot and cold water linear temperature profile is adjusted within each iteration based on the amount of heat transferred in the heat exchanger modules. The computer code is written in PL/1 for the IBM-360/91 and requires 130K bits of storage.

The major design-point conditions, including the seawater flows to the evaporators and condensers

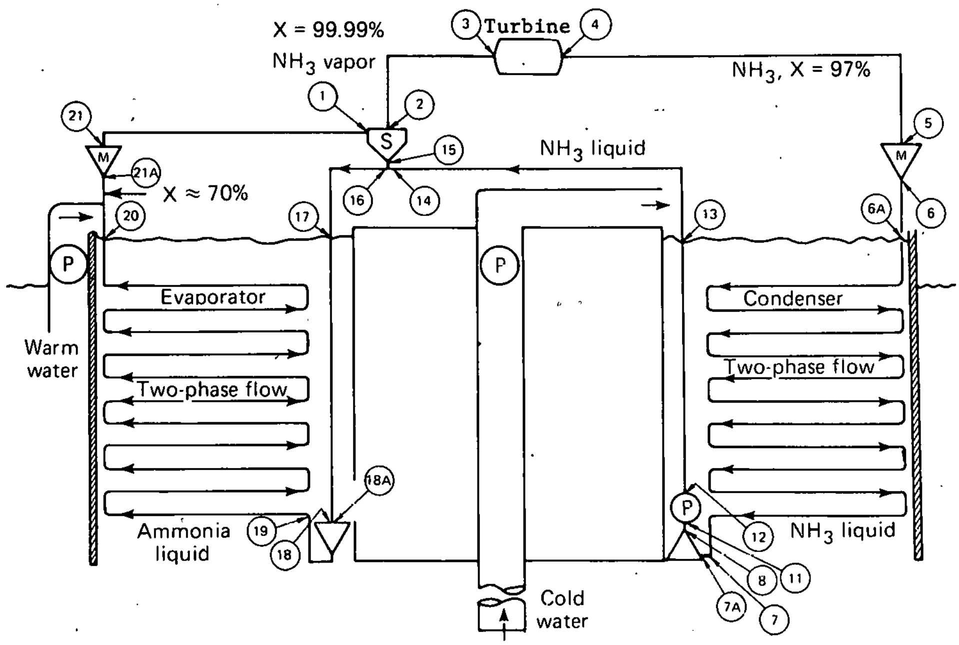


Fig. 2 Schematic of OTEC plant described for computer code.

Table I

OTEC 10-MW pilot plant off-design operating characteristics
Cold water temperature = 39.2°F

Warm water temperature, °F	84.2	82.2	76.2	71.2
Turbine-generator power, MW	15.28	13.76	9.49	6.14
Warm water flow, K lb/s	92	92	92	92
Cold water flow, K lb/s	95	95	95	95
Sea water pumps power, MW	2.856	2.856	2.856	2.856
Ammonia pump (4) power, MW	0.438	0.408	0.286	0.216
Aux. power, MW	0.155	0.151	0.144	0.135
Net electrical power, MW	11.831	10.345	6.206	2.933
Ammonia flow (lb/s)				
thru evaporator (4)	1262	1241	1032	917
thru turbines (2)	886.	850	731	610
Evaporator exit quality, %	70.2	68.5	70.8	66.5
Turbine				
inlet pressure psia	129.7	125.9	114.6	106.0
inlet temperature, °F	70.4	68.7	63.5	59.2
efficiency, %	88.0	88.0	87.6	82.8
exit pressure psia	90.0	89.3	87.1	85.0
exit temperature, °F	50.5	50.0	48.8	47.5
exit quality, %	97.4	97.6	98.1	98.6
Weight of ammonia, K lbs				
in evaporators (4)	339	345	346	370
in condensers (4)	236	236	236	236

which are kept fixed, are listed in Table 1. The design value of warm water temperature in Table 1 is 82.2°F. The off-design performance at each of three other warm water temperatures is also shown and is discussed later in connection with Fig. 5. Again, these values are subject to re-optimization as a result of greater experimental performance of the heat exchangers.

Plant Control

The high level operating logic is illustrated in Fig. 3 which depicts the interactions of the various component control units. A turbine-power profiler receives a primary signal from a software program based on the warm and cold water temperatures, indicating the theoretical power possible and

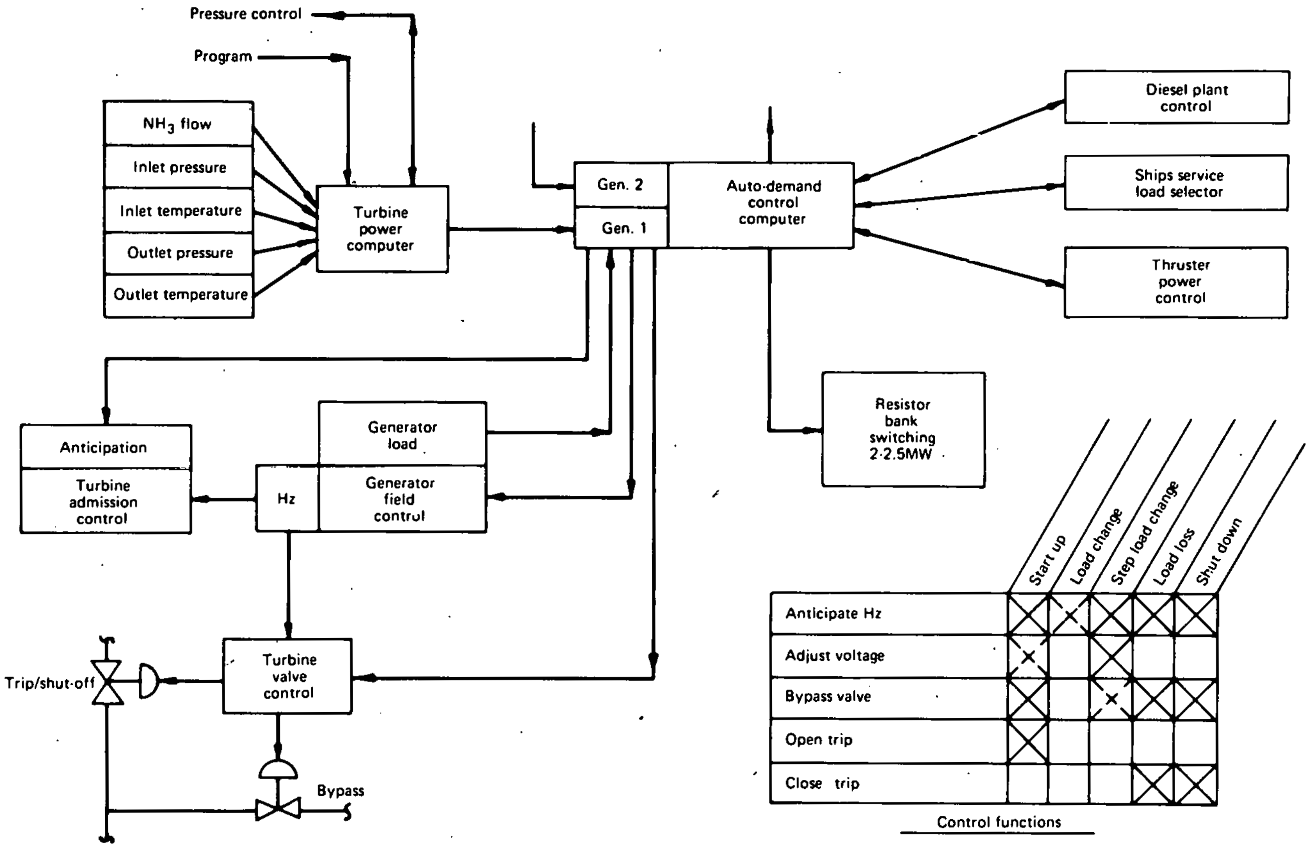


Fig. 3 Signal-flow diagram for turbine-generator controls.

computes the power available from the ammonia flow and turbine inlet and outlet temperatures and pressures. The computer seeks to optimize the power available via iterative interaction with a pressure program estimator, described below, to attain the maximum evaporator performance. An output signal indicating available turbine power is transmitted to an auto-demand-load-control computer which compares the available power with the generator load and maximizes the generator loading to the extent possible. This computer is required not only to perform as a basic load-control device, but also to balance the loads between the generators on line, and through the individual generator sections to provide turbine-generator supervisory control and load-change anticipation. The generator load is provided by resistor banks and by bus ties to the thrusters; thus the functional load-control requirement includes summation of individual turbine power vs. generator loads, and switching of resistor load banks with compensating modulation of thruster power to preset thruster power control limits. Coordinated control interlocks are required with the platform diesel generators control to provide for startup and load assumption by the OTEC generators, and diesel generator startup and load assumption on OTEC plant shutdown.

Additional load-control functions that may be beneficial or necessary upon detailed system dynamics analysis are: variation of generator voltage by up to (as much as) ± 5%, which would vary the resistor bank load by up to (as much as) ± 10%; and turbine bypass valve modulation on a large load drop. A turbine admission control anticipation feature is assumed to be a requirement to control operation of the turbine shutoff and bypass valves on startup and shutdown and is to be coordinated by the turbine-generator designer and controls manufacturer. The turbine-frequency-governor control would, as in normal practice, trip the shutoff and open the bypass valves on an overspeed condition of approximately 10 percent. The turbine-generator controls can be adaptations of presently available commercial mini-computers and auto-demand-load controllers to the specific requirements of the pilot-plant power module.

Figure 4 is a block diagram of the signal flow and processing to control the power system volume, flow and pressure control as discussed above. The overall system pressure and flow controls are not anticipated to require fast response or to respond to significant high-rate load swings, due to the long system flow and thermal lag times, and as discussed in the ammonia power cycle, excepting turbine step-load change or load loss which must be met by turbine-generator control response, the system dynamics are inherently mild due to the heat exchangers' pressure sensitivities. As shown from the system sensitivity study, turbine admission control operation affecting both vapor flow and evaporator exit pressure will react to change the evaporator exit quality directly and the total exit flow rate to a lesser extent. The evaporator-inlet-flow-control valve will respond, within any anticipated dynamic response requirement, to control the system flow to the optimum requirement at the new condition. Design of the overall control system as proposed will require complete dynamic modeling including the turbine-generator characteristics and computed flow time constants and heat-exchanger data from large-scale tests.

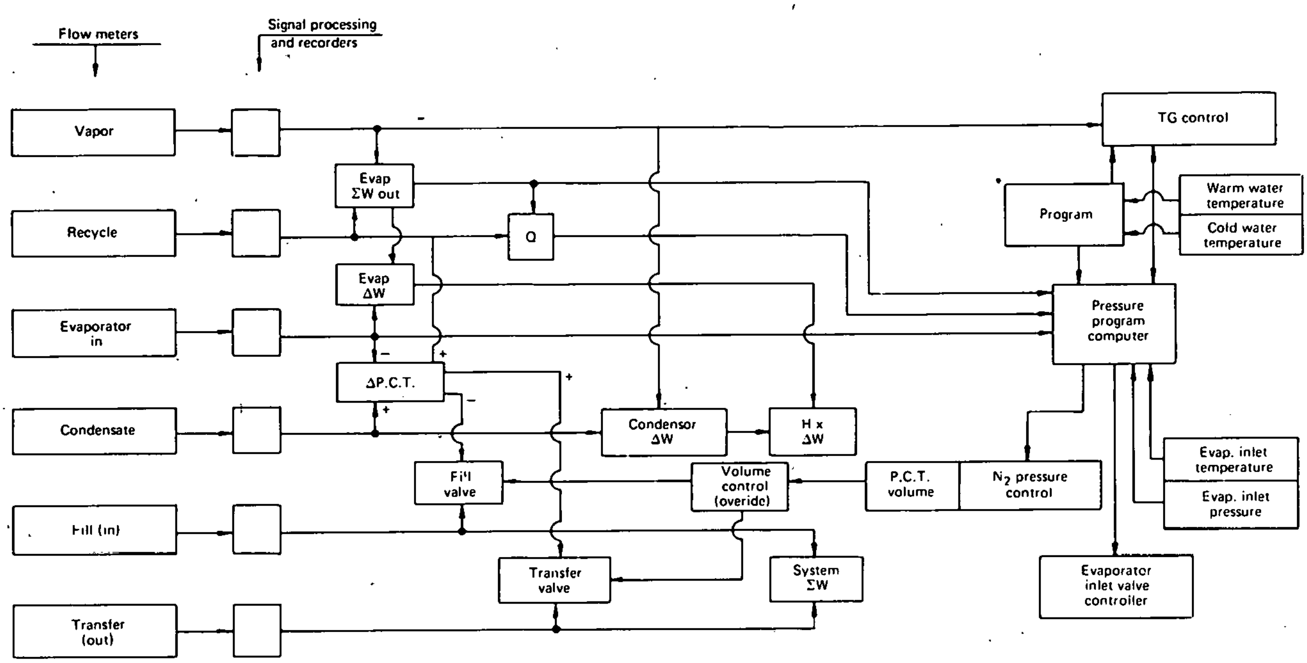


Fig. 4 Signal flow for flow metering, volume, and pressure control.

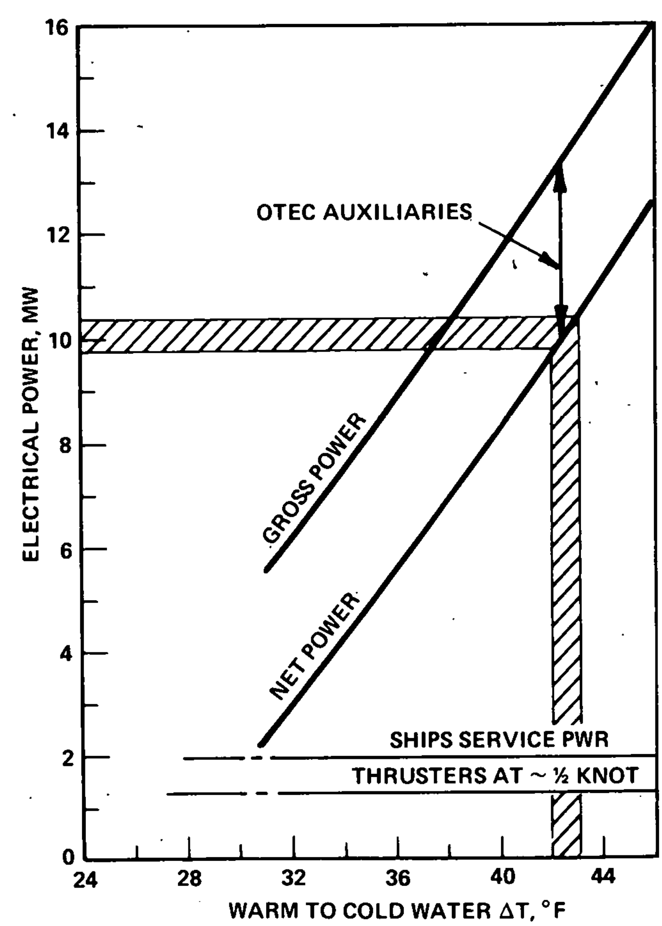


Fig. 5 OTEC 10-MW_e (net) pilot plant estimated power versus ΔT. Note that the improved performance measured in the tests of the evaporator at ANL indicate substantial improvements over this performance.

In addition to the primary volume, pressure, and flow-control parameters furnished by the flow meters, summation of the signals will provide data indicating the ammonia weight in evaporators and condensers, total system content, and rates of change. In addition, within assumed flow meter tolerances and bandwidths, recorded time observation of relative drift can be used for indication of ammonia leakage.

Off Design Conditions

Power System Performance

Estimates have been made of the power system performance both at nominal and off-nominal conditions, and of the system control requirements using the performance model of the OTEC power system described above. The off-nominal performance variations included variations in hot and cold water temperatures as well as variations in the ammonia system.

The gross power available from the OTEC plant is a strong function of the warm-to-cold water temperature difference (ΔT), while the parasitic power, because of fixed equipment size and fixed seawater pumping power, is not affected proportionately. In the present design the net power available at the busbars varies approximately as $\Delta T^{2.5}$; thus, in the grazing concept the plantship cruises to stay within the regions where the maximum ΔT is predicted to be available for maximum power generation. The variations of gross and net power output with ΔT are shown in Fig. 5. Figure 6 shows estimated net power generation throughout the year for the Atlantic-1 siting area east of Brazil where the annual average ΔT has been predicted to be near 43°F and where OTEC plant outage from weather conditions is minimized.

The ships service power plus the thruster power required for grazing for the present platform design is estimated at 2 MW_e, which would reduce the

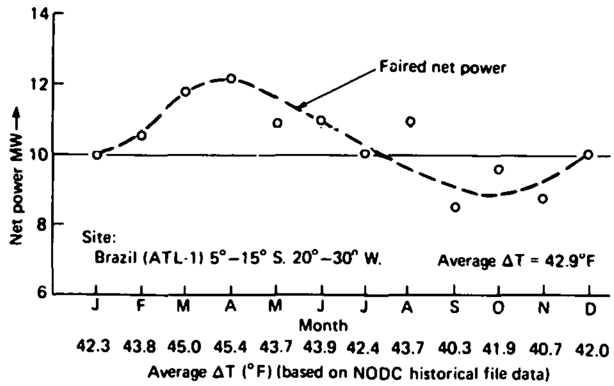


Fig. 6 Predicted power output of cruising modular experiment platform with two 5-MW_e (net) power modules.

net total platform power available. However, these requirements will not increase significantly with greater OTEC installed capacities, and for commercial plants may be < 1 percent of the net power.

Power control and utilization are anticipated to be well within normal industrial practice and capabilities for plants producing energy intensive products. With generator loads controlled to match the power capability and control margins required, via matching the product load, the maximum utilization of the thermal resource will be made.

Sensitivity Studies

An initial step in the design of the process control system is to develop a regulatory control subsystem to maintain system operation at a desired set point. Commonly in the design of conventional boiler-turbine power plants, the nonlinear equations describing the plant, if known, are linearized to yield a model valid for small-perturbation analysis. Alternatively, suitable dynamic models can be developed for a plant from fitting a model structure to experimental data. Data from test programs including the recent data for the APL heat exchanger will be incorporated into the development of these empirical models as they become available.

Information on major system sensitivities is useful in establishing the requirements for set-point regulations and assists in establishing system stability. To estimate these sensitivities, the evaporator module and the condenser module used in the complete heat engine computer simulation were used separately and subjected to perturbations in their major input quantities. Each perturbation represented a 5% shift from the established set point value for that parameter while holding the other parameters fixed. The results can be used to establish sensitivity coefficients suitable for a first-cut, steady-state analysis, and the results are presented below.

Turbine flow rate ($\dot{w}_t$) response to step changes in evaporator inlet flow ($\dot{w}_e$), inlet pressure (P_e) and inlet water temperature (T_{ww}) was evaluated (Table 2). Note that the condenser flow rate $\dot{w}_c$ is equal to $\dot{w}_t$, and the mass quality of the evaporator exit flow is $\dot{w}_t/\dot{w}_e$. A pronounced sensitivity to P_e is seen: for ± 5% variations in P_e (with $\dot{w}_e$ and T_{ww} kept constant), the average change in $\dot{w}_t$ s roughly ∓ 30% (total enthalpy changes by ± 30%). or comparison, ± 5% changes in $\dot{w}_e$ produce ± 3% hanges in $\dot{w}_t$. Approximately the same values for ressure and mass flow sensitivity coefficients are seen at two other water temperatures (81.2°F and 83.2°F). The responses to positive and negative perturbations are approximately equal and linear (see Fig. 7).

The lower part of Table 2 shows that 1°F changes in T_{ww} cause inverse changes of 12-13% in $\dot{w}_t$. Overall the evaporator is quite sensitive to pressure and water temperature variations. The control system must be designed to minimize or damp any disturbances in these parameters.

Table 2 Sensitivities of Turbine Flow Rate ($\dot{w}_t$, lb/sec) To Evaporator Exit Flow Parameters

T_{hw}, °F	$\dot{w}_e$ in, lb/sec	Turbine flow rate, $\dot{w}_t$, for evap. inlet P_e, psia: 121.9	128.32	134.74
82.2[a]	594	567.0	420.3	296.7
	625[a]	577.0	436.2[a]	310.4
	657	591.4	444.9	314.1
81.2	594	510.8	368.3	249.5
	625[a]	522.5	385.6	262.5
	657	538.7	394.2	269.4
83.2	594	b	475.2	297.0
	625[a]	b	487.5	350.0
	657	643.9	505.9	361.9

P_e, psia	$\dot{w}_e$ in, lb/sec	$\dot{w}_t$ for T_{hw}, °F: 81.2	82.2	83.2
128.32[a]	594	368.3	420.3	475.2
	625	385.6	436.2[a]	487.5
	652	394.2	444.9	505.9

a Design-point value

b Quality of 100% occurs at lower evap. inlet flow; simulation model non-convergent.

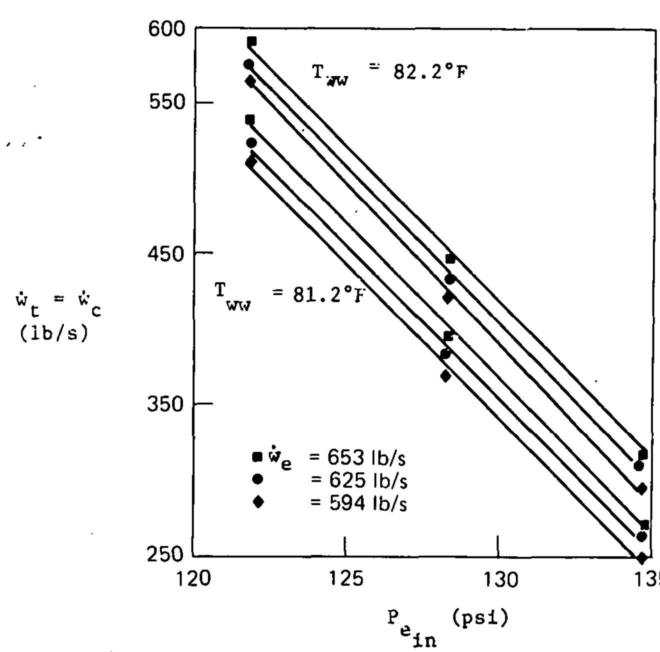


Fig. 7 Evaporator characteristics–effects of 5% variations in inlet flow (w_e) and pressure ($P_{e_{in}}$), and 1°F variations in inlet water temperature (T_{WW}) on turbine flow rate.

Since the key constraint in condenser operation is to maintain complete condensation, condenser performance was evaluated in terms of exit quality X_c (Fig. 8). Nominal parameter values at the established condenser set point were P_c = 89.26 psi, $\dot{w}_c$ = 432 lb/sec, T_{cw} = 39.2°F, and inlet quality = 0.9764. A 5% drop in pressure from its nominal set point value while holding mass flow and inlet quality constant caused X_c to shift from 0 to 21.4% as shown by the middle line in the left part of Fig. 8(b). This strong dependence on pressure was exhibited throughout the results at various cold water temperatures and mass flow rates. The relationship of pressure to mass flow to produce zero exit quality in the 25th pass is shown in Fig. 8(a) for each of three inlet temperatures. An overall observation is small perturbations in P_c, T_{cw} and $\dot{w}_c$ cause large changes in condenser response.

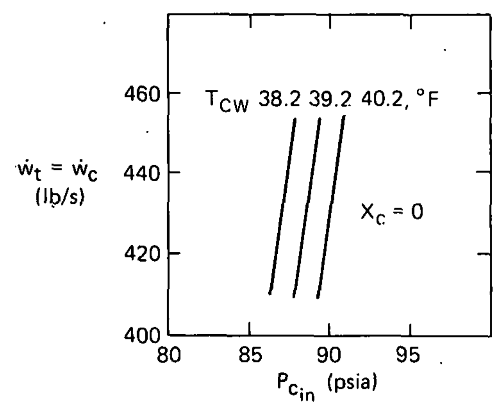
These data are used in establishing part of a preliminary control system representation with the overall control system hierarchical in organization, in which the lowest level acts as a regulatory subsystem to maintain operations at set points selected by the higher level of control based on exogeneous conditions. Nominal mass flow rates and pressures are established for evaporator and condenser; independent control loops are used to maintain component operation at these values. The condenser loop monitors condensate level at the outlet, with the turbine back pressure determined by the cold water temperature and optimum turbine mass flow. Regulation of the evaporator operations is accomplished by monitoring liquid flows out of the evaporator and the condenser to maintain entrance pressure to the evaporator. Simplified representations of component dynamics are based on the sensitivity data using the sensitivity coefficients as the gains in first-order lag models correlating steady-state output to step changes in each inlet parameter.

Control Considerations

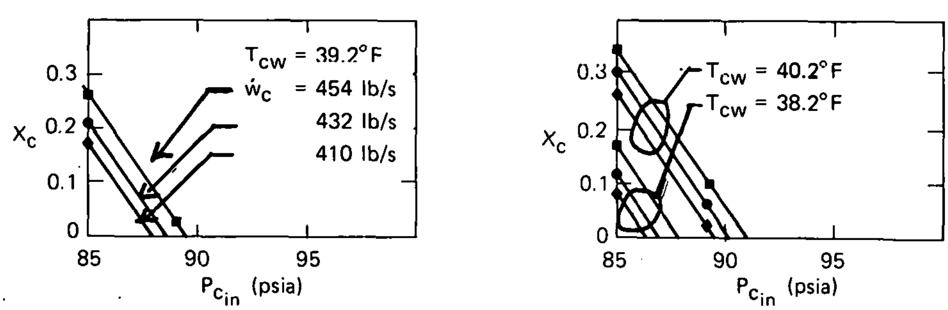
General Description and Requirements

The two major functions of the pilot-plant control and instrumentation systems proposed herein are 1) to control and optimize the OTEC power generation for demonstration purposes, and 2) to obtain system sensitivity and critical control parameter data as functions of input-output values, including perturbations from sea-state and platform motions, seawater-flow dynamics, and load variations. Hence, systems capable of in-place "tailoring" or change in parameter correlation and control output signal level including response time are required. Control of specific systems should be implemented through software in mini-computer controls, with critical hardwired controls to override in the case of rotating equipment. Optimization during power demonstration operation may also necessitate modification of such items as control valves and hardwired circuits, but a dominant "fail-safe" priority must be maintained throughout.

All of the instruments, control valves, etc. needed are commercially available. Excepting local equipment-mounted instrumentation and specifically required data for singular equipment design development, it is estimated that approximately 220 sensors are necessary per 5-MW power module. It is proposed



a) CONDITIONS FOR WHICH X_c = 0 IN 25TH PASS



b) EFFECTS OF AMMONIA INLET PRESSURE AND FLOW RATE AND COLD WATER TEMPERATURE ON EXIT QUALITY X_c

Fig. 8 Condenser parameter sensitivities.

that readout of these data be provided, via a central computer, both by printout and alpha-numeric CRT display. In addition, digital readout displays will be provided on the console mimic diagram. Sensors also are required for ancillary equipment including the power dissipation system. Continuous high quality recording is not necessary for these ancillary requirements; however, continuous monitoring recording will be required for some measurements.

For each 5-MW power module, alarm annunciation will be necessary for approximately 250 points not centrally indicated or recorded, including bearing temperatures, vibration frequency and amplitude, oil pressures, pump and fan failures, high coolant temperatures, etc. Another 200 alarms will be needed for ancillary and support equipment. These conditions will be reported through the computer printout and CRT display in the OTEC control room, with remote alarm and indication of critical functions in designated hotel areas including hazard alarms to the platform bridge and engine room.

To meet the control and data-output requirements described, the OTEC operational control is centralized and a separate data recording-analysis and computer room is located in the hotel area. With digital instruments and control switches combined with OTEC systems mimic diagram panels, up to four power-module control and monitoring systems can be located in the control room, together with ancillary system and alarms for which the space requirements can be reduced with computer printout-CRT display, with repeater alarm and printout in the recorder-computer room.

The control room proposed is a prefabricated, modularized design with all display and monitoring data transmitted from local signal-converter encoders for sensor groupings. All data and control signals can thus be made via low-voltage protected circuits simplifying transmission, identification, hazard protection, and possible modifications or change.

Data recording, signals, or a minimum 60 continuous concurrent channels per power module, can be transmitted to the recorder-computer room via converters and multi-channel shielded cables, where with suitable selection switching and recording equipment the data can be taped for later printout and analysis. While 60 data channels appear the minimum simultaneously required per module, a capability of at least 180 simultaneous data sets should be possible for comparative evaluation of alternate power module designs or modifications under identical operating conditions.

Sea State and Platform Motions

The control requirements discussed above consider only the "steady state" (calm sea) input vs. turbine generator loads. Additional control considerations include the possible perturbations from the sea state and resulting platform motions with associated water flow dynamics. In general, the platform motions up to the anticipated shutdown sea state 6C are not anticipated to have any significant effect on the ammonia cycle or on particular equipment *per se*. Maximum operating accelerations on the evaporator downcomer or condenser riser could produce pressure oscillations of $\leq \pm 0.5$ psi at 4.9 seconds peak-to-peak (half the wave period) at the evaporator inlet or consensate-pump discharge, and water flow rates through the heat exchangers can vary, primarily as a function of the relative sea pressure at the heat-exchanger exit but also due to pond head variation as a function of seawater pump flow variations. Because of the volume of the ponds the head perturbation time is much longer than the wave frequency and hence this effect should be minimal in comparison with combined heave and sea-wave pressure at depth, at the relative platform station. The variation of seawater velocity over the heat exchanger will perturb the overall heat transfer rate, and in combination with the acceleration pressures will induce a dynamic oscillation of evaporator and condenser pressures and flow rates at frequencies related to the wave periods.

Dynamic modeling for the control system design discussed below must include these sea-state effects including the sea-water-pump-motor-loads oscillation, and will require data from required warm and cold water system dynamics analysis. The relative phase of evaporator and condenser perturbations and pump loads with the possible excursion extremes may necessitate specific damping in the turbine control system, with generator load limiting to provide a control margin. Thruster power modulation for generator load compensation, as previously discussed, is not intended and should not be utilized in this mode, and a control interface device is required in the thruster tie. The proposed pressure and flow control system is not seen as being significantly affected at the probable limiting sea-state conditions and would appear to meet any control requirements, and in addition should have compensatory damping capabilities on the evaporator.

The limiting sea state 6C has been taken as that where the thermal resource of the mixed layers is degraded, significant platform motion starts, deck wetness becomes a factor, and deballasting would be required on worsening weather. Shutdowns due to worse weather should average less than 45 hours per year at the proposed Atlantic-1 site. The wave lengths and frequencies at this condition may not be the most severe for control-system design, and detailed system modeling may show that dynamic effects and required control response may be more critical at lesser sea states. Further investigation is needed to determine whether automatic draft and trim controls will be necessary for management of pond levels and seawater pumping power if significant changes in consumables or product load can occur, as will be the case in a commercial plantship.

System Control

The control system configuration will be organized in a hierarchical manner in order to accomodate the highly nonlinear and interactive nature of the heat engine components as well as the need to achieve optimum performance in responding to a range of operating conditions. The control system will perform the following tasks:

1. Operating Set Point Definition.
2. Operating Set Point Regulation.
3. Transient Control Between Set Points.
4. Start Up/Shut Down.
5. Failure Mode.

The basic strategy underlying the control system organization is to distribute the control functions down to the subsystem level, effectively isolating interactions between components in the nominal mode of set point regulation. Individual component controllers such as condensate sump level control, evaporator inlet flow control and generator load control operate locally to maintain set point conditions. Supervisory control establishes set points for the entire system. The primary control parameters are the ammonia flow rate and pressure and generator load setting, which are determined from the current exogeneous conditions such as the hot and cold water temperatures. Changes in environmental conditions will require the determination of a new set point and a "state profile" to be tracked during the set point transition to insure a graceful system adjustment.

The initial design effort has been directed toward identifying the requirements for set point regulation. This effort has resulted in the definition of a control system appropriate for set point control in a stable environment and sensitivity studies were used to identify the key parameters for measurement and control.

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9. OPEN-CYCLE OTEC POWER SYSTEMS AND OTEC CYCLE INNOVATIONS

OTEC 100-MWe ALTERNATE POWER SYSTEMS STUDY

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Abstract

Conceptual design studies were funded by DOE in FY78 for three alternate OTEC power systems: the open cycle with a surface condenser, the open cycle with a direct contact condenser, and the hybrid cycle. The major results of these studies are as follows:

1) The open cycle power systems are indeed feasible and potentially have a lower capital cost than the ammonia closed cycle OTEC power system; and

2) The hybrid power system is not cost effective.

The steam turbine has been the major concern and a previously stated objection to the open cycle systems. The adoption of helicopter-blade technology coupled with extensive steam turbine experience essentially eliminated all the feasibility and cost concerns of the turbine. The large quantity of air which must be removed from the system is adequately handled with high efficiency axial compressors. The analyses required to design the condensers are complex because of the large air quantities but they are not outside the scope of existing technology and computer capability. Existing multistage flash evaporator saline water conversion technology is the basis for the open cycle flash chamber design. The resulting power module designs are single vacuum housings containing all the components.

Introduction

In 1881, Jacques d'Arsonval first suggested that a steam thermal engine could be operated by using the temperature difference between the surface water of the tropical ocean and the deep water in these regions. George Claude[1] demonstrated the technical feasibility of this Ocean Thermal Energy Conversion (OTEC) concept on a limited scale in 1930 at Matanzas Bay in Cuba. Howe[2] further substantiated the feasibility with a land based plant. The long term cost effectiveness of commercial scale OTEC power systems has not as yet been thoroughly studied or demonstrated.

There are three possible OTEC power system cycles which can be employed for commercial power production; the closed, open and hybrid. The closed cycle uses anhydrous ammonia as the working fluid. Heat from the warm sea water is transferred in a large heat exchanger (ammonia evaporator) to the liquid ammonia and vapor is generated at approximately 125 psia. The ammonia vapor is expanded through a small turbo-generator to produce electric power and is condensed in another large heat exchanger, the ammonia condenser, where the cooling medium is the cold sea water. The ammonia condensate is then pumped back to the ammonia evaporator completing the closed loop. Both conceptual[3] and preliminary[4] studies sponsored by the U.S. Department of Energy have been completed and component heat exchanger testing is in progress.

The open cycle uses the ocean water as the working fluid. A portion of the warm sea water is evaporated (flashed) in a low pressure chamber. The resulting steam is then expanded through a very large turbine. The exiting steam from the turbine is condensed either by direct mixing with the cold sea water (direct contact condenser) or in a heat exchanger similar to the closed cycle ammonia condenser. The latter option permits the production of potable water. The open cycle type was employed briefly by both Claude and Howe to demonstrate the OTEC concept feasibility. The first conceptual studies[5,6,7] of this cycle type, sponsored by DOE, were of a preliminary nature and were not in the same detail as the closed cycle conceptual studies.

The hybrid cycle is a blend of both the open and closed cycles. Warm sea water is flashed to generate vapor. This vapor rather than the warm sea water is the heat source for a closed ammonia cycle. No previous detailed hybrid cycle conceptual study has, to our knowledge, ever been completed.

The purpose of this paper is to present an overview of the results of a recent conceptual design study[8] of the two open cycle and the hybrid cycle power module types. The level of analysis was similar to that of the closed cycle conceptual efforts and valid cost comparisons are therefore possible. Another very preliminary study of this nature[9] has been completed but not in the same detail.

The cost figures used for all cycle comparisons are the summation of the power module and platform capital cost. The costs of the cold water pipe, mooring and cabling are not included. The platform is an integral part of the power module components for the open cycles. As a result, this cost must be added to the summation of all power module component costs to obtain a valid comparison. Also the comparison cost figures are initial capital costs rather than life cycle costs. This approach is less questionable because the issues of escalation, interest, depreciation, maintenance cost, plant availability, etc. need not be addressed. A capital cost comparison is further justified because there is no fuel cost.

This paper is divided into four parts. In the first part, the computer cost optimization model descriptions are presented. In the second part, the major problems associated with each cycle are reviewed. The solutions to the open cycle feasibility and cost problems are then discussed, followed by the power module and platform capital cost comparison.

Cost Optimization Models

Computer cost optimization programs were developed for the closed cycle, the open cycle with a surface condenser, and the hybrid cycle. A similar model was not developed for the open cycle with a direct contact condenser; however, a reasonable estimate is possible because of the close similarity between the open cycle options.

Cost optimization programs contain the following parts:

- plant heat balance subroutine
- component design subroutines
- major component pricing subroutines
- main optimization program
- output subroutine

This modular approach was selected because algorithm improvement is more easily accomplished and because different individuals can simultaneously contribute.

Each of these parts will be briefly discussed; however, more detailed descriptions are presented elsewhere[10,3,8]. The plant heat balance first determines all the flows consistent with the specified constraints such as warm and cold seawater inlet temperatures and system variables such as the turbine inlet and outlet temperatures and exchanger outlet temperatures.

The component design subroutines then determine the specifications required to obtain their cost such as exchanger tube lengths, number of modules and the parasitic power losses such as seawater and working fluid pressure drops. After the completion of the plant heat balance and component design phases, the net power output is determined by subtracting the parasitic power requirements from the gross power output of the turbine-generator. The flow through the turbine is varied by the program to obtain the desired net power.

The power module and platform capital cost is then obtained by summing the individual component costs. Various types of cost estimates were used for the components depending on the availability of data. These include a single estimate, a factored estimate, a factored estimate with the capacity exponent, a summation of labor and material costs with an administrative cost multiplier, a curve fit through a range of supplier cost estimates, and price list data. Table 1 depicts the type of cost estimating used for each of the components of the power modules. A more detailed breakdown of the heat exchanger cost algorithms is presented in Table 2.

The optimization technique chosen was the pattern search method. This method does not require specific types of equations and is well suited for

Table 1

Component cost models.

Component	Constant $C=a$	Factored $C=bQ$	Capacity Exponent $C=C_1(Q/Q_1)^e$	Labor & Materials $C=f(L+M)$	Supplier Data Range	Supplier Price List
Heat Exchangers						
Internals				X		
Shell				X		
Tubes						X
Sea Pumps					X	
Condensate Pumps			X			
NH3 Turbine			X			
Steam Turbine				X		
Generator			X			
Chlorination			X			
Controls	X					
Auxiliaries		X				

NOTE: C = Cost
C_1 = Reference Cost
Q = Characteristic Capacity of Component
Q_1 = Reference Capacity

Table 2

Heat exchanger cost algorithms.

	Material	Labor
Tube	$/Foot	$/Foot
Tubeplate	$/Square Foot of Tubeplate	$/Hole & Constant
Tube Support Plate	$/Pound	$/Hole
Bundle Assembly		$/Tube Support Plate plus Constant (includes tube to tube sheet welding)
Tubeplate Support Lattice Structure	$/Foot of Structure	$/Foot of Structure
Internal Lattice Structure	$/Foot of Structure	$/Foot of Structure

Notes:

1) Material Cost is based on weight of material ordered not finished weight.

Major construction materials usage

Tube	Tube Sheet	Tube Bundle	Tube Support
90-10 Cu-Ni	Al. Bronze	C. Steel	C. Steel
Titanium	Titanium Clad C. Steel	C. Steel	C. Steel
Aluminum	Aluminum Clad C. Steel or Aluminum	Steel Aluminum	Aluminum Aluminum

2) A miscellaneous material charge of 5% is added.

3) Some representative material costs are:

C.S. Plate	.23 $/Pound
Al. Plate	.95 $/Pound
C.S. Structural	.32 $/Pound
Al. Bronze	2.0 $/Pound
Al. Structural	1.50 $/Pound

diverse varieties of thermodynamic, cost and design data incorporated in the optimization models. In the pattern search method, an arbitrary value is assumed for each of the system variables, and the cost and power output of the module are calculated using those values. Following this calculation, each of the variables, one at a time, is slightly changed. The power module is redesigned, in the search for a less costly system. After all of the system variables have been tested, the system variable which contributed to the lowest cost power module is changed from its previous value to the new value. The entire procedure is repeated until all perturbations of all system variables result in more costly power modules. This situation exists when an optimum module has been obtained.

Since reference has already been made to system variables and will continue throughout this paper, further discussion is in order. The cost and performance of the power system are functions of independent system variables. The number of system variables is determined by the equations used in the heat balance and design calculations. If the mathematical model consists of E equations and U unknowns, there are U-E system variables. While the number of system variables can be calculated using this method, the identity of the system variables is still open to judgment on the part of the analyst. For example, the seawater pressure drop may be calculated from the tube velocity, or the velocity may be calculated from the pressure drop. The optimization would yield the same result regardless of which system variables were chosen, so the choice depends on the order in which the equations are solved.

The system variables used in the optimization of the OTEC power modules are of two types: continuous and discontinuous. Continuous variables are those such as temperatures and velocities which can vary infinitesimally over the range to be studied. Discontinuous variables are variables such as tube diameter, tube material and the type of tube enhancement which exists only as discrete alternatives. The optimization technique was used to optimize the continuous variables for each combination of discontinuous variables.

The resulting capital cost figures obtained with the cost optimization programs are far more reliable than those previously obtained with assumed system values and simplified component costing techniques such as $/ft^2 for the exchangers. It is important to stress that the closed, open and hybrid cycle cost optimization programs used identical component design and cost subroutines when applicable in order to insure a valid comparison.

Known Cycle Problems

Table 3 lists the major problems or objections of the four power cycles. Ideally, a cost analysis is required to assess the significance of each. For the first two, however, the cost value would be very arbitrary and would be debatable until the incidents actually occured. Therefore, the environmental and hazard considerations will not be considered in the final cost comparison. The water production capability of the open cycle also is not considered in the cost comparison of the four cycles. The potential cost improvement of this capability is addressed at the end of this paper.

Heat exchanger fouling can be adequately handled in the cost evaluation by simply including the cost of chlorination and additional pressure drop due to mechanical cleaning devices. The heat ex-

Table 3

Known cycle problems.

Problems	CYCLE TYPE			
	Closed	Open		Hybrid
		Surface Condenser	Direct Contact	
1. Environmentalpollution	Yes	No	No	Yes
2. Fire and explosion	Yes	No	No	Yes
3. No water production	Yes	No	Yes	No
4. Potential warm water fouling	Yes	No	No	No
5. Potential cold water fouling	Yes	Yes	No	Yes
6. Large heat exchangers	Yes	Yes	Yes	Yes
7. Large vapor turbine	No	Yes	Yes	No
8. Low energy level turbine	No	Yes	Yes	No
9. Large amounts of non-condensible gases	No	Yes	Yes	Yes

changers for the open cycles (flash evaporator, surface condenser or direct contact condenser) are not usually considered as large but, they indeed are large. The volumes are always larger than the largest current power plant surface condenser. The flash evaporator and direct contact condensers do have a lower cost per unit volume when compared to the tube exchangers. The major problems of the open cycles are the large turbines with a low energy level or pressure drop and the dissolved gases liberated by the flashing warm seawater. The reason most often given for the initial rejection of the open cycle was the prohibitive size of the steam turbine. The statement was even made[11] that no practical proposal would consider a single turbine for a 100 MWe plant because of the turbine blade tip diameter being in the 150 ft. range. The turbine and air removal solutions will be discussed at length.

Open Cycle Turbine Solution

The major design problem of the massive turbine size was solved through the introduction of composite fiberglass blading and a hollow fabricated disc. The solutions to the low energy and resulting flow distribution problem were obtained through the use of a single vertical axial flow turbine. These unique combination of turbine concepts greatly enhanced the economic competitiveness of the open cycle. The optimized open cycle turbine efficiency and cost are 81.1% and 184 $/KW respectively.

Steam turbines using conventional alloy steel blades with lengths greater than three or four feet have not been practical primarily due to weight and the resulting high centrifugal forces. By employing composite blade technology, which has been developed for helicopter blades, greater lengths may be considered due to the high strength to weight ratios of composite materials such as fiberglass. The Boeing Vertol Company, as a subcontractor, has conceptually designed turbine blading as long as 40 feet and they have been shown to be economically attractive. The modifications needed to allow application in the open cycle steam turbine environment were investigated and present no apparent feasibility concern.

The typical disc in a modern steam turbine is of profiled or equal stress geometry and forged construction. The unavailability of forgings in the large diameters necessary for an open cycle turbine required consideration of a new disc fabrication technique. The hollow fabricated disc, resulting from consideration of new fabrication techniques, increased the allowable diameters of the open cycle turbine. The potential exists for even larger disc diameters through the use of a spider-spoked disc.

The approach to determine this optimum (i.e., a minimum cost power system) turbine-generator was a combined design and cost model capable of analyzing the tradeoffs encountered when using different concepts, materials, and fabrication techniques. The design model performs the physical sizing of the turbine-generator for a given efficiency and load requirement by performing the thermodynamic and mechanical design calculations. The cost model computes the major parts and assembly costs and then adds them to arrive at the overall unit cost. The optimum turbine-generator is determined through the computer cost optimization program.

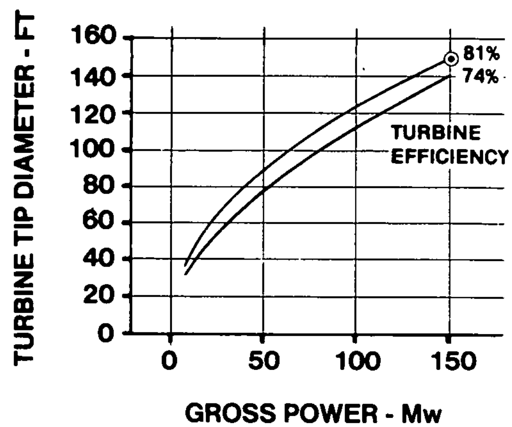

Fig. 1 Turbine tip diameter.

The design of the turbine is accomplished by performing the appropriate thermodynamic and mechanical calculations. First, the boundaries of the turbine flow field and turbine speed are established to meet the required load and efficiency. The calculation procedure used is a one-dimensional, axisymmetric method that considers the mean stream surface flow to be representative of the flow everywhere along a blade. An additional calculation, using a simplified radial equilibrium approach, is performed to assure good performance over the entire length of the rotating blade.

Mechanical adequacy is assured by sizing the individual turbine parts to have acceptable stress levels based on Westinghouse turbine experience and Boeing Vertol helicopter experience.

A parametric study of Westinghouse air-cooled and hydrogen cooled generators served as the basis for the generator design. The entire span for the generator product line was used to establish the data from which the appropriate generator was selected.

Turbine costs are computed on an individual part or assembly operation basis using appropriate costing techniques for the part or assembly operation being considered. The lumped parameter model used in previous investigations was avoided due to the difficulty of determining a common costing method that is valid in this uncommon range of interest. The long turbine blades have a strong influence on the cost not only because of the blades but also because of the effect of their centrifugal loadings on the disc. The blade material alternatives selected for analysis were steel, titanium, composite fiberglass and epoxy.

Figure 1 shows the range of turbine tip diameters that have been found to be cost effective designs. The computed turbine-generator costs associated with various turbine efficiencies are shown in Figure 2. Note that the turbine cost is about 150 $/KW gross for the entire load range. This cost does not include the turbine containment or shell cost since it is considered as part of the platform cost. These results show that the turbine design problems are surmountable at an economically attractive cost. Also shown are the turbine cost predictions obtained from previous investigations (shell not included). The substantial disagreement between

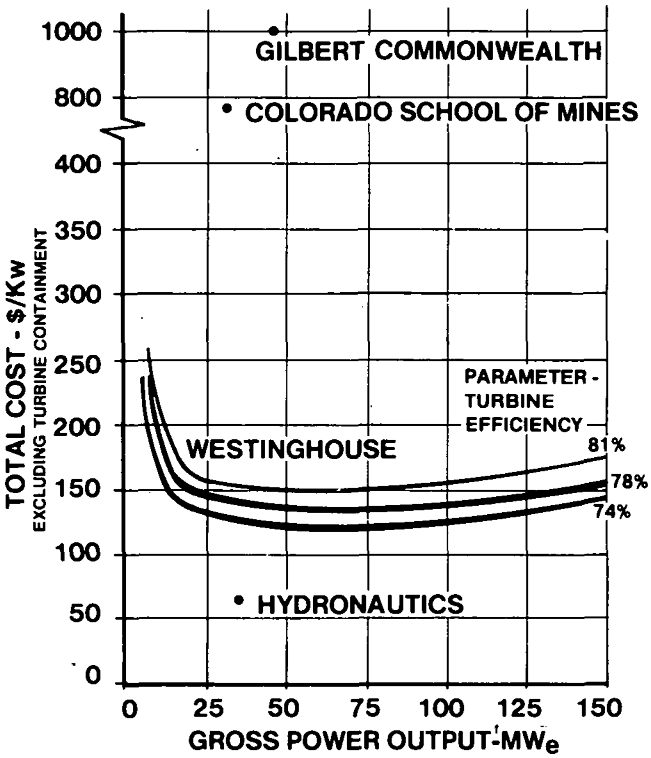

Fig. 2 Total turbine-generator and power conditioning equipment cost.

these studies was the result of a lumped parameter approach for the entire turbine cost. The very large turbine cost suggested by Gilbert Commonwealth[9] distorted their OTEC cycle cost comparison.

Interfacing of the turbine to the flash evaporator and condenser must be accomplished with minimal presure drop and maximum uniformity of flow because of the low energy level available for the turbine. Unnecessary pressure drop (energy loss) can not be tolerated in any individual component or connecting piping or passages. The concept of a single vertical turbine allows the power system to be disposed radially around the turbine to provide excellent flow distribution. The arrangement serves to reduce the pressure loss caused by turning the flow at the turbine inlet and permits a full arc of admission. These features are illustrated in Figure 3. It is interesting to note that both the French Company, Energie de Mers, and Howe designed their open cycle power plants with a single vertical turbine, an annular flash evaporator surrounding the turbine and the condenser below the turbine. Note that the power module design is a single vacuum housing containing all the components.

Air Removal Solution

Since dissolved noncondensible gases such as oxygen, nitrogen and carbon dioxide are present in seawater, some of these gases will be liberated when the warm seawater enters a component under vacuum such as the flash evaporator. Additionally, air leakage occurs at machinery shaft seals and by diffusion through the containment walls. In order to maintain system vacuum and performance, provisions must be made to remove these gases.

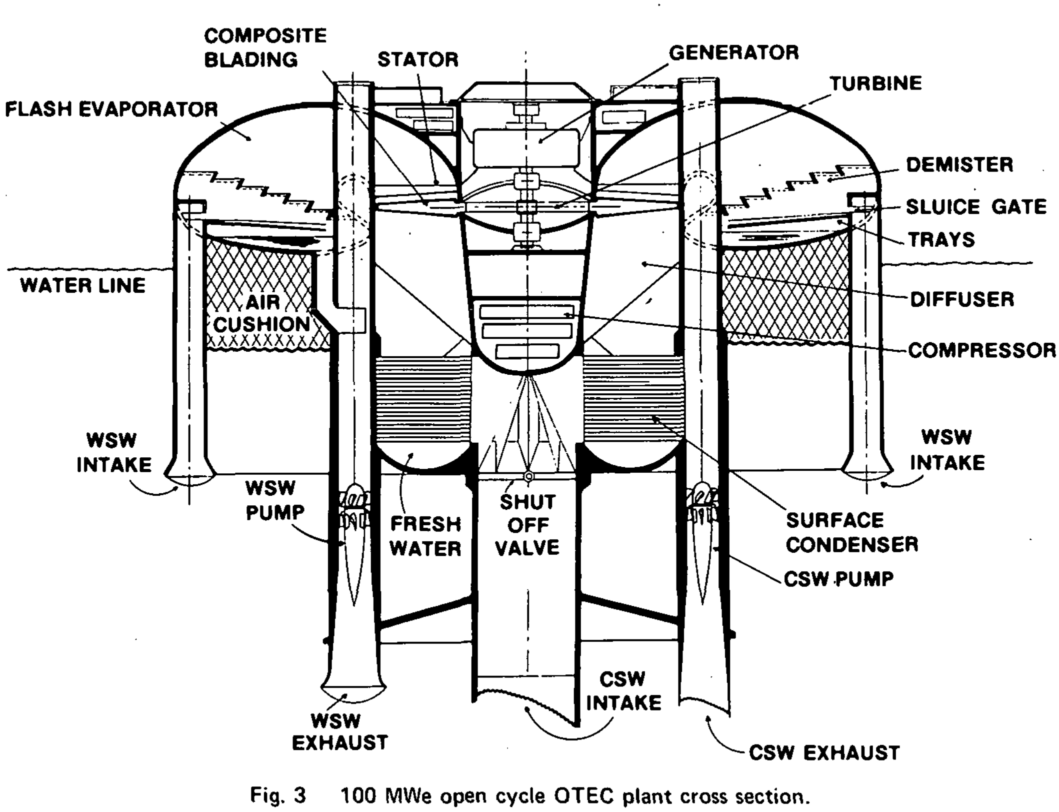

Fig. 3 100 MWe open cycle OTEC plant cross section.

For the open cycle with surface condenser and for the hybrid cycle, conventional shell and tube condensers are used. The gases in the cold seawater will remain in solution since the cold seawater is always contained in closed pressurized conduits. For these two cycle types, only the gases in the warm seawater and the air leakage must be removed. For the open cycle with a direct contact condenser, the gases liberated by the cold seawater must also be removed. The amount of gases expressed in parts per million dissolved in both the warm seawater (80°F) and cold seawater (40°F) are presented in Table 4 along with the values in previous open cycle investigations. Although there is favorable agreement in all the studies, the amount of oxygen and nitrogen dissolved in the seawater is somewhat dependent on local environmental conditions such as biological activity. There are no data which actually establish which gases exist or to what extent they exist in the cold deep seawater. Most investigators assumed that the gas content was that of the surface water, but at a reduced temperature.

A basic mechanism for promoting gas liberation from seawater is flashing. Flashing has been successfully used to deaerate seawater feed in desalination plants. An assumption made in this ope cycle analysis is that all the gases are liberated in the flash chamber. This assumption could be overly conservative because the flashing process is not identical to that used in saline water conversion plants. The low temperature level and small flash-down encountered in the open cycle flash chamber probably will not produce a large instantaneous population of bubbles which provide the interfacial area for the desorption of the dissolved gases. There is no current justification, however, to assume other than complete desorption in the flash evaporator even though an argument can be introduced[6] which counters this conservative position.

The noncondensible gas removal system, regardless of the type, increases the power module cost and increases the parasitic power requirements. The resulting cost optimized air removal system contains four stages of large high efficiency axial compressors with intermediate aftercondensers. All the gases are removed at the condenser since predeaeration was not found to be cost effective. The cost of all the equipment is 170 $/KW.

Compressors

Large axial flow compressors were chosen to continuously remove the gases since for large sizes, axial compressors are the most efficient[12] and are almost equivalent in capital cost to rotary positive displacement types. The compressor efficiency at all pressure levels was 86%. The optimum number of compressors was four with an average compression ratio of about 3.25. The cost was 1.6 $/CFM for the large units or first stage compressors (900,000 CFM/unit) and up to about 10 $/CFM for the highest pressure and smallest compressor.

Table 4

Seawater solubility of air, PPM.

INVESTIGATOR	SEA WATER	
	WARM	COLD
Westinghouse[8]	17.2	25.6
Colorado School of Mines[7]	16.6	23.7
University of Massachusetts[5]	21.6	-
Hydronautics[6]	25	25
Gilbert Commonwealth[9]	20	20

Table 5

Compressor cost summary.

ITEM	1.788×10^6 CFM					5.364×10^6 CFM				
	No.	CFM	Type	$/CFM	$	No.	CFM	Type	$/CFM	$
1st Compressor Auxiliaries Totals	2	894,000	A	1.45 1.608	2,600,000 275,000 2,875,000	6	894,000	A	1.45 1.608	7,800,000 825,000 8,625,000
2nd Compressor Auxiliaries Totals	1	204,000	A	2.45 3.02	500,000 117,000 617,000	1	612,000	A	1.80 2.37	1.100,000 351,000 1,451,000
3rd Compressor Auxiliaries Totals	1	49,410	A	4.04 6.09	200,000 101,000 301,000	1	148,200	A	3.71 5.76	550,000 303,000 853,000
4th Compressor Auxiliaries Totals	1	14,400	C	22.5 24.2	324,800 24,000 348,800	1	43,200	A	6.94 9.28	300,000 101,000 401,000
Instrumentation					350,000					1,050,000
Grand Totals				2.512	4,491,800				2.302	12,380,000

Type Key: A = Axial ; C = Centrifugal

Table 5 contains a cost summary based on information supplied by the Westinghouse Marine Division, a manufacturer of large axial and centrifugal compressors. Three points are worth noting:

- The $/CFM decreases with increasing capacity to about 900,000 CFM.
- The auxiliaries become a smaller portion of the total cost as the CFM increases.
- Except for small sizes, centrifugal compressors are more expensive.

This table shows that the incremental cost is indeed size dependent, a point not strongly emphasized in previous studies.

It is interesting to note that Claude's solution to the air removal problem also was high efficiency compressors. Colorado School of Mines considered a positive displacement lobe type (Roots) in their study; the maximum efficiency was 80% and the cost was about 1.3 $/CFM. The University of Massachusetts also considered compressors with an efficiency assumed to be 85% at all pressure levels, but the type and cost were not addressed.

A device called an ingestor deaerator[13,14] has been proposed for the removal of the gases. After a preliminary evaluation of this device, our conclusion is that it is not suitable for this application.

Aftercondensers

Most of the previous open cycle studies including another investigation[15] conclusively demonstrated the need for aftercondensers or vent condensers located between the compressor stages. Vapor in the steam-air mixture is removed by the aftercondenser, thus reducing the volumetric flow rate (CFM) to the downstream compressors. The high temperature difference needed for the heat and mass transfer is generated by the preceding compressor where typical outlet temperatures are in the 200°F to 250°F range due to the compression of the gases. The resulting temperature difference ranges from about 160°F to 210°F when the cold seawater at 40°F is used as the cooling medium. Typically, the parasitic power and cost are reduced by a factor of 2 and 1.5 respectively through the use of aftercondensers. Table 6 presents a power and cost breakdown for the air removal system. Noteworthy are the following:

- The steam flows and inlet CFM to each successive stage is substantially reduced due to the vent condensers.
- The heat exchanger costs are a small portion of the total cost.
- The additional pumping power introduced by the exchangers is small.
- The first vent condenser is very large and comparable in size to a typical power plant surface condenser.

The surface area of the aftercondensers can best be obtained by pointwise calculations using the Colburn-Hougen[16,17] method. It is important to realize that the vapor is highly superheated at the entrance of these exchangers. The computer algorithm which performed these heat transfer and also pressure drop calculations is rather lengthy and required a substantial computer run time. However, a simplified sensitivity analysis is presented in Figures 4 and 5 which shows that the vent condenser overall heat transfer coefficient, pressure drop and approach temperature do not impact strongly on the air removal system power consumption and cost.

Table 6

Air removal power and cost.

Air Flow	= 54,688. LB/HR
Steam Flow	= 164,065 LB/HR
Exit Pres.	= .149 PSIA
Cold H_2O Temp.	= 40.0 F
Warm H_2O Flow	= 2,947,000,000 LB/HR
Comp. Type	= Axial
Steam/Air Ratio	= 3.0
Pressure Drop - Vent Condenser	= 10.0%
Approach Temp. - Vent Condenser	= 10.0 F
Overall Heat Transfer Coef.	= 50. BTU/(HR-F-FT^2)

No. Stages	No. After Condensers	Power Required Total	Comp.	Exch.	Equipment Total	Costs Compressors	Exchangers
4.	3.	8.82	8.65	.17	16,974,748	14,762,424	2,212,325

Stage	Temp. In (F)	Pres. In (PSIA)	Steam In (LB/HR)	Comp. Power (MW)	Exch. Power (MW)	Exch. Area (FT^2)	Sea Water (LB/HR)	Temp. Out F	CFM	Pres. Out (PSIA)
1.	39.51	.15	164065.	4.96	0.15	278409.	1019156.	209.25	6597308	.47
2.	50.00	.42	20724.	1.56	0.02	31239.	88309.	255.63	655584.	1.48
3.	50.00	1.33	4641.	1.11	0.00	6398.	17319.	264.66	146802.	4.66
4.	50.00	4.20	1347.	1.02	0.00	0.	0.	267.67	42599	14.70

CONDITIONS:

4 AXIAL COMPRESSORS
WARM SEAWATER FLOW = $3.3X10^9$ LBM/HR
EXIT STEAM — AIR RATIO = 4
SAT. TEMP. = 50°F, CONDENSER PRESSURE DROP = 4°F
VENT CONDENSER APPROACH = 10°F

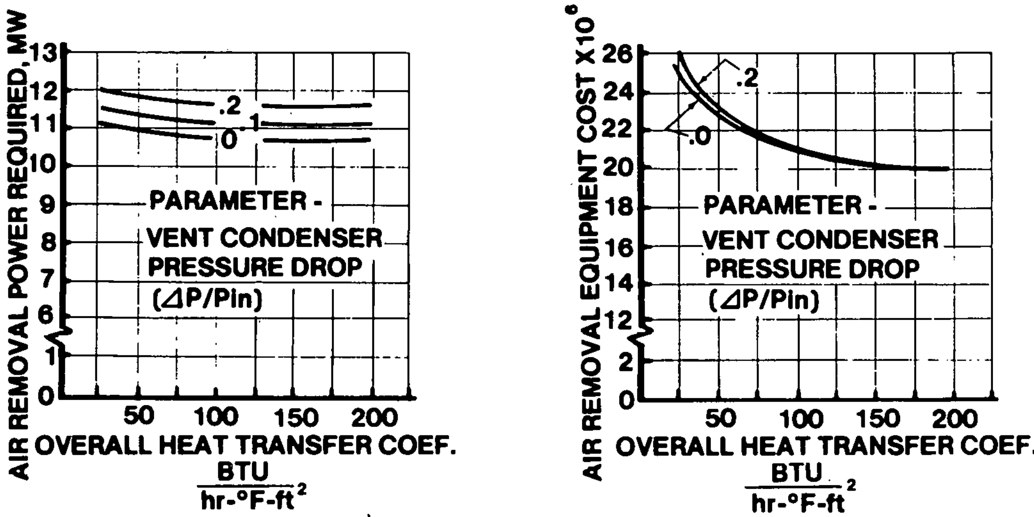

Fig. 4 Vent condenser sensitivity study.

Table 6 was based on an overall heat transfer coefficient of 50 BTU/(HR-F°-FT2), on a 10 percent pressure drop and on an approach temperature difference of 10°F. The pointwise Colburn-Hougen calculation method previously discussed was used to substantiate the magnitude of these three quantities.

Pre-deaeration

The surface condenser overall heat transfer coefficients measured by Howe[2] were about 200 BTU/(HR-F°-FT2) or less. Typical surface condenser heat transfer coefficients for comparable conditions are larger by about a factor of 1.5 to 2. Howe concluded that the effect of the air was responsible for the poor condenser performance and recommended that the air be removed prior to condensation. Interestingly, Claude had previously recognized this problem and recommended that a large portion of these gases be removed in a deaerator. Pre-deaeration was an integral part of two other open cycle studies[5,7]. Most of the gases in the warm seawater can be removed by one or more deaerators located upstream of the flash evaporator. The remaining gases plus the air leakage will pass through the system along with the steam and must be removed continuously from the condensers or ammonia evaporator. For the open cycle with the direct contact condenser, deaerators can also be used before the condenser in the cold water loops.

Pre-deaeration also held promise because the gases are removed from a higher initial pressure level (2.13 psia was recommended by Claude). As a result, less compressor power is required because of the reduced pressure ratio. Pre-deaeration, however, does introduce additional costs and parasitic power losses. Regardless of the deaerator type (packed, spray or thin film), pressure drop or free fall results in a power loss similar to the tube side pressure drop in any heat exchanger. Also a second air removal system must exist at the condenser to remove the gases not liberated in the deaerator and the air leakage into the system.

There was no previous study to our knowledge to show that pre-deaeration is indeed cost effective. A packed deaerator was selected to investigate this tradeoff for the following reasons:

- Most common for vacuum deaeration applications.
- Adequate test data.
- Lowest pressure drop.
- Lowest cost, in general

The particular packing type selected was two-inch ceramic rings, since large size permits the use of a large liquid mass velocity. The items which contribute to the total deaerator height in addition to the packing are the liquid distributor, packing supports, hotwell, chamber separators and gas collection areas. The deaerator internals, neglecting the packing, make a sizable contribution to the deaerator parasitic power loss because they introduce a fixed height of about five feet.

Figure 6 presents the deaerator sensitivity results. Observe that pre-deaeration is not cost effective regardless of the pressure level nor effectiveness. It is interesting to note that the pressure level for the single stage deaerator selected by Claude is identical to the optimum value in this figure or about 2 psia. The main reason why pre-deaeration is not cost effective is the parasitic pumping power loss of the deaerator itself. Pre-deaeration does reduce the air removal system parasitic power and condenser cost.

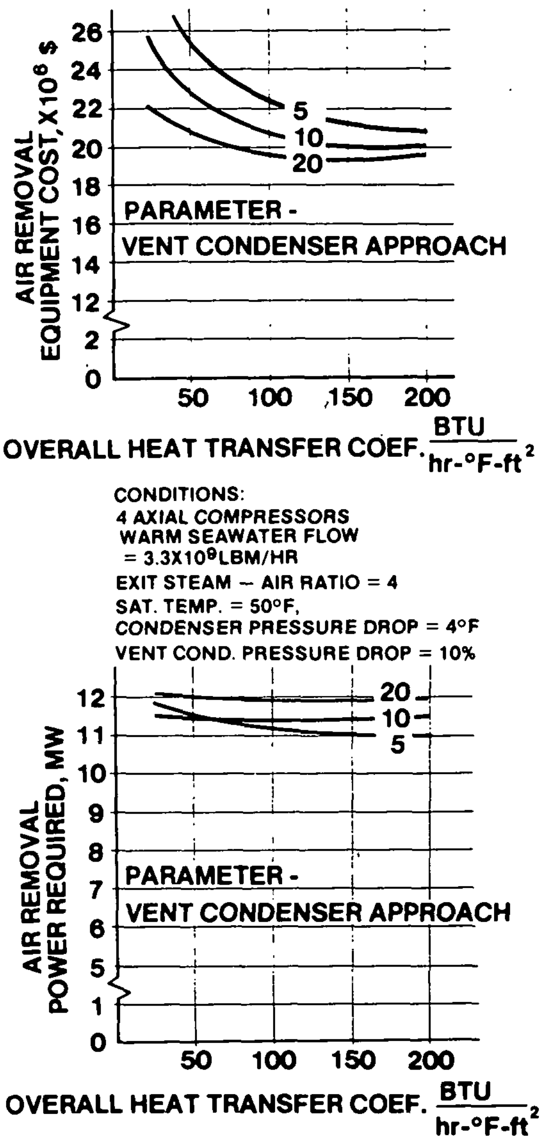

Fig. 5 Vent condenser sensitivity study.

This presentation does not conclusively prove that pre-deaeration is not cost effective. Other packings exist which are both cheaper ($\$/FT^2$) and allow a higher mass velocity ($lbm/HR\text{-}FT^2$). However, they were not considered because the performance data or HTU-liquid rate design curves were not available. Other types of deaerators could be employed; in particular, tray types which minimize the warm seawater pressure drop. Again, however, no test data or prediction techniques are available for these operating conditions. An important fact not considered in this cost analysis is the effect of the deaerator on the platform cost. Even if a cost effective deaerator type or internal packing is discovered, the cost advantage will be less with the hull consideration because of additional enclosure materials and space for the second compressor train since air removal equipment must be installed at the condenser regardless of the amount and type of pre-deaeration.

Surface Condenser

The surface condenser, with or without pre-deaeration, is not the typical surface condenser encountered in nuclear or fossil power plants mainly because of the large amount of noncondensible gases which enter the unit. Even if 90% of the air were removed by pre-deaeration, the inlet steam-air ratio would be about an order of magnitude lower than that commonly experienced in commercial power plant condensers. Additional reasons for this disparity are the low pressure level and the small temperature difference between the inlet vapor and cold seawater. The low pressure level or deep vacuum implies that the shell-side pressure drop impacts strongly on the condenser performance.

The Colburn-Hougen method, which was previously discussed in relation to the aftercondensers was used to account for the area increase due to the noncondensibles. A Westinghouse computer program[19] was used to develop design curves of pressure drop versus the condenser saturation temperature, percentage of the tube sheet area occupied by tubes, and the steam lane widths. These results, shown in Figure 7, demonstrate that surface condensers can be designed to perform at these low pressure levels but the percent occupied is lower and the steam lanes are larger than conventional power plant surface condensers. The combined area increase due to the noncondensibles and pressure drop is presented in Figure 8. Observe that the additional area required increases with a reduction of the inlet and exit steam-air ratios. At the baseline condition, the no-gas heat transfer coefficient (noncondensible gases and pressure drop neglected) was 556 BTU/(HR°F-FT²). The overall heat transfer coefficient considering these two effects was 366 BTU/(HR°F-FT²). This value is substantially higher

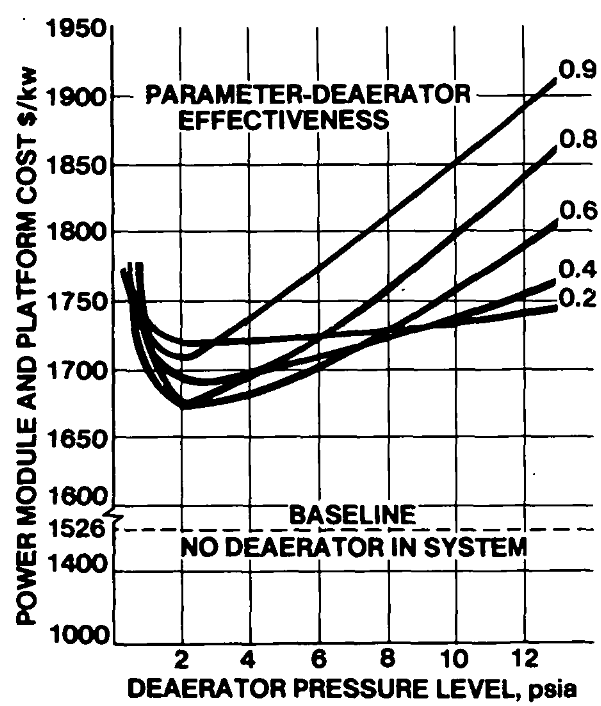

Fig. 6 Deaerator cost sensitivity study-open cycle with surface condenser option.

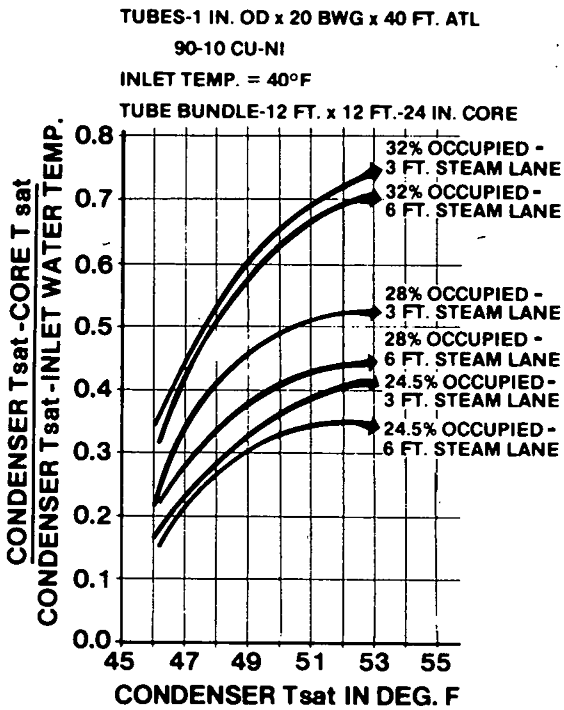

Fig. 7 Saturation temperature depression due to bundle pressure drop.

than the value obtained by Howe (200 BTU/HR°F-FT^2). for two reasons. Howe used plain tubes and a shell and tube heat exchanger. The cost optimized design used Korodense LPD enhanced tubing and a surface condenser which minimizes the adverse impact of the shell-side pressure drop.

The condenser rating analysis which considered both the noncondensibles and the pressure drop was pointwise rather than overall and resulted in a step change in complexity. The analysis could be even further refined to improve the accuracy since the technology exists and computer capability presents no particular constraint.

The exit steam-air ratio from the condenser (a system variable) has a very strong impact on the air removal system power and cost and ultimately on the power module and platform capital cost. The variation of the capital cost with the condenser exit steam-air ratio is shown in Figure 9. For large exit steam-air ratios, the compressor costs and combined parasitic power are the dominant contributions to the total cost. A minimum steam-air ratio exists; however, to reach this value the condenser surface area would approach infinity. Note that a very sharp optimum exists very close to the minimum value of 2.75. The minimum and optimum steam-air ratio are related to the inlet condenser temperature, condenser pressure drop and other condenser system variables.

Cost Comparison

Table 7 shows that the open cycle and closed cycle are comparable in capital cost and the hybrid is substantially higher. Not all items shown are directly comparable and are identified with footnotes. Another difference to be noted is that the open cycle and hybrid cycle results are for a conceptual level study whereas the closed cycle results are for a preliminary level study.

The heat exchanger tube materials and enhancements which yielded the optimum cost were different for each cycle. For the closed cycle, the tube material was titanium and no internal enhancement was cost effective. The Linde high flux surface was used on the outside of the evaporator tubes. For the open cycle, the tube material was 90-10 Cu-Ni and the surface condenser enhancement was Korodense LPD. The open cycle condenser defines the exit geometry of the turbine diffuser, and thus influences the efficiency of the turbine and the power system. The shorter enhanced tubes allow a shorter diffuser and a more compact plant, whereas the longer plain tubes require either a longer diffuser or a decrease in diffuser recovery, resulting in higher cost and/or poorer performance. In contrast, the geometry of the closed cycle condenser has no effect on turbine performance.

Table 7 shows that the steam turbine cost is about five times the cost of the ammonia turbine but yet not a prohibitively high value. Also note that the air removal system, which is principally the compressors, is the third most costly item in the power module. The platform in the open cycle is the most costly item because of its multifunctional nature (component containment, seaworthy platform, interface or piping between components). More details of the platform are presented elsewhere[8,18]. It is apparent that the major problems of the open cycle OTEC power module, the large size and low available energy turbine and the air removal system, are indeed adequately resolved.

More uncertainty does exist in the open cycle module and platform capital cost prediction. However, the possibility also exists for further cost improvements. Table 8 identifies the major contributions to both.

The greatest uncertainty regarding the power system is the thermal nonequilibrium in the flash evaporator. When a compressed or saturated liquid enters a chamber at a pressure where the saturation

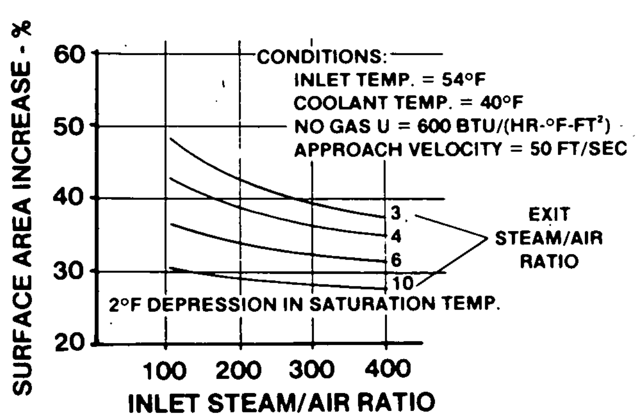

Fig. 8 Effect of noncondensibles and pressure drop on the condenser surface area.

Table 7

OTEC 100 MWe net module open cycle cost comparison (1977 $/KW).

	OPEN CYCLE		HYBRID CYCLE		CLOSED CYCLE	
Flash Evaporator	44.30		-		-	
Shell & Tube Evaporator	-		580.20		480.91	
Liquid/Vapor Separator	(in Evap)		35.48		34.93	
Surface Condenser & Hotwell	295.09		699.34		534.31	
Retubing (Present Value)	94.51		0		0	
Air Removal Vent Condenser	22.51		19.46		0	
Heat Exchanger Subtotal		456.41		1334.48		1050.15
Turbine	184.08		69.22		36.31	
Generator Exciter	33.93		40.43		29.54	
Seawater Pumps	110.61		121.40		93.00	
Condensate Pump	1.84		6.48		4.21	
Recycle Pump	-		6.58		1.98	
Air Removal Equipment	151.19		130.73		-	
Rotating Equipment Subtotal		481.65		374.84		165.04
Piping and Valves		-		20.34		6.34
Operating Control System		7.50		15.00		15.00
Fouling Control System		*48.09		**14.92		***34.37
Auxiliaries, Power Conditioning		42.41		57.20		53.19
Module Investment		1036.06		1845.28		1324.09
Platform Including Dock and Construction		490.00		388.00		391.00
Module & Platform Total		1526.06		2233.28		1715.09

* Mechanical Cleaning ** Chlorination in Condenser *** Chlorination in Condenser & Evaporator

temperature is less than the inlet temperature, flashing occurs. Thermodynamic principles state that the remaining liquid will exit the lower pressure chamber at the corresponding saturation temperature. Conventional thermodynamics is based on equilibrium conditions; in other words, the liquid must have an infinitely long residence time in the low pressure chamber. The residence time in an actual flash evaporator, however, is fixed by the size of the chamber and the velocity at which the liquid is moving. The thermal nonequilibrium is defined as the difference between the brine exit temperature and the equilibrium or saturation temperature in the chamber. A nonequilibrium temperature difference of 1.0°F was assumed to be an attainable value because of the low brine depth (9.1 in) and long stage length (77.8 ft) even though the temperature level is about 20°F lower than that experienced in multistage flash evaporator saline water conversion plants. A surface evaporation analytical thermal nonequilibrium model was predicting values much less than 1.0°F and is the basis for this assumption. A review of the existing empirical thermal nonequilibrium methods[20] and their applicability[21] did not resolve this uncertainty because no test data existed for the open cycle OTEC conditions. Very reliable data can be obtained at the low temperature levels and large flow rates with only a modest expenditure because the measuring techniques are developed and a scale model can be employed.

Table 8

Uncertainties and cost improvements of open cycle module and platforms.

COMPONENT	UNCERTAINTIES CAPITAL COST INCREASE, $/KW	COST IMPROVEMENTS CAPITAL COST DECREASE, $/KW
FLASH EVAPORATOR		
Thermal Nonequilibrium	0-400	
Reduced Diameter (Higher Δ P Mesh)		0-100
Desorption (Amount of Air Liberated in Flashing Process)		0-80
TURBINE		
Fiberglass Blades and Fabricated Disc Welding	0-250	
Concrete Stator Blades		0-70
DIRECT CONTACT CONDENSER		0-250
AIR REMOVAL SYSTEM		
Air Dissolved in Seawater	0-50	0-50
System Optimization	0-100	0-100

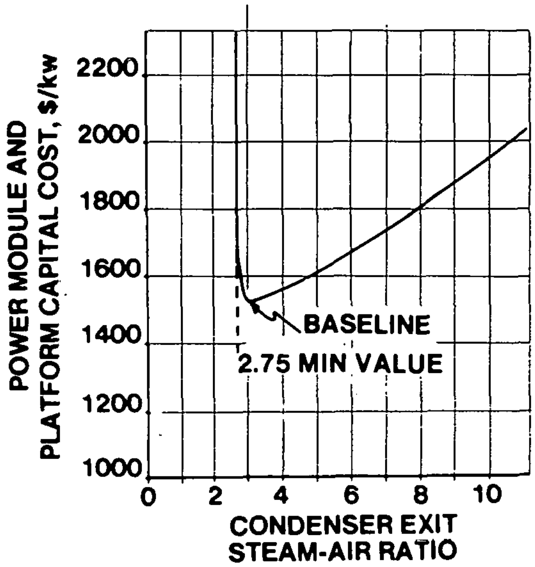

Fig. 9 Effect of condenser exit steam-air ratio on capital cost.

The second greatest group of uncertainties are turbine related. Erosion, mechanical property changes and outgassing have been identified as possible technical risks for the composite blading. The existence of moisture droplets in the turbine blade path may necessitate the use of erosion protection for the composite blading. Further studies into the actual prediction of where condensation will occur in the blade path will determine whether protection is needed. If protection is deemed necessary, tests to determine the best method of erosion protection should be conducted.

The change in mechanical properties of composite blading due to moisture absorption should be investigated. If the moisture present under open cycle conditions causes any change in the composite structure, this test can determine the effect on composite blade life. New binder material may also be developed and tested.

The low pressure conditions that exist in the open cycle turbine may cause a small percentage of uncured ingredients to be outgassed from the epoxy matrix. The effect, if any, of this outgassing can be determined from a test under simulated conditions. Additional turbine related uncertainties are blade tuning at low RPM and fabricated disc welding techniques.

Other uncertainties are the magnitudes of the dissolved air in the warm and cold seawater and minor details of various component design and cost algorithms.

The greatest possible cost improvement is obtained with the direct contact condenser. A preliminary analysis[8] demonstrated that this condenser can be positioned between the cold water pipe and the flash evaporator and essentially at the same elevation as the flash evaporator. The barometric level principle is again used to reduce the cold water pumping power. It is apparent, then, that the platform diameters are about equivalent for both condenser options but the platform height and volume are substantially reduced with the direct contact condenser. The cold water pumping power for both open cycle options are about the same but the air removal equipment cost and parasitic power are larger with the direct contact condenser option. A cost optimization program containing detailed cost and design algorithms for the direct contact condenser has not been developed so a more accurate cost comparison is not currently possible.

There are other possible cost improvements which are attainable due to system analysis efforts, in particular on the air removal system. Material changes such as a concrete turbine stator blading lead directly to a cost improvement. The exact magnitude of the other cost improvements can only be determined after basic test results are obtained.

Table 9 compares the cost to own and operate three OTEC plant types: closed, open, and open with water production. Note the dramatic impact the water production option presents to the OTEC concept. A value of $2.50/1000 gal. was used to compute the cost saving obtainable by selling the water. This value may be low in areas where saline water plants are located. The writers realize that

Table 9

Cost to own and operate OTEC power plants.

PLANT TYPE	OTEC CYCLE TYPE		
	CLOSED CYCLE	OPEN CYCLE	OPEN CYCLE AND WATER
Plant Size, (MW)	100	100	100
Plant Cost, ($/KW)	1715	1525	1525
Completed Plant Cost ($/KW), 1.5 Multiplier	2575	2300	2300
Capital Recovery Rate	.17	.17	.17
Operation and Maintenance, (mill/KW-HR)	3.0	3.0	3.0
Fuel Cost, (mills/KW-HR)	0	0	0
By Product, (mills/KW-HR) 2.5 $/1000 gal.	-	-	39.0
Cost to own and operate, ($/KW) Availability of 1	2730	2450	445

some or all of the economic assumptions can be argued, but an indisputable fact is that the open cycle with water production will always be the cost winner regardless how the numbers are compared.

Conclusion

This paper summarizes the results of cost studies for the open and hybrid OTEC cycles and presents a cost comparison between these cycles and the previously analyzed closed cycle. The important conclusions are that the open cycle OTEC power modules are indeed feasible and cost effective but that the hybrid cycle is not. The major problem of the open cycles, the large steam turbine cost, was solved with fiberglass blades and a fabricated disc. Another often mentioned objection to the open cycle, the removal of the dissolved gases in the seawater, was addressed and resolved by the use of high efficiency compressors, intermediate aftercondensers and no pre-deaeration. The open cycles have other advantages not considered in this cost comparison such as reduced biofouling and corrosion and the commercial potential of water production. The historians of OTEC development will recognize that many of the concepts presented herein have been previously introduced. A single vertical turbine surrounded by an annular flash evaporator within a reinforced concrete containment was suggested in this study and in almost all prior studies. High efficiency compressors with intermediate aftercondensers were usually recommended for the air removal system.

Acknowledgements

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DISCUSSION

Dr. Molini, CMU: The noncondensible gases are being discharged to the atmosphere. That means you are deaerating the water and then returning it to the ocean. Do you expect that the reduction in the oxygen content will be considered a pollution problem?

T. Rabas: No.

Dr. Molini: Table 6 shows the air removal power and cost where you discharge the noncondensibles to the atmosphere with four stages of compression. In the cost, did you consider returning the gases to their barometric leg of the return water from the flash evaporator? You would be returning them or discharging them to a lower pressure which would eliminate stages 3 and 4 and thereby reduce the cost.

T. Rabas: We did not consider discharging the noncondensibles into the flashed warm seawater. (After the question and answer period, Dr. Molini and I further discussed this question. Dr. Molini is indeed correct. A compression device of this type is called an ingestor and was tested at Palo Seco, Puerto Rico with the controlled flash evaporation system. This compression device is patented, and some problems could possibly arise in adapting it to the OTEC open-cycle concept.)

Dr. Molini: In Table 9, you show the completed plant cost in dollars per kilowatt. For the closed cycle, you show \$2575, and for the open cycle you show \$2300. My question is related to the fact that you show both the open cycle and open cycle with water production at the same cost.

T. Rabas: The completed open-cycle plant cost is identical with or without water production. The difference only occurs when you consider the operation, maintenance, and water production costs. The water production cost benefits were converted to a captal cost framework (cost to own and operate) with an assumed availability factor and capital cost recovery rate which are indicated in the table. The costs to own and operate the open-cycle power modules with and without water production on the bottom line are not identical. The water production option of the open cycle makes it an economic winner regardless of the assumed values of availability, recovery rate, and water setting price.

Question, General Electric: How do you envision controlling the steam-air ratio that is shown to be so critical. Is the ratio a well defined thing for a given design?

T. Rabas: The condenser is sized to obtain the existing steam-air ratio, and in principle it is well defined. The air entering the condenser is fixed by the warm seawater flow which contains about 20 ppm of liberated O_2 and N_2.

Question: The presence of noncondensibles themselves affect the condensing process.

T. Rabas: Yes, that is why Fig. 8 shows that the condenser area must be increased by about 30 to 40%. You have to consider both the noncondensibles and the pressure drop of the vapor. Also, I thought that I made the point that a very sophisticated three-dimensional analysis computer program is required to perform this calculation.

Question: You are proposing to take all the noncondensibles all the way through to the condenser before you get rid of them.

T. Rabas: Yes. At first, we thought that predeaeration would be the way to go but discovered that it was not cost effective. The major reason was the added parasitic pumping power in the warm seawater loop. It is interesting to note that many people initially thought that enhanced heat exchanger surfaces would be cost effective. However, it has been conclusively established that enhanced shell and tube heat exchangers — and in particular ID enhancements — are not cost effective because of the parasitic pumping power. OTEC power plants are very pressure-drop sensitive.

Question: What fraction of the total air came from air leakage through the enclosure?

T. Rabas: We assumed an air inleakage value of 4000 lbm/hr. The HEI standards of commercial power plant surface condensers show that the largest air inleakage is about 3000 lbm/hr. The assumed value for the OTEC condenser is probably conservative and also has a very small influence because 60,000 lbm/hr of air are entering the power system with the warm seawater.

Question: What is the open cycle? Is that with a surface condenser or with a direct contact condenser?

T. Rabas: With all open-cycle OTEC power systems, the working fluid is steam generated in the flash evaporator. Either a surface or direct contact condenser can be used with an open-cycle power module. Throughout this entire presentation, the direct contact condenser option was not addressed because I think it is a true statement that the state of the art of direct contact condensation is far behind that of large power plant surface condensers. As a result we did not thoroughly investigate the direct contact condenser in this study; however, we hope to do so in the future.

Question, Univ. Texas: I think you accept the concept that in the case of mixed production of both water and energy that two products are competing for the ΔT or available temperature difference. I wondered if you did any kind of alteration of optimization in between the ratio of fresh water production and energy production?

T. Rabas: We are negotiating with MarAd to study open-cycle water plants. The power produced will only be that required for the auxiliaries. We are in the position to cost optimize the system considering just water production but have not as yet done so.

Question, Univ. Texas: I think that if you optimize your power system considering water as the major product, you will probably reach the conclusion that it is economically more vluable than power.

T. Rabas: This has so far also been our belief.

Question: What information did you use to determine the air leakage through concrete?

T. Rabas: Tests conducted in our laboratory was the input in the study. I do not have the test results with me, but they are indeed test results and not assumptions. The air leakage is very low or about 200 lbm/hr for a 100 MW plant. Coatings are also available which will even dramatically reduce this value but were not considered because of the additional cost.

R. Bowles, Florida Power Corp.: Will partial removal of air by predeaeration change the economic picture?

T. Rabas: Figure 6 shows the power module and platform cost for a single deaerator operating at various pressure levels and effectivenesses. The effectiveness is defined as the ratio of the air removed to the total amount which is theoretically possible. Since the effectiveness values used in Fig. 6 are always less than unity, only partial removal of the air by predeaeration was considered. Figure 6 shows that, regardless of the quantity removed and the pressure level, predeaeration is not cost effective. The main reason why predeaeration is not cost effective is the additional parasitic pumping power introduced by the deaerator. If deaerator types were developed which introduce a substantially lower pressure drop, predeaeration could possibly be cost effective.

R. Bowles: (Inaudible Question)

T. Rabas: Figure 8 shows that the condenser cost can be reduced if the inlet and not exit steam/air ratio is increased. If the inlet steam/air ratio is increased, less air is entering the condenser. Figure 8 shows that the condenser cost increases with the exit team/air ratio. Figure 9 conclusively shows that an optimum exit steam/air ratio exists at a value of about 3. For larger exit steam/air ratios, the air removal cost and power dominated the power module cost. For lower values, the condenser cost dominates the total cost. As previously stated, the condenser exit steam/air ratio is controlled by the size of the condenser and the air inleakage is known rather than an unknown quantity.

Question, Franklin Research Center: (Inaudible)

T. Rabas: The cost comparison which considers the production of desalinated water is presented in Table 9 rather than Table 7. Since the results presented herein were the result of a DOE funded contract for OTEC power systems comparison, considering water production was outside the scope of the contract. That is why the production of desalinated water was not considered in the cost comparison presentation of Table 7.

RECENT DEVELOPMENTS IN THE FOAM OTEC SYSTEM

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Abstract

In the foam open cycle OTEC system warm water literally lifts itself several hundred feet. In preliminary experiments the foam behavior seemed to depend in a capricious manner upon the temperature of the incoming water, the pressure of the spray condenser, and the foam mass flow rate. Through a combination of analysis and experiment we have developed an equation which governs the steady state rise of a foam column. This equation correctly predicts the effects upon the foam behavior of all controllable parameters. These parameters include, in addition to these mentioned above, the diameter and height of the foam column. We show that foam generated in an appropriate manner flows laminarly rather than turbulently, and has a well defined drag function. This drag function decreases as the inverse square of the column diameter.

Introduction

During the past year Carnegie-Mellon University has made great strides in its Foam Open Cycle OTEC system.[1,2] In order that everyone may understand the significance of our progress, I shall review briefly the essential features of the foam system.

The purpose of foaming the warm surface water is to transform it into a binary phase of finely dispersed liquid and vapor. This finely dispersed state has an extremely high ratio of surface area per gram of water. Evaporation therefore proceeds under essentially isentropic conditions. As a consequence, a drop in temperature follows a vertical path in the T-S diagram, as shown in Fig. 1. The enthalpy ΔH released in a drop from temperature A to C is therefore the area of the ABC triangle.

$$\Delta H = \begin{cases} \frac{C_p}{2T}\Delta T^2 \\ 0.0072(\text{joules/gram})\Delta T^2 \end{cases} \quad (1)$$

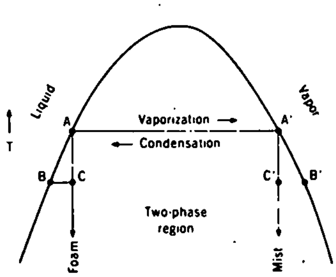

Fig. 1. Typical temperature-entropy (*T-S*) diagram.

A finely dispersed foam has a second important property. Each element of vapor is entrapped within a liquid polyhedron cage. As the vapor expands through further evaporation, the entrapping liquid cage follows this expansion. The liquid thus necessarily follows the motion of the expanding vapor.

As an example, suppose the expanding foam is confined to a vertical cylinder. The foam can then only expand if it rises. The more it expands, the higher will it rise, and the lower will its temperature drop. By the time its temperature has dropped from 25 to 5°C, Eq. (1) tells us that it will have reached 960 feet.

With sufficient ingenuity we should be able to make practical use of this property of foamed warm water to raise itself to a great height. At least in principle we can construct a power plant by providing (1) a foam generator, (2) a foam breaker, (3) a cold water spray which removes the vapor freed from its foam cage by the foam breaker, as well as removing the noncondensibles, (4) a tall feedstock into which the freed liquid follows and eventually passes through a hydraulic turbine-generator, (5) an economically viable surfactant. A possible arrangement of these components into an OTEC system is illustrated in Fig. 2.

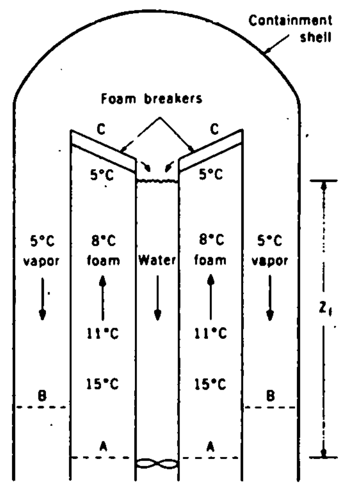

Fig. 2. Schematic diagram of a foam SSPP designed to generate a hydrostatic water head. The transition between the pure liquid and the liquid-vapor mixture (foam) in the warm seawater intake at 25°C takes place at *A*. This point is at about 30 feet (9 m) above sea level. The foam breaker at *C* separates the liquid from the vapor in the foam. At *B* the vapor is condensed by thermal contact with deep ocean water at 5°C. The liquid is channeled down the central pipe through a standard hydraulic turbine at the bottom. The height of the water head above *A* is represented by Z_f. Points *A*, *B*, and *C* correspond to those similarly labeled in Fig. 1.

We have been intrigued by the possibility of converting the concepts envisioned in Fig. 2 into a practical reality. We have systematically first studied those features of our concept which appeared the most problematic. You are not to interpret our ignoring other features as being unaware of their problems. Nor are you to interpret our preoccupation with the dome concept of Fig. 2 as indicating we believe this concept will ultimately be the most practical embodiment of the foam concept. Rather, we at Carnegie-Mellon University feel that our particular talents are more profitably spent in developing the dome concept than in developing alternative concepts. One such concept is the conversion of the enthalpy released by the warm water directly into kinetic energy, and from thence via a turbine into electrical power.

The foam OTEC system and the mist OTEC system being developed by Ridgeway have identical thermodynamic properties. The two have, however quite different technical problems.

Foam Making and Foam Breaking

Since foam making and foam breaking initially seemed to be the most difficult functions to perform, they were singled out for intensive investigation. In solving these two problems we found it necessary to enlarge our understanding of the dynamics of a foam column.

These two problems could not be solved separately. Obviously we could not learn how to break the foam rapidly without learning to make it rapidly. Equally clear, it was impossible to learn how to generate the foam rapidly without knowing how to rapidly eliminate it from our system.

The locations of the foam generator and foam breaker in our system are indicated in Fig. 3.

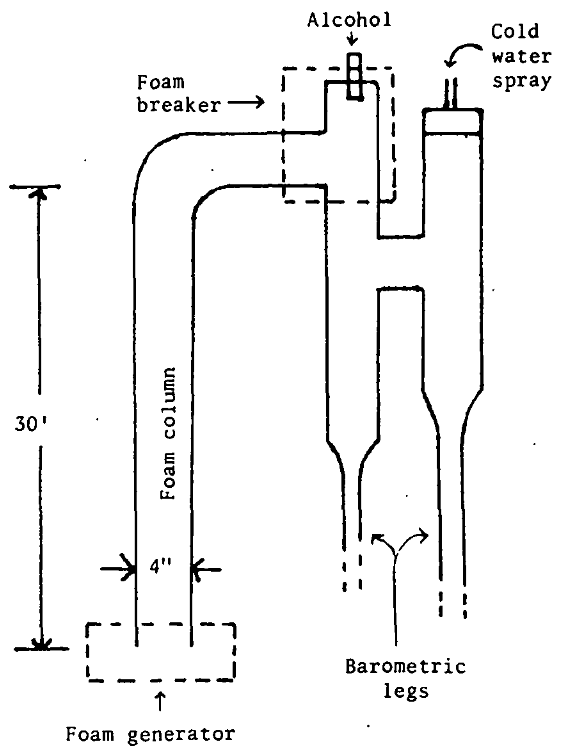

Fig. 3 Laboratory Foam System

The details of the current foam generator are given in Fig. 4. Both warm water (~25°C) and air are injected into the bottom of our 4" diameter column. The air is injected via a forest of some 1200 capillary tubes. These capillary tubes extend upward from an air chamber through a water chamber into the bottom of our foam column. We provide a fairly wide clearance for these capillaries in passing through the top partition of the water chamber. An upward flow of water thereby surrounds the upper part of each capillary tube. This water flow allows each tube to generate bubbles at a rate of at least 30 per sec. The capillaries have an 8 mil bore, a 0.1" spacing . The ratio of bore to spacing is such that if the capillaries ejected bubbles in synchronism, the bubbles would form a close packed lattice of spherical bubbles.

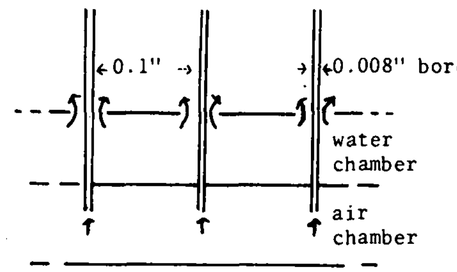

Fig. 4 Foam generator 1200 capillary tubes in 4" diameter chamber.

Independent valves control the warm water input and the air input. Although these are controlled independently, we obtain the steadiest operation if the water input and the air input are closely correlated to one another by the control curve of Fig. 5. This control curve is close to the curve which corresponds to a 26% of water volumetric input, a 74% of air volumetric (at the vapor pressure of 25°C water) input.

After trying many foam breaking systems with only partial success, we have finally found a completely successful system. An alcohol simply drips into a downward bend of the surfactant column, as illustrated in Fig. 3. A couple of drops per second is sufficient to disperse a foam raising 1.5 gallons of water a minute. Pentanol or hexanol work equally well. No attempt has so far been made to obtain an optimal dispersion of the alcohol.

The concept back of our foam breaker was to hit the foam with a fine alcohol aerosol spray. As an alcohol particle would hit a cell wall, its spreading action would burst the wall. Such spreading action would of course be a manifestation of the

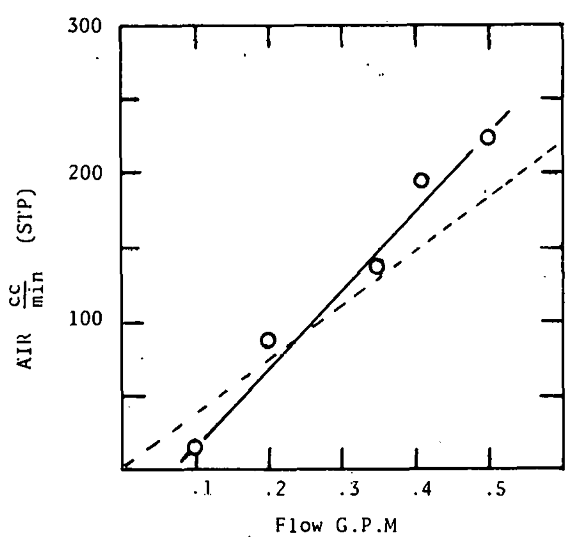

Fig. 5 Control curve for air input to foam generator. Dashed line represents theoretical 3:1 volumetric ratio for close packed spherical bubbles

Maragoni effect. To our surprise, physical contact of the alcohol with the foam does not appear necessary. The foam appears to burst merely through bombardment of the alcohol vapor.

Our foam breaking system gave us an additional bonus. We had anticipated having to send the broken foam through special equipment to separate the liquid from the vapor. However, the broken foam segregates into such large aggregates of water that the water offers no impediment to the sideway flow of vapor into a parallel tube acting as a cold water spray condenser.

The foam column joins the foam generator with the foam breaker plus cold water spray. These three components form an intimately related system. With proper use, this system can produce a wide range of foam which starts as large polygonal cells. Under all conditions the matured foam at the top of the column has a polygonal structure. We have been able to acquire this ability of controlling the foam structure only after obtaining an understanding of the basic equations which govern the rise of the foam column. An understanding of these basic equations has given us the additional bonus of being able to develop scale-up laws to be applied to larger foam systems. I shall now briefly review the basic equations of a foam column, as well as point out the general properties of their solution, and finally shall show that the observed properties indeed conform to theory.

Governing Equation and its Solutions

The governing equations for the rise of a foam column may be expressed most succinctly in terms of the Bernoulli function Eq. (2). If no losses were present, this function would remain constant for an element of mass as it rises in the column. In the presence of wall losses, the Bernoulli function of an element of mass decreases as the mass rises according to Eq. (3), where τ is the retarding wall shear stress. The literature gives us no information about τ for our foam. As a start we assumed it either obeys the standard turbulent friction law for a homogeneous fluid, Eq. (4a) or the laminar flow law Eq. (4b). Because of the conservation of the mass flux density M, as expressed by Eq. (5), both expressions for τ depend in the same way upon the density ρ, and hence upon the temperature.

$$B \equiv \Delta H - \frac{1}{2} v^2 - gZ \qquad (2)$$

$$\frac{dB}{dZ} = \frac{4\tau}{\rho D} \qquad (3)$$

$$\tau = \begin{pmatrix} \frac{1}{2} f\rho v^2 \\ \frac{\mu v}{\delta} \end{pmatrix} \qquad \begin{matrix} (4a) \\ (4b) \end{matrix}$$

$$\rho v = \dot{m} \qquad (5)$$

$$1/\rho = 43{,}400 e^{.056 \cdot \Delta T} \qquad (6)$$

$$\frac{d\Delta T^2}{dZ} = (0.0014 + A\Delta T^2 e^{.112\Delta T}) \qquad (7)$$

$$A \sim \left(\frac{4''}{D}\right)^{1 \text{ or } 2} e^{.112(25^\circ C - T_{inc})} \dot{m} \qquad (8)$$

In writing Eq. (4a) and (4b), we at least tentatively assume that τ is independent of cell size. Then our governing equation (3) has only one dependent variable, ΔT, and one independent variable, Z. Recognizing that ρ is represented with high accuracy by Eq. (6) over our range of interest, and that in our range of interest $1/2\ V^2$ is small compared to ΔH, our governing Eq. (3) may be written simply as Eq. (7).

The solutions to Eq. (7) are given by Fig. 6 for a set of drag parameters A covering our range of interest. The lowest curve, for A = 0, is of course just the curve of ΔT versus Z for a foam column rising quasi-statically. These curves vividly display the influence of the drag coefficient.

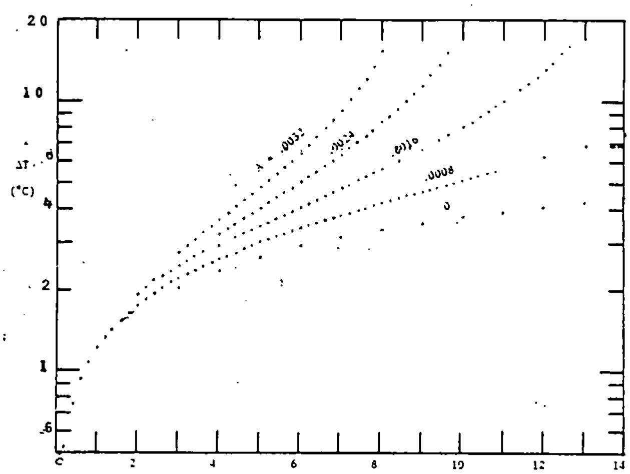

Fig. 6 Family of Solutions to Eq. 7 satisfying the the boundary condition $\Delta T = 0$ @ $Z = 0$.

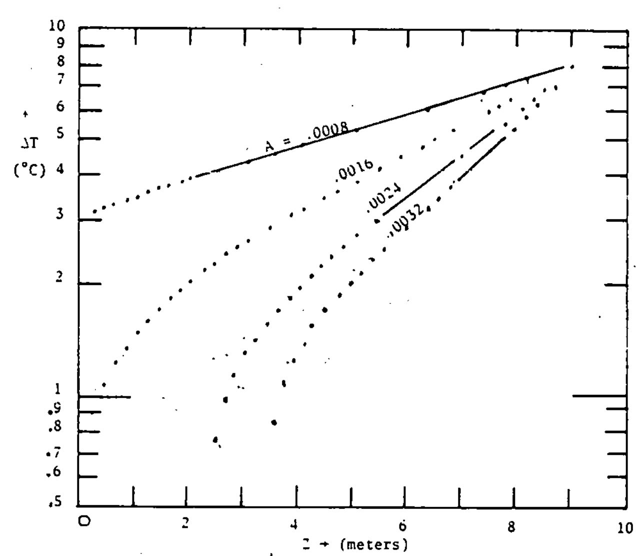

Fig. 7 Family of solutions to Eq. (7) satisfying the boundary condition $\Delta T = 8^\circ C$ @ $Z = 9$ meters.

All the curves of Fig. 6 are drawn to satisfy the simple boundary condition that ΔT be zero at the bottom of the foam column. We can also construct a family of solutions to Eq. (7) which satisfy the boundary conditions that ΔT have a specified value at the top of the column. Fig. 7 gives such a family of solutions. This family represents the solutions to the boundary condition $\Delta T = 8°C$ at $Z = 9$ meters. No further calculations are required to obtain such a family of solutions. Our governing Eq. (7) contains Z only as the differential dZ. Hence any solution remains a solution after it is shifted horizontally by an arbitrary amount. Thus, the solution for $A = 0.0008$ in Fig. 7 is identical to the solution for the same A given in Fig. 6, save for a shift to the left by 5.5 meters.

Interpretation of Experimental Data

Appropriate boundary conditions

The bulk of our experiments have consisted in measuring the temperature at various ports along the column, the temperature of the input water and the mass flux being held constant. Plots were made of ΔT vs Z, where ΔT represents the drop in temperature of the foam at Z below the temperature of the incoming warm water. Should we compare such experimental curves with the theoretical curves of Fig. 6 or of Fig. 7?

An unambiguous answer to this question comes by assembling a set of curves taken under similar conditions save for the mass flow rate. Such a family of curves converge toward a common point in the upper right hand corner of the diagram, as in Fig. 7. None of the experimental data actually reached this common point because the last measurement was a couple of feet before the foam breaker.

The above correlation demonstrates that the top foam temperature is determined primarily by the cold water spray, rather than the bottom foam temperature being determined by the temperature of the incoming warm water. One cannot speak of the spray temperature itself, since the temperature within a spray droplet is lower than that in the surrounding vapor. One can however speak of the spray pressure, since this is essentially constant throughout the spray chamber. The pressure within the spray must also equal the final pressure within the foam. But within the foam, the pressure at any position is equal to the vapor pressure at that position. The conditions within the spray chamber must therefore determine the top foam temperature.

Super heated incoming warm water

Once we accept that the top foam temperature is determined by the spray condenser, we must recognize that for some conditions, as for $A = .0008$ in Fig.7, ΔT is positive at the bottom of the foam. In such cases the first layer of bubbles in the foam column is colder than the warm water which lies below. How is this possible? Puzzling through the answers to this apparent paradox has given us the clue to those factors which determine the foam structure.

A finite ΔT at the bottom of the foam implies that the pressure of the foam is less than the saturation pressure of the warm water. Once we recognized this implication, we instrumented the column to measure the pressure at the capillary tips. The measurements are reported as Fig. 8. We see that under the constraints of these measurements, the pressure lies below the water saturation pressure as the mass flow rate falls below ~ 0.25 grams/cm' sec. At a flow rate of 0.1, the pressure lies 11 mm Hg below saturation pressure. The incoming water is thus 10°C superheated. Such a superheat has a profound effect upon the foam structure. As an incipient bubble expands on the tip of a capillary tube, it literally explodes until constrained by neighboring exploding bubbles. An attempt is made in Fig. 9 to represent the bottom layer of bubbles. A thin transition layer separates these bubbles from the warm uprising water below. This thin transition layer consists of an ever widening temperature gradient. Simultaneously the surface of the water is rising, and finally breaks away from the capillary tips. A fresh set of bubbles immediately explodes from the capillary tips. Water is of course entrapped in the edges where the new bubbles meet.

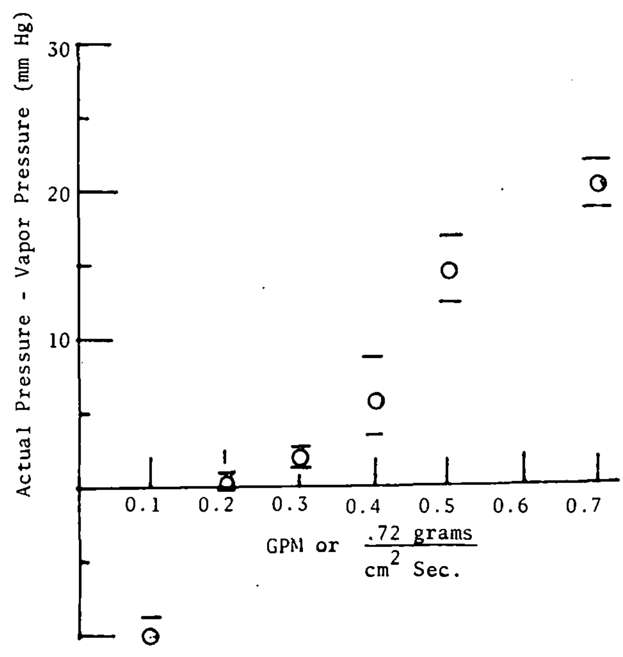

Fig. 8 Dependence of superheat of incoming warm water vapor $\dot{M}$.

Super pressurized incoming warm water

Referring again to Fig.7 , we recall that as the drag coefficient is raised, as e.g. by a rise in mass flow rate, a ΔT vs z curve gets steeper, and eventually reaches a zero ΔT at a positive value of z. Such behavior would imply that the foam rises in the lower part of the column without a drop in temperature. Is this physically possible?

It is conceptually possible, but only if air is present. Hypothesize a foam whose bubbles contain only air. Such a foam would rise, and without an observable decrease in temperature. Such a foam would operate as a simple air lift pump. Once the foam attains such a level at which the pressure is lower than its vapor pressure, the foam could lift itself without the assistance of any air.

In a real foam both air and water vapor are present. A detailed analysis then shows that the curves for a drag coefficient greater than A_{cr} start off with the slope

$$\frac{d \ln T}{dz} = g/p_o V_o ,$$

rather than with a zero slope. Here p_o is atmospheric pressure, V_o is the volume of the injected air per gram of water, referred to atmospheric pressure. Our observations are consistent with this equation.

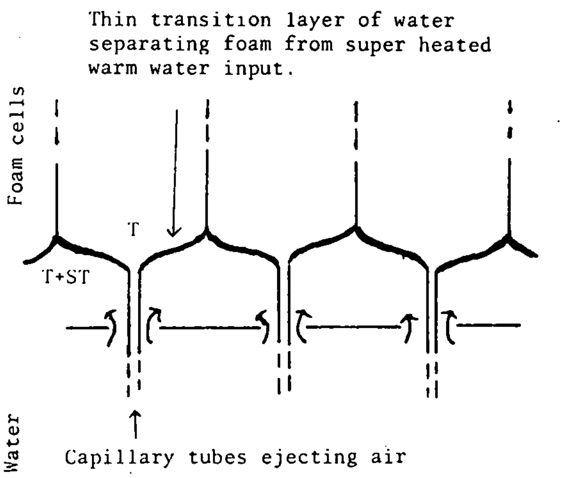

Fig. 9 Bottom of foam when the controls are in superheated range

Effect of temperature of incoming warm water

We now know we must correlate our experimental data with the family of theoretical curves in Fig. 7, rather than in Fig. 6. We next examine to what conclusions this correlation leads. Suppose we change the temperature of the incoming warm water. An analysis of the derivation of the governing equation (7) shows that the drag coefficient decreases as we increase the incoming water temperature. The predicted relation of Eq. (8) is completely verified. A rise in incoming temperature by 10°C decreases A by a factor of 1/3, a decrease by 10°C increases A by a factor of 3.

Laminar Vs turbulent flow

An exhaustive study of the effect of mass flow rate $\dot{m}$ upon the drag coefficient shows that A is proportional to $\dot{m}$, Eq. (9). This proportionality is consistent only with laminar flow. Turbulent flow would give a quadratic dependence. From the derivation of Eq. (7) we conclude that laminar flow implies that the drag coefficient is inversely proportional to the square of the column diameter, at least for moderate increases in D. Going from 4" to 1' diameter will therefore reduce the drag coefficient by $\sim 1/9$. Upon combining (9) - (11) we obtain (12), giving A as a function of T_{inc}, D, and m.

$$A \sim \frac{1}{\rho^2 (T_{inc}} \sim e^{0.112(T_{25°C} - T_{inc})} \qquad (9)$$

$$A - 0.010\,\dot{m},\ T_{inc} = 25°C \qquad (10)$$

$$A \sim 1/D^2 \qquad (11)$$

$$A = 0.0010 e^{0.112(T_{25°C} - T_{inc}} \left(\frac{4''}{D}\right)^2 \dot{m} \qquad (12)$$

A in units of $(°C)^2$ / cm

$\dot{m}$ in units of gram / cm^2sec

Control Surface

In Fig. 6 we have sketched a family of solutions of the equation governing foam flow, Eq. (7), for a range of drag coefficient A. We have learned how, by appropriate horizontal shifting, to transform these solutions of Fig. 6 into solutions which conform to the boundary condition of a usual experiment. Here we know beforehand the difference ΔT of incoming warm water and existing foam temperature, as well as the column height Z. An illustration is given in Fig. 7 of this transformation for the particular case

$$\Delta T = 8°C$$
$$Z = 9 \text{ meters}$$

For each particular case a critical value A_{cr} of the drag parameter A exists such that the associated curve remains unchanged by the above transformation. As a reminder that A_{cr} is determined by the parameters ΔT and Z, we write

$$A_{cr} = A_{ct}(\Delta T, Z)$$

We now distinguish between two types of foam. In the polygonal (p) type, the foam has polygonal cells from the very first layer. In the spherical (s) type, the bottom of the column has primarily spherical cells. All the cells, save possibly those at the surface, become polygonal only some distance up the column. The following surface determines which type of foam is generated

$$A(\dot{m}, D, 25°C - T_{inc}) = A_{cr}\ (\Delta T, Z). \qquad (13)$$

A given by (12)

$$\text{if} \quad A < A_{cr}, \quad \text{we have P foam} \qquad (14)$$

$$\text{if} \quad A > A_{cr}, \quad \text{we have S foam} \qquad (15)$$

The preceding discussion of Eq. (7) implicitly neglected the contribution of injected air to the entropy released by the rising foam. The surface (13) will be influenced by injected air. We are at present developing techniques for greatly reducing this air injection. We anticipate this development will bring Eq. (13) into still better agreement with observations.

Surfactant Requirements

Minimum surfactant concentration

In order to calculate the minimum required surfactant concentration, we must first understand why a surfactant is required. Consider an element of a vertical wall. Under steady state conditions, all external forces acting upon an element of a vertical wall must exactly balance. Gravity is of course acting upon every element of volume. Since the pressure within the vertical wall must everywhere equal the gas pressure on either side, no pressure gradient can exist to balance the downward pull of gravity. The only remaining possible force is a gradient in surface tension. But the surface tension of pure water is constant. A surfactant is required to enable the surface to have a variable surface tension. In this section we find the surface concentration of surfactant molecules necessary to counteract gravity.

In order to simplify our calculation, we shall replace the actual polygonal cell by a spherical cell. By equating the tangential component of the gravity pull to the gradient in the surface pressure Π due to the surfactant molecules, we obtain (16). The solution of this force balance equation is given by (17), and by Fig. 10. The average of Π, (18), is that average surface pressure required to balance gravity. At the low surface surfactant concentrations we are interested in, the surface concentration Γ and surface pressure are related by (19). When all the surfactant is on the surface, this average surface concentration is related to the original volume concentration by (20). Upon multiplying (18), (19), (20) together, we obtain an expression for C, (21).

$$\frac{1}{r}\,\frac{1}{\sin\phi}\,\frac{d}{d\phi}\,\Pi + \rho g\,\frac{w}{2}\,\sin\phi = 0 \tag{16}$$

$$\Pi \gtrsim \rho g\,\frac{w}{2}\,r\left(\frac{1}{2}\,\phi - \frac{1}{4}\,\sin 2\phi\right) \tag{17}$$

$$\bar{\Pi} \gtrsim \frac{\pi}{4}\,\rho g\,\frac{w}{4}\,d \tag{18}$$

$$\bar{\Gamma} = \bar{\Pi}/kT \tag{19}$$

$$C \gtrsim \frac{\bar{\Gamma}}{w/2} \tag{20}$$

$$\left.\begin{aligned} &\gtrsim \frac{\pi}{8}\,\frac{\rho g d}{kt}\ \text{molecules/cm}^3 \\ &\gtrsim \frac{\pi}{8}\,\frac{\rho g d}{NkT}\cdot(\text{Mol.Wt.})\cdot 10^6\ \text{ppm} \end{aligned}\right\} \tag{21}$$

We now estimate the numerical value of C. For this purpose we shall take Octanol as a typical surfactant, with Mol. Wt. of 130, specific gravity of 0.82. We shall also take 2 cm for the final cell diameter, representing a typical 1,000 fold volume expansion. Substitution of the cgs values

ρ (water) $= 1$

$g = 980$

$N = 6.02 \times 10^{23}$

$k = 1.37 \times 10^{-16}$

$T = 298$

gives $C > 4.0$ ppm

Estimated surfactant Cost

The foam power system will be a serious competitor to the closed cycle system for OTEC only if the surfactant cost is reasonable. In order to estimate this cost, we must make assumptions regarding (1) work obtained per unit of warm water input, (2) concentration of surfactant, (3) cost of surfactant per unit quantity. Since the estimated cost of surfactant per unit work is a product of these three factors, one may readily see the effect upon cost of any other set of assumptions.

1 gram of warm water input generates 1 joule of work.

A 20°C drop in temperature of the warm water input would give 2.87 joules as a maximum, a 15°C drop in temperature would give 1.6 joules. 1.0 joules seems a conservative estimate.

(2) We assume

5ppm of surfactant is required in the warm water input.

We calculated that 4ppm would be required for fresh water. It is generally believed that sea water is more foamable than fresh water. The assumption of 5ppm for ocean water therefore seems conservative.

(3) We assume

The surfactant cost will be \$12/barrel.

In the absence of better estimates, we take the cost of surfactants to be identical to that of petroleum.

These three assumptions lead directly to the following conclusion:

The surfactant cost will be 2 mils/kwhr.

Surface slippage

From Fig. 11 we see that the bottom of a cell interior has a high surfactant concentration, whereas the further side of the cell wall, which is the top interior of the cell below, has a small surfactant concentration. This existence of surfaces of high and low surfactant concentration separated only by a thin wall, will give rise to a downward drift of surfactant molecules, with a corresponding downward motion of the interior surfaces. The surface interior thus acquires a slip velocity with respect to the individual cells.

We obtain the slip velocity v_s from the five independent Eqs. (22) - (26). Eq. (22) equates the downward flux of surface surfactant molecules across the equator of a cell interior, to the outward diffusive flux across the lower half of the cell wall. Eq. (23) gives the foam density ρ in

$$\pi\,d\,\Gamma\left(\frac{\pi}{2}\right)V_S = \frac{\pi}{2}\,d^2\,\frac{D}{w}\,\overline{\Delta C} \tag{22}$$

$$\frac{\rho}{\rho_w} = \frac{3w}{d} \tag{23}$$

$$\rho V_f = \dot{m} \tag{24}$$

$$\Gamma = \lambda C \tag{25}$$

$$\Delta C = \frac{1}{\lambda}\,[\Gamma(\pi-\phi)-\Gamma(\phi)] \tag{26}$$

$$\frac{V_S}{V_f} = \frac{3\rho_w D}{\dot{m}\lambda} \tag{27}$$

terms of the d/w ratio. Eq. (24) states the constancy of the mass flux along a column of constant cross section. Eq. (25) expresses the constant ratio between the surface surfactant concentration and the volume concentration in equilibrium with the surface. This relation is valid at the lower concentrations we are concerned with. Eq. (26) is derived from (17). These five equations lead directly to (27). This equation tells us that

$$V_S << V \text{ only if } \lambda >> \rho_w D/2\dot{m}$$

In cgs units

$$\rho_w = 1$$

$$\dot{m} = 1$$

$$D = 10^{-5}.$$

We have

$$V_S << V \text{ only if } \lambda >> 3\ 10^{-5}.$$

We present Table I to illustrate how the aliphatic alcohols are more effective in retarding surface slip the longer their carbon chain.

Table I

Alcohol	Butanol	Hexanol	Octanol
λ (in cm)	4×10^{-6}	4×10^{-5}	4×10^{-4}

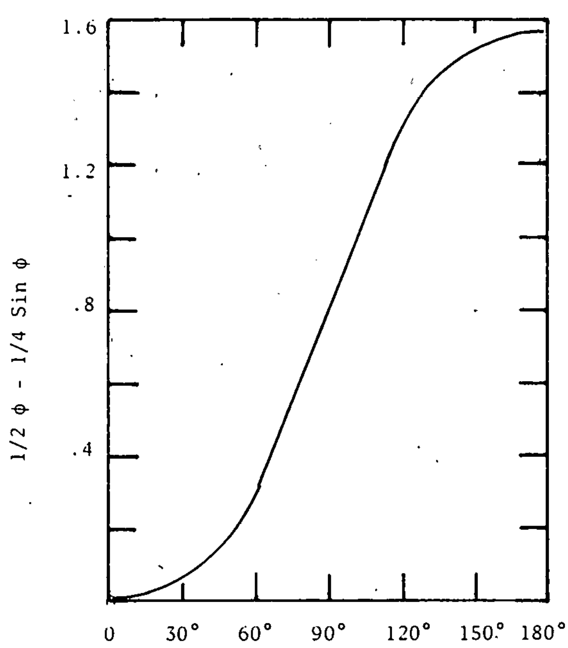

Fig. 10. Surfactant distribution function

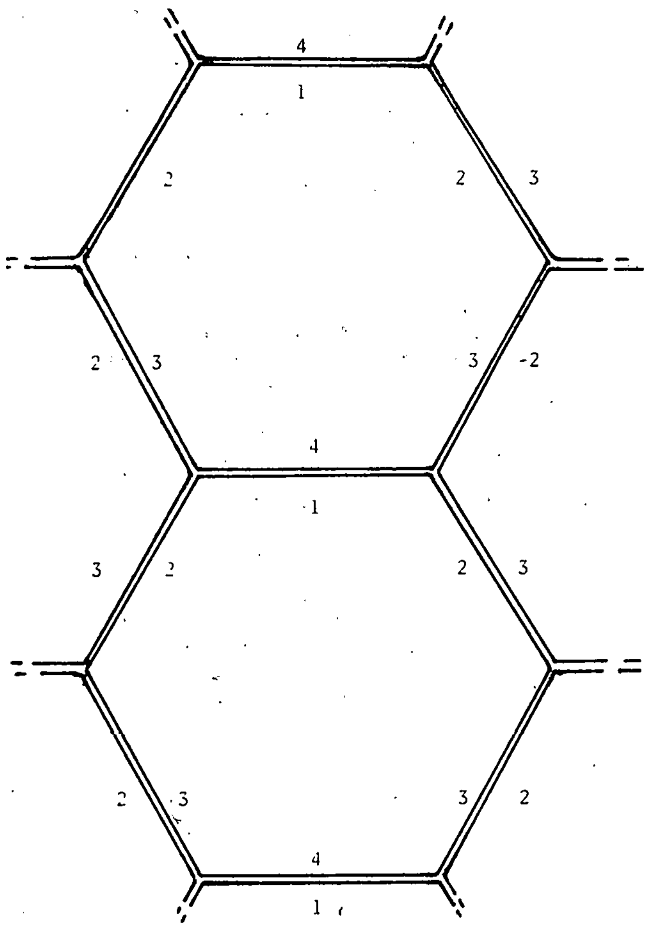

Fig. 11. Illustration of surfactant distribution in a foam, and of its concentration gradient across the foam walls. Numbers are surface concentration in arbitrary units.

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DESCRIPTION AND STATUS REPORT OF A PROGRAM TO DEFINE SEAWATER-SURFACTANT INTERACTIONS IN RELATION TO THE FOAM SYSTEM

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Abstract

The scope of a project to investigate surfactant-sea water interactions with the objective of minimizing quantities of surface active materials is described. The second phase of the project is a determination of the surface tension variability of the upper layers of sea water. The first surface tension measurements showed some sample variability at the 10 meter level. This was noted in two different types of measurements and is probably due to "biological products". The foam generator has run for several hours, including one sea water run.

Nomenclature

γ_0 = Surface tension of component 1.
γ = Surface tension
π = $(\gamma_0-\gamma)$ = film pressure
s Superscript denoting surface.
C Concentration in moles/liter or moles/cm^3 (maintain consistency)
R Gas Constant
T Absolute Temperature
G Gibbs function. In this case, free energy of desorption of the surfactant into the solution.
t Stability time in sec.

Introduction

The problems that we are addressing are: What is the variability of sea surface tension? How will that variability affect foam transport, and can sea water efficiently support foaming? What is the least amount and least expensive surface active agent that can be used in a Foam Plant?

As is well known, surface tension may be expressed as free energy per unit area. The chemical potential or partial molar free energy defines the equilibrium distribution of species in the system. Thus, as the hydrodynamic forces change the shape of the surface, the response of the surface through the Young-Laplace equation will cause surface tension and surfactant adsorption gradients which can lead to free energy driven mass flows.

The chemical potential in the surface[1] is a simple function of mole fraction and film pressure, π. This surface potential may be equated to well known bulk chemical potentials to obtain equilibrium distributions. In essence, inhomogeneities in chemical potential will tend to vanish with time. Rates, however, will be critical. These will be limited by diffusion coefficients and activation energies.

The problem will be to make certain that the correct concentration of surfactant is available on the foam cell surface at all times. A 10° drop in water-water vapor for a Beck[2] cycle leads to a 1000 fold increase in specific volume of the system. If the number of cells does not change, the surface area will increase by a factor of 100. The solution volume will not change.

At the top of the column Zener[3] has estimated that the fully expanded foam will need an adsorption of 10^{-4} moles per liter of solution below. Using the foam column estimated bubble dimensions and the above expansion, we estimate that an objective should be $\sim 10^{-11}$ moles/cm^2. If we compress the film through two orders of magnitude 10^{-9} moles/cm^2 (17Å^2/surfactant molecule; A solid film) would be found. Surfaces begin to saturate at about 3 x 10^{-10} moles/cm^2 (50 Å^2). In expanding the foam one does not want to have to break Van der Waals bonds i.e. the highest adsorption wanted should probably be about 10^{-10}. It would be desirable to stay in the ideal gas range if possible.

The above guidelines indicate that a fairly soluble surfactant may be needed. The distribution of surfactant between the bulk and surface is approximated by:

$$C^s = C \exp \Delta G/RT$$

The free energy of desorption may be estimated[4] from existing data for many types of compounds. ΔG increases linearly with the number of carbon atoms.

The main difference between this system and many foam applications is that here the surface must seriously deplete the solution if quantities of surfactant are to remain in an economic range.

Sea Water

On going to sea water, we find reports of lowered surface tension in biologically active regions (Kelp beds for example). There are further reports[5] of the development of fatty acid-like film pressures from standing or sparged open sea water. In the foam system one will start with water layers of perhaps 100μ and decrease to about 1μ thus the concentrations of films from macroscopic layers will not be available i.e. the needed surface tension for sea water is that which will develop between $\sim$.01 sec. and 10 sec. The latter is more significant since the need for adsorption may become critical at higher film areas. At this low volume/Area ratio it is unlikely that open sea water impurities will have a significant effect.

The thermodynamics of sea water can be handled on several levels of approximation. First as a 0.5 molar salt solution (surface deficit, = -1.3×10^{-11} moles/cm^2 at 0.5 molar), secondly as an ionic sea water solution with activity coefficients dependent on salinity using ionic strengths for each species.

The first needed measurements are thus the standard surface tension of a sector of sea water from inside the ocean at appropriate warm water depths (0-50 meters). Whatever surface active material is available in the micron range should appear.

Experiments

Fig. 1a. shows the bottom of CEER's foam generator. It was constructed at the CEER sea water facility about 7 miles from the main laboratory. 1b. shows the 30 foot column.

We have solved the problems of foam breaking (Isoamyl alcohol, Ethyl methyl ketone) and trapping. New calculations

Fig. 1b. Overall view of Foam Column.

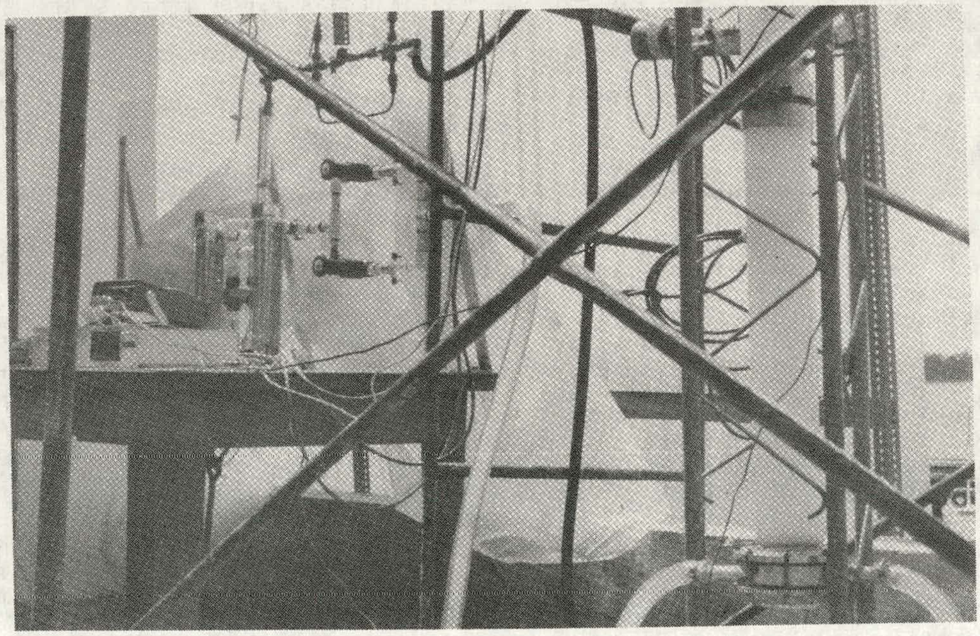

Fig. 1a. Bottom of UPR Foam Column showing Foam and Generator Proper

by Zener[5] indicate that to obtain continuous expansion of foam we must limit foam flow. Maximum flow rates at which Beck cycle conditions can be maintained will be estimated at a given ΔT and then the minimum concentration necessary to support such foam flow will be measured. The column has operated in a steady state mode for several hours. The condenser system will be modified to better control ΔT.

Sea Water Variability Experiments

We have carried out initial experiments on the preservation and handling of sea water. We were able to make initial measurements within about two hours of collection. We see no change in water for about 24 hours. Tension then becomes time variable (the scale of minutes) with, however, fresh surfaces still showing high readings for another period. Initial water was collected from the CEER dock (highest salinity) and about two miles into the bay, unfortunately in a river plume (low salinity).

We have also made the first OTEC site measurements. Data for samples from the Vieques and Punta Tuna sites are given in Table 1.

Sea Water Foam Stability

To obtain immediate, on board measurements, the bubble height of sea water in a tube was measured at given nitrogen flow rates.

$$t = \frac{\text{Bubble volume}}{\text{Air volume/sec.}} = k\,\frac{\text{Height of Foam}}{\text{rate}}$$

Samples collected and measured at sea showed that sea water gives times of under or about 0.5 sec. as opposed to <0.2 sec. for laboratory distilled water and 0.8 sec. for old bay water. Results are shown in Table 2. We do not expect much help from organics in the sea. One sample, however, did show a 50% higher stability which corresponded to the low and variable reading in Table 1.

Table 1 Surface Tension Measurements from OTEC Site Water

Samples were cooled during transport, Vieques water was measured 1 day after collection; Punta Tuna 2 days. The symbol $\pm$ gives deviation from the average of 2 readings. From callibration attempts, the method probably has an error of about 0.2 dyne/cm.

$$\text{(Surface Entropy)}S = -\frac{d\gamma}{dT} = 0.144 \text{ dyne/cm for clean sea water}^{7}$$

$\Delta(5)$ gives undisturbed surface tension change after 5 min. All measurements were made with water that was permitted to stand for 20 min.. Then the surface was swept clean and measurements made on the cleaned surface.

Data are corrected to 20°C. Temperature measurements were made using a very small responsive thermocouple. T varied from 20 to 21°C.

Punta Tuna γ; $\Delta(5)$	Depth in meters	Vieques γ; $\Delta(5)$
74.09 + 0; -0.81 (2 pm)	0	74.17 $\pm$ 0; -0.19 (11 am)
72.62 $\pm$ .93; -2.2 (3 pm)	10	74.00 $\pm$ .08; -0.33 (4 pm)
74.14 $\pm$ .09; 0 (10 am)	25	74.08 $\pm$ .10; 0 (1 pm)

Note: Readings were taken on 4/23 at given time.

More extensive measurements from duplicate casts at 0, 10, 20, 30, 50 meters depth off Punta Tuna and Vieques during the June cruise gave about 74.1 dynes/cm with little variability over a three week period. We conclude that the surface tension of clean open ocean water does not change much and does not show much variability. The comparison with near shore measurements indicates that we should try to bring water in from deep sites to avoid land generated contaminants.

Table 2 Foam Stability in Seconds

Air Velocities were between 750 and 1400 ml/min., P = 1.04 Atm.. T $\approx$ 30°C. Lower flow rates did not cause complete liquid surface coverage.

DEPTH	PUNTA TUNA APRIL	JUNE		VIEQUES APRIL	JUNE	
0 Meters	.50	.44	.41	.50	.45	.43
10	.50	.43	.41	.47	.50	.43
20		.41	.41		.45	.43
25	.69 (AM)* / .50			.54		
30		.41	.41		.45	.43
50		.45	.47		.45	.43

* This sample corresponds to low surface tension sample in Table 1.

Biologists have suggested that plankton migration or biological gelatinous planktonic animals trapped in the sampler could have caused the anomalous measurement. Multiple casts were made in June. These did not show any high stability times or low surface tension.

Our preliminary data indicates that it would be feasable to set up a rugged tensiometer (we have a Rosano that seems fairly reliable.) on shore and go out and obtain water that is needed since the water seems to retain its tension for over a day.

Choice of Surfactants

Surfactants have hydrophillic heads with hydrophobic tails. The tail must be long enough to keep most atoms that arrive in the surface ($\Delta G/CH_2$ = 600 or 700 cal/mol). The approach will be to vary tail length for various alcohols and oxyethylene alcohols $R(O\text{-}CH_2\cdot CH_2)OH$. It is likely that non ionics will be least affected by sea water. Once an idea of optimum straight chain length is found and the needed head group is, determined one may look at the economics of other types of groups.

The solubility and diffusion requirements both seem to point to short groups but, C_5 to C_7 alcohols are, at least at high concentration, antifoaming agents. It may be that small amounts of surfactant could be added to a bubble surface with a relatively small alcohol in the solution. Approaches such as this will, however, have to be more quantitatively examined.

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Acknowledgement

I would like to thank Prof. C. Zener and T. Fort for aid and encouragement.

The tower was built and operated by Mr. Alvaro Brusi who has been indispensable. Surface tension measurements were carried out by Mr. José López Merced, Technical Assistance was lent by Mr. G. Benítez and E. Morales. Mr. T. Morgan carried out the onboard measurements.

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DISCUSSION

Question: Regarding the concern, raised by Dr. Ridgway, that in a pipe of large diameter, the wall shear forces may not be sufficient to prevent the foam from falling on one side of the pipe while rising on the other side, I would like to ask the following questions: (a) Is there any theoretical reason to believe that an instability of that kind will not occur? (b) Would an experimental test of this possibility be feasible? (c) If instability occurs for square ducts of side < S, could you suppress the instability by dividing the large pipe cross section into a number of square ducts of side < S? That might require that an entrance flow resistance be provided for each duct, to ensure that every duct has upward flow.

M. I. Kay: At low mass transport rates less than about 0.05 gpm we do see such a flow pattern, especially, I believe, when ΔT is not too high. As the flow rate rises, we still see some difference in flow, i.e., true cylindrical symmetry does not exist, but no down flow. I cannot answer (c) at this time. I also note that the runs which I am describing were carried out in July 1979. These results occurred with surfactant concentrations of 500 to 1000 ppm. Further seawater seems to stiffen neodol foams and may alleviate the problem if indeed there is one at faster flows.

DESIGN OF LAND-BASED, FOAM OTEC PLANTS FOR BOTTOMING CYCLES

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Abstract

Open cycle Ocean Thermal Energy Conversion (OTEC) technology suggests new concepts for the embodiment of commercially feasible bottoming processes to recover energy from, and simultaneously minimize the environmental impact of, hot industrial effluents. Preparatory to demonstrating a Foam Energy Recovery Open Cycle System (FEROCS) at a 1 MW - 10 MW scale, a structural design was initiated for a unit 380 ft. high visualized as an inverted, vertical, reinforced concrete U tube of 36 ft. I. D. with walls 11 in. thick. The structure is feasible based on present construction practices with reinforced concrete in Puerto Rico. It would cost approximately \$1.4 MM and consume 3,800 yds^3 of concrete and 860 tons of reinforcing steel. To accelerate the demonstration of FEROCS, it is proposed to utilize artificially created temperature differences that can be readily obtained between industrial thermal effluents, for example flue gases at 250°F from fossil fuel fired steam generating plants, as the heat source and ambient air as the heat sink. The concept is presented including use of different scrubbing-working fluids.

Introduction

The technology being developed for open cycle OTEC systems open new fields for the embodiment of commercially feasible processes to recover electric energy and simultaneously minimize the environmental impact of hot flue gases from fossil fuel fired furnaces. The approach would make their pollution abatement more economically attractive yielding "clean, low-temperature" flue gases while recovering a significant portion of their thermal energy as electricity. The ease with which the embodiment of the idea appears to be applicable would open for use lower quality fossil fuels **e.g.,** those of higher ash and higher sulfur content, without fear of the air pollution caused by their flue gases. A relatively simple scrubbing step in properly modified, commercially available scrubbing towers would clean and cool the gases, yielding a hot liquid which after proper cleaning would serve as the working fluid for open cycle systems using ambient air as the heat sink.

The Foam OTEC[1] system makes testing of this concept possible. The concept could be applicable just as well in a foam tower to raise the potential energy of the scrubbing fluid or in a convergent-divergent nozzle to increase the kinetic energy of the foam to move a water wheel that would drive a generator.

Contrary to the OTEC scheme which visualizes using the ocean surface waters as the heat source and the ocean deep waters as the heat sink, the approach described here visualizes utilizing the hot flue gases from fossil fuel fired furnaces and other industrial thermal effluents as the heat source and atmospheric air as the heat sink by the use of judiciously selected scrubbing fluids with presently known direct contact scrubbing and heat transfer technology.

A potential scrubbing fluid could be plain water, but higher-boiling organic liquids or oils will probably permit higher energy recovery rates. Glycols or other high-boiling liquids like silicone oils and high boiling plastizicers might serve the purpose just as well if not better than water in the presence of small amounts of water to help the foaming, cooling tower operations and the separation of the phases. Because of the nature of the scrubbing open-cycle working fluids being considered, and the temperature levels of the process in addition to the foreign matter that the fluids will tend to accumulate, it is doubtful that it will be necessary to use a non-condensable gas and detergents to help generate the foam necessary to couple the vapour and liquid phases especially in convergent divergent nozzles.

Figure 1 shows the embodiment of the process.

Description of the Process

The following description refers mainly to the recovery of energy from hot flue gases. The concept is equally applicable to other thermal effluents. The principal change would be in Unit I,

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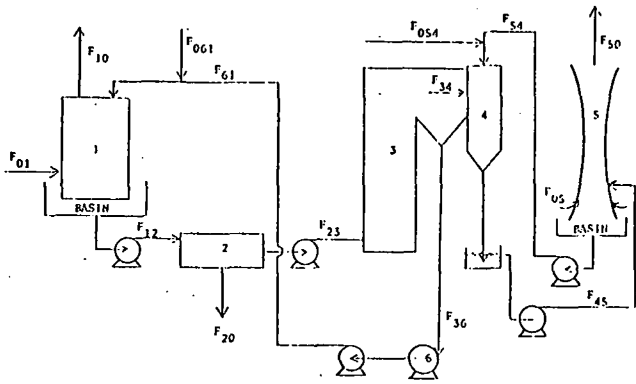

Fig. 1 Open cycle to recover energy from hot flue gases and to minimize their impact upon the environment.

where the waste energy is transferred, preferably by direct contact, to the open cycle scrubbing-working fluid.

Unit I

A scrubbing/cooling tower where the hot flue gases F_{01} will be scrubbed of solid particles, sulfur and NO_x compounds and cooled to approximately 95°F - 100°F. The temperature of F_{01} will range upward from approximately 250°F depending upon the amount of sulfur in the fuel. Higher sulfur contents will call for higher F_{01} temperatures to prevent corrosion of the preheaters and economizers of the fossil fuel fired furnaces because of condensation of H_2SO_3 and H_2SO_4. F_{10} will leave the tower at 95°F - 100°F "cleansed of sulfur, NO_x compounds and solid particles" containing some water and a small amount of organic scrubbing liquid. The scrubbing agent F_{61}, plain water or higher-boiling liquids of the most thermally stable kinds, will enter the top of Unit 1 in a fashion counter-current to the flue gases and collect at the tower basin as a liquid at 212°F if water, or higher temperatures if higher boiling fluids are used. The flow F_{12} will contain the solid particles, H_2SO_3, H_2SO_4 and some of the NO_x compounds. F_{061} and F_{054} will be make up streams.

Unit 2

A system used to treat F_{12} to remove the sulfur compounds, solid particles and possibly NO_x compounds by sedimentation, filtration or a combination with other procedures. F_{20} could be treated to recover the high temperature scrubbing fluids.

Unit 3

This unit could be a foam tower to utilize the enthalpy released from F_{23} to raise the liquid, thereby increasing its potential energy. The raised liquid is used to drive the water wheel - Unit 6 to move an electric generator. Unit 3 could also be a convergent-divergent nozzle to "flash" the working fluid into a foam and utilize the enthalpy released from F_{23} to accelerate the foam and increase its kinetic energy, which could be used to drive the water wheel-generator, Unit 6. The foam density is much higher than the density of vapor so the size of the equipment will be greatly reduced from the large diameters being considered at present for the Claude-Westinghouse OTEC System.[2,3] This technology is available.

Unit 4

A direct contact spray condenser operating at 95°F - 100°F barometric leg.

Unit 5

A cooling tower to serve as heat sink. F_{50} will leave at 95°F - 100°F saturated with water and minute amounts of the high-boiling working fluid used. F_{05} will be air. F_{45} will be cooled by the evaporation of the water.

Energy Recovery Potential

Case 1

Basis. Low sulfur content liquid fossil fuel fired steam generating plant of 500MW.
Steam cycle and turbine efficiency of 37%.
Boiler efficiency of 86%.
Flue gases at 280°F.
Thermal energy content of F_{01}.

$$E_{01} = \frac{500}{0.37} \times \frac{1}{0.86} (0.14) = 220 \text{ MW Heat}$$

Heat transferred to F_{61} at Unit 1 when using a "high boiling" scrubbing fluid, to be used as an open cycle working fluid.

$$E_{12} = 220 \times \frac{(280 - 95)}{280}$$

$$= 145.4 \text{ MW Heat}$$

(assuming the Cp of the gases remain essentially constant)

Which means:
% transferred = 66%
Assuming that 3% of the working fluid is "lost" with effluent F_{20}.

Energy available in F_{23}

E_{23} = 141 MW Heat
T_{F23} = 280°F; T_{F45} = 95°F

$$Eff_{cycle} = \frac{E_{released}}{E_{gained}} = \frac{C_p \Delta T^2/2T_H}{C_p \Delta T}$$

$$= \frac{185}{2(740)} = 0.125$$

A recovery of 50% of cycle is an extremely conservative value.

$$E_{recovered} = (141)(0.125)(0.50) = 8.8 \text{ MW}$$

which is equivalent to a recovery of 4% of the flue gas waste energy or a 1.8% increase in the over-all generating capacity of the power plant.

The results become more attractive in the case of power plants burning coal as Case 2 shows.

Case 2

Basis. Coal fired steam generating power plant of 1000 MW.

Steam cycle and turbine efficiency of 37%

Boiler efficiency of 80%.

Flue gases at 500°F.

Total energy content of F_{01}

$$E_{01} = \frac{1000}{0.37} \times \frac{1}{0.8} (0.20) = \underline{675 \text{ MW Heat}}$$

Heat transferred to F_{61} at Unit 1.

$$E_{12} = 675 \times \frac{(500 - 95)}{500} = 546.75 \text{ MW Heat}$$

% transferred = 81%

Energy available in F_{23}.

$$E_{23} = 531 \text{ MW Heat}$$

$$TF_{23} = 280°F; \quad T_{F45} = 95°F$$

$$Eff_{cycle} = \frac{280 - 95}{2(740)} = 0.125$$

Assuming a recovery of 50% cycle,

$$E_{recovered} = (531) \times (0.125)(0.50) = 33.2 \text{ MW}$$

for a recovery equivalent to 4.9% of the flue gas waste energy or a 3.3% increase in the over-all generating capacity of the power plant. These results would be obtained with a scrubbing-open cycle working fluid with a thermal limitation of 280°F that limits the cycle efficiency to 14.1%. Using a fluid thermally stable at 500°F (260°C) would have a very marked effect on the cycle efficiency and possibly upon the percent of the waste energy recovered. In such a case:

$$Eff._{cycle} = \frac{500 - 95}{2(960)} = 0.211$$

Assuming again a recovery of 50% of cycle:

$$E_{recovered} = 531 \times (0.211)(0.50) = 56 \text{ MW}$$

for a recovery equivalent to 8.3 % of the flue gas waste energy or 5.6% increase in the over-all generating capacity of the power plant. This would most probably make the pollution abatement of the flue gases more economically attractive. Also, it would open for use the lower quality fossil fuels, those with higher sulfur and ash contents, with less fear of the impact of their hot flue gases upon the environment.

Table 1 gives an idea of the economics involved, especially in Puerto Rico.

Table 1 Order of magnitude of venture economics when applied to flue gases of a 500 M W thermoelectric power plant.

Investment/kw	Investment $MM	Cost of $* / hr.	Sales** / hr.
$	$	$	$
250	2.49	49.8	696
500	4.97	99.4	696
750	7.46	149.0	696
1000	9.94	249.0	696
1500	14.90	298.0	696
2000	19.90	398.0	696

*Interest and finance-related factors ≈ $\$2 \times 10^{-5}$/hr - $ invested.

**In Puerto Rico at $.07/kwh.

The results made it highly desirable to obtain a cost estimate of the structure required to apply the technology of the Foam OTEC system[1] as a bottoming cycle for a fossil fuel fired steam generating power plant. Since all the experimental work until now has dealt with converting the enthalpy released during the controlled flashing operation into potential energy it was deemed most proper to obtain the cost estimate for a foam tower.

A second study will be done soon to obtain a cost estimate of the structure required to apply the technology of the Foam OTEC system to the recovery of energy from low grade sources via convergent-divergent nozzles.

Design of the Foam Tower

The design of the foam tower was initiated as a step preparatory to demonstrating the Foam OTEC system in Puerto Rico at a scale of 1 MW. The size-up of the structure was based on initial experimental results[3] which indicated a power generating density or capacity of 1 KW per square foot of foam generating cross-sectional area when operated at temperature differences of ∿ 12°C between the foam generator and the condenser. At these conditions the foam would rise ∿ 340 ft where it would be broken and separated into its vapor and liquid phases. The vapor would continue down to the condenser and the liquid to a surge tank from which it would go to a water turbine to drive an electric generator. In August 1976 it was realized that the supply of the cold deep ocean water was years in the future. This realization dictated the need for artificially created temperature differences readily obtainable in the island, at the required water flow rates. These requirements fixed the site near one of the liquid fossil-fuel fired steam generating plants by the ocean. Such a site would provide process steam and ambient ocean water to create the temperature differences needed.

Since the unit was to be the tallest reinforced concrete structure in the island, the design was to incorporate all safety factors, especially seismic stresses. Fig. 2, Fig. 3 and Fig. 4 amply justify this design criteria.

Figure 2 shows the topography of the ocean floor around Puerto Rico[4] showing the island to be the crest of a very steep mountain more than 5000 meters high.

Figure 3 shows the location of earthquakes relative to Puerto Rico.

Figure 4 shows the frequency of earthquakes that have affected Puerto Rico since 1915.

In essence, the unit consisted of a structure 380 ft high visualized as an inverted, vertical, reinforced concrete U tube of 36 ft. I. D. and walls 11 in. thick. A water storage tank (surge tank) with a capacity of ∿ 60,000 gals. was located at the top between the two vertical cylinders at a height of 333 ft from ground level. Fig. 5 shows the elevation of the Foam Tower.

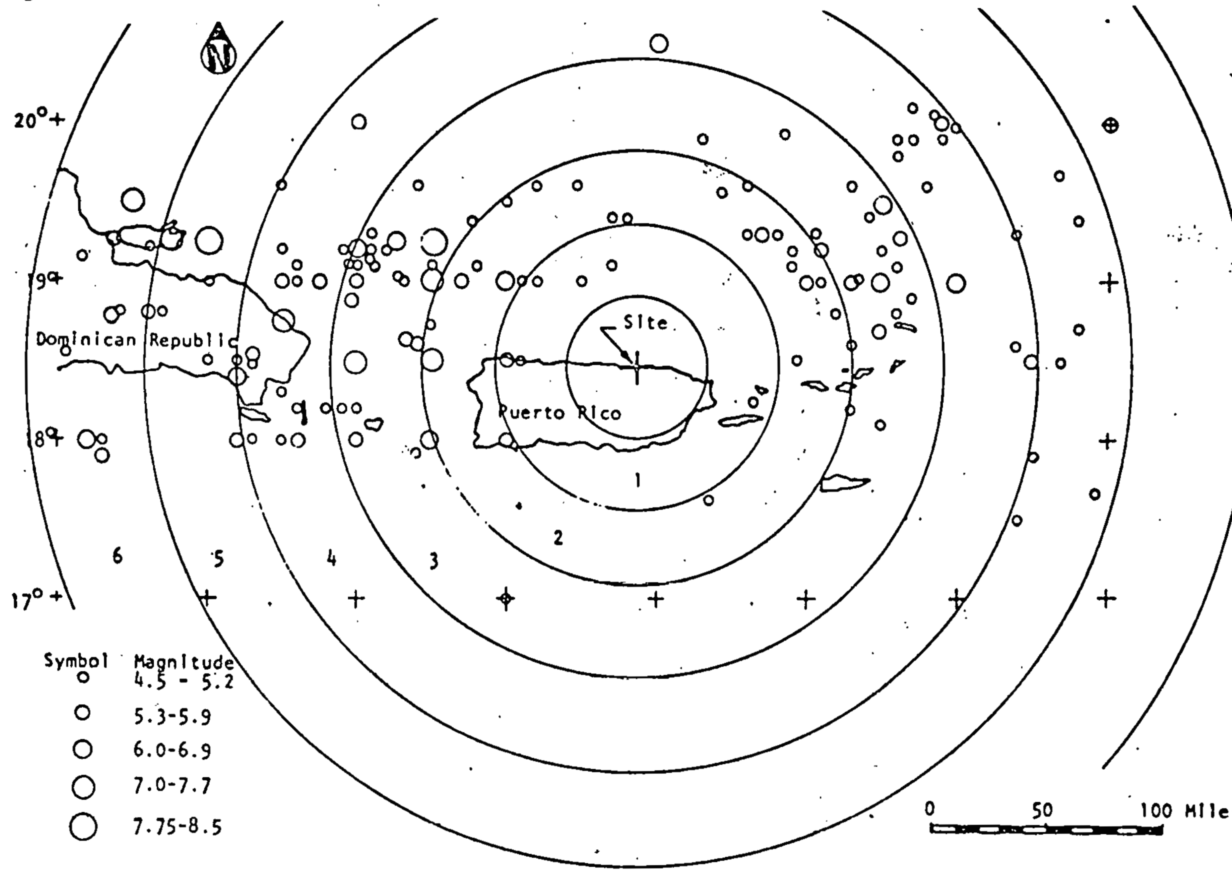

Fig. 3 Map of epicenters for San Juan

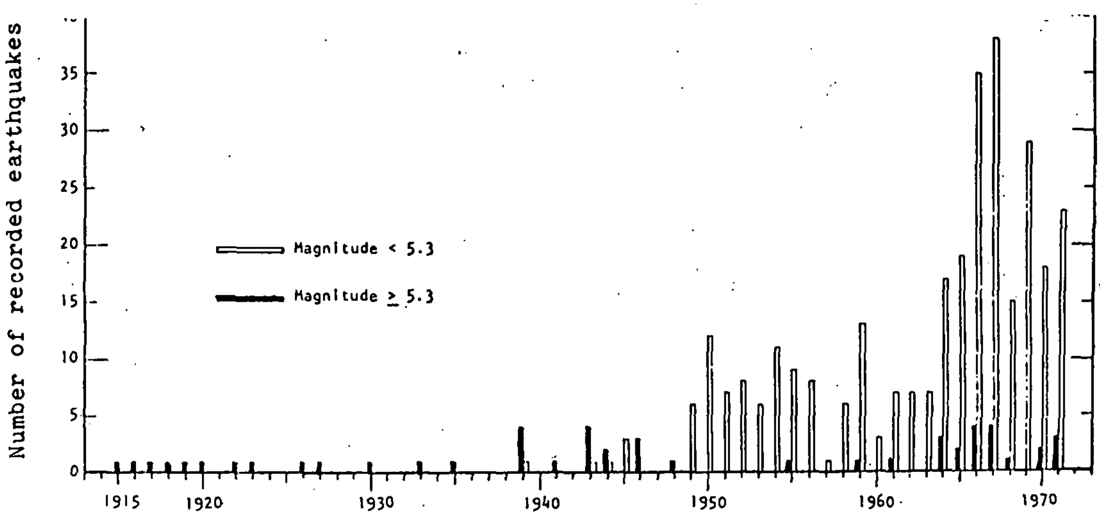

Fig. 4 Frequency of earthquakes within 200 miles of Puerto Rico.

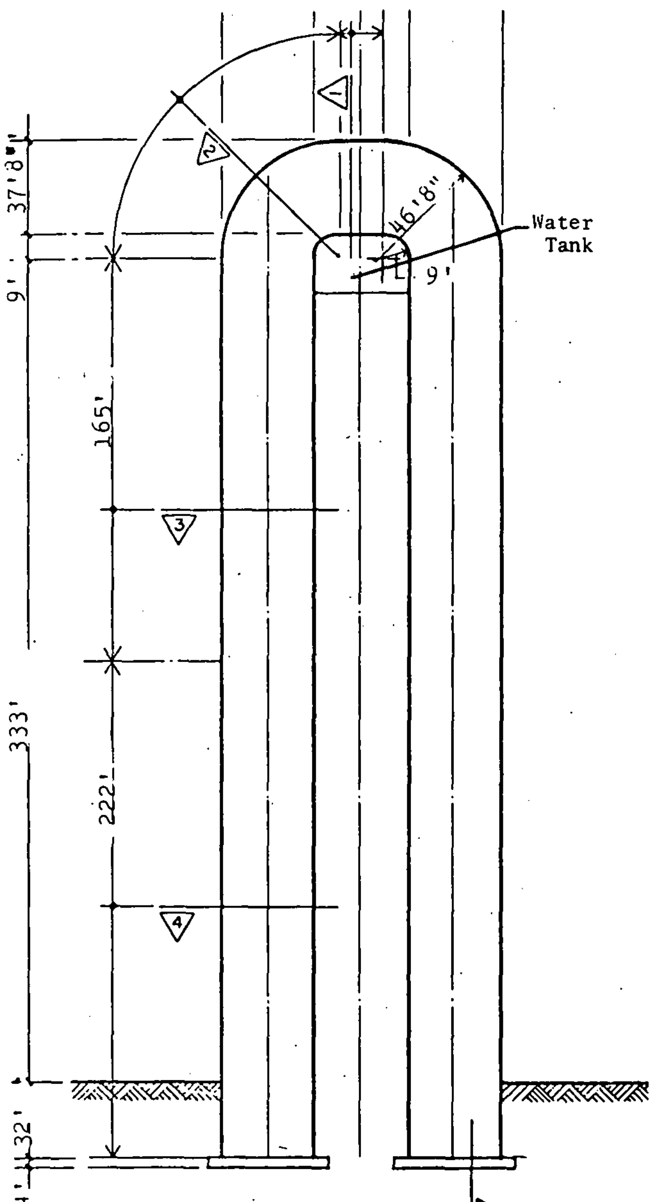

Fig. 5 Foam Tower

The structure is feasible based on present construction practices with reinforced concrete in Puerto Rico and would cost approximately $1.44 MM, consume 3,800 yds.^{3} of concrete and 860 tons of reinforcing steel for a construction cost of ∿ \$372/yd^3 of concrete and a capital investment of ∿ \$1440/KW. The high investment per unit of power is a reflection of the heavy reinforcement required to withstand the seismic stresses and the low power density achieved from the low ΔT of OTEC which we were initially trying to simulate. But, the application of the controlled foam flashing concept to the recovery of waste energy, or from flue gases, would be done at much larger ΔT which will result in a marked increase of the power generated per unit cross sectional area for foam generation. This will most probably place the investment/KW at a very attractive level in Table 1.

Summary

Application of the open cycle OTEC technology to the recovery of energy from poor thermal sources appears to be feasible. The concept proposed in this report would:

- Permit the use of the cheaper fuels without fear of the environmental impact of their flue gases.
- Facilitate pollution abatement of the flue gases from fossil fuel fired furnaces.
- Eliminate the need for expensive, lined, tall smoke stacks.
- Result in the recovery of at least 5% of the energy presently lost in flue gases and/or augment the power output of steam-electric generating stations by at least 2%.
- Recover a large amount of energy from poor thermal sources.

Convergent-divergent nozzles appear to be more desirable than foam tower for the controlled foam flashing of the scrubbing-open cycle working fluids because of the much higher initial temperatures than originally envisioned for the Foam OTEC system demonstration.

Acknowledgments

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MULTIPLE STAGING OF THE COLD WATER IN THE OPEN CYCLE OTEC SYSTEMS

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Abstract

Using the cooling water of an open-cycle Ocean Thermal Energy Conversion (OTEC) system in a multiple-stage fashion results in its most effective utilization. Such use increases by 2-1/3 times the power production capability per unit mass of cold water, thus reducing the cost of the most expensive single item of an original installation. Later stages could utilize the effluent from previously installed stages. Also, the warmed effluent from the last stage could be utilized to enrich the nutrient value and CO_2 absorption capacity of the ocean waters near the surface. A form of the "six-tenth factor rule" was used to estimate the cost of multiple-stage installations using as a basis the cost of the initial unit. Results are presented relating the cost of the initial cold water supply system, number of stages, and power output per unit mass of original cold water at constant cost per unit of power.

Introduction

Presently envisioned OTEC systems propose to use warm ocean surface waters at approximately 25°C as the heat source and cold deep ocean waters at approximately 5°C as the heat sink achieving a temperature difference of approximately 20°C to drive a heat engine. The heat source is readily available at a low cost. The expensive item is the heat sink or the cold water. Nevertheless, when discussing such systems, reference is almost always made to the energy recovered per unit mass of warm water, the low cost item; apparently taking the availability of the cold deep ocean water for granted. The cold deep ocean water (the expensive item) is used on a "once through" basis, returning the resulting effluents to ocean depths at temperatures corresponding to the temperature of the effluents to reduce possible thermal pollution. The temperature of the deep cold water is raised by approximately 10°C before returning it to the ocean.

This report shows that using the cold water on a "once through" basis, is not the most effective way of utilizing the heat sink. Using the cooling water of the OTEC system in a multiple stage fashion gives better results especially for open cycle OTEC systems. Staging increases by 2-1/3 times the power production capabilities of the open cycle OTEC system per unit of cold water at the same cost per unit of power, thus greatly reducing the cost of one of the most expensive subsystems of an original open cycle OTEC installation, the cold-water supply subsystem.

In addition to increasing the power production per unit mass of cold water, staging the heating of the cold water in multiple units in series should permit returning the effluent from the final unit nearer the surface and to enrich the nutrient value of the surface waters.

We are aware also of the concern existing about steadily increasing CO_2 levels in the atmosphere and the effect that OTEC might have upon it.[1] The potential greenhouse effect should be minimized since staging the heating of the cold water will require less cold deep ocean water for the same total energy recovered.

Nomenclature

m_h = warm water input, mass flow rate

ΔT_h = change in temperature of warm water input

m_c = cold water input, mass flow rate

ΔT_c = change in temperature of cold water input

W = work = enthalpy released by warm water

c_p = specific heat of warm and cold waters

m_{ex} = resultant effluent, mass flow rate

T_h = original temperature of warm water

T_c = original temperature of cold water

T_{ex} = resultant temperature of effluent

C_{cw} = total cost of what can be included as the original cold water system.

Advantage of Staging

Staging of the closed cycle OTEC systems' overall ΔT increases the power output by 2.7 per cent.[2] Our proposal could increase the power output of the open cycle system more than 200 per cent by staging the use of the cold water at the optimum increase of the cold water temperature per stage (ΔT_c/stage), to yield the maximum amount of

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power per unit mass of cold water, maximum $\sum\left(\frac{W_n}{m_{c1}}\right)$

A similar proposal for Rankyne cycle power plants was made by Van Hemlryck[6] from thermodynamic considerations. We relate the cost of the initial cold water supply system, number of stages and power output per unit mass of original cold water at constant cost per unit of power.

Theoretically, an infinite number of stages yields a total power four times larger than a single "once through" unit operating at the optimum ΔT_c. Five stages, as proposed here, yield 2.36 times the power. The optimum number of stages depends upon the reliability of the cost scale-up factors when adding units.

The embodiment of this concept should advance the open cycle OTEC system concept. It should make land based installations much more economically attractive because of the readily available space for subsequent units at sites where the ocean bottom slopes very steeply from the shore; i.e. in Puerto Rico.

Estimating the cost of the initial cold water system has a very high element of risk when going deeper than 500 ft and it must be based on a successful pipeline placement on the first attempt. Bids on a guaranteed installation for the original cold water system will be perhaps more than 5 times the cost of any estimate that can be made at the present time for the cost of the initial cold water system.[3]

Constraints Imposed Upon the Concept of Multistaging the Utilization of the Cold Water

1. The ΔT_c/stage must yield the maximum total power, $\sum\left(\frac{W_n}{m_{c1}}\right)$ maximum per unit mass of cold water regardless of whether a "once through" stage or series of stages use the original "once through" cold water supply.
2. The flow of the original "once through" cold water supply remains constant for one or for multistages at the optimum ΔT_c/stage to yield the maximum $\sum\left(\frac{W_n}{m_{c1}}\right)$.
3. The unit cost of the total power generated cannot be higher with the multistages in series than the cost with the "once through" original installation.
4. An exponent of 0.8 was used to scale the cost of the stage under consideration using the following as a basis:
   a) The total overall cost of the previously installed stages less the total cost of the cold water supply system (as defined).
   b) The overall capacity of the previously installed stages in terms of sum of flows of cool and hot water for which they were designed and fabricated.
   c) The overall capacity of the stage under consideration in terms of the sum of the flows of cool and hot water for which it is designed.
5. The decision on whether to add another stage to the installation is made based upon:
   a) The cost of the cold water supply system expressed as a fraction of the overall total cost of all the stages.
   b) The cost per unit of total power must not exceed the cost of the power obtained from the previous units.
6. The overall cost of the original "once through" stage is defined as:

(Total cost) = (Total cost of the installation) minus (the total cost of what can be included as the original cold water supply system = C_{cw}) plus C_{cw}

$= 1 + C_{cw}$.

As shown in Fig. 1, from the First Law of Thermodynamics there is obtained

$$-W = m_h \Delta T_h c_p + m_c \Delta T_c c_p \quad . \qquad (1)$$

For a reversible flashing:[4]

$$W = \frac{c_p}{2\,T_h}(\Delta T_h)^2 \quad . \qquad (2)$$

Therefore,

$$-\frac{(\Delta T_h)^2}{2\,T_h} = m_h \Delta T_h + m_c \Delta T_c \quad . \qquad (3)$$

For unit m_h ,

$$m_c = -\frac{\Delta T_h}{\Delta T_c} - \frac{(\Delta T_h)^2}{2\,T_h \Delta T_c} \quad . \qquad (4)$$

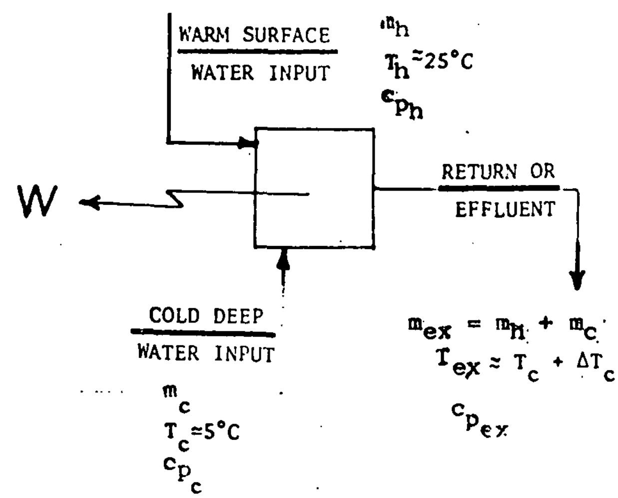

Fig. 1. Foam OTEC System Based on the "Once Through" Cold Water Use

Table 1 Relative Effects Upon m_c

$-\Delta T_h$ °K	ΔT_c °K	$\frac{-\Delta T_h}{\Delta T_c}$	$\frac{(\Delta T_h)^2}{2\,T_h \Delta T_c}$	m_c	% m_c off neglecting work term
15	5	3	.08	2.92	2.74
10	10	1	.02	.98	2.04
8	12	.666	.01	.656	1.52
6	14	.430	.0043	.426	1.01
4	16	.250	.0017	.248	.68

The results of Table 1 show that the second term on the right side of Eq. (4) has such a small effect upon the value of m_c at $\Delta T_H < 10^{\circ}C$ that it is neglected in the following analysis. Then

$$\frac{m_h}{m_c} = \frac{\Delta T_c}{\Delta T_h} \quad . \qquad (5)$$

Combining (2) and (5):

$$\frac{W}{m_c} = \frac{W}{m_h} \cdot \frac{m_H}{m_c} = \frac{c_p}{2\ T_h} (\Delta T_H)^2 \frac{\Delta T_c}{\Delta T_H} \qquad (6)$$

Since $\quad \Delta T_h = T_{ex} - T_h = (T_c + \Delta T_c) - T_h \quad , \qquad (7)$

then, $\quad \frac{W}{m_c} = \frac{c_p}{2\ T_h} [(T_c - T_h)\ \Delta T_c + (\Delta T_c)^2] \quad . \qquad (8)$

$$\frac{d\left(\frac{W}{m_c}\right)}{d(\Delta T_c)} = \frac{c_p}{2\ T_h} [(T_c - T_h) + 2\Delta T_c] \equiv 0 \qquad (9)$$

Therefore $\quad \Delta T_c = \frac{T_h - T_c}{2} = \frac{25-5}{2} = 10^{\circ}C \quad .$

For maximum energy recovery per unit mass of cold deep ocean water, the increase in temperature from the cold water input to the effluent temperature should be $\sim 10^{\circ}C$ for a "once through" installation for a maximum $W/m_{c1} = 0.702$ Joules/gram of cold deep ocean water.

Figure 2 shows the embodiment of the multiple staging of the cold water in the open cycle OTEC systems.

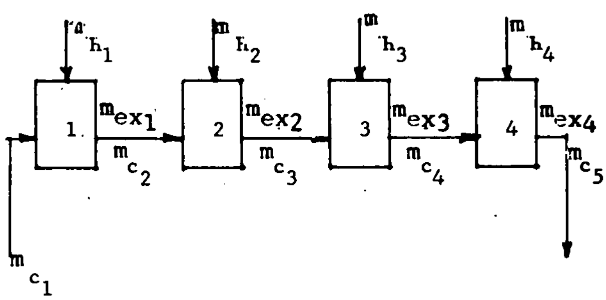

Fig. 2. Multiple Staging of the Cold Water in the Open Cycle OTEC System

For Stage n:

$$\Delta T_h = T_{ex} - T_h \quad ; \quad \Delta T_{h_n} = T_{ex_n} - T_{h_n}$$

$$\Delta T_c = T_{ex} - T_c \quad ; \quad \Delta T_{c_n} = T_{ex_n} - T_{c_n}$$

m_{h_n} and m_{ex_n} will vary from the first stage to stage "n" depending upon the optimum ΔT_c/stage dictated for maximum $\sum\left(\frac{W_n}{m_{c1}}\right)$ for the installation of "n" stages. The stages will increase in size and their cost must be scaled up to determine if it is worthwhile to add another stage. The "six-tenth factor rule"[5] was used based on the total cost and capacity of the previously installed units but using an exponent of 0.8 to be on the conservative side.

The flow rate of cold deep ocean water remains at the same value as for the original single "once through" stage.

Figure 3a shows the optimum ΔT_c/stage at which the installation must be operated to achieve the maximum $\sum\left(\frac{W_n}{m_{c1}}\right)$ from the installation.

Figure 3b shows the maximum obtainable power per unit mass of cold deep ocean water as a function of number of stages when the installation is operated at the optimum ΔT_c/stage and holding m_{c1} constant at the original "once through" unit level.

Figure 3c shows when it is worth economically to add the unit under consideration.

Table 2 shows the sample calculations to determine when it is worthwhile to add the stage under consideration; in this case, the second stage.

Table 2 Staging the Cold Water Two Times in the Foam OTEC System ΔT_c/Stage = 6.85°C

Stage	m_h	m_c	$\frac{W}{m_{c1}}$	Total Cost	(C_{cw})
				0.8 exp.	% of total cost of the two stage installation
1	0.5209	1	0.632	1 + C_{cw}	
2	1.654	1.521	0.460	1.477	
			1.092	2.477 + C_{cw}	39.64

$$\frac{\text{Total Cost}}{\text{Total Joules}} = \frac{\text{Total Cost of "Once Through" Stage 1}}{\text{Power Production of "Once Through" Stage 1}}$$

$$= \frac{1 + C_{cw}}{0.702} = 1.4245 + 1.4245\ C_{cw} = \text{constant}$$

$$\text{Cost of Stage 2} = 1\left(\frac{3.175}{2}\right)^{.8} = 1.447$$

Total cost of 2-stage installation = $2.447 + C_{cw}$

$$1.424 + 1.424\ C_{cw} = \frac{2.447 + C_{cw}}{1.092} = 2.241 + 0.916\ C_{cw}$$

$$C_{cw} = \frac{0.817}{0.509} = 1.607$$

$$= 39.64\% \text{ of total cost}$$

of the entire two satage installation. Thus we can say that:

It is worth while to install stage 2 if C_{cw} is $>$ 0.40 of the total cost of the two-stage installation.

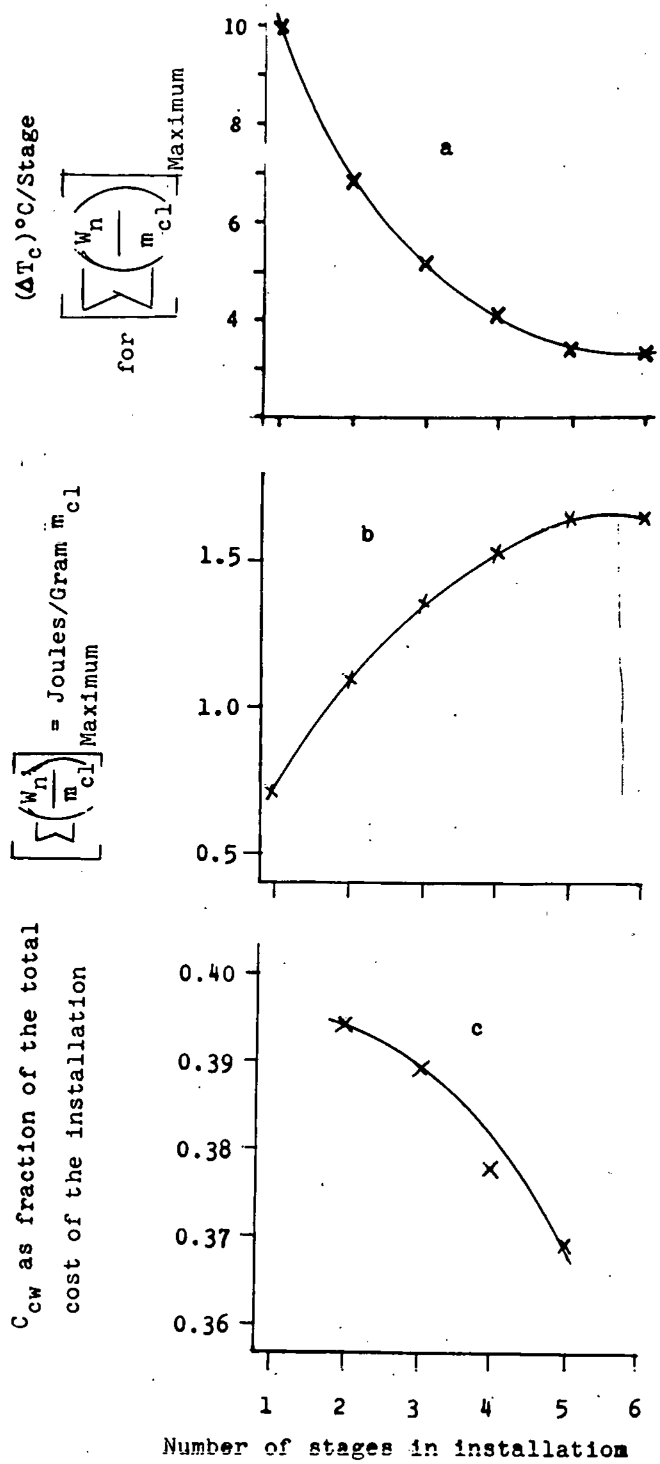

Fig.3 Effect of multiple staging the cold water.

Summary

Staging the overall temperature difference of the open cycle OTEC systems in five steps, at the optimum ($\Delta \dot{T}_c$/stage) to obtain the maximum $\sum\left(\frac{W}{m}\right)_{cl}$ increases by 2-1/3 times the total power production capability per unit mass of cold water at the same cost/unit of power of a "once through" installation.

Acknowledgement

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THE MIST-TRANSPORT CYCLE: PROGRESS IN ECONOMIC AND EXPERIMENTAL STUDIES

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Abstract

A preliminary design and cost analysis of an OTEC power plant utilizing the mist transport cycle has been conducted. Arbitrary choices for the expected lift tube losses were made; other losses estimated from conservative engineering practice. The plant cost is lift efficiency insensitive, and is of the order of $1500/kW. An experimental facility has been constructed by UCLA for the purpose of studying the essential fluid dynamics of the mist generation and mist transport process--a 3-ft-diameter, 24-ft-tall, vertical-flow, two-phase tunnel with supplies of vacuum, and warm and cold water. An orifice-plate mist generator is at the bottom, and a vapor condenser is at the top. The equipment shakedown phase is nearing completion, and data collection is beginning.

Introduction

The mist-transport process for ocean thermal energy has been described in detail in other papers [1,2]. In brief, the plant (shown in Figure 1), consists of a vertical cylindrical hull about 200 meters deep and extending 10 meters or more above the surface. The hull contains an evacuated vertical channel in which the mist lifting process takes place. Warm surface water is filtered and dropped through a penstock to the lower part of the hull, where energy is extracted by a hydraulic turbine and the water passes in fine jets into the evacuated lift tube. The jets break up upon entrance into uniform small droplets and about one percent of the water evaporates. The vapor flows rapidly to a contact condenser at the top of the tube which is supplied with cold water from the depths. The moving vapor lifts the mist droplets by aerodynamic drag and both vapor and mist mix with the cold water stream for return to the sea at a convenient level.

The process is unique among open cycle concepts in that it requires no separation of phases at any point, and lends itself naturally to a spar-buoy type hull-form which is extremely seaworthy and forms an ideal, heave-free support for the long cold water pipe.

An experimental mist-tube facility is undergoing initial test at UCLA as described later. The successful development of an inexpensive mist-generator nozzle plate has been an important part of this effort. The data from the facility will permit more accurate estimates of lift-tube operation modes and indicate limiting values of design variables.

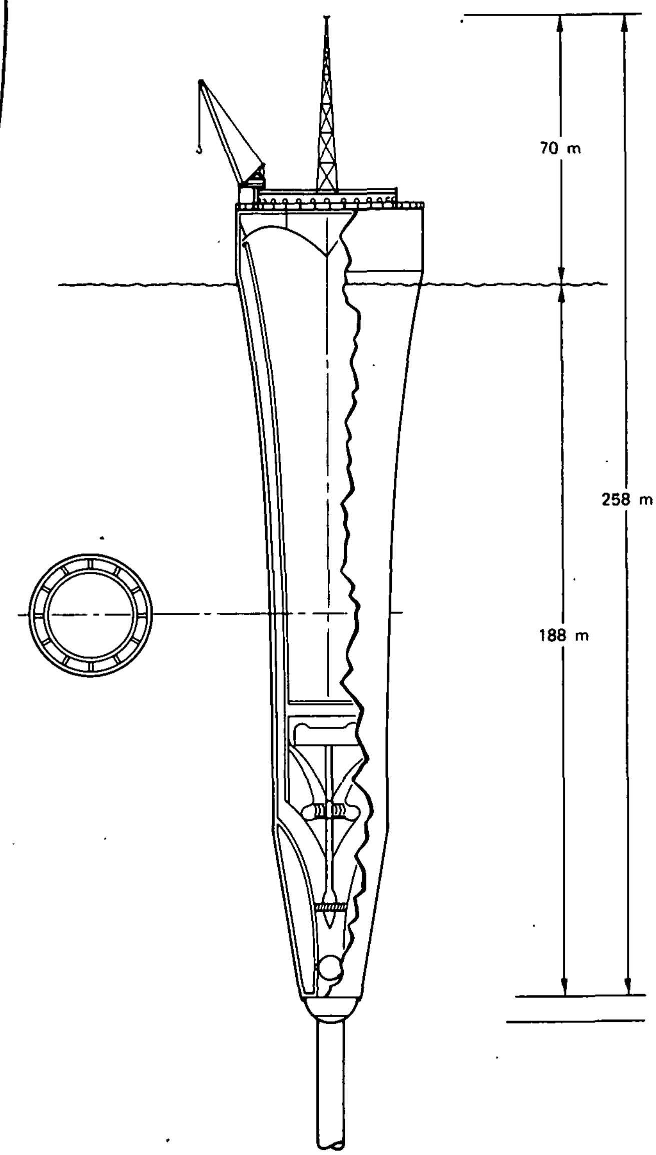


Fig. 1 Conceptual design of 10 MWe (net) mist-flow power plant.

Thermodynamic and engineering analysis of the process has been particularly fruitful in showing that the efficiency of the basic lifting process is not a sensitive factor in overall performance or in cost. This characteristic permits a preliminary design of a sea-going plant to be undertaken with some confidence that later experimental data will not invalidate it. The current design concept is shown in Figure 1 and the first cost estimate is shown in Table 3.

Plant Design Considerations

The problem of designing an economical power plant revolves around the question of where the losses in the thermodynamic cycle are allowed to fall. Some losses may be unavoidably substantial, others may be reduced by spending more money. Once the loss picture is understood, the design parameters can be chosen to minimize the plant cost.

Lift Tube Losses

There are three major losses in the lift tube. The first is the flashdown loss, which is an essential part of the mist generation process. Based on analyses of the requirements for a stable vertical mist-transport, a baseline specific volume of 370 cm^3/gr has been chosen at the start to keep the droplets separated. The flashing takes place irreversibly at constant enthalpy, using the sensible heat stored in the water. The equivalent lift loss is equal to 11.5 meters (a temperature drop of 4.0°C).

Second is the slip loss. The droplets are lifted by the drag work of the moving vapor. The work done by the vapor is the force it exerts on the droplets times the distance the vapor moves. The work done on the droplets is this force times the distance the droplets move, and the difference of these two works is the slip loss. This loss is the one that we have the most difficulty estimating at present.

Third is the exit loss. The flow must leave the lift tube and enter the condenser at finite velocity. Accepting an exit loss has two benefits, first that the size of the structure can be reduced, and second, that the flow may prove to be more stable.

Minor losses include turbulence loss and rain-out loss which results from non-uniformities in droplet size, permitting some droplets to grow by accretion to a size which cannot be lifted.

Mechanical Losses

Filtering loss. To protect the orifices in the mist generator, the warm water taken into the plant must be filtered. The cold water stream requires only trash screens. Head loss in filters can be reduced by using larger areas and low velocities, for which an economic optimization must be made.

Pumping losses. The arrangement of the mist-flow process eliminates the need for a warm-water pump. Filter and penstock losses are subtracted from the turbine inlet head. For the cold water, friction in the long intake pipe and the head lost in the condenser spray nozzles must be provided by a pump. The relative flow of cold to warm water is a crucial design factor which determines, on the one hand, the condensate temperature and hence the available lift, and on the other hand, the size and cost of pumps and condenser. It is likely that direct drive from the turbine shaft will be preferable to motor drive for the cold water pump.

Vacuum pumps must be provided to remove non-condensible gases from the lift tube. The energy lost can be estimated from experience in the operation of sea-water desalinization evaporators.

Turbine-generator losses. The performance of hydraulic turbines and generators suitable for a mist-flow plant is well-understood and its cost and efficiency can be accurately predicted from current hydropower practice, with overall efficiency in the 85-90 percent range.

Summary of Mist Flow Plant Losses

Table 1 gives calculated and estimated losses for an overall plant design used later in this report for cost estimating purposes. It must be noted that important trade-offs between capital cost and efficiency have been only approximated in this design. The detailed work leading up to the parameters chosen is outlined in Reference 5. It is shown there that the cooling water flow must be at least three times the warm water flow, since total water flow reached a minimum at about this mix for a fixed net power output. The study also showed that high lift-tube efficiency is not a prerequisite for a successful mist-flow plant, since the lower specific volumes and vertical heights of a low-lift plant produce construction savings which tend to offset the greater water throughput required. Even with modest lift-tube efficiencies which can almost surely be attained, the mist tube process extracts several times as much energy from a ton of warm water as other OTEC processes. For example, the Lockheed ammonia cycle requires 2.9 tons/sec per MW of net electric output which is to be compared to 1.3 tons/sec per MW for the present cycle.

The analysis to date indicates a large degree of flexibility in the design of the system. In the next section of this paper a brief summary is given of the preliminary cost estimates which have been made using the system parameters given in Table 1.

Projected Cost Estimates

Since much of the equipment required in a mist flow plant is standard in the hydropower industry, and the basic hull structure is stable, inherently seaworthy and simple to construct, preliminary cost estimates can be undertaken with reasonable confidence. The plant design upon which is based the cost estimates for a small demonstration plant producing 10,000 kW is shown in Figure 1, and Table 2. The cost breakdown is given in Table 3, assuming that the cost of the slip forms and tooling is spread over ten identical units. The net cost comes in at a little over $1,500/kW in 1980 dollars. The cost of transmitting the power to shore or of converting it to a portable product is not included in this total nor are development or research costs and first-of-a-kind extras.

Table 1 Summary of Losses Expected in Mist Flow Process

The following data are based on a mist-flow plant using 25°C warm water, 5°C cold water, and 10°C mixed outlet temperature. (Cold/warm flow 3:1.)

Theoretical Lift Height (meters)		164
Lift-Tube Losses (meters)		
Flashdown Loss	11.5	
Slip Loss	23.0	
Exit Loss	10.2	
		-44.7
Mechanical Losses		
Filter and Deaerator Losses	10.0	
Cold Water Pumping Losses	20.0	
Turbine-Generator Losses	10.0	
		-40.0
Net Output (meters)		79.3

Table 2 Reference Characteristics of 10 MWe Mist Flow Ocean Thermal Plant

Hull Diameter at Surface (m)	60
Depth at Juncture of Hull to Cold Water Conduit (m)	200
Displacement (metric tons)	189,000
Freeboard to Upper Deck Level (m)	25
Structure Weight (metric tons)	50,000
Equipment Weight (metric tons)	10,000
Ballast (metric tons)	129,000
Warm Water Intake Flow (tons/sec)	13
Filter Area (m^2)	130
Cold Water Intake Flow (tons/sec)	39
Cold Water Conduit Diameter (m)	7
Cold Conduit Depth (neutral buoyancy pipe) (m)	1,500
Pumping Load (MWe)	2.5
Housekeeping Electric Load (MWe)	1.0
Turbine-Generator Gross Rating (MWe)	14
Net Output (MWe)	10

The UCLA Experimental Facility

The experimental mist-transport tube, being completed at the School of Engineering at UCLA, is seven meters (23 feet) long and 0.9 m (35 inches) in diameter. It is 8 to 10 times shorter than the projected prototype. Therefore, it can be expected to yield only preliminary answers with regard to problems which depend on the residence time of the mist in the flow, such as drop coalescence, and the development and growth of instabilities. The mist generation process--the uniform atomization, and the thermal and dynamic equilibration of the two-phase flow can be fully studied in the space available. The characteristic length for approach to thermal equilibrium between droplet and vapor is estimated to be of the order of 0.3 meters; dynamic equilibrium will be reached within 0 to 5 meters, depending on the water injection velocity relative to that of the fully developed flow (a function of the quality of the mist, i.e., the mist-tube pressure and the water temperature). The facility can be used to a certain extent to simulate any section of the transport process by adjusting the simulated thermal driving potential. It will hopefully yield significant results regarding the behavior of density nonuniformities and velocity wakes produced at the atomizer/injector. On completion of the current program, the mist transport tube length will hopefully be extended to 25 meters.

Table 3 Preliminary Cost Breakdown 10,000 kW Mist Flow Unit

		$ Millions
Major Structures:		5.26
Basic Hull Structure 18,800 m^3 @ $200/$m^3$	3.76	
Cold Water Intake 1000 meters @ $1500/m	1.50	
Process Structures:		1.47
Mist Generator Plate & Support	.85	
Condenser Structure, Piping, Nozzles	.62	
Machinery:		1.35
Hydraulic Turbine, Tailpipe & Control; 14,000 kW @ $30/kW	.42	
Alternator & Accessories 11,500 kW @ $37/kW	.43	
Cold Water Pump 2500 kW @ $30/kW	.08	
Deaerator Vacuum Pumps	.07	
Auxiliary Power Generators 500 kW @ $225/kW	.12	
Cranes and Elevators	.23	
Miscellaneous Equipment:		1.15
Hull Fittings, Valves, Ballast Pumps, Safety Gear	.15	
Filters, Valves, and Controls	.15	
Wiring, Communications Equip., Control Equipment, Lighting	.23	
Crew Quarters, Facilities, and Equipment	.30	
Ballast; 40,000 tons @ $8/ton, Installed	.32	
Total Direct Cost		9.23
Tooling & Forms (share of 10 units)		1.20
Contingencies @ 10%		.92
Indirect Costs @ 20%		1.84
Total Construction Cost		13.19
Interest During Construction @ 15%		1.98
Project Cost		15.17

Overall views of the facility are shown in Figures 2 and 3, which also identify the major components. To reduce the cost of the experimental facility, the tests will be run with the thermodynamics transposed to a higher temperature level. City water (about 20°C) is used as coolant in the contact condenser. Part of the condensate is reheated to a maximum of 70°C by the steam heater and re-injected. The maximum injection capability is 150 GPM; the maximum cooling water flow rate is 400 GPM.

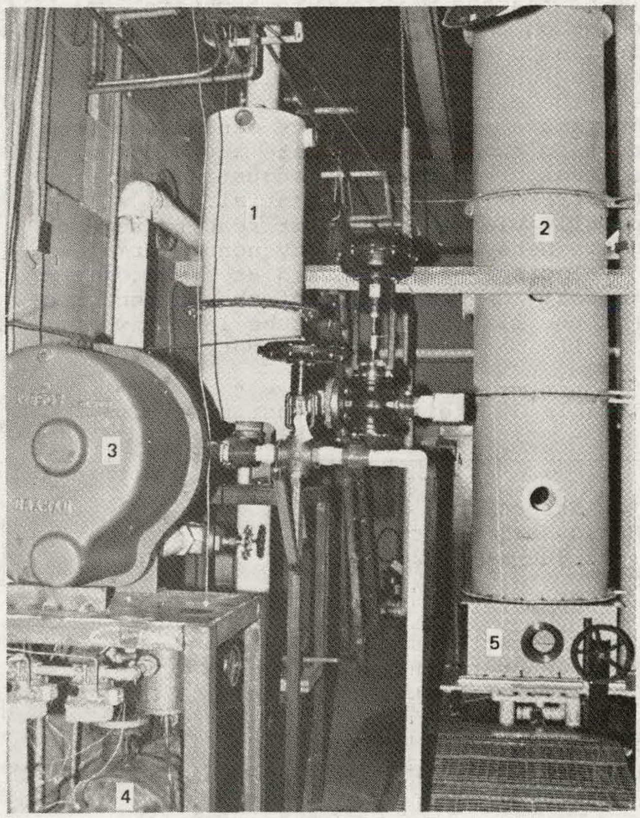


Fig. 2 Lower portion of Mist Lift Tube.
1. surge tank
2. lift tube section (4-ft long)
3. heater
4. circulating pump
5. injector section.

The Injector

A functional sketch of the injector is shown in Figure 4. The water is admitted to a square stagnation chamber, 32 x 32 inches in section and 3 inches deep, through a distribution manifold. This section is thermally insulated from the base; the inlet piping is plastic to prevent heat losses.

The current design calls for six rectangular injector modules, each in an independent frame. These modules are bolted together, and efflux channels are formed between them which permit water that may accumulate on the orifice plate during start-up (and due to fallback) to flow off into a drain. Each module consists of parallel simple, end-supported beams 0.050 inch thick and 1.25 inches deep. These strips are bolted together on 3-inch centers and separated by 0.156-inch washers. The orifice diaphragm is copper-brazed in a hydrogen atmosphere to the top of the beam matrix. The overall active area of the injector is 711 square inches (4587 square cm). All the components are made of type 304 stainless steel.

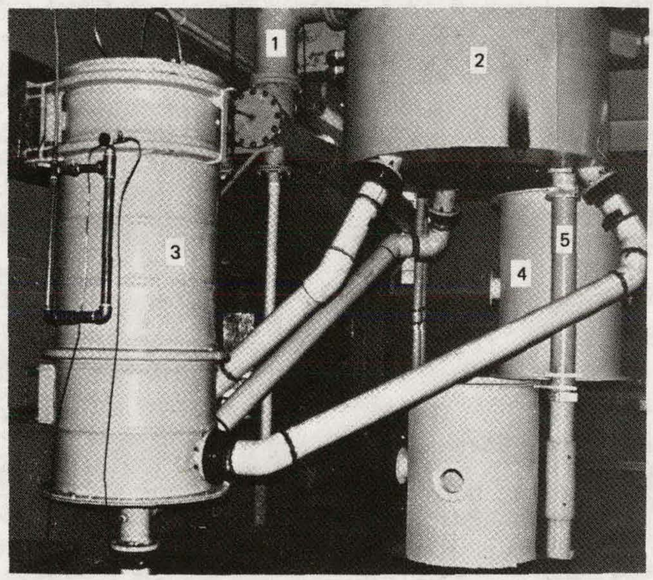


Fig. 3 Top portion of Mist Lift Tube. Last section of tube shown swung out.
1. vacuum line
2. condenser
3. accumulator tank
4. lift tube section (swung out)
5. tube section support column.

The orifice plate is a sheet of 0.01-inch thick stainless steel perforated by electron-beam punching (The Farrel Company, Ansonia, Connecticut) with a matrix of holes having an inlet diameter of 0.15 mm and exit diameter of 0.1 mm (.004 inches). A microphotograph of the exit of one orifice is shown in Figure 5. The cylindrical jets of water issuing from the orifices break up due to Rayleigh instability; Figure 6 shows the phenomenon for a sample row of orifices used in preliminary experiments. The diameter of drops produced by this process is theoretically 1.89 times the jet diameter. Two typical histograms of actual droplet size measurements is shown in Figure 7.

The available theory and experimental data on critical flow out of orifices and short nozzles [4], as well as our own experimentation, indicates that for as large a degree of subcooling as in the present case, internal phase change will not limit the discharge rate; it is only significantly affected by the value of the usual, single-component, incompressible flow coefficient.

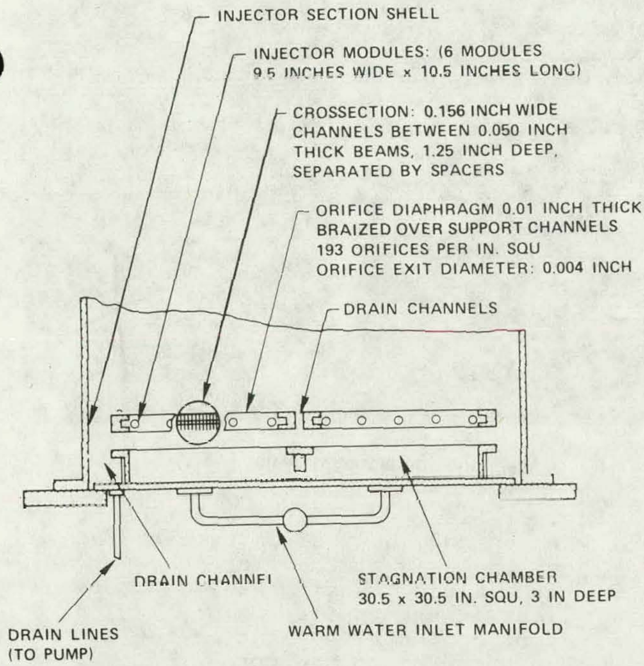


Fig. 4 Functional sketch of the orifice array injector.

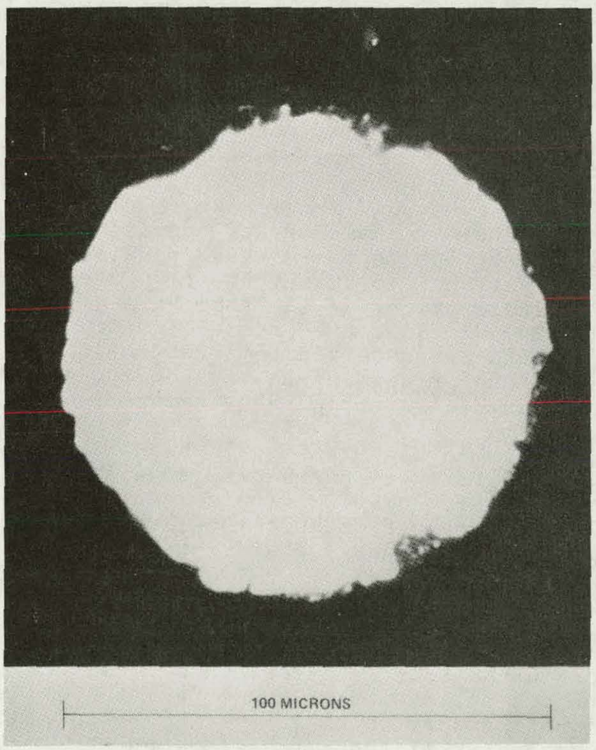


Fig. 5 Microphotograph of typical orifice exit.

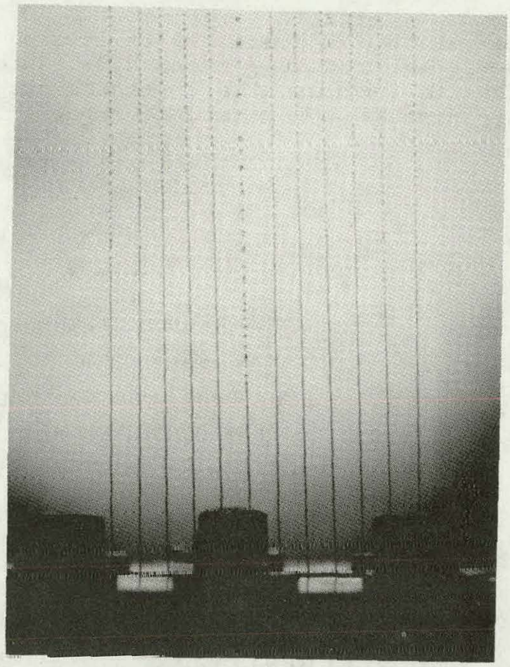
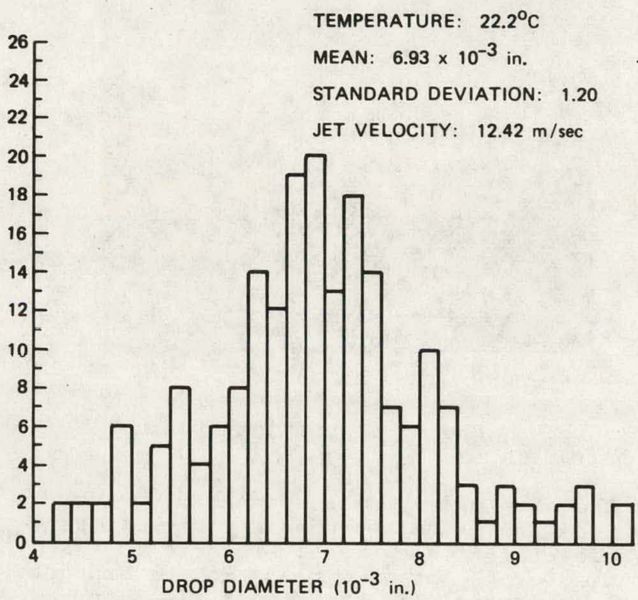


Fig. 6 Photograph of jet breakup.

The development of a successful practical injector required considerable effort and several trials, mostly related to the bonding of the orifice plate to the support. The problem of aligning the orifices and flow channels to given tolerances in manufacturing was solved in the following scheme. The density of orifices is made larger, allowing for a certain number of orifices to fall over the support strips. The rows of orifices are oriented at an angle to the strips, so that the projection of their spacing on a line normal to the strips is equal to the thickness of the strips. It can be shown that under these conditions the total number of orifices falling on the strip is independent of their straightness, and insensitive to the position of the orifice plate on the support frame.

The Condenser

The condenser consists of two stages. The first stage also plays the important role of a water-drop collector. The problem is to prevent centrifugal phase-separation of the mist during the turning at the exit, which would yield a growing water film on surfaces facing the up-stream area of the tube and significant water fall-back into it. This problem was solved by terminating the mist-transport tube by an inverted conical sheet of water generated by the spreading of a downward-directed axial jet to a properly shaped flow-inverting cup. This is shown in operation in Figure 8. The drops impinging on this sheet of water are absorbed by it and carried outward by its radial momentum. Vapor also condenses on the sheet so that it acts as a nearly perfect sink for the two-phase flow directed against it. The angle of the water cone, the thickness of the sheet, and the velocity of the water in it are design variables which can be adjusted to optimize its function and its stability; they are discussed in a project report [3].



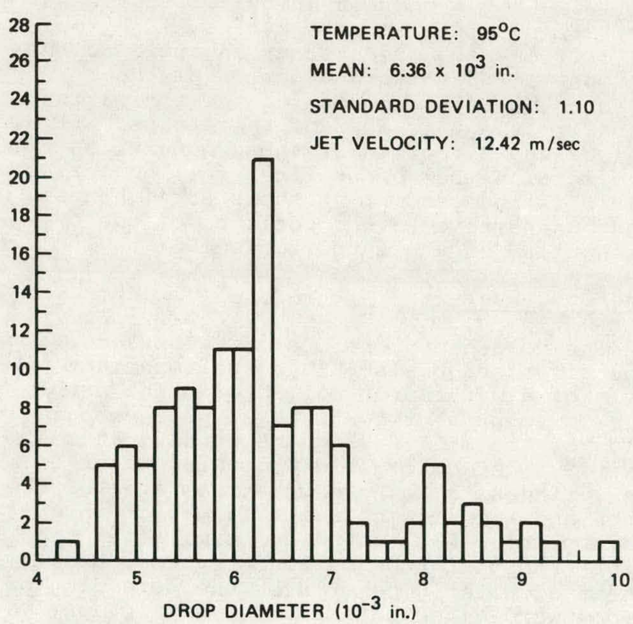
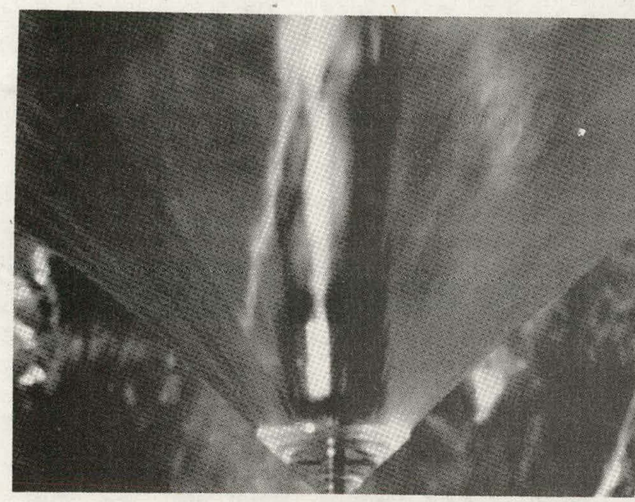


Fig. 7 Histograms of diameters of drops formed from water jets from .004 inch diameter orifices. The ordinate gives the number of drops in each size category.



Mist-tube condenser - "water bell".

Fig. 8 Side view of the water bell. A conical sheet of water centered at the axis of the mist tube serves as a fluid "turning vane". Droplets in the up-flowing mist which tend to centrifuge out as the mist is turned into the condenser are collected on the water-bell sheet and carried out of the tube without disturbance to the flow.

Based on such studies, the ratio of the mass flow of coolant water in the sheet to that of the mist of approximately one was selected. A second stage of contact cooling consists of an array of spray nozzles; the maximum total rate of coolant flow in the condenser is three times the rate of mist flow.

Instrumentation

The mist tube is provided with six-inch access and viewing windows at opposite diameters every four feet. A traversing mechanism is available to survey the flow along the entire diameter of the tube at any section. In addition, the entire circuit is fitted with pressure and temperature (thermistor) probes recording the system operation continuously on a strip-chart recorder.

Diagnostic measurements in two-phase flow are difficult and require sophisticated instrumentation coupled with careful analysis. Two types of measurements are being prepared for the early phases of the testing.

Measurements of pressure and/or temperature gradients along the flow. The problem here is one of accuracy. The pressure variation in the short section of the mist-transport process is of the order of 0.02 psi (1 mm of mercury column). The corresponding change in temperature (the mist can be assumed to be locally in thermodynamic equilibrium) is of the order of 1°C. Instruments capable of this kind of accuracy are available; differential pressures will be measured with a capacitance-type transducer

(Setra Model 239) which has a full-scale sensitivity of 0.1 psi. Differential temperatures will be measured with precision thermistor bridges.

A special instrument is being developed to simultanesouly measure mist density and mist velocity, which we call the spin decay anemometer. It consists of a pair of short cylinders (1.5 cm long and 1.2 cm in diameter) made to spin about the axis of the pair. This "dumbbell" is mounted on the shaft of a miniature magnetic clutch which is driven by a permanent magnet motor when engaged. The axial nacelle containing this machinery is about two centimeters in diameter; the radius to the center of the cylinders is 4 cm.

The dumbbell is spun up to 5000 rpm and disengaged. The history of the decay of the spin-rate contains information necessary to determine the velocity and the density of the particles in the mist separately.

In early May the experimental facility was complete and in operational status for shakedown and calibration of instruments. Preliminary data taking will proceed to some extent concurrently. It is expected that some adjustment and modification of the system may be required during the experimental program to accommodate new objectives. This may include changes in the internal profile of the lift tube by adding contour adapters to the inner wall. Improvement of diagnostic instruments will be a continuing effort. Most of the basic installation will be utilized in the much taller version which will follow.

In September stable vertical mist transport for times as long as a minute was achieved. Quantitative data is hoped for soon.

Acknowledgment

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LAND-BASED APPLICATION OF AN OTEC OPEN-CYCLE POWER SYSTEM*

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Abstract

Ocean Thermal Energy Conversion (OTEC) power-component technology may be applied to many land-based, low-temperature heat resources including rejected heat from thermal plants and federal nuclear facilities, as well as freshwater thermoclines. For utilizing rejected heat from large thermal power plants where OTEC-range cold water is economically available, it seems to be more cost-effective to apply OTEC open-cycle steam turbine technology for additional turbine stages for the conventional thermal power plant cycle rather than to add a separate bottoming cycle system. For utilizing rejected heat from federal nuclear facilities, the open-cycle power system has its advantage because it does not need isolation heat exchangers to avoid cross contamination between the working fluid and the hot water supply. A simple thermal performance model and the thermal analysis of a conceptual rejected heat open-cycle power system corresponding to the binary cycle system studies by Exxon for the Oak Ridge Gaseous Diffusion Plant are presented. Rejected heat open-cycle power systems may operate at a higher thermodynamic state than OTEC systems and are closer to existing state-of-the-art technology. Thus, such systems may serve as intermediate steps toward the development of OTEC open-cycle power components and subsystems.

Introduction

Ocean Thermal Energy Conversion (OTEC) is a low-temperature heat utilization technology. It may be applicable to many land-based, low-temperature heat resources, besides ocean thermal gradients. In the review of the OTEC program within ERDA (now DOE) in August 1977, one of the recommendations for the program made by the Marine Board of the National Research Council[1] was to "Consider the utilization of shore-based rejected heat sources as the first step for long-term evaluation of OTEC components and subsystems." In view of the rapid progress in OTEC technology, the prospects of employing the open-cycle technology in land-based, low-temperature heat resources will be explored.

Among the different kinds of land-based low-temperature heat resources, only the rejected heat from large thermal power plants and federal nuclear facilities may have a sufficient quantity of low-grade heat for significant power generation. The thermal energy stored in freshwater thermoclines exisitng in hydroelectric dams and lakes has also been point out[2] as a possible area in which OTEC technology may be applied.

The heat rejected from thermal power plants is usually at a temperature in the 100°F to 120°F range that corresponds to a power plant condenser pressure of 3 to 3.5 in. Hg. There seems to be no technology barrier to the design and operation of a thermal power plant with a condenser pressure of 1 in. Hg that will yield a rejected heat temperature of 80°F, at most. Thermal power plants with a design condenser pressure of 1 in. Hg can be found in operation in northern states; many of the power plants that cannot be designed to operate at this condenser pressure are mainly limited by the economically available cold water sources (heat sinks) or institutional constraints.

Almost all the OTEC power cycle concepts may be employed for freshwater thermocline power generation and as bottoming cycles for the rejected heat from thermal power plants, if cold water is economically available. Large thermal power plants use water as working fluid. Instead of employing a bottoming cycle using a different working fluid such as NH_3, it would seem to be more effective for a power plant to apply OTEC open-cycle steam turbine technology for additional turbine stages. The additonal very low pressure (VLP) turbine stages will produce power from the further expansion of the 100°F range steam to that of the OTEC condenser pressure and temperature range. Applying OTEC open-cycle turbine technology as an integral part of a thermal power plant where OTEC-range cold water is economically availabe will result in greater cost savings than bottoming-cycle options, because it does not require an evaporator heat exchanger for the secondary working fluid.

The federal nuclear facilities are large electricity users, and 90% of the used electricity is rejected in the form of hot water around 140°F to 150°F. These rejected heat resources, as summarized in Table 1, have been subject of may feasibility studies[3,4] including agro-industrial complexes, district space heating systems, electric power generation, and temperature augmentation for process heat. A binary (closed) cycle system for power generation was analyzed by Exxon for the possible application at the Oak Ridge Gascous Diffusion Plant (ORGDP). A corresponding OTEC open-cycle power system for the rejected heat source will be assessed.

Table 1 Summary of Heat Resource at the Gaseous Diffusion Plants

Site	Heat Available (MW)	Maximum Flow (gpm)	Maximum Temperature (°F)
Paducah, KY	2812	330,000	155
Portsmouth, OH	2250	360,000	135
Oak Ridge, TN	1968	290,000	145

*Research sponsored by the Division of Central Solar Technology, U.S. Department of Energy, under contract W-7405-eng-26 with the Union Carbide Corporation.

Exxon Binary (Closed) Cycle Analysis

A rejected heat utilization study on the ORGDP site was conducted by the Exxon Nuclear Co., under a contract with the Department of Energy. Electric power production from the temperature difference between the ORGDP rejected heat streams and the available ambient heat sinks (i.e., the air and the Clinch River) were analyzed. In the study, 40,000 gpm of 140°F water from one of the mechanical draft cooling towers was the heat source of a closed-cycle power system. A Rankine cycle design program developed by Mechancal Technology Inc. (MTI)* was employed in the power system design analyses. Many different working fluids were analyzed for power generation. Since the operating pressure of the evaporator in a closed-cycle power system is above atmospheric pressure and significantly higher than that of the ORGDP cooling water, isolation heat exchangers are required to prevent working fluid leakage into the GDP cooling water loop and to protect the GDP equipment. The isolation heat exchangers will reduce the available heat source temperature by 3°F and impose additional equipment capital cost and operating power losses. Two heat sinks were considered, a once-through cooling system with available cooling water from the Clinch River at 60°F, and a wet cooling tower system with a design ambient wet bulb temperature at 55°F. The conceptual plant layout was defined in which the tie-in to the GDP for hot water supply and return required 3200 ft of 54-in. pipe each for the hot water supply and return systems. The length of the once-through cooling water loop is 3000 ft with a topographic hydraulic head of 35 ft, and the length of the wet tower cooling water loop is 1000 ft with an additional tower head requirement of 41 ft.

A screening analysis to determine the relative power system costs for binary working fluids with different water condensing temperatures (cooling options) was performed. It was found that the plant cost of an ammonia closed-cycle plant with no isolation loop was about one-half the cost of a corresponding R-113 power plant. The cost of the ammonia closed-cycle plant would be tripled when the isolation heat exchanger loop was added.

The binary cycle R-113 power system with no isolation loop was the base-line design of the Exxon study. The design parameters of the base-line binary cycle system with both cooling options are presented in Table 2.

Thermodynamic Process of An Open Cycle

An open-cycle power system comprises a flash evaporator, vapor expansion turbine and generator, steam condenser, noncondensilbes removing equipment, and deaerator (Fig. 1). A thermodynamic temperature-entropy (T–S) diagram of the open cycle is shown in Fig. 2 where the numerals are the thermodynamic states which correspond to those shown in Fig. 1. Comparing with Fig. 1, state 1 in Fig 2 is the warm seawater at the heat-source temperature, about 80°F. The warm seawater at atmospheric pressure flows to the inlet of a deaerator at state 2 where the pressure is suddenly reduced to a value slightly above the saturated vapor pressure at the corresponding temperature. The majority of the dissolved gas may be released due to the sudden pressure drop. The

*Subcontract by Exxon.

Table 2 Exxon Closed-Cycle Power System Design

Cooling Mode	Once-through	Wet cooling tower
Warm water loop		
Flow rate, gpm	40,000	40,000
Inlet temp., °F	140	140
Pipe diam, ft	4.5	4.5
Pipe length, ft	6400	6400
Evaporator surface area, 10^5 ft^2	3.80	3.75
Parasitic power, kW	560	560
Turbine/generator		
Working fluid	R-113	R-113
Vapor temp., °F	120	120
Condensate temp., °F	75	90
Power output, kW	6325	4160
Parasitic power, kW	80	72
Cold-water loop		
Flow rate, gpm	61,400[a]	39,000[b]
Inlet temp., °F	60	70
Exit temp., °F	70	85
Pipe diam., ft	5	4.5
Pipe length, ft	3000	1000
Wet bulb temp., °F	—	55[c]
Condenser surface area, 10^5 ft^2	2.50	2.07
Parasitic power, kW	1700	2078
Net power, kW	3985	2078
Efficiency		
Turbine/generator, %	87	87
Net power cycle, %	4.2	2.3
Cost, $/kW net	3620	6000

[a]From Clinch River.

[b]From cooling tower.

[c]Design value.

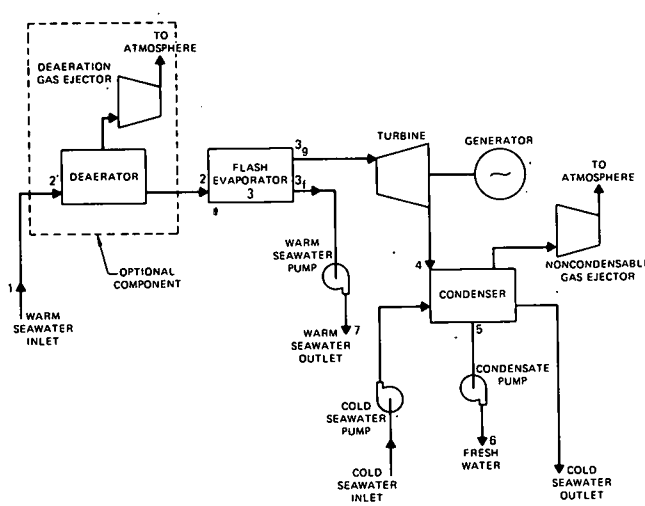
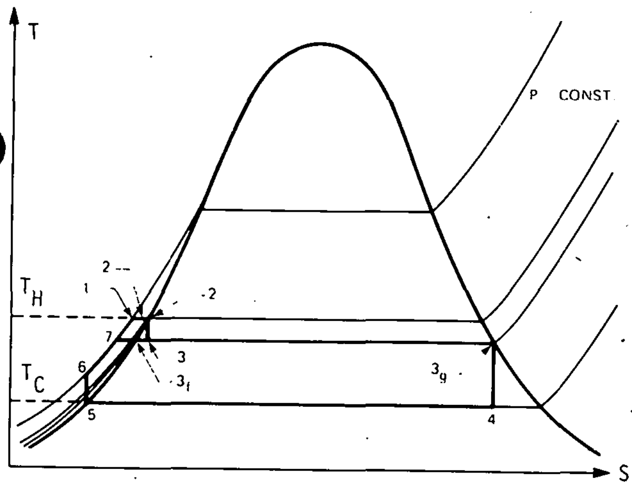


Fig. 1 Schematic of an open-cycle OTEC power system.



SOLAR THERMAL ENERGY IS ABSORBED ALONG STATE 6 TO STATE 1. UNFLASHED WARM SEAWATER IS PUMPED ALONG STATE 3_f TO STATE 7. VAPOR TRAVELS ALONG STATES 3_g TO 4 AND 5. DRAWING NOT TO SCALE.

Fig. 2 A typical open-cycle T–S diagram.

vapor release in the deaerator will be minimized because the pressure in the deaerator is kept above saturated vapor pressure. The deaerated seawater then flows into the flash evaporator at state 2 where ambient pressure is dropped to the saturated vapor pressure equivalent to the outlet steam temperature; this pressure drop is the driving force for evaporation. A slight amount of working fluid is flashed into steam with flashdown temperature drop corresponding to the vapor latent heat requirement. The bulk thermodynamic state in the flash evaporator is represented by state point 3 in Fig. 2. The vapor phase of the working fluid at state 3_g with a mass flow rate of $x\dot{m}$ (where $\dot{m}$ is the total warm seawater mass flow rate and x is the vapor mass fraction) expands in a turbine where the thermal energy is converted into mechanical work. The slightly concentrated seawater with a mass flow rate of $(1 - x)\dot{m}$ at state 3_f is pumped back to the ocean at state 7 where its temperature is lower than the inlet warm seawater temperature. This temperature difference is the "flashdown" of a flash evaporator. The turbine exhaust at state 4 is condensed by giving up heat to the cold seawater and ends up at state 5 as saturated condensate before it is pumped back to the surroundings at state 6. The condensate in a surface condenser is fresh water that can be utilized as a by-product of an open-cycle OTEC power plant. If the condensate is put back into the ocean, it will eventually be mixed with the seawater and travel along the process path from state 6 and 7 to state 1 through a natural convection mechanism originated from the absorption of solar thermal energy by the earth's atmosphere and the oceans.

Thermal Performance Modeling

To investigate the proper thermal design of an open-cycle power system, a simple thermal performance model is derived. It is applied to the ranges of operating conditions that have been established by recent open-cycle studies. The open-cycle thermal performance may be measured by the net power output divided by the condenser heat transfer surface area,[5] and it is dependent on the design of the condenser, system parasitic losses, as well as the design of other system components. The system temperature and mass flow distribution in an open cycle consisting of the flash evaporator, turbine/generator, and surface condenser is shown in Fig. 3.

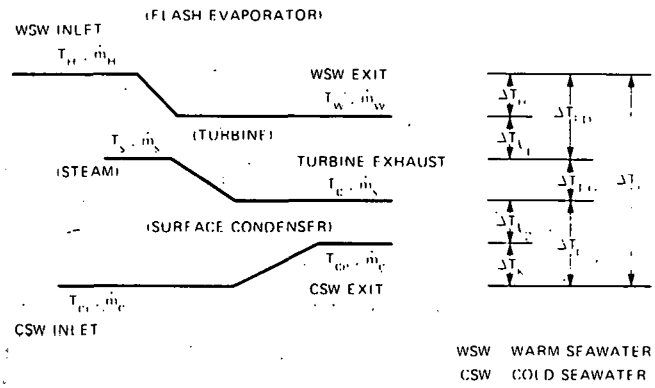


Fig. 3 System temperature (T) and mass flow ($\dot{m}$) distribution.

Combining the heat and mass balance in the flash evaporator and the turbine, neglecting the small sensible heat effect in the steam production, the rates of heat transfer and work production in the power system may be expressed as[6]

$$Q_{in} = \dot{m}_H \, c_p \, (T_H - T_w), \tag{1}$$

$$P_g = \eta_T \, Q_{in} \, (1 - T_c/T_s), \tag{2}$$

$$Q_{out} = Q_{in} - P_g, \tag{3}$$

and

$$P_n = P_g - P_{csw} - P_{wsw} - P_{misc}, \tag{4}$$

in which $\dot{m}_H$ is the warm water mass flow rate; T_H, warm water inlet temperature; T_w, warm water exit temperature; T_w and T_c, steam and steam condensing temperatures; c_p, specific heat; η_T, turbine efficiency; P_n, net power; P_g, gross power; P_{cws}, P_{wsw}, and P_{misc}, parasitic power loss in cold and warm water loops and miscellaneous equipment, respectively.

The log-mean temperature design approach[7] is employed to describe the surface condenser in which the cold water mass flow rate and heat transfer area can be related to condenser water temperature rise, tube water velocity (V), tube lnegth (L), tube size, cold water inlet temperature (T_{ci}), flashdown temperature and water water temperature and mass flow rate.

The optimal thermodynamic state of an open-cycle system can be derived by maximizing an objective function defined as the net power output divided by the condenser heat transfer area (i.e., P_n/A). For given seawater hydraulics, miscellaneous equipment hydraulic losses, and condenser tube size, the objective function will be dependent upon T_H, T_w, T_c, T_{ci}, V, and L. Analytical representations can be obtained for the maximum of the objective function; however, the function values of the optimum state will have be solved numerically.

In optimal performance modeling, different objective functions and optimization schemes may lead to different optimum states. Besides optimization of the net power per unit heat transfer area, a sequential optimization scheme was employed by Dunn[8] in a study of OTEC systems. The sequential scheme maximizes the gross power output first followed by the optimization of the net power per unit heat transfer area.

The simple thermal performance model derived above is applied to an open-cycle power system design with relevant parameters approximately corresponding to one case of the recent system studies in which $\dot{m}_H$ = 758,000 lb/s, T_H = 80°F, T_{ci} = 40°F, ρ = 64 lb/ft^3, c_p = 1 Btu/lb/°F, turbine efficiency = 0.81, pump efficiency = 0.7, hydraulic losses in warm seawater loop = 9.7 ft, in cold seawater loop other than tube bundle = 3.9 ft, and in miscellaneous equipment = 9.0 ft, and smooth condenser tube of 1 1/8 in. BWG 17 is employed. Among the many unknowns which the objective functions depend upon, the tube length (L) is limited by the dimensions of the turbine exhaust area for proper system integration.[9] A tube length of 45 ft is chosen in this case. Under the above specified conditions, the optimal thermal performances of the power system derived from two different optimization schemes are presented in Table 3 and their trajectories in policy space are depicted in Fig. 4. Each of the closed contours in the figure represents a constant value of the net power per unit heat transfer area. The peak of these contours corresponds to the optimum state where the net power per unit heat transfer area is maximized; i.e., the Case 1 in Table 3. Applying the sequential optimization to the same system, the maximization of the gross power introduces an extra constraint which limits the movement of the optimization of the objective function (P_n/A) in the policy space along the dotted curve in Fig. 4.

Table 3 Thermal Performance Modeling Results

Parameter	Case 1	Case 2
Condenser tube water velocity, fps	10	10
Power, MW		
Gross	121.3	181.7
Net	75.3	100.9
Mass flow rate		
Warm seawater, 10^5 lb/s	7.58	7.58
Steam, 10^3 lb/s	3.62	8.62
Cold seawater, 10^5 lb/s	6.15	14.9
Temperature, °F		
Warm seawater inlet	80	80
Warm seawater exit	75	68
Steam	74	66.4
Condensate	54	54
Cold seawater exit	46	46
Cold seawater inlet	40	40
Condenser surface area, 10^6 ft^2	2.29	5.55
Net power cycle efficiency, %	1.85	1.11

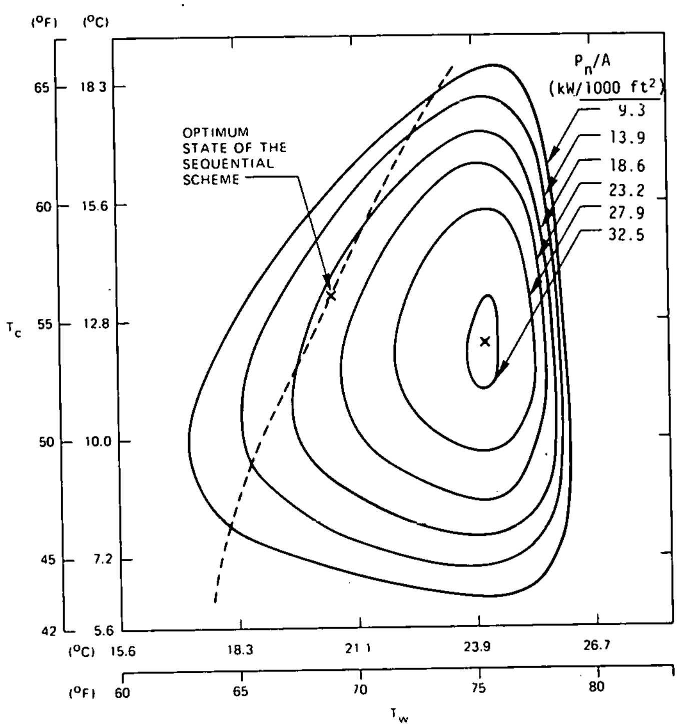


Fig. 4 Trajectories of objective functions in policy space.

The gross power reaches a maximum value at every point on the dotted curve when it is being approached both horizontally and vertically in the policy space. This dotted curve cuts through the isopleths of P_n/A, but does not pass the peak point. The optimization of P_n/A in the sequential scheme leads to a different optimum state as shown in Fig. 4, and the values of the state variables are tabulated in Case 2 of Table 3.

Rejected Heat Open-Cycle Application

The major power components of an open-cycle power system are operated at soft vacuum environment. The application of the OTEC open-cycle concept to ORGDP rejected heat sources will result in a simplified tie-in system. Any leakage in the warm water supply system will proceed from the GDP cooling water loop toward the open-cycle power system. Since the open-cycle working fluid is warm water, any water leakage will not likely cause concern of liquid cross contamination. The elimination of isolation heat exchangers in the warm water supply loop will not only reduce capital investment but also improve power plant thermal efficiency.

To investigate the possible thermal design of a rejected-heat, open-cycle power system, the same thermal performance model is applied in the ORGDP case studied by Exxon. The location of the proposed power system, the associated water supply and discharge systems, warm water flow rate, heat source and sink temperatures, and cooling options of the Exxon base line closed-cycle design are adopted. The conceptual thermal design of land-based ORGDP open-cycle power systems with both cooling options are presented in Table 4. For the same heat source conditions, the open-cycle power system will generate about the same amount of net electricity as the closed-cycle system.

Table 4 Thermal Design of Land-Based Open-Cycle Power System

Application: ORGDP Rejected Heat Utilization		
Cooling Mode	Once-Through	Wet Cooling Tower
Parameter		
Mass Flow Rate		
Warm water, lb/s	5570	5570
Warm water, gpm	40,000	40,000
Steam, lb/s	87	76
Cooling water, lb/s	8340	4960
Cooling water, gpm	59,900	36,000
Temperature, °F		
Warm water inlet	140	140
Warm water exit	124	126
Steam	121.6	123.9
Condensate	75	90
Cooling water exit	70	85
Cooling water inlet	60	70
Surface Condenser		
Tube size (BWG 20), in. OD	7/8	7/8
Tube length, ft	46.5	54.6
Surface area, ft^2	65,450	39,033
Tube water velocity, fps	6	6
Power, MW		
Gross output	6.1	3.9
Warm water loop	-0.56	-0.56
Cooling water loop	-0.94	-0.57
Cooling tower	0	-0.52
Miscellaneous	-0.22	-0.22
Net output	4.41	2.01
Efficiency, %		
Turbine/generator	81	81
New power cycle	4.7	2.4
Condenser surface area/net power, ft^2/kW	14.9	18.0

However, due to the higher heat transfer coefficient between steam and water that can be achieved in the surface condenser than that between R-113 and water, the heat transfer area required in an open-cycle system is considerably less than the corresponding closed-cycle system. Cost estimates of the major power components of the land-based open-cycle system have not been made. Since the cost estimate of an OTEC open-cycle power system is comparable to that of the closed-cycle system, and the land-based, open-cycle steam turbine operating conditions is in the range of low-pressure turbine technology, the cost estimate of the land-based, open-cycle system should be comparable to or possibly less than Exxon's estimate of the ammonia closed-cycle system without the isolation heat exchanger, which was about one-half the cost of its base-line R-113 power system design.

Conclusions

Development of OTEC power system technology, though primarily for the utilization of renewable ocean thermal resources, could conceivably be applied to other types of low-temperature heat resources. For utilizing rejected heat from large thermal power plants, applying the open-cycle steam turbine technology for additional very low-pressure turbine stages as an integral part of the thermal power plant would appear to be more cost-effective than a separate bottoming cycle system, where OTEC range cold water is economically available.

For utilizing rejected heat for power generation from federal nuclear facilities, the open cycle has an advantage since it does not need isolation heat exchangers to avoid cross contamination of GDP cooling water. A conceptual thermal analysis of such open-cycle power system is presented. An open-cycle power system utilizing 140°F rejected heat from federal nuclear facilities may not be economically competitive, presently, if the system cost is one-half of the Exxon base-line design with wet-cooling tower option. Nevertheless, it does look promising if rejected resource is made available at 150°F. Land-based, open-cycle power systems utilizing rejected heat, though they may be operating at a higher thermodynamic state than OTEC systems, may serve as intermediate steps in the technology extrapolation toward a "low-temperature" OTEC technology.

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WASTE HEAT FROM OTEC CONDENSER WATER TO MELT ICEBERGS FOR IRRIGATION WATER

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Abstract

Towing icebergs to arid regions to provide inexpensive irrigation water would help cultivate much of the world's 120,000,000 hectares of potentially arable desert, alleviating future food problems. However, it may be more costly to melt the iceberg once it reaches its destination than to tow it from Antarctica. Using warmed condenser water from OTEC power plants to melt icebergs and then recirculating some of the cold melt water to the OTEC condensers would greatly enhance iceberg melting rates and, because the melt water would be colder than deep ocean water, increase OTEC efficiency. The cold water pipe of conventional OTEC installations could be eliminated. Several possible OTEC-Iceberg couples are described, employing either open or closed cycle OTEC systems. An open cycle system using direct-contact condensation of water vapor with cold, fresh iceberg melt water is preferred, for such a system could produce up to 30% more fresh water than is contained in the iceberg. Ice melting rates and water and power production rates are calculated for suitable OTEC-Iceberg couples. Prospective global areas for such systems are pointed out.

Iceberg towing has been proposed as a way of providing cheap water for arid regions, but the problems of melting the iceberg and delivering the water once the iceberg has reached its destination are formidable. The melting operation requires up to 2000 times the energy (as heat) required to tow the iceberg to its destination.[5] The economics of the melting operation will probably be the most significant factor in the determination of the final cost of iceberg melt water to the user.

Utilization of a portion of the enormous quantity of waste heat contained in warmed condenser water from an OTEC facility could help solve the iceberg melting problem. Since OTEC operates on small temperature differences (15-25°C), it has a Rankine Cycle efficiency of less than 10%, and actual efficiencies are expected to run between 2% and 3%.[6] In spite of low efficiencies of OTEC, the potenital has been estimated to be as great as 40 billion MW(e), providing many sources of waste heat for iceberg melting.

If the huge amounts of waste heat from the low-efficiency OTEC power plant could be used to melt icebergs and the iceberg melt water used as a heat sink for the OTEC plant, several advantages besides inexpensive iceberg melting could accrue. Floating OTEC plants could be located closer to shore and in regions where access to sufficiently cold water would not otherwise be available or would only be available with a very long expensive cold water pipe. Using iceberg water (at close to 0°C) as a heat sink instead of cold sea water at 5-10°C could increase the temperature difference in the working cycle and thus increase the plant efficiency. Alternately, employing the same temperature differences as in a conventional OTEC plant, the range of OTEC installations could be extended to regions of the earth where surface water temperatures would not normally be sufficiently high to provide temperature differences sufficient for economic OTEC operations.

The purpose of this paper is to briefly review the iceberg towing and OTEC proposals and at least partially examine the potential and feasibility of the Iceberg-OTEC Couple to meet the need for irrigation water and the consequent need for power and fertilizer.

Introduction

The most productive agricultural regions tend to be arid regions that obtain their agricultural water from humid regions. They are rich in solar energy, and their soils are rich in nutrients. Increased irrigation is necessary to significantly extend crop acreage and to make possible the full scale use of technology in farming, all needed to greatly expand world production of crops for food, fiber, and energy. It has been estimated that once water demand exceeds 20% of the annual surface flow in any given area, water supply becomes the absolute limiting factor for economic development.[1] It has also been estimated that by the year 2000, the world population will require about 25% of the annual runoff; the world is in desperate need of supplementary sources of fresh water.

Currently, less than 3% of the world's annual river water runoff is used to irrigate about 160 million hectares, about 12% of currently cultivated land but only 5% of the 3.2 billion hectares considered to be potentially arable.[2] The use of river water for irrigation is limited by its uneven distribution. South America, with less than 15% of the world's land, provides a third of the runoff, while Southwest Asia, North Africa, Mexico, Southwest United States, temperate South America, and Australia, with 25% of the world's land, account for less than 5% of the runoff.[2] Transporting runoff water great distances to agricultural areas (interbasin transfer) may cost as much as \$85/k$\cdot$m^3,[3] a prohibitive price compared to estimates of \$10-25/k$\cdot$m^3 for the highest economic price for irrigation water.[4]

Icebergs

History of the iceberg proposal and extent of the resource

Annual world iceberg yield is estimated at 1200 km^3 of fresh water,* although the total iceberg accumulation is about six times this quantity, corre- iceberg is about six times this quantity, corresponding to an average iceberg life of about six years.[8] The annual Antartic iceberg yield, if it

* This compares with an estimated total annual world river water runoff of 38.8 x 10^3 km^3.[1]

were delivered and used for agricultural water, would provide sufficient water to irrigate about $9 \cdot 10^7$ hectares, about 40% of the area presently irrigated.[9] Exploitation of just 10% of the annual yield would have an economic impact of over $10 billion annually.[3]

The first serious proposal for iceberg transport and utilization was written in 1954 but not published; the idea was later published in popular form.[10,11] Subsequently, researchers with the U. S. Army Cold Regions Research and Engineering Laboratory found the idea to be "...highly attractive for selected locations in the Southern Hemisphere." It was calculated that iceberg melt water could be delivered and distributed to Australia or Chile for less than $8/k·m³ (1973) with only 15 to 25% of this cost actually applied to towing.[12,13] It has also been estimated that an iceberg train could deliver water to California for around $24/k·m³ or less.[9] This is within estimates of economic limits on the costs of irrigation water. In 1969, it was estimated that it was unlikely that irrigation water could attain a value greater than $24.3/k·m³, for large scale irrigation.[3] On the other hand, transferring runoff water large distances to arid areas (interbasin transfer) could cost as much as $100/k·m³.[14]

During 1973 and 1974, numerous papers and reports dealt with various aspects of the iceberg towing proposal.* [3,8,12,13,15-18] A French consulting firm has proposed towing a small iceberg (7.5-13.5×10^7 m³ of ice) 12,000 Km to Jeddah, Saudi Arabia.[5] They proposed that the sides and bottom of the iceberg be insulated with a huge "blanket" to prevent convective heat transfer, and that the top of the iceberg be protected with a lake of meltwater, where 80% of the sun's heat would be utilized in evaporation, greatly reducing the rate of melting of the underlying surface. The "blanket" surrounding the iceberg would not have to be watertight to be effective, for the purpose of such a protective covering would only be to limit convective heat transfer by providing a stagnant layer of water next to the iceberg as the iceberg moved through the water. The "blanket" could just be overlapping layers of plastic. Their calculated economics for towing icebergs are shown in Table 1.

* All serious iceberg towing proposals advocate towing of Antarctic icebergs only. Arctic icebergs are usually small, irregular in shape, and exhibit a distressing tendency to turn over and break up. Antarctic icebergs are usually tabular in form. That is, they have flat tops, having broken off from large ice shelves which extend many miles over the ocean, and their surfaces are usually quite large compared to their depths. Thus, they are quite stable in comparison to Arctic icebergs. If some care is taken to select Antarctic icebergs which have large length and width compared to depth, there should be little danger of the iceberg turning over.

The melting problem

Since even the desert sun of Saudi Arabia would only melt ice to a depth of 10 m/yr, inexpensive alternative methods must be found to melt a berg in a reasonable time. A number of methods have been proposed, although none has been studied in detail.

Ablative heat transfer through direct heat exchange with ambient sea water. Examples include pumping chopped ice and melt through pipes immersed in warm sea water currents or dumping slices into shallow membrane barges with large bottom areas and using wave and tidal action to stir up greater convective heat transfer.[3]

Indirect heat transfer with air. Includes spraying-water from a pool on top of the iceberg to increase its temperature and melt ice.[5]

Low grade waste heat from costal electric power plants. Includes pulverizing ice and pumping the slurry to the plant, or running warm waste water to the iceberg.[3]

Using the heat sink capacity of an iceberg to cool a warm fluid whose heat would otherwise be wasted, using the heat to melt ice, might be the cheapest method for melting an iceberg. Condenser water from a shore-based conventional fossil fuel or nuclear power plant could be used, but conven-

Table 1 Estimates of Iceberg Delivery Costs**

	Iceberg Size		
	Large 10^9 m³	Medium 5×10^8 m³	Experimental 0.75×10^8
Daily water production	2.2×10^6 m³/day	1.2×10^6 m³/day	0.2×10^6 m³/day
Investment for:			
Iceberg water	$ 657.5 M*	$ 405.5 M	$ 88.4 M
Desalination plants	1,800 M	1,064 M	200 M
Cost to deliver water (as ice) (cents/m³ of water)			
To Aden	6	7.8	31.2
[Cost of slicing]	14.2	15.4	21.5
To Jeddah	20.2	23.2	52.7
Slicing cost as % of total	70.3%	66.4%	40.84%

* M = million

** Data from CICERO[5] (Centre d'Informatique Commerciàle é Economique é Recherche Operationelle) tional power plants are so much more efficient than OTEC power plants that the quantity of waste heat available per unit of power produced would be much less that with an OTEC facility. A huge power plant complex of 10,000 MW(e) capacity would be required to melt 1.23 x 10^6 $k \cdot m^3$ (10^6 acre feet) of water-equivalent ice in one year.[3] As an added cost, the warm condenser water to melt the iceberg would have to be pumped from shore to the iceberg.

OTEC

Most recent interest in OTEC has been in the "closed" cycle, where the working fluid (ammonia or chlorinated and/or fluorinated hydrocarbons) is heated through heat exchange with warm surface sea water. After expansion through a turbine, the vapor is condensed in a heat exchanger with cold sea water. The heat exchangers are patterned after the compact plate type developed primarily for aeronautical requirements in the late 1940's.

Today, closed cycle OTEC has advanced beyond purely theoretical studies, and a pilot plant of 10 MW(e) capacity should be operating by 1980, with commercial plants deployed in the mid 1980's.[19,20]

Although the closed cycle OTEC system is suitable for use in a system to melt icebergs, the open cycle OTEC system, with its working fluid being water vapor generated directly from warm sea water, should be particularly favorable for the OTEC-Iceberg couple proposed here. The water vapor working fluid of the open cycle can be condensed and collected as fresh water to supplement the fresh water produced from melting the iceberg.

Design problems associated with the open cycle approach to OTEC have caused it to be neglected in comparison with the attention and financial support given to closed cycle OTEC. Open cycle is basically simpler than closed cycle, although the steam turbine and steam passages become very large by conventional standards, and a soft vacuum of 15-23 mm of mercury must be maintained. A 100 MW(e) open cycle plant could require a 78 ft. diameter steam turbine. However, fans on cooling towers have these dimensions, and recent innovations would reduce the turbine size required.[21]

Since 1951, the Sea Water Conversion Laboratory of the University of California has been investigating open cycle systems for power and water production from low energy sources such as sun warmed waters and industrial waste heat. In 1955, they built and operated a dual-purpose power/water plant using a metal surface condenser and capable of producing 450 Kg/hr of steam from sea water when operating on a 3°C temperature drop during flash evaporation.[22]

With the open cycle it is possible to use the more efficient direct contact heat transfer (e.g. barometric condensers) which eliminates the use of expensive heat transfer surfaces and their inherent surface fouling problems. If the condensing fluid was cold sea water, such systems would not produce fresh water, but direct contact heat transfer would reduce capital costs about 15%.[23] If the condensing fluid was cold fresh water, as is used in the vapor reheat system of multi stage flash evaporation,[24] fresh water could be produced as condensate.

In 1975, an analysis of a 100 MW(e) (Net) open cycle system was carried out by Hydronautics Incorporated of Maryland.[25] The system which was analyzed used water turbines on the evaporation and condensation side of the vapor turbine to minimize pumping losses. Falling film flash evaporation was suggested and a falling film for the condenser side to minimize spray carry-over and air build up. Hydronautics concluded that the open cycle solar sea power plant was both technically and economically feasible and that the cold water pipe, not the large low-pressure vapor turbines and vapor passages, would be the major cost item. The final estimate for capital costs was \$450/kw.

The low Rankine cycle efficiency of OTEC would provide large quantities of waste heat per unit of power produced; it would require a much smaller OTEC power plant to melt the same amount of ice as a more efficient fossil fuel or nuclear power plant. In fact, a 283 MW(e) OTEC plant should be capable of melting the same amount of ice as a 10,000 MW(e) fossil fuel or nuclear plant.

OTEC-Iceberg Couple

In the proposed system, fresh cold iceberg melt water is pumped to the OTEC plant for use as condenser water replacing cold water drawn from ocean depths. Warmed condenser water from a closed cycle plant or condenser water plus condensate from an open cycle plant (at a temperature of about 6°C) is pumped back over the iceberg to melt more ice and be cooled again. Figures 1 and 2 show how this

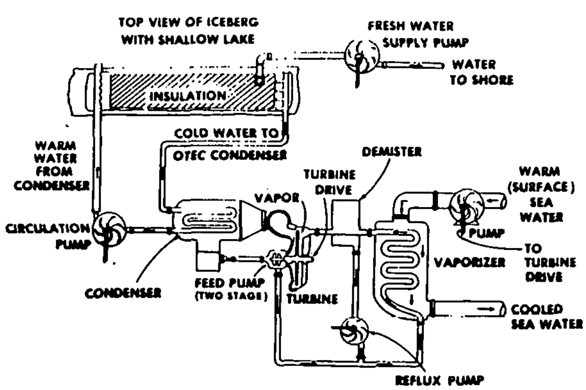

Fig. 1 Closed cycle OTEC-Iceberg Couple using flowing lake.

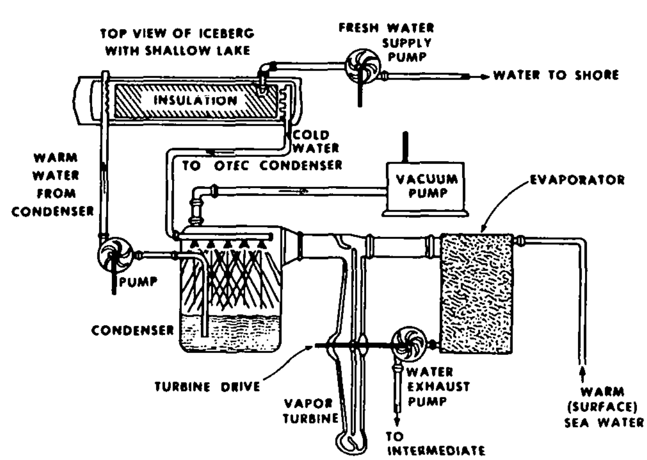

Fig. 2 Open cycle OTEC-Iceberg Couple using flowing lake.

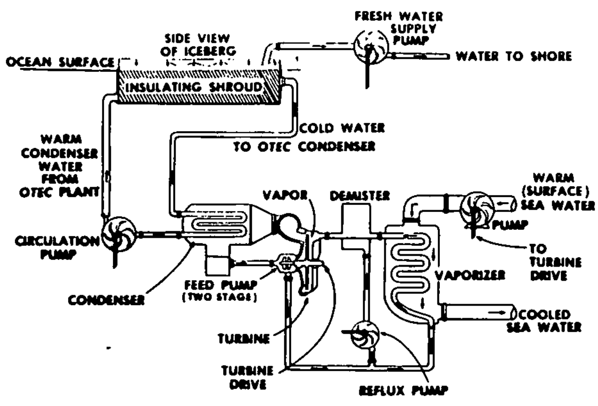

Fig. 3 OTEC-Iceberg Couple using insulating shroud.

might be accomplished using a "lake" on top of the berg with closed and open cycle systems, respectively. At the far end of the berg, part of the water is pumped back to the OTEC condensers, and part is sent to shore as irrigation or urban fresh water. This scheme would eliminate the huge, costly, and troublesome cold water pipe.

An alternative proposal is to recycle the condenser water under the quilted shroud proposed to protect the iceberg from convective heat transfer during transport (Fig. 3). For convenience, only a closed cycle OTEC system is shown in this alternative, but an open cycle system could be used equally as well. The condenser water is introduced at a point under the iceberg. The warm water flows along the underside of the iceberg, melting ice and being cooled, and the cooled water is pumped out from the other side of the berg for recirculation and as consumable fresh water. For this particular application, a watertight shroud would be preferable to a simpler protective shroud suitable for insulation during towing.

The former plan has a possible advantage in that the water is primarily melted from the top of the iceberg, which at the beginning might be 30-50 meters above the ocean surface. The potential energy available to help pump water to shore is theoretically equivalent to one or two percent of the total power available from the OTEC installation.

Since the cold water taken from a melting iceberg is colder than water taken from ocean depths, efficient use of OTEC installations might be expanded to include more temperate ocean areas with cooler surface water temperatures than tropical areas presently being considered. Figure 4 shows the world's arid regions with arrows indicating some of the potential sites for OTEC-Iceberg Couples based on the need for water plus annual minimum surface water temperatures and required water depth to float an iceberg close to shore.

An iceberg containing 10^9 m^3 of ice is considered economically feasible to tow. In a practical process such an iceberg would be melted within one to two years. Can such conditions be

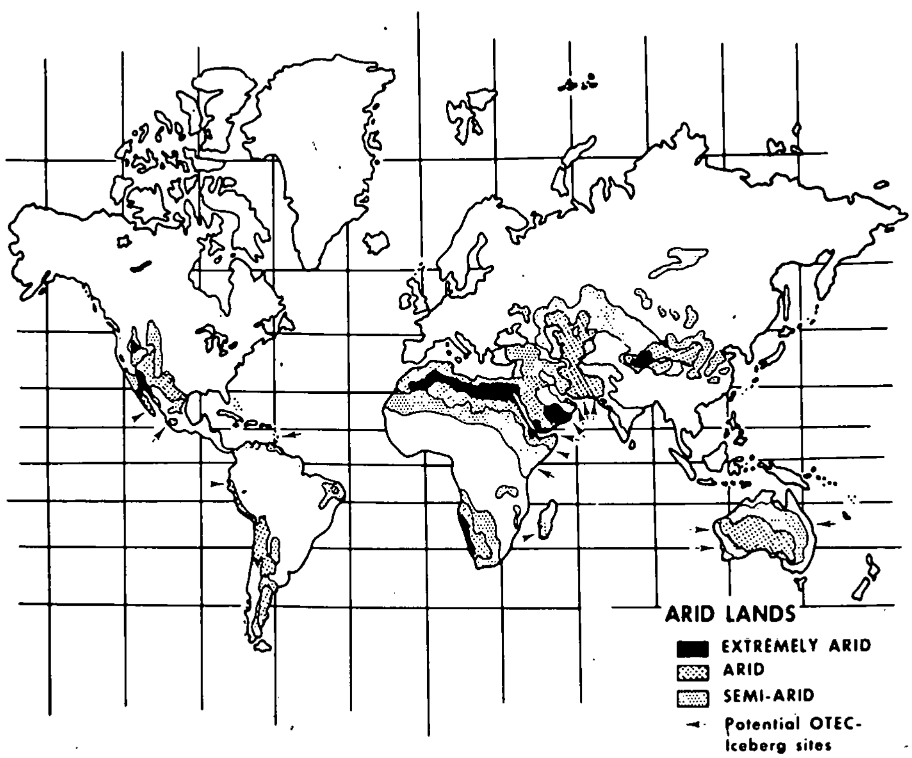

Fig. 4 World's Arid Lands with Potential OTEC-Iceberg Sites.*

*Made from a map of arid lands found in G. F. White (ed), The Future of Arid Lands, Pub. No. 43, AAAS, Washington, D.C., 1956 and World Atlas of Sea Surface Temperatures, Hydrographic Office, U.S. Navy, second edition, 1944, plus miscellaneous navigational charts. Arrows (potential sites) are located near arid regions where water temperature does not fall below 19 C and water depth is at least 250 meters within 10 miles of shore.

met, even theoretically, by flowing OTEC condenser water over the top of the iceberg to melt ice?

Assume the following:

- Iceberg dimensions: 1 km wide x 5 km long x 200 m deep.
- Condenser water to OTEC installation: @1000 m^3/sec (velocity = 1 m/sec)
- Outlet temperature of condenser water to iceberg (t_1): 6°C
- Depth of "lake" at water takeoff point: 1 meter
- No external input of heat to the iceberg.

Turbulent flow greatly enhances heat transfer rates. For flow of water over a flat plane, turbulent flow begins at Re = 2.7×10^6, and flow becomes completely turbulent above Re = 3.7×10^6.[26] Define Re as $LV\rho/\mu$, where L = length of plane over which fluid flows, V = velocity, ρ = density, μ = viscosity. For the above conditions, flow becomes completely turbulent at 5.6 meters from the water inlet, and the laminar region can be ignored.

The head (F) needed to produce a velocity of 1 m/sec at the end of the iceberg can be found from the Fanning equation:[27]

$$F = 4fLV^2/2gD \tag{1}$$

where f is the Fanning friction factor; D = 4S/P, or four times the hydraulic radius; S = cross sectional area of fluid flow = 10^7 cm^2; and P = wetted perimeter = 10^5 cm. From the above values of S and P, D = 400 cm. The corresponding Reynolds number is 2.5×10^6, and f = 0.0025.[27] From equation 1, F = 0.634 m. Thus, a depth of lake at the input of 1.63 meters would be sufficient to produce a velocity of 1 m/sec at a depth of 1 meter at the end of the iceberg, 5000 meters away.

The heat transfer factor (j) is defined[28] as:

$$j = (h/c_pV\rho)\,(c_p\mu/k)^{2/3} = S_t Pr^{2/3} \tag{2}$$

where S_t = Stanton number, Pr = Prandtl number, k = thermal conductivity, c_p = specific heat, h = heat transfer coefficient.

Since, $$S_t = (S)\,(t_1 - t_2)/\Delta tm\,(A) \tag{3}$$

$$j = (S)\,(t_1 - t_2)\,(Pr^{2/3})/\Delta tm\,(A) \tag{4}$$

where t_2 = temperature of water leaving system; Δtm = mean temperature difference across the film; and A = surface area over which fluid flows. For water at about 5°C, Pr = 11.0, and $Pr^{2/3}$ = 4.95. For conditions at the chilled water end of the iceberg, Re = 3.29×10^9 and j = 4.50×10^{-4}.[28] For $\Delta tm = (t_1 - t_w)/2 + (t_2 - t_w)/2$, where t_w = temperature of the wetted wall (ice at 0°C), t_2 = 3.83°C. Thus, the temperature of the water to be pumped back to the OTEC condensers is 3.83°C, and the total heat flow (Q) from the condenser water to the iceberg is 9.08×10^9 J/s. The heat of fusion of ice is 30.1×10^7 J/m^3, so the melting rate is 30.1 m^3 of ice/sec, and for a total iceberg volume of 10^9 m^3, the time required to melt the iceberg is estimated to be 1.05 years. The daily production of water from iceberg melting for agricultural or other use would be 2,210,000 m^3 (1792 acre-ft).

For the assumed rate of flow of recirculated condenser water, the quantity of heat which is removed from condensing working fluid in the OTEC installation is the same as the heat available to melt ice, 9.08×10^9 J/s, assuming no change in the temperature of the water from iceberg to condenser or from condenser back to the iceberg. The total energy available is then 9080 Mw. For an OTEC efficiency of about 2%, the actual power output obtainable from an OTEC plant, using the cooling water rates and temperatures given in this paper is 182 MW(e). Although the efficiency of the iceberg-OTEC couple will be somewhat higher than for a typical OTEC plant because of lower rejection temperatures for the couple, the very conservative figure of 2% will be used to prevent possibilities of exxageration.

Based on design analyses from the Sea Water Conversion Laboratory of the University of California,[23] an open-cycle OTEC plant producing 10 MW(e) net power could produce 36,000 m^3/day of condensate. A modular 182 MW(e) open cycle power plant with the same condensate to power ratio of the 10 MW(e) plant would augment the daily 2,210,000 m^3 of fresh melt water with 655,200 m^3 of condensate giving a total of 2,865,200 m^3 (2320 ac ft) of fresh water daily or about 30% more fresh water than the closed cycle plant.

Actual heat input from solar radiation and surrounding warm ocean water would melt the iceberg faster than calculated above and complicate the situation. However, based on the above estimates, the annual iceberg yield of 1.2×10^3 Km^3 of water represents a potential power yield of 2.5×10^{12} kw-hr per year. A closed cycle 182 MW(e) OTEC-Iceberg Couple would annually supply \$20 million in fresh water, at \$25/k·m^3, and \$32 million in power at \$0.02/Kw-hr. The open cycle 182 MW(e) couple would supply \$26 million in water and \$32 million in power annually. The potential for the simultaneous production of inexpensive fertilizer ammonia[29] which could be in the water should make further study of such a scheme for the agricultural development of arid coastal regions even more compelling.

Although much more study is required in the area of environmental effects, it should be pointed out that the total mass of the sea ice that forms every year is about ten times the mass of icebergs. The moderating influence of sea ice on the climate (together with that of the continental icecap) dominates over that of icebergs so that little climatic effect would be expected even with the removal of total annual iceberg yield.

The local environmental effects at the OTEC-Iceberg melting site would require detailed investigation for each specific proposed site, but the modification of the marine environment should be no worse than could be expected from a small change in latitude. Local fog statistics might be altered somewhat.[3] A reduction of the ocean surface temperature in vicinity of the iceberg would actually bring about a greater absorption of solar energy, since the rate of reradiation of energy is a function of temperature to the fourth power.

Conclusion

The engineering difficulties of pumping 1000 m^3/sec of water onto one end of an iceberg, forming a 5×10^6 m^3 lake on the iceberg, and pumping the water off the other end while the berg is melting are formidable. It is possible that other water-ice contact schemes for the iceberg-OTEC Couple would be superior, but the basic concept may have virtues that outshine other melting proposals for many site locations. Some of the benefits seen are:

A. Elimination of the long vertical cold water pipe of the conventional OTEC plant.

B. The attainment of comparable OTEC efficiencies from cooler surface waters.

C. The extension of the range of potential locations of OTEC plants because of reasons given in A and B.

D. Open cycle production of fresh condenser water without solid wall condensers.

E. The reduction of the cost of water delivered, since the melting operation now becomes an energy source rather than a energy sink.

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DISCUSSION

R. Lyon, Oak Ridge: Your concept is intriguing because it points up what many people have already realized, that OTEC not only uses the warm water on the surface, but what may be more important, it "mines" cold water from a depth of 3000 ft or so as the heat sink. I made a little calculation some time ago that indicated that the availability of warm, equatorial water, with respect to the cold water in an OTEC plant, is of the order 980 ft-lb/lb of cold water. But a pound of ice has an availability of about 11000 or 12000 ft-lb/lb, so you could lift that water a couple of miles, and I have often wondered whether anybody had looked at the possibility of using that energy for moving an iceberg as well as generating power at the iceberg's end-use site.

J. Randall: There have been a lot of schemes developed for moving the iceberg, of which the most common one is to pull it with a big tugboat. There have also been schemes for using the temperature difference between surface water as the heat source and cold air as the heat sink at a site such as Alaska.

R. Lyon: Yes, I know that people have investigated various concepts, but your concept does emphasize that OTEC is really mining the cold water, and the cold water pipe (CWP) is what we will spend a lot of the money on. If you saw the French movie just shown by P. Marchand at this conference, you know that a lot of work is required to put the CWP on boats and go through many operations to deploy it where the cold water is. Your concept avoids the need for a long CWP, and I can imagine that you could even use an OTEC plant, which would be more like a ship or tug without the long CWP, to move the iceberg once it gets into equatorial waters.

G. Wachtell, Franklin Research Institute: I noticed that you use 2 to 3% as the OTEC thermal efficiency. Won't the efficiency be much higher with such a low rejection temperature?

J. Randall: It probably would be somewhat higher; the degree of improvement would depend on what the reference OTEC plant's cold water temperature was. Apparently the cold water temperature may vary from 4 to 8°C for conventional OTEC plants depending on the OTEC plant site and pipe length.

[Editor's note: The effect could be very significant. The net power output varies with the available ocean ΔT to a power near 2.5. For a site normally having a 22°C ΔT, a 4°C increase to 26°C would increase the plant efficiency and output by a factor of $(26/22)^{2.5} = 1.52$, or 52%. A typical overall plant thermal efficiency of 2.4% would be increased to 3.7%.]

EXAMINATION OF A GRAVITY-OPPOSED HEAT PIPE FOR OTEC APPLICATION*

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Abstract

Replacement of the cold-water pipe (CWP) of a closed-cycle ocean thermal energy conversion (OTEC) plant by a "gravity opposed heat pipe" (GOHP) is considered. The GOHP condenser is placed at what would have been the CWP inlet depth. The power cycle rejects heat to the GOHP evaporator. The GOHP working fluid (ammonia) vapor from the evaporator flows down to the same depth to entrain the condensate and lift it through a pipe to a separator. The liquid from the separator is returned to the evaporator, and the vapor flows down a third pipe to the deep condenser. In an alternative configuration, three possible locations for the power turbine are considered. It is concluded that an OTEC plant using the GOHP concept is not as good as one using the usual CWP. However, the concept may still be of interest for other applications in shallower water.

Introduction

Ocean Thermal Energy Conversion (OTEC) plants use the temperature difference between warm surface water and cold water pumped from 2000-4000 ft depth to drive a heat engine. For a typical OTEC closed-cycle plant with a 3000-ft-long cold water pipe (CWP), the power required to pump the cold water to the surface (overcoming CWP wall friction and the density change head) is equal to about 4% of the gross power or 5% of the net power produced.† The "gravity-opposed heat pipe" (GOHP) concept considered here was intended to eliminate that cold-water lifting loss by replacing the CWP with two working-fluid (ammonia) vapor downflow pipes and a 50% quality mist upflow pipe as sketched in Fig. 1. The reviews and appended Discussion of the earlier form of this paper as it was presented at the 6th OTEC Conference[2] have indicated that this approach would be inferior to the CWP in performance and cost. This amended paper is presented for the record with the thought that some variation of the concept may merit consideration for other systems involving smaller water depths.

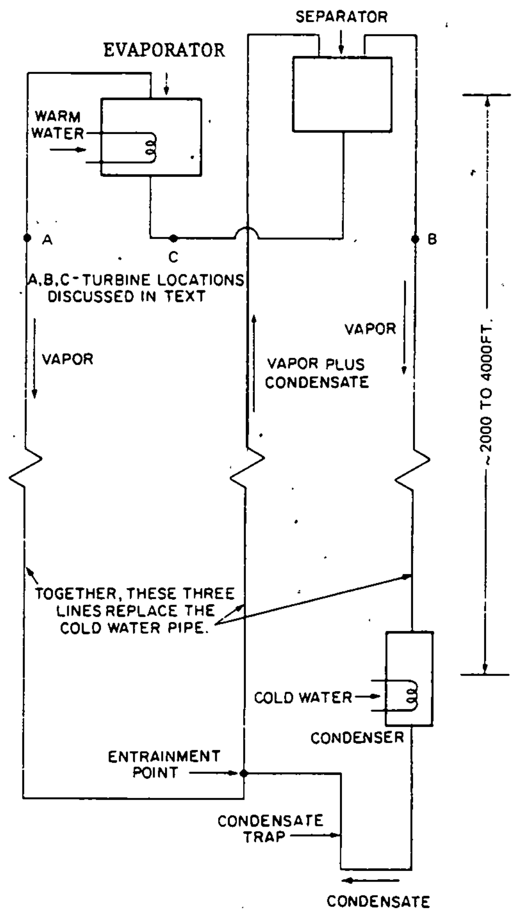

Fig. 1 Schematic diagram of the concept.

In the GOHP concept (Fig. 1) ammonia vapor from the evaporator flows down to entrain the liquid from the condenser and lift it to the separator, from which the liquid is returned by gravity (or a pump placed at point C) to the evaporator. The vapor from the separator flows down to the deep-water condenser and then through a condensate trap to the entrainment point.

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*The idea reported here is an outgrowth of work supported by the Systems Development Branch of Conservation and Solar Applications, Department of Energy, concerned with self pumping schemes for solar collectors.

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†Judged by the difference between cold- and warm-water pumping power in, e.g., Refs. 1a and 1b.

Table 1

Properties of Saturated Ammonia

Property	Condenser, $50^{o}F$ Liquid	Condenser, $50^{o}F$ Vapor	Evaporator, $70^{o}F$ Liquid	Evaporator, $70^{o}F$ Vapor
P, psia	89.2		128.8	
$h_{\ell v}$, Btu/lb	527.3		508.6	
$\mu \times 10^7$, $lb\text{-}sec/ft^2$	0.1417	2.23	0.1724	2.31
ρ, lb/ft^3	39.0	0.304	38.0	0.433
$\sigma \times 10^5$, lb/ft	161	--	145	--

The GOHP may be used in either of two ways:

1. Heat rejected by the condenser of a conventional open or closed OTEC power cycle located near the ocean surface may be transported to the deep-water condenser, where it is rejected to the cold ocean water. The "warm water" in Fig. 1 then represents the heat rejected from the power cycle. This way of using the GOHP is the basis for the name, "gravity opposed heat pipe." Because there is a temperature difference of about $10^{o}F$ between the cold water and the ammonia in the deep-water condenser, there is a loss in power cycle efficiency that is not compensated by the reduction in cold water pumping power. This method of using the GOHP is therefore inferior to the use of the usual CWP.

2. The GOHP can be used as the power cycle itself by placing a liquid turbine at C or a vapor turbine at A or B in Fig. 1. As for a conventional OTEC plant with warm and cold seawater temperatures of $T_{ww} = 80^{o}F$ and $T_{cw} = 40^{o}F$, the evaporator will operate at $70^{o}F$ and the condenser at $50^{o}F$.

Nomenclature

f = friction factor

g = gravitational acceleration, 32.2 ft/sec^2

H = turbine head, ft of ammonia liquid or vapor

H_C = reversible Carnot cycle head = $h_{\ell v}\, J\, \eta_C$ = 28,800 ft

$h_{\ell v}$ = latent heat of vaporization of ammonia, 509 Btu/lb at $70^{o}F$

J = mechanical equivalent of heat, 778 ft-lb/Btu

L/D = length/diameter ratio

P = ammonia saturation pressure, psia

V = vapor velocity, ft/sec

T_{cw}, T_{ww} = absolute temperatures of cold and warm water, assumed to be $500^{o}R$ ($40^{o}F$) and $540^{o}R$ ($80^{o}F$), respectively

η_C = ideal Carnot cycle efficiency, $(T_{ww} - T_{cw})/T_{ww}$ = 0.0741

η_{HX} = fraction of $(T_{ww} - T_{cw})$ remaining after accounting for driving ΔT's for heat exchange in the evaporator and condenser; $\eta_{HX} \simeq 0.5$

η_R = Rankine cycle efficiency = $\eta_{HX}\eta_{TG}\eta_{misc}\eta_C$; $\eta_R \simeq 0.4\ \eta_C$ for conventional OTEC plant

η_{misc} = factor to take into account the ammonia pump work and various piping losses within the closed cycle: $\eta_{misc} \simeq 0.95$

η_{net} = net thermal efficiency = $\eta_R\ \eta_{wp} \simeq 0.024$ for conventional OTEC plant

η_{TG} = efficiency of turbine-generator $\simeq 0.84$

η_{wp} = fraction of gross power remaining after accounting for cold- and warm-water pumping; for conventional OTEC plant $\eta_{wp} \simeq$ 0.8

μ = viscosity $lb\text{-}sec/ft^2$

ρ = density, lb/ft^3

σ = ammonia surface tension, lb/ft

Subscripts

ℓ, m, v = liquid, mist flow, and vapor

Lifting of Condensate

Before addressing effects of turbine location at point A, B or C, let us consider the requirements for lifting the condensate from the entrainment point in Fig. 1 to the separator. All of the condensate must be lifted; otherwise, a progressive net loss of liquid from the evaporator would eventually cause it to run dry. The flow regime of interest here is two-phase mist-annular flow, in which some liquid is carried upward as mist droplets by the vapor stream and some liquid forms a film on the pipe wall. To be sure that all liquid is eventually carried up, two requirements have to be met:

1. All liquid carried in the vapor stream must be in the form of droplets having a terminal velocity of falling, relative to the vapor, that is less than the vapor velocity V.

2. All liquid present in the liquid film on the pipe wall must either (a) move upward as a result of shear stress at the interface between liquid film and vapor, or (b) enter the vapor stream.

With small volumetric liquid loading and $\rho_\ell >> \rho_v$, the criterion for lifting the mist droplets is (Ref. 3, Eq. 12.40, page 385):

$$V > 0.2\ (\rho_v/g)^{-\frac{1}{2}}\ (\sigma\ \rho_\ell)^{\frac{1}{4}} \qquad (1)$$

For this vapor velocity, drops that are large enough to fall with a velocity higher than V relative to the vapor are unstable and break into smaller droplets as a result of shear forces. Consequently, no drops can long remain with a net downward velocity. For ammonia at 50°F (Table 1), V must exceed 1.03 ft/sec. Thus, a rather low velocity is sufficient to carry up the mist droplets, meeting the first requirement.

An approach to meeting the second requirement is to have V high enough to ensure complete entrainment of the film, leading to homogeneous mist flow rather than mist-annular flow. For low liquid volumetric loadings, substantial entrainment occurs when (Ref. 3, Eq. 12.43, page 390 and Fig. 12.10, page 393):

$$V\mu_v \, (\rho_v/\rho_\ell)^{1/2}/\sigma \simeq 0.001 \tag{2}$$

For ammonia at 50°F this leads to V = 82 ft/sec. To estimate the friction factor f, suitable average properties for a homogeneous mist flow may be assumed (Ref. 4). This leads to $f \simeq 0.008$ for Reynolds numbers of the order 4×10^7. In a pipe of length L and diameter D, the resulting friction pressure drop is

$$\Delta p = f(L/D)\, \rho_m \, V^2/2g = 0.508 \; L/D \quad \text{(psf)} \tag{3}$$

where the mixture density is $\rho_m = 2\rho_v = 0.608$ lb/ft^3 at 50°F.

For a 40 MW$_e$ plant, assuming an efficiency of 2.4%, the mass flow rate of ammonia is 3107 lb/sec. At 50°F with V = 82 ft/sec, the pipe diameter is 12.6 ft. The pressure drop, with L = 3000 ft, is 121 psf or 0.84 psi in accordance with Eq. (3).

The pressure drop due to the acceleration of condensate as it joins the vapor stream at the entrainment point is the increase in momentum flow per unit cross section. Since the average density of the dispersed condensate after entrainment is equal to the vapor density, ρ_v, the pressure drop due to entrainment is $\rho_v V^2/g$ = 63.5 psf or 0.44 psi.

The vapor down-flow ducts may have cross-sectional areas similar to that of the mist lift pipe. If these vapor ducts are annuli, concentric with the central pipe that carries the mist, sized so that the sum of their frictional pressure drops equals that of the central mist lift pipe, then, neglecting wall thickness, the inner annulus has an outer meter of 19.0 ft and the outer annulus has an outer diameter of 25.5 ft. Of course, the OD of the outer pipe must be much more than this, in order to support the severe compressive load imposed by the sea water at 3000 ft depth. The OD will be comparable to that of a cold water pipe for a 40-MW$_e$ OTEC plant, and the duct system will cost more than the CWP it would replace.

Irreversible effects include pipe friction, momentum exchange due to entrainment, and slip flow in the mist lift pipe. Apart from such effects, flow in the ducts is reversible (assuming local thermodynamic equilibrium between vapor and droplets in the mist lift pipe). Consequently, in the absence of a turbine at A, B, or C, the temperatures in the evaporator and condenser must be equal, if irreversibilities may be neglected. Since the vapor is saturated in the evaporator and condenser, the pressures must also be equal. It follows that to first order, the gravity effects (pressure variations due to elevation differences) cancel out. This may be seen in Fig. 1 if we neglect variations in ρ_v due to the pressure variations resulting from elevation differences. Starting at the evaporator, the pressure is increased by $\rho_v L$ at the entrainment point, decreased by $2\,\rho_v L$ in going from the entrainment point to the separator, and increased by $\rho_v L$ in going from the separator to the condenser.

The effect of slip may be estimated by noting that the droplet velocity in the mist lift pipe is 82 - 1.03 ft/sec. This raises the mean liquid density from $\rho_\ell = \rho_v$ (no slip) to $\rho_\ell = 1.013\,\rho_v$. Therefore, ρ_m is increased by $0.013\,\rho_v$, causing an increase in the pressure drop in the mist lift pipe of 0.01 psi. The sum of the frictional, momentum exchange, and slip flow irreversible pressure drops is 2.1 psi. There are certainly other losses in addition to these. For example, there is a pressure drop in the separator, and there is an irreversible exchange of heat between the vapor and the liquid at the entrainment point. It is probably quite optimistic to estimate that the total pressure difference due to all the irreversible effects is as little as 2.5 psi.

Comparison to Conventional OTEC System

As noted in the Introduction, the power requirement for lifting the cold water through a conventional CWP is approximately 4% of the gross power output. The pressure drop through the turbine corresponding to evaporation at 70°F and condensation at 50°F is approximately 39.6 psi (based on the corresponding saturation pressures). Thus, the 2.5 psi loss optimistically estimated above for the GOHP system represents a reduction in gross power by 6%, so that there is a small net penalty in performance in using the GOHP.

To complete the picture, let us consider the differences due to ammonia power turbine location at point A, B, or C in Fig. 1. If the turbine is placed at A, the condenser and separator operate at 50°F, the evaporator operates at 70°F and the turbine at 70°F inlet and 50°F outlet temperatures. Thus the vapor in both down-flow pipes is at 50°F, and all of the calculations in the preceding section apply. The liquid can be made to flow from the separator to the evaporator by elevating it to 146 ft above the evaporator to provide the 39.6 psi pressure drop that is to be taken through the turbine, plus additional smaller elevation increments to provide for the pressure drop through the evaporator and the connecting plumbing. The net power output will be further reduced because of the additional height of the vapor lift pipe, but this is a tradeoff with the ammonia pump in the conventional OTEC system (which also could be used here at C instead of elevating the separator).

If the turbine is located at B, everything will be similar to the preceding case except that the vapor in the downcomer from the evaporator will be at 70°F and 129 psia at the top of the left vapor duct, and the pressure will increase by $\rho_v L$ = 0.433(3000) = 1299 psf = 9.0 psi. With a 0.4 psi friction loss, the net pressure rise is 8.6 psi to the bottom of the duct. The condenser will be elevated 178 ft above the entrainment point to deliver liquid to it at this same pressure. Thus, liquid at 50°F will be entrained by vapor at 70°F. Thus, with the turbine at B, the irreversible exchange of heat is especially severe.

If the power turbine is placed at point C in Fig. 1, it is a liquid turbine. Its head H is the difference in elevation between the liquid levels in the separator and evaporator (neglecting the small piping losses). If the reversible cycle work in ft-lbs/lb is H_C, corresponding to the Carnot cycle efficiency η_C, then

$$H = H_C \times \eta_R/\eta_C \simeq 11300 \text{ ft} \qquad (4)$$

which is not feasible.

Concluding Remarks

These calculations indicate that use of the GOHP concept confers no advantage over the conventional OTEC plant with a CWP. This is true whether the GOHP is used simply as a heat pipe or as the power cycle. Its cost will also be greater due to the cost of the three pipes that would replace the CWP and due to the increased costs of building and maintaining the condenser and the cold water pump which would be located at approximately 3000-ft depth. Perhaps the GOHP concept will warrant consideration for other kinds of undersea processing plants located at shallower depths.

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DISCUSSION

D. Aronson, Consultant: The author is to be commended for offering still another alternative to the many already proposed for utilizing the temperature differences between warm surface ocean waters and the deep waters. One of the problems in the development of a practical heat-power system is that of designing a cost-effective means of rejecting heat to the cold water located about 0.9 km below the surface. Most proposals have been based on pumping the water from the depths to the surface or close to it. The difficulty is associated with the great length of the pipe and its extremely large diameter. The large diameter is called for to provide sufficient flow area for the flow of water. The amount of heat to be rejected is enormous, yet the allowable temperature change of the water which serves as heat sink is in the order of 1.5° to 3°C (3° to 6°F).

The basic idea of the author's concept is to avoid the need for elevating this tremendous amount of water. Instead, the condensible vapor, probably ammonia, would be sent down to the ocean depth, there to be condensed into liquid, which would then be pumped to the surface. The volumes involved are a small fraction of those for the case of cooling water being pumped to the surface.

There are at least four major problems associated with this system. They may have solutions and certainly deserve consideration. The problems are:

1. The design of a surface condenser of high reliability to operate with the cooling water at a pressure about 1200 psi higher than that of the condensing vapor.

2. The design of water circulating pumps to operate at the depth of 0.9 km.

3. The design of a pumping system for returning condensate to the boilers located close to the surface.

4. The construction of leak-tight piping for the vapors going down to the condenser and the liquid boiler feed being pumped back up.

There is an interesting phenomenon which is not a problem but should be recognized as a change in cycle characterisitcs. The turbine exhaust will be at a considerably lower pressure than with systems having condensation take place close to the turbine. The high column of vapor will compress adiabatically, so that the pressure at the condenser will be appreciably greater than at the surface. The extra work obtained from the turbine, then goes to pumping the condensate from the lower depth to the surface.

The problems may all have suitable solutions. The circulating pumps could be driven by submersible electric motors or by hydraulic motors with a suitable fluid pumped at the surface power plant. A similar pump or set of pumps could be used for pumping the condensate from the condenser up to the boiler close to the surface.

Instead of a pump of conventional design, the author proposes a novel scheme which he entitles a "gravity opposed heat pipe." The actual proposal does not match the title. The system described is actually a vapor lift pump. Such pumps are often advantageous when the complexity of a mechanical pump is not justified. In general, the work involved is small, so that despite the low efficiency of the vapor lift, it offers a practical solution.

In the past, when suitable mechanical pumps were not available for pumping liquids from considerable depths, the vapor lift pump was sometimes used despite the low efficiency. Special conditions may also favor vapor lift, as for example, the Frasch process which melts sulfur in situ, then forces the molten material up as a multi-phase, multicomponent fluid.

Such special considerations are not applicable to the present case where efficiency is a crucial matter, and the pump head is in the order of 0.9 km. The use of a vapor lift pump would result in the entire heat-power cycle failing to produce net power. Further, it should be noted that the nature of the system defeats the reason for using it. The large volume flow of the pumping gas going down, and then returning as a two-phase stream, would require a major increase in the size of the vertical fluid handling column(s) as compared to a conventional OTEC cold water pipe.

Overall, if other condensate pumping means are considered, then the reviewer feels that consideration should be given to the alternative of condensing the vapor at the ocean depth. Study may show that it is not justified, but the exercise would, at least, resolve the question.

The "gravity opposed heat pipe" may have other applications, but the discussor is convinced that it is not justified as a replacement for a mechanical pump in the application suggested.

G. P. Wachtell: Mr. Aronson's discussion is greatly appreciated. The author agrees with all of Mr. Aronson's comments except his expectation that the inefficiency of a vapor lift pump would result in the entire heat power cycle failing to produce net power. That expectation arises because of the large pumping head, on the order of 0.9 km. It should be remembered, however, that the fluid being lifted has a much lower density than liquid ammonia, since half its mass is vapor.

Mr. Aronson questions the use of the title, "Gravity Opposed Heat Pipe" because the system actually described is a vapor lift pump. The title, however, refers to its function, not to its mechanism. Indeed, its mechanism incorporates a vapor lift pump. But its function is the same as that generally served by heat pipes; namely, to transfer heat from one end to the other in a way that involves no mechanical moving parts and is entirely passive as far as the user is concerned.

It has to be admitted that the gravity opposed heat pipe requires a total cross section area sufficient for three vapor streams, two of them flowing down and one carrying the condensate up, and that only about 70% this total cross section area is needed if the condensate is returned by means of a mechanical pump, as suggested by Mr. Aronson. It should be noted that the pump has to deliver a head equal to L, about 3000 ft. Possibly more than one pump would have to be used in series.

Mr. Aronson suggests various arrangements for supplying power for the circulating pumps needed for the submerged ammonia condenser and for returning condensate to the boiler. In addition, one may consider using a vapor turbine located near the condenser. This turbine would be sized to supply practically all of the parasitic power associated with the gravity opposed heat pipe (or pumped condensate alternative) system, so that the turbine near the ocean surface produces essentially only useful power.

Replacing the OTEC cold water pipe is evidently an inappropriate application for the gravity opposed heat pipe, although at first glance it appeared to be an ideal application. There may be other, more appropriate applications to ocean engineering, however, for a passive device that transports heat downward through a distance of many feet; for example, in systems stationed on the sea floor at a much shallower depth of up to a few hundred feet.

W. Shippen and G. L. Dugger, JHU/APL: The objective of this proposed arrangement is to save the pumping power required to raise the cold water up to the water surface, which for existing OTEC power system designs represents about 5% of the net power or 4% of the gross power. To accomplish this, the cold water pipe is replaced by two down-flow vapor pipes and one return flow pipe of the same 3000-ft length. For 10-MW_e power modules the vapor pipes are about 6 ft in diameter to keep the friction loss down, whereas the cold-water pipes being designed for 40-MW_e pilot plants are 30 ft in diameter. The additional length of the heat pipe tubes would increase the vapor pipe diameter required to at least 7 ft for 10 MW_e or 14 ft for 40 MW_e giving a total circumference of $\Pi \times 14 \times 3 = 132$ ft compared to $\Pi \times 30 = 94$ ft for the cold water pipe. Moreover the walls would be thicker because of the large collapse or buckling load imposed by the large hydrostatic external pressure compared to the internal pressure of ammonia vapor. Thus, there would be a predictably greater cost for the vapor tubes vs the cold water pipe.

From the power production standpoint, the vapor lift pumping would require a high vapor velocity at the "entrainment point" in order to break up the liquid which would generate a high back pressure in line A. If the turbine is at point A, this back pressure plus the friction loss would reduce the available turbine pressure ratio. The loss in turbine power output would exceed the 4% saved by removing the CWP. There are also the obvious disadvantages (increased capital cost and greatly increased costs of maintenance, cleaning, or repair) or placing the condenser and cold water pump at maximum depth.

It also appeared from the preprint that the author had neglected the driving ΔT's required between seawater and working fluid in the heat exchangers; i.e., if a 40^{o}F ocean ΔT is available, 20^{o}F or more of this ΔT will be used for the heat exchangers and only 20^{o}F or less will be left for the OTEC power turbine. [This point has been recognized in the present amended paper.]

G. P. Wachtell: The comments by Messrs. Shippen and Dugger are certainly appreciated. The present amended text reflects many of the points made by them, but I would like to reply to the point regarding entrainment losses. The velocity needed to entrain the condensate and break up the liquid is only that required for fluidization, 1.03 ft/sec, given by Eq. (1). Thus, the 82 ft/sec velocity that results in entraining any liquid film on the pipe wall is more than adequate to break up the liquid entering at the entrainment point.

THERMOELECTRIC OTEC

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Abstract

A novel thermoelectric OTEC concept is proposed and compared with the ammonia closed-cycle designs. The thermoelectric OTEC is a much simpler system which uses no working fluid and therefore requires no pressure vessel, working fluid pumps nor turbogenerator. These components are replaced by power modules which are simply compact heat exchangers integrated with thermoelectric generators. The thermoelectric OTEC offers several potential advantages including: simpler and more easily mass-produced components; higher reliability system performance through the use of a high level of redundancy and long-lived, solid-state thermoelectric generators; greater safety for crew and environment by elimination of the pressurized working fluid; and the possibility of achieving comparable system costs.

These comparisons are discussed and plans for future work are presented in the paper.

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Introduction

Over the past few decades, the art and science of thermoelectric energy conversion has gradually evolved to a high level of performance. Thermoelectric generation has become the preferred method for producing electric power where reliability and maintenance-free operation are essential. Radioisotope heated thermoelectric generators power orbiting satellites, remote radio transmitters, space probes (Pioneers 10 and 11 and Viking Mars Landers) deep sea sonar sounding bouys, etc. Fossil fueled thermoelectric generators provide reliable cathodic corrosion protection for remote pipelines, bridges, etc.; and large scale industrial thermoelectric cooling is becoming a commercial reality in France.[1]

The objective of an extensive thermoelectrics R&D program during the 1960's was to develop materials and systems which were competitive with the steam turbogenerator. Although this goal was never met and thermoelectric research suffered a decline, it is important to recognize that thermoelectrics were judged by a comparison with steam turbines in the temperature range where steam is most efficient and economical. For much lower temperature heat sources steam turbines are much

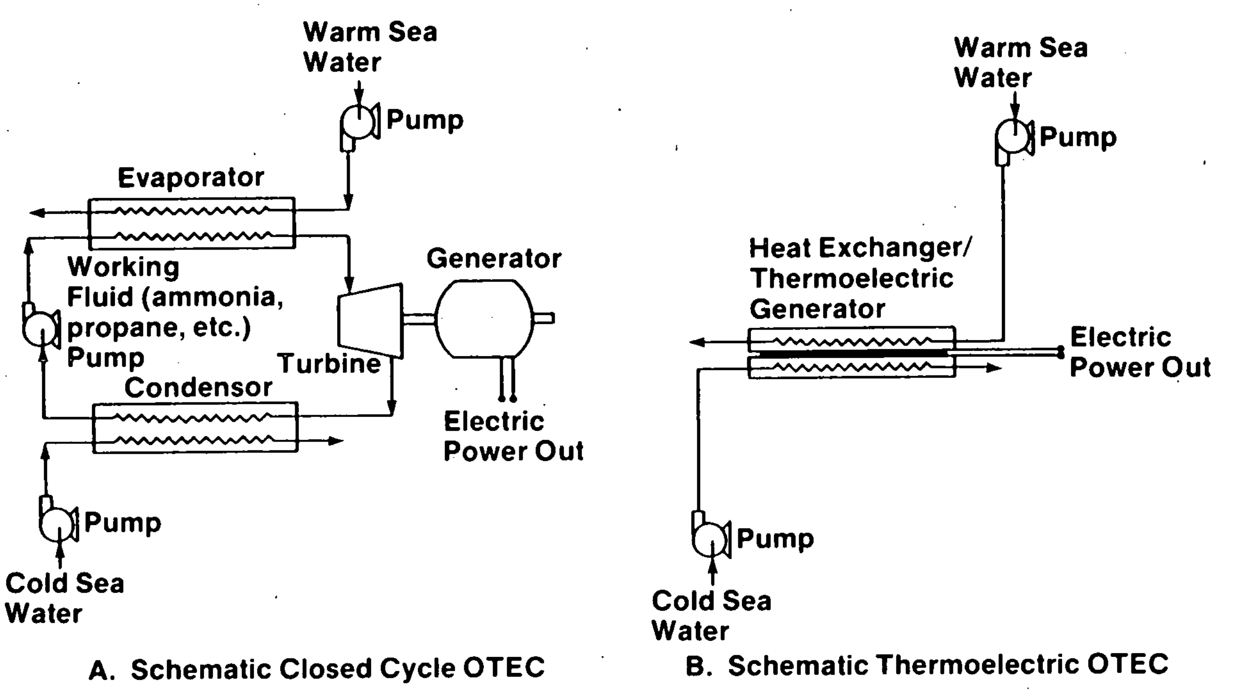

Fig. 1 Comparison of system schematic designs for closed cycle OTEC and thermoelectric OTEC.

less efficient and uneconomical; and mass produced thermoelectric generators can compete very favorably.

The two major reservations about thermoelectric devices during the 1960's were degradation of the materials and bonds under high temperature thermal cycling and electrical contact resistance. These problems do not exist at low temperatures such as are encountered in the thermoelectric OTEC. Low temperature thermoelectric devices have exhibited 35 year mean-time-between-failures.[2]

New materials now being developed by the thermoelectrics industry exhibit energy conversion efficiencies far superior to presently available thermoelectric materials. These new materials may represent a major improvement in thermoelectric technology and may offer the possibility for numerous new applications. The thermoelectric OTEC is one such new application that has been proposed by the authors and which is now undergoing preliminary evaluation at SERI[1].

Discussion

Basic Principles

The thermoelectric OTEC is a simple system which uses a thin layer solid-state generator rather than a working fluid. Consequently, no evaporator, condensor, working fluid pump, pressure vessel or turbogenerator are required (Fig. 1). A compact heat exchanger is used to transmit heat through the thermoelectric generator.

Thermoelectric generation makes use of a bulk phenomenon--the Seebeck effect (the same phenomenon that makes a thermocouple operate). A temperature gradient across any material tends to drive charge carriers from the hot side to the cold side and produce a voltage, V, which is proportional to the temperature difference, ΔT. The proportionality constant, the Seebeck coefficient, is a characteristic of the material:

$$\alpha = \frac{V}{\Delta T} \qquad (1)$$

For most metals, α is quite small, a few microvolts per degree; but specially designed semiconductor thermoelectric alloys produce several hundred microvolts per degree Kelvin. Using such materials, thermal energy can be converted to electrical energy at about 20 percent of the maximum (Carnot) theoretical efficiency.

Efficient thermoelectric materials exhibit a high Seebeck coefficient, α, high electrical conductivity, σ, and low thermal conductivity, k. A figure of merit,

$$Z = \frac{\alpha^2 \sigma}{k}$$

is a good indicator of a thermoelectric material's performance.

The efficiency of electric power generation, η, is a fraction K of the Carnot limiting efficiency η_{Carnot} and depends on Z approximately as

$$\eta = \left[\frac{(1+ZT_{AV})^{\frac{1}{2}} - 1}{(1+ZT_{AV})^{\frac{1}{2}} + \frac{T_{cold}}{T_{hot}}} \right] \cdot \eta_{Carnot} = K(Z)\eta_{Carnot} \qquad (3)$$

where $T_{AV} = (T_{cold} + T_{hot})/2$, the average temperature of the thermoelectric generator. At a given T_{AV}, the efficiency is roughly proportional to the figure of merit, Z.

Figure 2 shows how Z varies with temperature for the most common commercial thermoelectric materials.[3] Special alloys of bismuth, antimony, tellurium, and selenium now used in the thermoelectric cooling industry have Z values as high as 3.5×10^{-3} at room temperature. New alloys of bismuth, antimony and tellurium, which are under development by the industry, have recently exhibited reproducible values of $Z = 6 \times 10^{-3} K^{-1}$. Such materials could provide a conversion efficiency as high as 30% of the Carnot limit. In a thermoelectric OTEC, the use of such an advanced material could produce a gross conversion efficiency of 2% — an efficiency of 2/3 that of the closed cycle ammonia OTEC. Other new materials such as amorphous semiconductors and organic semiconductors also have promise for achieving high thermoelectric energy conversion efficiency in the OTEC temperature range.

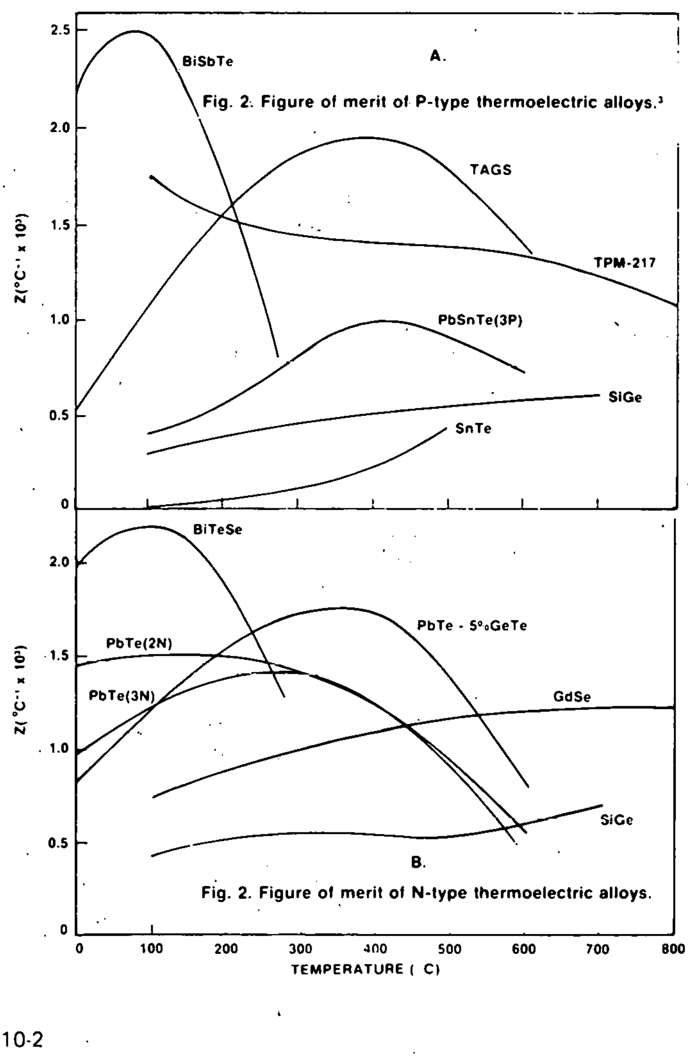

Fig. 2. Figure of merit of P-type thermoelectric alloys.[3]

Fig. 2. Figure of merit of N-type thermoelectric alloys.

Heat Exchanger and Thermoelectric Generator

The thermoelectric OTEC concept permits the use of a particularly simple design for the power module which serves both as heat exchanger and electric power generator. Figure 3 shows a preliminary design concept for such a module which would produce about 30 kWe (net) in an OTEC with $\Delta T = 20^{\circ}K$.[4] If advanced thermoelectric materials were used, the net output from the module could be increased by about 50%. A 400 MWe (net) plant would use about 20,000 such modules. The power module is basically a compact heat exchanger of parallel plate or plate and fin design with the thermoelectric generators sandwiched between the flow channels as shown. The material of choice in the heat exchanger is a copper-nickel alloy which can provide several advantages:

- o excellecnt resistance to biofouling,
- o low corrosion rate (less than $2.5x10^{-5}$ m/yr or 1MPY),[5]
- o high thermal conductivity,
- o high electrical conductivity, and
- o easy fabrication.

Unlike the closed-cycle OTEC designs, the thermoelectric OTEC is forgiving. For example, a corrosion pit which would shut down an ammonia OTEC module (because of concern for accelerated corrosion and/or scaling) would only damage a small number of thermoelectric elements, but would not diminish the power output from the affected power module perceptibly and would not require repair. One of the consequences of this forgiving design is that only minimal corrosion tolerances need be used in the heat exchanger.

The thermoelectric generator device is a novel design which makes use of thick film technology already well developed in the electronics industry. The thermoelectric material could be deposited on the copper base as a paste by lighographic techniques, fired in place to form a durable coating, a suitable pottant added to fill the interstices; and finally the cover plate could be soldered in place. All of these steps are amenable to automation.

The thermoelectric generator design is particularly simple because only a single type (that is either n-type or p-type) semiconductor need be used. Each thermoelectric generator device will then consist of many identical thermoelectric elements connected in parallel; and these devices in turn will be interconnected in series to proivde a high current, low voltage power module. Modules can be suitably insulated from each other and connected in series-parallel arrays as needed to produce the desired system current and voltage output.[4]

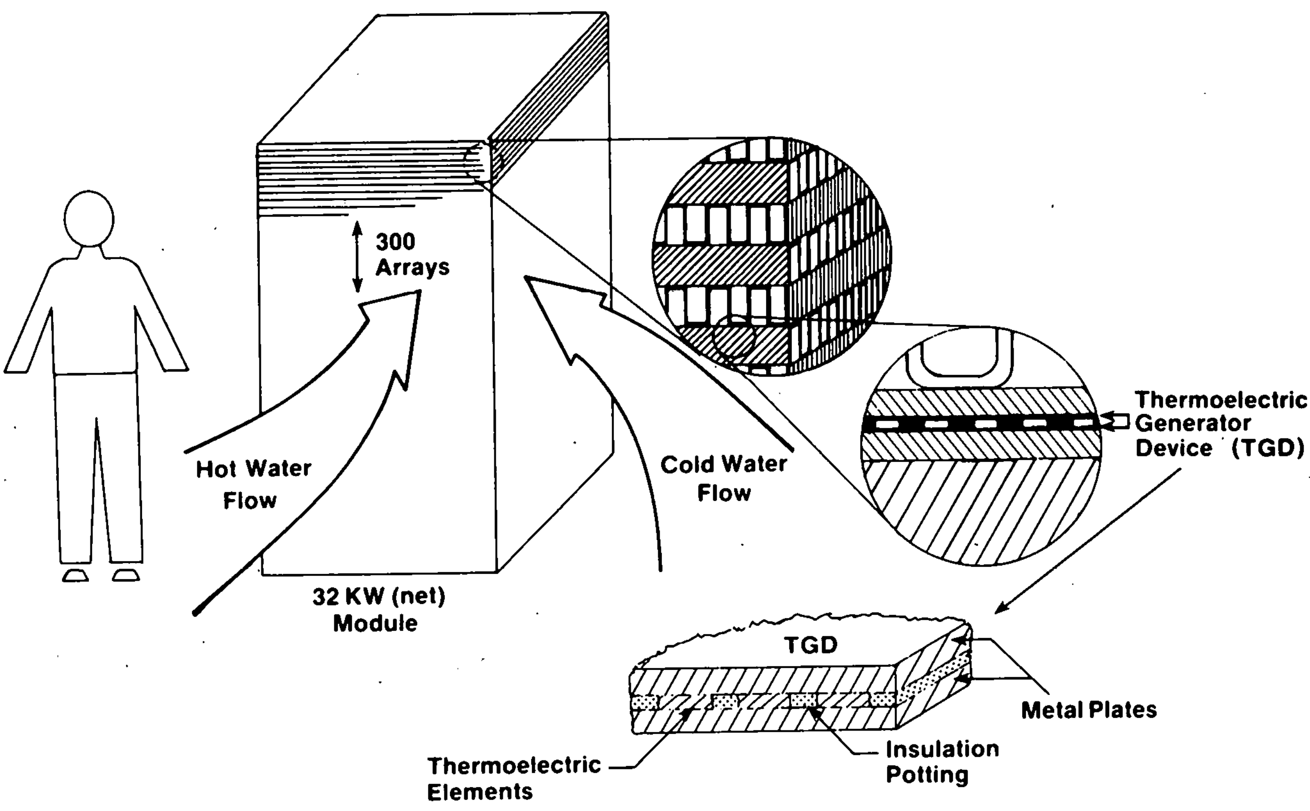

Fig. 3 A preliminary power module design is based on a parallel plate cross-flow heat exchanger in which the thermoelectric generator devices (TGD) are sandwiched between the plates. The TGD and the heat exchanger are well suited to automated mass-production.

The particularly simple design of the thermoelectric power modules suggests that they may be suitable for large scale automated mass-production. This possibility is in contrast to the tube in shell heat exchangers for closed cycle OTEC which require expensive, labor intensive assembly.

Environment and Safety

The ammonia closed-cycle OTEC poses several safety and environmental risks that would not be present with the use of a thermoelectric OTEC. Small, chronic leaks of ammonia may pose a health hazard to the operating crew as well as causing accelerated corrosion and scaling of the heat exchangers. Larger leaks, which would release large quantities of ammonia before they could be repaired would be a serious health hazard, a potential fire and explosion hazard, and have serious impacts on neighboring marine life, (particularly in combination with hypochlorite used to control biofouling).

The thermoelectric OTEC would have none of the ammonia hazards; but its use of copper alloy heat exchangers would significantly raise the copper and nickel ion concentration near the OTEC. These ions are harmful to certain mollusks. However, a study of the problem has shown that the copper and nickel ion concentrations would probably not exceed the EPA standards for marine water quality.[6]

Heat Exchanger Design

To determine the performance which could be expected from a thermoelectric OTEC system, a preliminary performance analysis was made of a simplified design. Included in the design effort was development of an expression for electrical power output per unit base area of the heat exchanger, selection of the correct thickness of thermoelectric material, calculation of parasitic losses, and minimization of costs for a given plant output power.

The heat exchanger considered has parallel flow channels of rectangular cross section (Fig. 3 & 4). Cold water and hot water are pumped through alternate flow channels with thermoelectric material sandwiched between the channels.

The power output of the thermoelectric module per unit area may be written*

$$P''_e = Kq_h\left(1 - \frac{T_\ell}{T_h}\right) \qquad (4)$$

The heat transferred from the hot fluid to the cold fluid is

$$q_h = \frac{T_{hot} - T_{cold}}{R_{te} + 2R_f} = \frac{T_h - T_\ell}{R_{te}} \qquad (5)$$

The power output may be expressed in terms of the various thermal resistances (Eq. 3) by replacing T_ℓ and T_h in Eq. 4 and rewriting it as:

$$P''_e = \frac{KR_{te}}{T_{hot} - R_f\left(\frac{T_{hot} - T_{cold}}{2R_f + R_{te}}\right)}\left(\frac{T_{hot} - T_{cold}}{2R_f + R_{te}}\right)^2 \qquad (6)$$

Since the temperature differences are small,

$$\varepsilon = 1 - \frac{T_{cold}}{T_{hot}} \ll 1 \qquad (7)$$

Equation 5 can be linearized to give

$$P''_e = \frac{K\varepsilon^2\, T_{hot}}{R_{te}(1 + 2R_f/R_{te})^2} \qquad (8)$$

The local electric power generation P''_e may be maximized by adjusting the thermal resistance of the thermoelectric to $R_{te} = 2R_f$, giving

$$P''_e = \frac{K\varepsilon^2\, T_{hot}}{8R_f} \qquad (9)$$

The power density P''_e is a function of distance along the flow direction within the passage (T_{hot} and T_{cold} vary in the flow direction) and must be integrated along the flow length to determine power output per flow channel. This calculation is similar to determining heat exchanger effectiveness as a function of NTU (number of transfer units) for a given flow configuration [7]. However, since P''_e is proportional to the square of the temperature difference, the calculation of power generation per flow passage must be carried out for each flow configuration; and the results of heat exchanger effectiveness calculations are not useful. The power generation calculation was done for a parallel flow and a counter flow arrangement, and the results show that NTU for power generation is four times that for heat transfer. Therefore the number of generation units NGU can be defined as

$$NGU = \frac{4ULW}{C_p\dot{m}} = \frac{L}{GC_pR_fb} \qquad (10)$$

In addition, for small values of NGU ($\sim$1) the flow configuration is not important in determining the power generation per flow channel and the change in fluid temperatures is small.

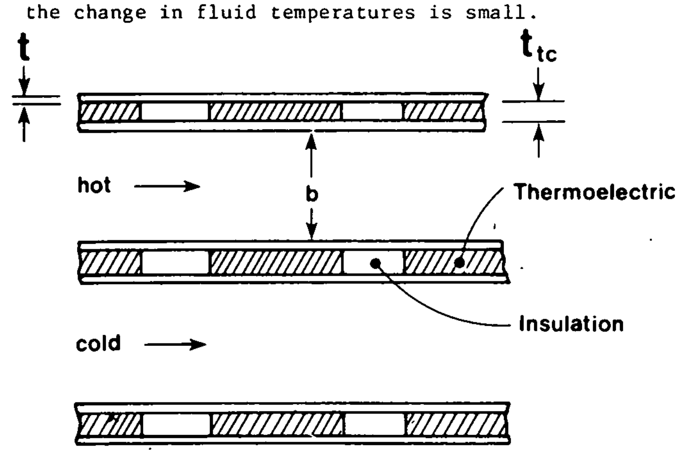

Fig. 4 Detail of heat exchanger/thermoelectric generator configuration.

*Nomenclature is defined in the Appendix.

For NGU << 1 the power generation of a flow channel is

$$P_c = P''_e\,WL. \qquad (11)$$

but as NGU increases, the power generation of the channel per unit channel length decreases as [1-exp(-2NGU)]/2NGU. In general we found that for NGU <0.7 this expression was valid for parallel flow or counterflow heat exchangers, and we expect it will be valid for most other flow arrangements.

The power generated by the channel is given by

$$P_c = \frac{0.95\ \Gamma\ K\varepsilon^2 T_{hot}\ WL}{8K_f} \qquad (12)$$

where $\Gamma = \frac{1-\exp\ (-2NGU)}{2NGU}$ (13)

The factor 0.95 accounts for an expected 5% loss due to electrical contact resistance at the heat exchanger/thermoelectric junction.

Pumping the fluid through the flow channel requires pumping power [8]:

$$P_{hx} = f_{hx}\ \frac{L}{b}\ \frac{G^3 bW}{\rho^2 g_c} + \text{minor losses}$$

$$f_{hx} = 0.079\ \ Re^{-1/4}$$

$$Re = \frac{2bG}{\mu} \qquad (14)$$

If the cold water pipe feeds N/2 heat exchanger channels, a pumping power P_p will be required:

$$P_p = f_p\ \frac{L_c}{d_c}\ \frac{(GbWN/2)^3}{2\rho^2 g_c A_p^2} + L_c\ \frac{\Delta\rho}{\rho}\ \frac{GbWN}{4} + \text{minor losses} \qquad (15)$$

The first term in Eq. 14 is the frictional loss, while the second term is the work required to pump the cold water up against the density gradient.

The cold water pipe Reynolds numbers are on the order of 10^8, and we may assume fully turbulent flow. Assuming a dimensionless roughness for a cold water pipe of 0.0003 gives $f_p = 0.015$. The density gradient term was derived by neglecting compressibility and salinity effects and by assuming a linear density profile with depth. Minor losses refer to losses in channel inlets or outlets.

Assuming the pumping power can be supplied with efficiency η_p, the net power output of a plant consisting of N channels is

$$P_o = NP_c - \frac{1}{\eta_p}\ (P_p + NP_{hx})$$

We have neglected power required to pump the warm water to the heat exchanger inlet.

Dividing Eq. 15 by 2NWL, we find the net output power per unit heat exchanger plate area:

$$k_p = 0.433\ \text{W/cm°C}$$

The optimization procedure for a net 400 MW_e yields:

$b = 1.43$ cm
$G = 70\ g/cm^2 s$
$L = 9.31$ m (NGU = 0.5)
$t_{te} = 0.66$ mm
$d_c = 41.7$ m

Required area of heat exchanger plate = $27.6 \times 10^6\ m^2$

Required flow rate (hot plus cold) = 14.8×10^6 kg/s

Cost Estimates

The economics of the thermoelectric materials presently considered for OTEC benefit from two factors. The elements involved (primarily bismuth and tellurium) are byproducts of lead, gold, and copper processing and are available in abundance.[4]

Since the mining costs and most of the processing costs are borne by the primary products, the byproduct costs are determined by the costs of capital equipment, operation, and maintenance of the byproduct separation and purification process. At present, these costs must be recovered with the sale of a relatively small volume of byproduct. As larger quantities are demanded, production equipment can be scaled up, its efficiency improved and its costs spread over a larger volume of byproduct. The result of this situation is an inverted supply-demand curve. The greater the demand, the lower the unit costs.

Information obtained from a major producer of bismuth and tellurium indicates a very substantial reduction in material costs as the demand increases.[3] The following trend is predicted for the price of Bi_2Te_3:

Demand	Unit Cost (1979 dollars per pound)
10^3	2.96
10^4	2.37
10^5	1.78
10^6	1.19

The conceptual design for a thermoelectric OTEC is not yet well enough developed to justify any attempt to make detailed cost estimates. However, it is possible to make an order of magnitude estimate of the system costs for comparison with closed cycle system estimates.

Two major subsystems probably account for most of the thermoelectric OTEC--the power modules (heat exchanger, thermoelectric generator combination) and the seawater pumps. With automated mass production, the power modules should approach about 1.35 times the costs of the materials required. Seawater pumps for OTEC have been surveyed and an algorithm developed relating cost to pumping capacity:[9, 10]

Cost per unit capacity $313\ Q^{-0.616}$ (17)

where Q is m^3/sec of seawater and costs are given in thousands of (1979) dollars per m^3/sec capacity. Utilizing these simplifying assumptions, the approximate costs are estimated as follows.

The thermoelectric generator for the 400MWe (net) plant discussed in the preceding section produces about 30 W/m^2 net power. The power module surface area required is $1.38 \times 10^7 m^2$. The masses of the required materials are estimated as the product of their density and thickness times the total heat exchanger area (4):

- flow channel plates of 90 copper-10 nickel alloy with thickness 0.508×10^{-3} m (0.02 in.) and density 8900 $Kg\text{-}m^{-3}$ (8.9 $g\text{-}cm^{-3}$): mass = $8900 \times 0.508 \times 10^{-3} \times 1.38 \times 10^7 \times 2 = 125 \times 10^6$ Kg (276×10^6 lb).
- thermoelectric device plate of copper with thickness 0.13×10^{-3} m (0.005 in.) and density 8900 $Kg\text{-}m^{-3}$ (8.9 $g\text{-}cm^{-3}$): mass = $8900 \times 0.13 \times 10^{-3} \times 1.38 \times 10^7 \times 2 = 32 \times 10^6$ Kg(71×10^6 16).
- thermoelectric material covering 0.3 of the surface area and a thickness of 0.66×10^{-3} m (.026 in.) and density 7700 $Kg\text{-}m^{-3}$ (7.7 $g\text{-}cm^{-3}$): mass = $0.66 \times 10^{-3} \times 7700 \times 0.3 \times 1.38 \times 10^7 = 21 \times 10^6$ Kg (46×10^6 lb.).

Two flow channel plates and two thermoelectric device plates are required for each unit of power module area. Materials requirements and cost estimates are summarized in Table I.

The sea water pumping capacity for a 400MWe (net) thermoelectric OTEC is $1.48 \times 10^4 m^3/s$. In the interest of increased reliability through a high level of redundancy, it may be desirable to use 74 large pumps of 200 m^3/s capacity each rather than a smaller number of even larger capacity pumps even though the costs are projected to be less if special, very large pumps are used.[10] The algorithm (Eq. 17) yields a cost estimate of $\$2.4 \times 10^6$ per pump for a total pump cost of $\$178 \times 10^6$ or \$444/kWe (net) electric power output.

It is important to note how sensitive the thermoelectric OTEC costs are to thermoelectric conversion efficiency. The power density achieveable is proportional to the 3/2 power of the conversion efficiency (Equation 17); so that if the advanced materials now under development were used in these estimates instead of current, commercial thermoelectric materials, all of the costs would be multiplied by a factor (efficiency old/efficiency new)$^{3/2}$ which equals (0.20 Carnot/0.3 Carnot)$^{3/2}$= .54. Thus, the successful development of the new, more efficient thermoelectric materials will have a strong impact on the economic viability of this concept. Similarly, the development of a more corrosion resistant copper alloy for the heat exchanger could reduce materials requirements and costs dramatically. The capital cost estimates are summarized in comparison with closed cycle estimates in Table II. The greater simiplicity and reliability of the thermoelectric OTEC should have a favorable influence on the life cycle costs--fewer replacments or major repairs of rotating machinery, less costly biofouling control measures, larger availability factor, etc. These improvements may offset initial capital cost disadvantages.

Potential Market Penetration

The preliminary nature of the cost estimates for thermoelectric OTEC do not warrant a detailed market penetration study at present. However, it is possible to estimate the relative impact of reduced OTEC capital costs on its probable market penetration.

The MITRE system for Projecting the Utilization of Renewable Resources (SPURR Model)[13] was used to compete OTEC base load power systems with various capital costs against conventional power sources and other solar options. The market was limited to the southern US; and OTEC sites were limited to the Gulf of Mexico.

TABLE I SUMMARY OF POWER MODULE MATERIALS REQUIREMENTS AND COSTS FOR A 400 MW_e (NET) THERMOELECTRIC OTEC

Component	Material Required (10^6 kg)	Material Required (10^6 lb)	Unit Cost[a] (\$/kg)	Total Cost (10^6 \$)
Heat exchanger channel plates (90 copper - 10 nickel alloy)	125	275	2.2	275
Thermoelectric device (copper plates)	32	70	1.8	57
Thermoelecrric material (alloyed Bi_2Te_3)	21	47	2.6	55
			Total materials cost =	387

Total materials cost x 1.35 = fabricated power module costs = 522

Cost per kW_e net power = \$1306

[a]Costs estimated from current (1979) commodity prices for electrolytic copper (\$0.75/lb) and nickel (\$2.20/lb) with 1.10 multiplier to cover alloy processing [12].

Figure 5 shows the resulting projections through the year 2000. Notice that even a small decrease in OTEC capital costs below the base case cost range ($1371-$2020/kW$_e$, 1979 dollars) can dramatically improve the market penetration. Such projections offer encouragement to our efforts to design a lower cost thermoelectric OTEC.

Conclusions

The concept of a thermoelectric OTEC offers many potential advantages in simplicity, reliability and safety. Its economic competitiveness appears to depend on successful development of new thermoelectric materials and power module designs. The potential for decreased OTEC costs and increased market penetration are promising.

Ongoing SERI research will include continued evaluation of newly developed thermoelectric materials and exploratory research which is designed to discover additional thermoelectrics among new classes of materials such as amorphous and organic semiconductors. Thermoelectric device design and fabrication R&D is also being pursued. Further analysis of the thermoelectric and other potential solar applications will be made as developments warrant.

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TABLE II ULTIMATE CAPITAL COST ESTIMATES FOR OTEC SYSTEMS (400 MW$_e$ PLANT)

	Ammonia Closed Cycle[a]		Thermoelectric OTEC	
Module	Aluminum HX (1979 $/kW)	Titanium HX (1979 $/kW)	Commercial Thermoelectrics (1979 $/kW)	Advanced Thermoelectrics (1979 $/kW)
Heat exchangers (and thermoelectrics)	482-844	567-989	1306	731
Demisters	8-48	8-48	NA	NA
Turbogenerators	84-135	84-135	NA	NA
Seawater pumps	103-241	115-241	444	240
Other power systems	139-235	130-235	130-235	130-235
Platform	60-362	60-362	60-362	60-362
Cold water pipe	86-96	86-96	86-96	86-96
Mooring/deployment	60-238	60-238	60-238	60-238
Electric cable	121-543	121-543	121-543	121-543
Total	1143-2743	1232-2887	2207-3224	1428-2445
Average	1943	2064	2716	1937

[a]On a module basis, these represent the highest and lowest estimates by four DOE contractors and DOE personnel as reported during February 1978. The contractors' names corresponding to the estimates made are priority information and are not presented for that reason. These estimates are for sites 3 miles to 200 miles from shore [13]. Costs have been inflated to 1979 dollars using the Chemical Engineering Plant Cost Index and ref. [11].

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Nomenclature

A - cross-sectional area of thermoelectric material

A_c - cross-sectional area of heat exchanger passage = bW

A_p - cross-sectional area of cold water pipe = $\pi d_c^2/4$

b - height of heat exchanger flow passage

C_p - fluid (water) specific heat

d_c - diameter of cold water pipe

f_p - friction factor in cold water pipe

f_{hx} - friction factor in heat exchanger flow passage

G - mass flow rate in the heat exchanger flow passage ÷ bW

g_c - acceleration of gravity

h - heat transfer coefficient at fluid/heat exchanger interface

I - electrical current in thermoelectric device

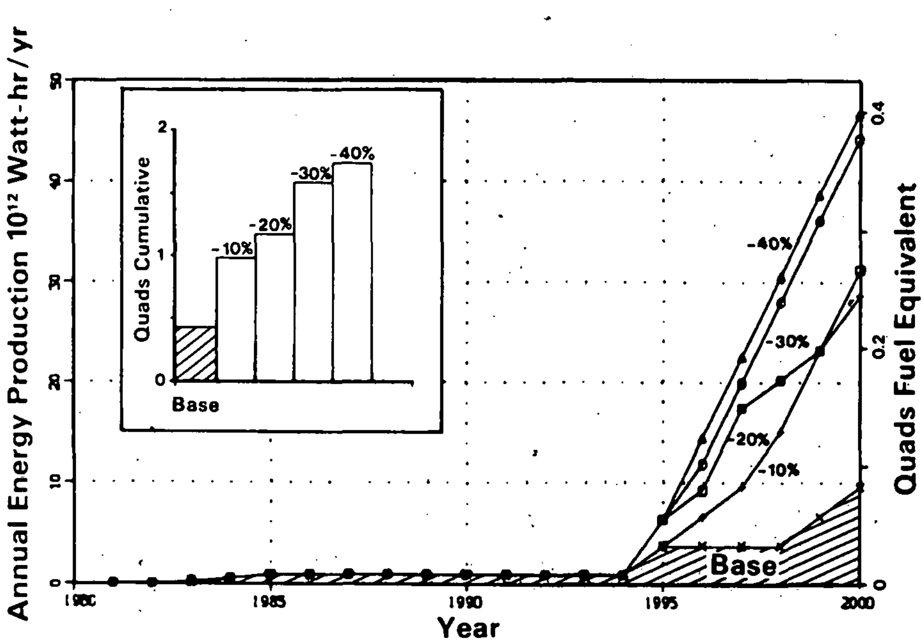

Fig. 5 The market penetration of ocean thermal electric systems based on the results of a simulation model (14).

K - efficiency of thermoelectric material as fraction of Carnot efficiency

k_{te} - thermal conductivity of thermoelectric material

k_{eff} - effective thermal conductivity of thermoelectric material

k_p - thermal conductivity of the heat exchanger plate material

ℓ - length of thermoelectric material in an elementary device

L - length of heat exchanger flow passage in flow direction

L_c - length of cold water pipe

$\dot{m}$ - mass flow rate in heat exchanger flow passage

$\dot{m}_c$ - total mass flow rate of cold water

N - total number of flow passages in the heat exchanger

NGU - number of generation units

P''_e - local electrical power generation per unit base area of heat exchanger

P_c - electrical power generated by a flow channel

P_{hx} - power required to pump fluid through a flow passage

P_p - power required to pump fluid through the cold water pipe

P_o - net electrical power produced by the system

P''_o - maximum value of P_o (wrt G) per unit heat exchanger plate area

Pr - fluid Prandtl number

q - heat flux in thermoelectric device

q_h - heat transferred per unit area from hot fluid to cold fluid

R - electrical resistance of thermoelectric device

R_f - thermal resistance from the fluid to the thermoelectric material (including plate resistance and fouling)

R_{te} - thermal resistance of the thermoelectric material including Peltier resistance

Re - Reynolds number in heat exchanger flow passage

t - thickness of heat exchanger plate

t_{te} - thickness of thermoelectric material

T_ℓ - cold junction temperature of the thermoelectric

T_h - hot junction temperature of the thermoelectric

T_{hot} - temperature of the hot fluid

T_{cold} - temperature of the cold fluid

U - thermal conductance from the hot fluid to the cold fluid

V - voltage generated by an elementary thermoelectric device

W - width of heat exchanger flow passage

α - Seebeck coefficient of the thermoelectric material

β - fraction of area covered by thermoelectric material

Γ - heat exchanger effectiveness parameter (Eq. 12)

σ - electrical conductivity of thermoelectric material

ε - Carnot efficiency

η_p - efficiency of water pumps

μ - fluid viscosity

ρ - fluid density

$\Delta\rho$ - $\rho(T_{cold}) - \rho(T_{hot})$

DISCUSSION

Question: If the efficiency is 30% or so, how does that compare with a Rankine cycle?

D. Benson: The efficiency of a thermoelectric OTEC is estimated to be 20 to 30% of the maximum Carnot efficiency, which is about 6.7% for a ΔT of 20C. Therefore, the maximum net conversion efficiency would be in the range of 1 to 2%, which is somewhat less than an ammonia Rankine cycle can achieve.

Question: The thermoelectric OTEC requires much more water than a closed-cycle ammonia design. So I am puzzled by the comparable cost estimates.

D. Benson: You are correct, the thermoelectric OTEC would require as much as two times the seawater that an ammonia closed-cycle OTEC needs. This increased pumping requirement was accounted for in the performance and cost calculations. The reason that the costs seem to balance out is that the heat exchanger and the rest of the thermoelectric OTEC system are so much simpler and are amenable to mass production. The real costs therefore, boil down to materials costs multiplied by some factor (1.35) to account for the fabrication costs. We have new thermoelectric materials under development at SERI (and under subcontract with the thermoelectric device manufacturing industry) that appear to have 50% higher conversion efficiency. If such improved materials were used, then the size of the heat exchanger, the pumps, and the volume of water that has to be pumped would be reduced by a factor of two.

R. Lyon: A lot of people have looked at thermoelectric methods of getting power. When I have done this, I have run into the limitations of parasitic heat loss due to direct heat transfer through the thermoelectric material, a heat transfer which is parallel with the Seebeck effect. Making the thermoelectric device thicker decreases the parasitic heat loss; but also increases the electrical resistance. It seems that you are limited by the Wiedermann-Franz relationship between electrical and thermal conductivities. Does this relationship apply in your design?

D. Benson: There is a heat loss through the thermoelectric material that limits the energy conversion efficiency. This loss was accounted for in the performance analysis. The Wiedermann-Franz ratio, L, expresses a relationship between temperature, T, electrical conductivity, σ, and thermal conductivity, k

$$L = \frac{k}{\sigma T} \sim \text{constant}$$

$$(\sim 2 \text{ to } 5 \times 10^{-8}\ v^2\ {}^oC^{-2} \text{ for most materials}).$$

This ratio is closely related to the thermoelectric figure of merit, Z. In fact, the thermoelectric energy conversion efficient k is roughly proportional to ZT where

$$ZT = \frac{\alpha^2 \sigma T}{k} = \frac{\alpha^2}{L}$$

The reason that the Wiedemann-Franz "law" does not inherently limit thermoelectric conversion efficiency to the low values obtained 30 years ago with metal thermopiles is that the Seebeck coefficient, α, in semiconductors can be an order of magnitude greater than in metals (e.g., 220 $\mu V/{}^oC$ for Bi_2Te_3 versus 3 $\mu V/{}^oC$ for copper).

HYBRID OTEC AIR CYCLE AVOIDS INDIRECT HEAT EXCHANGERS

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Abstract

With the objective of making OTEC even more attractive, alternatives were evaluated which would avoid the need for indirect heat exchangers. One such system was found that gives large savings in investment using air as the working fluid. The air is alternately warmed and cooled by direct contact with water to produce useful work from the expansion and contraction. The paper describes methods to decrease buoyancy and size of equipment, as by increasing the compression beyond that caused by the temperature change alone.

No water pumps are used since changes in pressure within the system produce the desired water flow. The system is especially well suited to combinations with industrial manufacturing and with aquaculture.

Based on a conceptual design the air cycle system makes electric power for 2.9¢/kWh compared to 3.9¢/kWh by conventional technology burning coal. A cost of about 3.4¢/kWh is estimated for advanced coal systems now in the development stage or for OTEC using indirect exchangers. A present worth of over $600 million is calculated for only 2000MW of capacity based on estimated savings for the air cycle design over advanced coal power plants.

Introduction

OTEC has shown attractive costs compared to conventional power generation, with little development risk and with less adverse environmental effects. However, OTEC is competing with a moving target since savings of about 20% are predicted for advanced energy systems such as fluid bed combustion, combined cycles,,etc. using coal (1). OTEC is still on the early part of the learning curve and it is timely to look for innovative improvements. A first place to look is at exchangers since they account for over half of the investment and are subject to fouling, leaks and corrosion. Some improvement has already been made by avoiding shell-and-tube designs and by using aluminum construction (2).

A lower OTEC plant investment is the key to lower power cost, since the costs are primarily related to investment. OTEC has no operating cost for fuel; therefore efficiency and parasitic losses are secondary factors except as they affect investment. Low investment implies smaller equipment size, a convenient measure of which is buoyancy or displacement.

Open cycle systems are still considered promising and avoid indirect exchangers, but the equipment is large because of the low vapor pressure of water. An alternative is direct contact of warm water with a fluid to vaporize it but no suitable fluid could be found. For example, propane is too soluble in water. Direct contact with a working fluid such as air was found to have great promise so it was explored in depth. Success depended on solving several design problems to get more useful net work output per mole of gas and to decrease equipment size, as covered in the subsequent discussions.

Basic Principle

The air cycle avoids indirect exchangers by using air as a working fluid which is alternately contacted with warm and then cold water to cause expansion and contraction. First, a simplified picture will be given to clearly illustrate the principle of operation, before looking at a specific design.

Air is confined in a cylinder as shown in Figure 1. Starting with the piston on the left, the air on the left side is cold while the air to the right of the piston is warm. Warm water is now introduced on the lefthand side to warm up the air while cold water is added on the righthand side, causing the piston to move to the right. Then, reversing the warm and cold water flows moves the piston back to the left. It will be seen that useful work can be produced from each cycle.

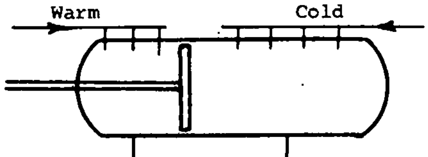

Figure 1.
Hybrid Principle

For OTEC this simple system has a volume change of only 6% so the equipment size and buoyancy are excessive. Gas volume can be decreased by raising the pressure to say 10 atmospheres, corresponding to a depth of 330 feet. Further improvement is needed and is achieved by forcing the piston to move over a greater stroke than that caused by the change in temperature. This requires that work be put in but the net result is a large increase in net work output per mole of gas and a corresponding decrease in equipment size. A flywheel can be used to store energy and return it as needed. Net work output can be taken from the flywheel or some of the high pressure air can be depressured through a turbine.

Water pumps are not needed as the changes in pressure are used to induce the required water flows. Parasitic work for pumping water is not saved and is a major parasitic loss. Thus, if the water falls through 10 feet of air there will be correspondingly less net work output.

Good contacting is needed between water and air for close temperature control and is provided by a spongelike packing that fills the cylinder. The packing is a foamed plastic that is resilient so as to accommodate the volume changes. Good temperature control is achieved with the wetted packing. The area for heat transfer is comparable to that provided with indirect exchangers; however, the plastic surface is much cheaper and a fouling film does not interfere with heat transfer from the gas to the surface. Sprays could be pictured instead of the packing for contacting using a rotating disc atomizer instead of pressure atomizing sprays, but the packing is preferred to assure adequate surface while minimizing pressure loss on the water. Buoyancy is decreased by going to higher operating pressure levels but air solubility increases at the same time and can be a serious debit above 10-20 atmospheres. There are ways to overcome the problem so that the hybrid plant can be located on the ocean floor at say 3000 feet depth and operate at an equivalent pressure.

Conceptual Design

There are many ways to apply the basic principle to OTEC designs depending on the methods selected for contacting with temperature control and the type of work storage used. Also, there is the choice of locating at the ocean surface, at mid-depth or on the bottom where the operating pressure might be 100 atmospheres. Only a few alternatives have been screened and a conceptual design has been made of one that looks attractive. It is based on a mid-depth location (300-600 ft.)

The design is shown in Figure 2 and the description will apply only to that part of the cylinder to the right of the piston since the part on the left operates in a similar manner except for timing. Operation is similar to the Figure 1 description but modified to handle the water flows with minimum complication and energy losses. Cold water flow is straightforward as it is always introduced when the cylinder pressure is at 10 atm. which is the same as ambient pressure. Warm water is needed at 20 atm. but pumping is avoided by using a warm water reservoir. The reservoir is filled at a time when the entire system is at 10 atm. pressure and the displaced air is returned to the cylinder. At the same time cold water is flowed over the packing within the cylinder to cool it, while the water discharge tank is being emptied.

The system is now ready for the compression stroke. The water valves are closed off and the piston moved to the right, raising the pressure to 20 atm. Since the warm water reservoir is full and connected to the cylinder by a balance line it is pressurized with no significant energy consumption.

At the end of the compression stroke the piston is stopped while the packing within the cylinder is warmed up. To do this, valves are opened to flow warm water from the reservoir over the packing and into the water discharge tank. Good staging is needed to minimize the water consumption. When the warm reservoir is emptied it becomes filled with air. In order to get maximum work out of this air in the subsequent expansion it is kept warm by a spongelike packing that fills the reservoir.

A flywheel is used to absorb power during the first part of the expansion from 20 atm. and return power during the last part of the expansion stroke. The stroke and the net power output are threby greatly increased. It will be seen that the flywheel could be replaced by a free piston which would accomplish the desired result.

The design basis used in making the conceptual design is summarized below:

Warm water F	80
Cold water F.	40
Air temp. expansion ^oF	75
Air temp. compression ^oF	45
Cycles per hour	500
Pressure @ 75 F. max. atm.	20
Pressure @ 75 F. min. atm.	10
Water head loss ft.	15
Sponge packing lb/cu. ft.	2

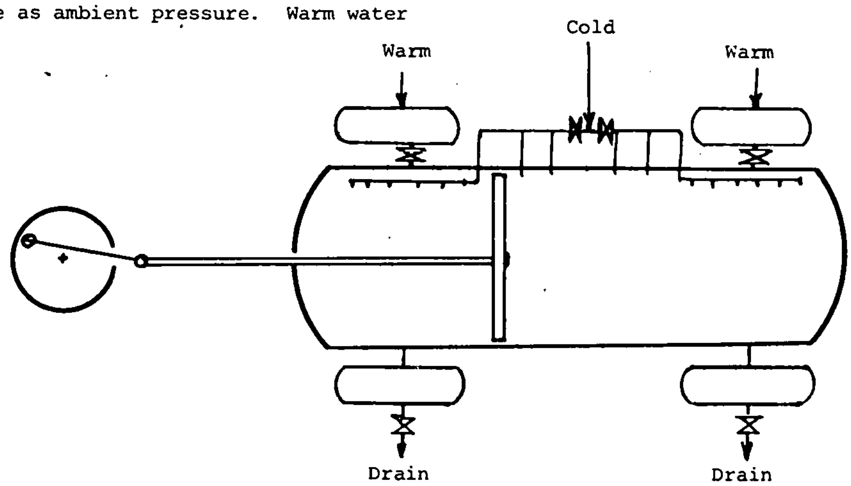

Figure 2.

Conceptual Design

In calculating water flow rate a temperature change of 5 F. was used on the work of compression or expansion and 25 F. on the sensible heat changes, plus a 20% safety factor.

Derivation of the work output is given below along with the parasitic losses:

	Btu/mol air
Expansion work	738.0
Compression work	696.6
	41.4
Losses:	
warm water	3.6
cold water	3.4
mechanical	4.8
air replacement	3.5
	15.3
Net output	26.1

Head loss of 15 feet on the water includes 7 ft. for packing height, 1.5 ft. on entrance and discharge tanks plus 1 ft. each for risers, velocity head and flowpath, leaving 2 ft. for imbalance, etc. Part or all of the power used to replace air lost due to solubility in the water may be offset by CO_2 released when the cold water is depressured.

It is of interest that a 1 MW pilot operation could use 10 tanks 7 ft. diameter by 21 ft. long with a piston stroke of 7 ft. Water flow would be about 100 cubic feet/second of cold water.

A brief comparison versus designs using indirect exchangers is shown in Table 1. The important differences are eliminating the exchangers and the large decrease in buoyancy. Plastic construction can be used wherever desired and is not limited by heat transfer. Water flow rates are lower for the air-cycle.

Table 1. Comparison of Designs.

	Lockheed	TRW	Air-Cycle
Area of exchangers, sq. ft./kW	150	79	none
Water temperature change, F	2-3	3.8-4.4	4.4
Cold water flow CFS/MW	400	131	100
Buoyancy, Tons/kW	1.9	2.1	0.5

Cost Comparisons

Economics of OTEC need to be compared to alternatives, especially coal which may have to supply our energy growth in the next 10-20 years. Fortunately a very extensive study was recently completed for the government to evaluate advanced energy systems using coal (1). This Energy Conversion Alternatives Study (ECAS) uses a carefully defined basis and ground rules to compare a conventional furnace versus combined cycles, fluid bed combustion, fuel cells, MHD and topping and bottoming cycles. It provides a sound basis for evaluations. Investments include escalation and interest during construction. The original ECAS numbers were modified somewhat in view of subsequent studies (3).

For OTEC the base case was estimated by Lockheed and TRW (4, 5) and adjusted for inflation. Table 2 compares investment and cost of electricity (C.O.E.). A conventional coal furnace with limestone scrubbing of the flue gas has an investment of $725/kW and a C.O.E. of 38.7 mills/kwh. The next column allows for improvements such as combined cycles and fluid bed combustion that are expected to save 15-20% as a result of ongoing R&D. The next column for oil-fired gas turbine with steam bottoming represents the cost of electricity today at island locations, or the cost for peaking at a mainland power plant.

The OTEC base case uses titanium tubes in standard shell and tube exchangers and is costly, but competitive today with oil-fired systems now used to supply electricity at island locations. Considerable saving is shown in the next column using aluminum instead of titanium and by avoiding the conventional shell type construction. Cost of the exchangers is cut from $10/sq. ft. to $4/sq. ft. The C.O.E. of 34.8 mills/kWh is attractive versus conventional technology but about a standoff with advanced coal systems.

The air-cycle case is shown in the last column. It is appreciably better as would be expected because of the smaller equipment and lower buoyancy. Investment is about the same as a conventional furnace but fuel cost is eliminated. There are good prospects for further savings especially by modifying the piston-flywheel arrangement and by designing for shorter cycles.

Table 2. Comparison of Costs (1979 dollars)

	Coal Furnace	Advanced Coal	Oil Fired Gas Turbine	OTEC Cases Base	OTEC Cases Improved	OTEC Cases Air-cycle
Investment, $/kW	725	600	450	1500	900	750
Cost of Electricity, mills/kWh						
Capital charges	22.9	19.0	14.2	47.4	28.5	23.7
Operating & Maint.	5.1	4.2	3.2	10.5	6.3	5.2
Fuel	10.7	9.6	48.8	-	-	-
	38.7	32.8	66.2	57.9	34.8	28.9

Further details of the economics and cost basis are given in Table 3. The 100 MW plants uses 120 FRP tanks 90 ft. long, including 30 ft. of length designed to accommodate the flywheel system. Air working fluid occupies 60 ft. of length at 15 ft. diameter and the piston stroke is 20 ft. Additional volume is provided for the electrical sub-system using a turbine/generator driven by air withdrawn at a high pressure part of the cycle and returned to tanks when they are at a low pressure point in the cycle. The number of tanks is such that steady output is achieved on the turbine/generator. Extra tank systems have been included so that several can be taken out of service for cleaning or maintenance without incurring a service factor debt. No process development contingency is included in any of the cases since the study is intended to define potential savings for an innovative system and the related incentive for R&D efforts.

It should be noted that the economics in Table 2 are based on 65% load factor as is representative of a land based public utility. However, for an OTEC industrial application the plant availability should be 90% or more which will decrease the C.O.E. by a factor of 0.7 versus the numbers in Table 2, compared to a factor of 0.8 for the conventional coal fired furnace.

Environmental

Environmental aspects deserve special mention. OTEC avoids the environmental concerns or costs related to mining, air pollution, CO_2 buildup in the atmosphere as well as the disposal of coal ash and waste from flue gas scrubbing. Available studies have shown no appreciable adverse environmental effects for OTEC.

Outlook

Savings projected for the hybrid system were used to calculate present worth. For a single 1000MW plant the saving is $20M/year over advanced coal systems giving a present worth of more than $300M. It is clear that greatly increased effort is warranted on OTEC as one of the most promising ways to use solar energy. OTEC effectively solves two difficult problems on solar power: first, the high cost of collectors, and second, the need for storage at night and on cloudy days.

Integration with industrial manufacturing is promising in that OTEC supplies power, cooling water and shipping facilities. With suitable land locations becoming more difficult to obtain, it offers an attractive alternative.

The air-cycle design shows potential savings and is especially suited for combinations with mariculture, which can be worth more than the electric power. In addition, further study of combinations using solar collectors may show that the warm ocean water source can be dispensed with so that OTEC becomes applicable in many more locations, such as California. Locating the plant on the ocean bottom would allow operating at high air pressure to increase power output when incorporating design changes of offset increased air solubility.

Table 3. Economic Basis

Inflation rate 6.5%, Interest 10%/yr.
Load factor 65%, Capital charges 18%/yr. on investment
Operating and Maintenance (O&M) 4%/yr. on investment.
Coal cost $1.10/M BTU and oil $5/M BTU HHV.
1979 dollars with escalation and interest during construction.

Plant Investment Details

Subsystem	$/kw Installed	Remarks
Tanks	93	1 in. thick FRP at $3/lb.
Flywheel	75	$15/kW energy input
Piston	47	Estimate
Valves	27	Incl. 440 lid type water valves
Packing	39	Foam plastic 2 lb/CF at $1/lb.
CW pipe	90	3000 ft. at $3000/ft. installed
Ballast	32	Concrete at $50/ton
Turbine-Generator	60	Literature x 1.5 installed
Inst. & Controls	10	Representative plant
Electrical	30	Comparable power plant
Shops & Quarters	54	10% of basic displacement & equipment
Mooring	30	Representative literature
Other	33	5% of preceding
Contingency	130	20% of subsystem costs
	750	

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DISCUSSION

G. Wachtell, Franklin Institute: You mentioned oxygen as a means of transporting energy for manufacturing. Could you elaborate?

C. Jahnig: An oxygen plant requires air at roughly a 100 lb/in^2. OTEC can supply air at this pressure, which is passed through a drier and then through an expansion turbine, followed by air fractionation to separate oxygen and nitrogen. This way bypasses electric power generation, whereas the usual oxygen plant has an electric drive on the air compressor. OTEC can use ocean energy to compress the air.

G. Wachtell: Would that comment apply to any OTEC plant?

C. Jahnig: Yes and no. The usual OTEC plant generates shaft power that can drive an air compressor instead of making electric power. However, the air-cycle system can be used to replace an air compressor. By having several systems in series, air can be compressed to 100 psig without having gone through an air compressor or shaft power.

Question: How large would the equipment be?

C. Jahnig: A 10-MW plant would require 10 vessels, 7 ft in diameter by 21 ft long. The displacement of these cylinders would be smaller than that of the ammonia heat exchangers in a conventional OTEC plant. And if the construction materials are fiberglass reinforced plastic and concrete, hopefully it can be cheaper. The equipment volume is less, and it should cost less per cubic foot than metal systems.

R. Lyon, Oak Ridge: Maybe I missed the principle of this, but it sort of takes me back to the Newcommen engine days, where Watt says you don't have to condense in the cylinder, you can condense outside, and it leads me to think that perhaps what you have is a form of Brayton cycle. Is that correct?

C. Jahnig: The simplistic operation is just isothermal expansion and cooling. Efficiency is equal to a Brayton cycle but the operation is more like a Carnot cycle. The simplistic operation requires equipment which is 10 times too big, but by forcing the piston to move over a greater distance than the temperature would cause it to; this is what allows reducing the equipment size to make a practical system.

D. Aaronson: I can see the system moving one stroke, I can't see how net work is produced. You have a momentary expansion and then, so far as I can see, you are stuck there.

C. Jahnig: I think I can answer that. Start with the piston stationary with warm air on one side and cold air on the other and in equilibrium. Now reverse the temperatures, and where it was warm before make it cold, and where it was cold before make it warm. Periodically reversing the warm and cold water flows is going to force the piston to move back and forth in cycles, and in doing so it will produce work. Another way of looking at it is to compare the isothermal work of compression versus expansion. The expansion work is RT ln compression ratio. The compression work is the same thing, RT ln compression ratio, so the net work is R times the ΔT times the compression ratio, and you can calculate your net work from that. We cannot increase the ΔT to increase the net work, but we can increase the compression ratio. That is the trick here. You increase the compression ratio, take the same ΔT, and you get 5 or 10 times as much work per pound of gas.

G. Wachtell: The spongelike packing keeps the expansion and compressions approximately isothermal. A true Carnot cycle and its theoretical efficiency could be approached if adiabatic steps can be included in the final part of the expansion and compression strokes. Is this feasible?

C. Jahnig: Yes, it is feasible with proper design. Picture the cylinder with no packing, using simple water sprays for temperature control. Now, simply shutting off the sprays changes the operation from isothermal to adiabatic at the desired points in the cycle.

G. Wachtell: Doesn't heating and cooling of the packing and cylinder walls increase the water consumption and parasitic losses?

C. Jahnig: This loss is significant and is allowed for in the design and economics. The packing provides good staging to maximize temperature change on the water used during sensible heat changes.

10. OTEC APPLICATIONS, ECONOMICS, AND INTEGRATION

POTENTIAL FOR OCEAN THERMAL ENERGY CONVERSION ELECTRIC POWER GENERATION IN THE SOUTHEAST REGION

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Abstract

This paper is based upon the results of the recently completed U. S. Department of Energy funded "Southeast Regional Assessment Study." The study evaluated the operating characteristics and probable costs of dispersed and central station solar electric power generation options through the year 2000 in the Southeastern United States. Ocean Thermal Energy Conversion (OTEC) was evaluated in detail using the system characteristics of the Florida Power & Light Company (FPL) together with a resonable, although hypothetical, 22 year generation expansion plan. A computer analysis of system generation costs using conventional fossil and nuclear generation was used as a basis for comparison with a system in which a small percentage of coal-fired capacity was replaced by OTEC capacity sufficient to maintain the same degree of system reliability. From this analysis a break-even capital investment (value to the utility) for OTEC was determined and compared to a range of estimated OTEC costs from previous studies. Sensitivity of the break-even value to assumptions regarding forced outage rates and fuel escalation rates was also studied. The results presented ~~here~~ show that, in comparison to other renewable energy sources available for central station power generation in the Southeast, OTEC holds the greatest promise.

Introduction

The information presented in this paper is abstracted from a much broader study entitled "The Southeast Regional Assessment Study"[1] funded by the Department of Energy (DOE) under contract No. EG - 77 -C -06 - 1018 and just recently completed. The Southeast Study was an assessment of opportunities for solar electric power generation in the Southeastern United States through the year 2000. For central station power generation it evaluated wind, solar thermal, ocean thermal energy conversion (OTEC), photovoltaic, solar thermal hybrid, and wood biomass. While not important to this paper, the Southeast Study also included dispersed user applications of wind, photovoltaics, and solar total energy systems.

Participants in the study were ten electric utilities, two industrial organizations, a university, a state solar energy center, a research institute, and Stone & Webster Engineering Corporation, an architect-engineer. A unique feature of the study was that it also included a four member Consumer Advisory Panel and a State Government Advisory Panel made up of representatives from each of the ten states in the region plus the Commonwealth of Puerto Rico. Of the ten utility participants, three were chosen as representatives of the region's solar characteristics for modeling their systems with central station solar generating plants. These were Baltimore Gas & Electric Company (BG&E), Tennessee Valley Authority (TVA), and Florida Power & Light Company, (FPL). Solar thermal central receiver plants were modeled on all three utilities, photovoltaic plants on BG&E and FPL, and retrofit hybrid and OTEC plants were modeled only on the FPL system.

OTEC Electric Power Generation

OTEC derives its energy from the temperature differences between cold water from great ocean depths and the warm surface waters that occur in tropical and semitropical climates. It is the action of the sun in heating the surface waters that makes OTEC part of the broad spectrum of solar technologies. OTEC plants use a working fluid, such as ammonia, that will boil at temperatures slightly below the ocean surface water temperatures and be liquified slightly above the temperature of the cold water brought from the ocean depths. OTEC's energy is derived from directing the vapor of the working fluid through a turbine that is connected to an electric generator. The exhaust vapor from the turbine is then liquified in a condenser. The condensed working fluid is then pumped to a vaporizer where the cycle begins again.

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OTEC, unlike other solar technologies such as solar thermal, wind, or photovoltaic, derives its energy from a resource that is potentially available at all times. For this reason, OTEC appears to have promise as an economic alternative to other conventional forms of electric power generation for base load use.

Conceptual Plant Design

The OTEC plant concept used for comparative evaluations in this study was based on a 400 MWe station moored in the Gulf of Mexico 150 mi from the west coast of Florida where a 38°F (21°C) water differential temperature was available. Nominal plant rating was 405 MW. The plant was equipped with rectifiers to convert the ac generation to bipolar dc at approximately 200 kV. Inverter equipment was placed onshore to convert the dc power back to ac for normal dispatch. Total losses for ac-dc-ac conversions and I^2R cable losses at full power output were estimated at 25 MWe, resulting in a net generation for dispatch of 380 MWe.

The plant platform consists of a barge-shaped semisubmersible hull of reinforced concrete with 815 ft (249 m) length, 340 ft (104 m) Beam, 180 ft (55 m) draft. Displacement is 415,000 long tons (421,640 metric tons). A two-point anchoring system is used at an assumed mooring depth of 4,600 ft (1,400 m). The cold water pipe is suspended prestressed concrete pipe 3,300 ft (1,000 m) long; 18 in. (0.5 m) thick walls; 129 ft (39 m) ID (at the top) with flexible joints.

Turbine generators consist of three ammonia-powered turbines per module, each rated at 32 MW shaft power. Each turbine drives an 1,800 rpm, 13 kV generator. An artist's conception of a 400 MW OTEC central station is shown in Figure 1.

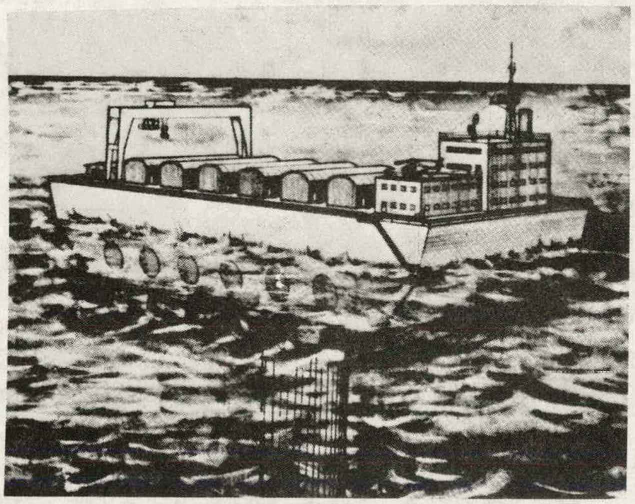

Fig. 1 400 MWe Ocean Thermal Energy Conversion Plant

The estimated capital costs of the plant (1978 dollars) ranged from \$1,500 to \$2,600 per installed kW and were based on the one-hundreth plant in a

Table 1 Capital Cost Estimate for 100 MW OTEC Generating Plant

Category	1978 \$/kW First Plant (Low/High)	1978 \$/kW 100th Plant (Low/High)	Basis
Platform	30/178	30/178	DOE/Mitre[2]
Cold Water Pipe	83/92	83/92	DOE/Mitre[2]
Heat Exchangers	651/1,134	450/850	DOE/Mitre[2]
Fluid Transfer Subsystem	213/450	213/450	DOE/Mitre[2]
Turbine Generator	128/205	128/205	DOE/Mitre[2]
Auxiliary Systems	62/112	62/112	DOE/Mitre[2]
Mooring	43/167	43/167	DOE/Mitre[2]
Cable	245/294	245/294	DOE/Mitre[2] (140 mi off Tampa, assumes 3,000 MW plant complex)
Deployment	3/4	3/4	DOE/Mitre[2]
Total Direct Cost	1,458/2,636	1,257/2,352	
Indirects	73/132	63/118	5% of Direct Cost
Total Plant Cost	1,531/2,768	1,320/2,470	
Owner's Cost	42/69	34/60	2.5% of Plant Cost
AFUDC	150/91	130/74	10.5% - 6% (escalation) applied to 40% of 24-month construction period
Total Capital Cost	1,797/2,995	1,483/2,605	
Rounded Off	1,800/3,000	1,500/2,600	

series of like plants, mass produced for cost efficiencies. The range in costs is based on optimistic/pessimistic spread for the different plant components found in DOE studies[2] performed by other contractors. The wide range of values are due to the considerable uncertainty of a new technology. The basis of this cost estimate is shown in Table 1.

Full plant output is available when the water temperature differential is in excess of 38°F (21°C). Temperature differences less than 38°F will decrease plant output roughly by the temperature squared (Pnet = Pgross x $(\text{Tactual}/\text{Trated})^2$ -. Ploss). Data obtained from Tetratech[3] indicate monthly average temperature differentials in excess of the rated 38°F for all months except February and March, during which it is reduced to 37°F (20.6°C) in February and 36.75°F (20.4°C) in March.

The reduced output for less than optimal temperature differential was simulated by adjusting the number of days of maintenance to obtain a reduced availability rather than a derated output. The normal plant expected down time for maintenance is 8 days/yr based on estimates by MITRE.[2] Since less than full plant output occurs in February and March, maintenance was scheduled for this period and increased to 12 days/yr.

Conventional Plant Costs and Economics Assumptions

Conventional coal fuel plant capital costs and economic assumptions for the FP&L system are given in Table 2.

Table 2 Economic Assumptions Used for Calculating Value of Solar Electric Facilities to FP&L

Capital Cost of Conventional Coal Plant Capacity (1978 $ including owner's costs and IDC)	808/kW
Coal Fuel Costs, 1978 $	0.0147/kWh
Coal Plant O&M Costs, 1978 $	8.00/kW-yr
Discount Rate, %	9.37
Inflation Rate for Capital Costs, %	6.0
Inflation Rate for Fuel Costs, %	6.0
Assumed General Inflation Rate, %	6.0
Levelized Annual Capital Cost Factor, %	18.92

For the year 2000, FP&L has projected that their installed generating capacity would be approximately half oil-fired units and a quarter each of coal and nuclear. It must be recognized that utility generation planning is a dynamic process and subject to change due primarily to fluctuations in load growth, plus new financial, environmental, and regulatory constraints.

Table 3 presents, for the year 2000, the projected generating mixes of the FP&L system for the base case conventional generation expansion plan. This table shows the proportions and costs of total energy supplied by oil, coal, and nuclear fuels.

For the FPL System, the total estimated generating capacity in the year 2000 is 22,883 MW.

Table 3 Projected Generating Mix for Conventional Expansion Plan for Year 2000 on FP&L System

Fuel	Energy Supplied, %	Average Fuel Cost, Mills/kWh
Oil	23.7	67.9
Coal	36.0	52.9
Nuclear	40.3	15.1
Total System	100.0	41.2

Methodology For Computation of Value of OTEC Plant

The first commercial OTEC central stations are not expected to be available until the 1990s and will have small impact on utility economics before the year 2000. For this reason, generation costing studies focused on the single year 2000, with adjustments made to include the effects of fuel cost escalation to the year 2020.

The production costing model used to determine the economic value of OTEC uses the Booth-Baleriaux simulation, which is widely accepted for system planning studies. In this method, forced outages and planned maintenance cycles are simulated by decreasing the availability of the plant. In general, this is a good simulation for OTEC since forced outages and maintenance will primarily involve complete plant shutdown. Failures such as cold water pipe break, underwater transmission cable failure, or mooring failures resulting from ocean storms could cause outages of long duration.

The value of OTEC central station can be attributed to three components - capacity replacement value, fuel replacement value, and excess operation and maintenance (O&M) costs.

Capacity replacement value results from the reduction in the size or number of conventional units when OTEC units are added to an expansion plan. Reliability is used as the basis to determine the capacity replacement value.

The reliability of a system is measured in terms of "Loss of Load Hours" (LOLH), which represent the number of hours in a year that simultaneous outages of generating capacity will exceed the installed reserve. By adding OTEC capacity to a conventional generating expansion plan, the system reliability improves, reducing the LOLH. Then, removing just enough conventional generation will result in obtaining the same reliability, or LOLH, associated with the conventional plan. The investment in this conventional capacity replaced by OTEC is defined as the capacity replacement value of the OTEC capacity added. The savings in capital costs of this displaced capacity is calculated based on an estimated $808/kw in 1978 dollars.

Since the coal fuel costs are 3.5 times greater than nuclear fuel costs for the FP&L system, it was assumed that OTEC would replace coal-fueled units planned for construction in the 1990s. As a result, the fuel cost simulations show that a large part of the fuel replaced is coal. A smaller portion of the fuel savings is attributable to the reduced operation of oil-fueled existing capacity.

The fuel replacement value of OTEC facilities is primarily the saving in the cost of fuel that would have been consumed in the coal plant that was replaced by OTEC facilities. There also are small savings in fuel burned in other coal and oil plants. The fuel cost escalation rate is used to compute a levelized fuel savings for the period 2000-2020. The levelized fuel savings represents the fuel savings for the year 2000, escalated each year for 20 yr, discounted to the year 2000, and made into a levelized uniform savings. This in effect accounts for the end effects of 20 yr of fuel escalation. Dividing the levelized fuel savings by the fixed charge rate provides a capitalized value of fuel savings in the year 2000.

O&M costs for OTEC stations are expected to be greater than for conventional facilities. O&M costs for conventional facilities were estimated as 1 percent of the conventional capital cost, which for a coal plant is approximately $8/kw/yr in 1978 dollars. OTEC O&M costs were estimated to be 1 1/3 percent of capital cost or $35/kw/yr in 1978 dollars. The annual O&M penalty for OTEC is obtained by subtracting the O&M cost of the replaced conventional capacity from the OTEC plant O&M. This annual O&M penalty is then divided by the fixed rate to produce a capitalized O&M penalty.

Overall plant availability of OTEC is predicted to be as high as 90 percent by TRW[4] and MITRE.[5] This corresponds to a forced outage rate of nearly 10 percent. A major uncertainty of OTEC is the availability of power delivered to the ac system ashore. To account for this uncertainty, various forced outage rates in the range of 10-40 percent were simulated to determine their effect on the break-even investment.

Value of an OTEC Plant to FP&L

Based on the analysis methodology described above, the break-even value (or the amount of capital investment that would be justified to break even with a coal-fired plant) for several different fuel escalation rates and several different forced outage rates is given in Table 4 for a penetration of 5 percent (OTEC plants comprising 5 percent of the system generating capacity). Computations were also made for OTEC penetrations of 10 percent, but the differences in break-even value were less than 1 percent and are not of any consequence.

Table 4 Break-even Value of OTEC Plants, 1978 $/kW

Rate of Fuel	Forced Outage Rate, %			
Escalation, %	10	20	30	40
6	$1,340	$1,160	$ 980	$ 810
10	$3,060	$2,700	$2,330	$1,960

OTEC is essentially a "free fuel" source of energy. Escalation rates of conventional fuels will have a dramatic effect on the break-even investment of OTEC as shown in Figure 2. The break-even investment of OTEC for a fuel escalation rate of 9 1/2 percent is twice the break-even investment with a fuel escalation of 6 percent.

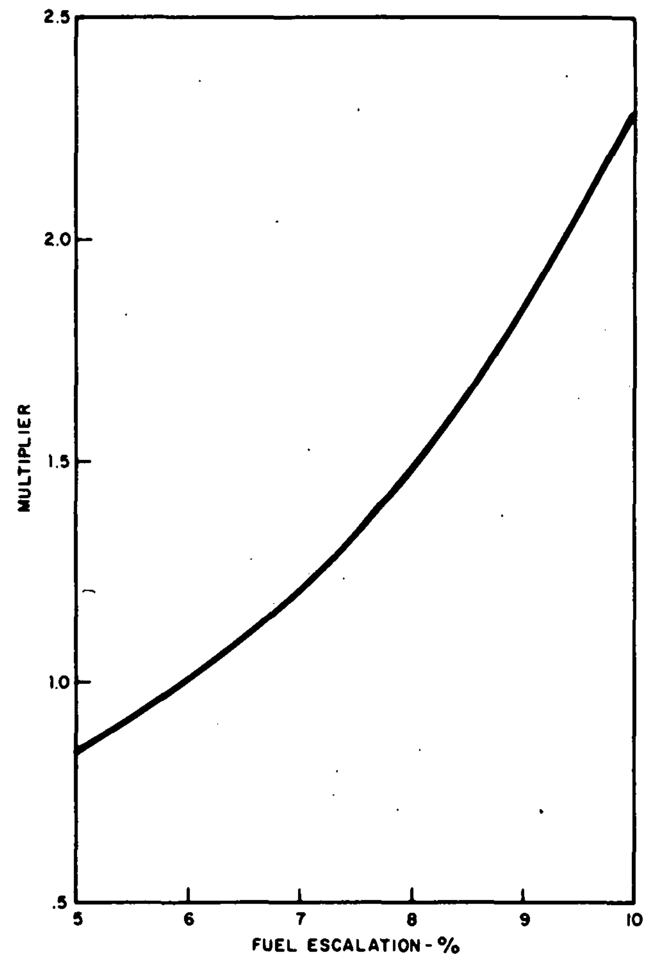

Fig. 2 Multiplier to Correct Solar Breakeven Investment for Changes in Fuel Escalation Rate (1980 to 2000) on FPL System

Break-even costs are also significantly affected by the cost of competing conventional fuels and other economic factors specific to an individual utility. Low fixed charge rates, which are more frequently found in publicly owned utility systems due to their lower or nonexistent taxes and their ability to borrow capital at lower interest rates, permit higher capital investments to achieve break-even. High fuel costs permit higher fuel credits to offset the high OTEC capital costs. Puerto Rico would be an example where both of these considerations would favor OTEC. Hawaii would be an example where high fuel costs would play a major role.

The relatively high allowable investments to achieve the break-even shown in Table 4 result primarily from the high capacity factor of the OTEC plant. With a scheduled downtime for preventative maintenance of 8 days per year and a forced outage of 10 percent, the capacity factor is 87 percent. Increasing the forced outage to 30 percent results in a capacity factor of 68 percent and a lower allowable investment.

Figure 3 shows graphically the relative value of all OTEC plants on the FP&L system to the high and low range capital cost estimates for the plant in 1978 dollars. Based on the analysis for the FP&L system, it appears that mature design OTEC plants could be a viable economic option for certain electric utilities in the Southeast by the year

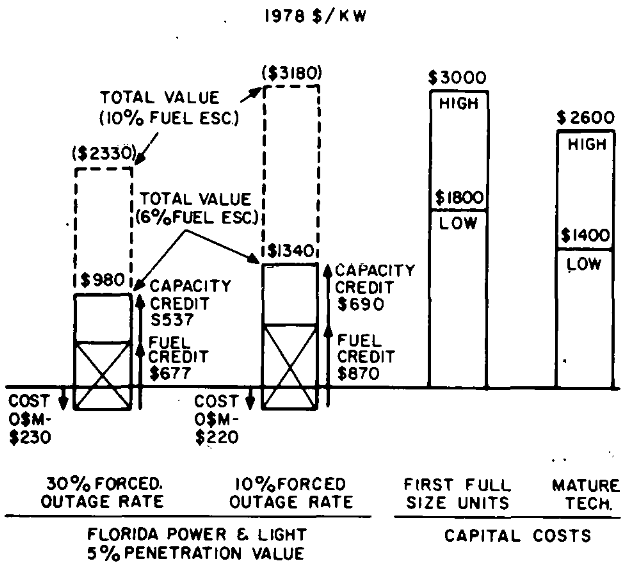

Fig. 3 Comparison of Cost and Value of OTEC Central Stations On FP&L System

2000, based on current DOE program goals. The key factors to making OTEC a viable utility option are:

- Achieving or reducing the current capital cost goals
- Obtaining a highly reliable OTEC plant that will have a low forced outage rate
- Escalation of fossil fuels above general inflation rate
- Minimizing O&M costs
- Demonstration of large-scale OTEC plants as soon as possible

Comparison of OTEC with Other Solar Electric Central Station Options for FP&L

OTEC is only one of several types of solar electric generating plants modeled in the Southeast Regional Assessment Study on the FP&L system. The other solar electric plants studied, using the same methodology described above, were solar thermal (central receiver), photovoltaic, wood biomass, retrofit hybrid, and new hybrid. Hybrid plants use a central receiver solar collector in conjunction with a conventional fossil-fuel-fired boiler so that the electric output of the plant is available by combustion of fossil fuels when the solar portion of the plant is not in operation. The following table indicates the relative value of the several solar central station options.

As can be seen above, none of the solar electric central station options indicate the chance of achieving the economic break-even that OTEC shows. It is clear that if the Southeast region is to have a solar electric central station option available to it, high priority should be given to achieving the DOE cost and reliability goals for OTEC in as expedient a manner as possible.

Table 5 Summary Comparison of Cost and Value of Solar Central Stations

Plant Type	1978 $/kW (for year 2000) Mature Plant Costs(1)	Value at 6% Fuel Escalation(2)	Value at 10% Fuel Escalation(2)
Central Station			
Solar Thermal (6 hr storage)	2,000-2,500	350	830
Photovoltaic	1,300-1,700	390	820
Retrofit Hybrid	900-1,300	(50)	330
New Hybrid	Not Estimated	520	510
OTEC (10% forced outage rate)	1,400-2,600	1,340	3,180
OTEC (30% forced outage rate)	1,400-2,600	980	2,330

Notes: 1. Assumes DOE cost goals for development of solar components are achieved.

2. Fuel inflation rates include 6 percent general inflation; 10 percent fuel escalation is equivalent to 4 percent greater than general inflation.

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DISCUSSION

Question: Mr. Sutherland should clarify whether, if OTEC displaces oil consuming power plants, it has in fact replaced oil that would otherwise have been consumed to produce the same electrical energy.

P. Sutherland: In the Southeast Regional Assessment Study, we assumed that OTEC units would take the place of planned new generation. Since no new oil generation is permitted (except for peaking units) under the Powerplant and Industrial Fuel Use Act of 1978, then OTEC cannot be assumed to displace new oil fired generation.

It is possible that OTEC could be used to replace prematurely retired existing oil fired units. In that case the value of OTEC in terms of capacity credits would be reduced to zero since the required capacity was already available from the existing oil unit. However, the reduction in capacity credit would be offset by an increase in the value of fuel displacement. This case was not specifically considered in the study so the net effect is uncertain. However, it is obvious that this alternative would increase the financing requirements on the utility over and above the already capital intensive case considered in the Southeast Study.

G. Dugger, APL: First I want to congratulate you on the paper as a whole. Of course we are happy to see that OTEC comes out better than the other solar option. I have a couple of questions though. How can you justify the 15.1 mills/kWh for nuclear power in the year 2000?

P. Sutherland: We're talking about fuel costs there, and the inflation assumption is just 6%/yr, equal to general inflation.

G. Dugger: Is that realistic considering that uranium also is a diminishing resource?

P. Sutherland: In fact our system planning people assume a decreasing real cost of uranium due to things like uranium mining from phosphate mines as a by-product and new uranium discoveries in Australia and things like that. So, we think that is a realistic assumption.

G. Dugger: That is interesting because I looked at uranium from phosphate ores in Florida when I was working for IMC in 1954, and it did not look very promising then.

P. Sutherland: We have a contract with IMC now to buy uranium from them.

G. Dugger: My other question is: Why do you think the outage rate will be $\geq$20%?

P. Sutherland: Because the outage rate on conventional power plants is more than that.

G. Dugger: I would suggest that your home refrigerator is very reliable, and we believe OTEC plants will be, too. We think that 10% is a good figure; it may be less than that.

P. Sutherland: We'd be glad to see one operate at that reliability level.

G. Dugger: One final short thing, you said that the utilities would not be interested in working with a captain of a plant. I have seen in newspapers recently the suggestion that that is exactly what the nuclear plants need — to put a captain in charge who knows what's going on, is responsible, and can run the plant.

R. Cohen: Why are we talking about fuel costs — aren't we talking about newly installed capacity? Why don't you throw in the capital costs as well as the fuel costs?

D. Guild: The capital costs for coal (shown in Table 2) and OTEC (shown in Fig. 3) were included in the analysis but were not shown in Table 3 for simplicity.

Question: In your OTEC transmission costs, are you assuming a redundant cable? Or a sea return?

D. Guild: Costs are based upon a 3000 MW plant complex with six pairs of paper-impregnated bipolar (±300 kV) cable with a seawater return. I think it is important to keep a study like this in perspective. The different forced outage rates we had were to show what the sensitivity of the economics would be to forced outage rates. If an OTEC plant can achieve 10%, that will be great. The utilities will be lining up to buy OTEC plants after you show them that they work. Whether the outage rate is 10, 20, or 30% really doesn't make much difference. If you can achieve a good forced outage rate by designing a reliable plant, you will have a going concern.

R. Cohen: I congratulate you also. I think that's an excellent study, very broad-minded in most respects.

A CASE STUDY OF OTEC PLANT FINANCING FOR THE MIDDLE SOUTH UTILITIES

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Abstract

This paper presents the results to date of a case study of Ocean Thermal Energy Conversion (OTEC) financing for Middle South Utilities, Inc. (MSU), an investor-owned public utility holding company with 100 ~~percent~~ % of the common stock of Arkansas Power and Light, Arkansas-Missouri Power Company, Louisiana Power and Light, Mississippi Power and Light, and New Orleans Public Service, Inc. Selection of MSU for this study is appropriate because of its proximity to potential OTEC sites in the Gulf of Mexico, its interest in non-traditional forms of electricity generation, its capacity addition outlook, and its innovative solutions to the many financial problems plaguing all utilities these days. ~~Specifically,~~ MSU has formed Middle South Energy to finance and own large, capital-intensive, baseload units ~~(such as OTEC)~~ for the System, ~~thereby~~ relieving the individual operating companies of the pressures associated with financing these programs on their own. ~~The study has led to several preliminary conclusions:~~ Incentives designed to increase the availability of financing will not speed OTEC integration until the MSU System is assured of OTEC technical performance and costs. Once the System decides that OTEC performance and costs justify OTEC integration, financing will be no more difficult to obtain for OTEC than for the rest of its construction program as long as there is sufficient lead time to incorporate OTEC into its generation and financing plans. ~~Under these circumstances,~~ financing incentives will be needed only to the extent necessary to make the first few OTEC plants cost-competitive with nuclear and coal plants, including consideration of such potential beneficial factors as shorter construction times and fewer environmental problems with OTEC plants. ~~If sufficient lead time cannot be given,~~ special financing arrangements may be required to induce the utility to purchase the demonstration plant once its reliability is proven.

Introduction

This paper presents the results to date of a case study of OTEC financing for the Middle South Utilities, Inc. (MSU). This work is being done under the sponsorship of the Department of Energy, Division of Central Solar Technology, as part of the contract ET-78-C-02-5092 entitled "Study of Integration Issues to Realize OTEC Market Potential." Selection of MSU for this study was appropriate not only because of its proximity to potential OTEC sites in the Gulf of Mexico but also because of its interest in alternative methods of electricity generation including OTEC and its innovative approaches to the many financial problems facing utilities in general. Described herein are the Middle South System, its financing requirements and methods (including how they would relate to OTEC), and barriers and incentives to the integration of OTEC within the Middle South System.

Middle South Utilities, Inc. is an investor-owned public utility holding company which owns 100 percent of the common stock of five operating companies:

1. Arkansas Power and Light Company (AP&L), regulated by the Arkansas and Tennessee Public Service Commissions;

2. Arkansas-Missouri Power Company (ARK-MO), regulated by the Arkansas and the Missouri Public Service Commissions;

3. Louisiana Power and Light (LP&L), regulated by the Louisiana Public Service Commission;

4. Mississippi Power and Light (MP&L), regulated by the Mississippi Public Service Commission; and

5. New Orleans Public Service, Inc. (NOPSI), regulated by the New Orleans City Council.

The two other principal subsidiaries are:

1. Middle South Services (MSS), a service company; and

2. Middle South Energy, Inc. (MSE), which provides financing and ownership of certain baseload generating units within the System and is regulated by the Federal Energy Regulatory Commission as are the wholesale sales of each of the subsidiaries.

System Fuels, Inc., subsidiary of AP&L, LP&L, MP&L, and NOPSI, is responsible for fuels procurement, transport, storage and handling, and gas and uranium exploration for the System. The Associated Natural Gas Company is a gas distribution subsidiary of ARK-MO. The name Middle South Utilities System refers to each of the companies described above in addition to Middle South Utilities, Inc.

Membership in the System is advantageous for the operating companies since the System enjoys a diversity of load which is not available to the companies individually. In addition, the System can build larger, more efficient plants and take advantage of economies of scale in management and financing. Finally, if an operating company experiences difficulty, the financial community will be more receptive to requests for financing when the company is under the umbrella of the holding company.[1]

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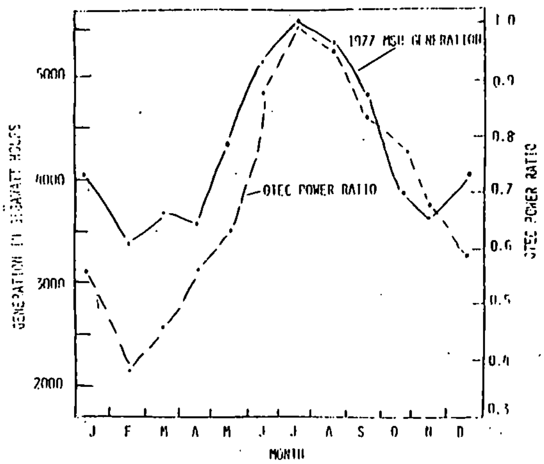

Figure 1. MSU seasonal demand pattern and OTEC seasonal power ratio.

Operating Characteristics

The power load for the System peaks in the summer with a summer to winter peak load ratio of 1.4:1. The 1978 summer peak demand for the System was 10.7 GW. The load is largely made up of industrial customers, who accounted for over 40 percent of energy sales in 1977. Residential customers made up one-third of the load, and commercial and other users the remainder. The System has a no-cost sharing arrangement, extending through the year 2000, with TVA in which TVA provides power to help MSU meet its summer peak and MSU supplies equivalent amounts of power to help TVA meet its winter peak. In 1977, 592 GWh were exchanged under this arrangement and sales by the System to outside utilities amounted to 6289 GWh.

The System appears to be compatible in at least two respects with the use of an OTEC plant. First, as Figure 1 shows, its seasonal demand pattern[7] is similar to the seasonal power potential of an OTEC plant, which was determined[2] as a function of monthly OTEC temperature-difference (surface water temperature minus temperature at 100-m.depth) data[3]. The OTEC power potential (or power ratio) is defined for a specific unit as the ratio of the maximum monthly output capability to the designed maximum output of the unit. Second, the System is a suitable candidate for the integration of one or more OTEC plants because of its projected growth in capacity (Table 1). This estimate is based upon an estimated average growth in peak load of 6.1 percent per year through 1994, and 4.8 percent through 2005. The estimates of growth until 1994 were made by MSU and estimates for 1995-2005 were made by General Electric-TEMPO, assuming a 16 percent reserve margin.

The generation mix (Table 1) of the System is anticipated to shift considerably over the next 18 years from gas and oil to nuclear and coal-fired capacity. The MSU System is also working with the Federal Government to study the feasibility of compressed air energy storage. Excess off-peak energy (such as that which could be generated by an OTEC) would be used to pump air into underground salt domes which are numerous in the System's

Table 1. Projections of capacity and generation mix of the MSU System.

Year	Installed	Generation Mix %			
	Capacity (GW)	Gas	Nuclear	Oil	Coal
1978	12	43	11	46	0
1995	34	7	26*	20	47
2005	54	--	--	--	--

service area. A decision will be made in 1980 after completion of the design project on whether to go ahead with the development of a pilot compressed air storage plant.

System Financing Requirements and Procedures

Construction expenditures for additions to the System's facilities, 1977-83, are shown in Table 2. System estimates for required capital expenditures for the 1984-88 period show no significant departures from this trend of upward growth although the rate of increase is likely to be less than that over the 1982-83 period. Financing strategies for MSU are discussed in the following subsections with respect to two major components: determination of asset ownership and selection of appropriate financing methods.

Table 2. Recent and projected near-future capital expenditures by the MSU System (Millions of current dollars)

Year	1977	1978	1979	1980	1981	1982	1983
$	692	901	911	948	830	1019	1568

Determination of Asset Ownership

In large part, MSU additions to generating capacity are operated and owned by the operating company experiencing the load growth. There are, however, at least two exceptions to this policy:

1. If an operating company is located in a geographic area giving it a clear cost advantage in generating electricity by a certain technique, that company will own plants of that type and sell to the other companies in the System. Examples of this would be the presence of a rich coal field or hydro potential in a company's service area. For example, in the past, Louisiana has had available relatively low cost gas. In the long run, however, an operating company would not own capacity exceeding its own requirements.

2. Because of the growing financing requirements associated with (a) new capital-intensive technologies and (b) the increased minimum efficient size of baseload plants, an operating company may find both its anticipated load growth and

financing capability unable to support such plants. In this event, Middle South Energy (MSE) can own the plant and allow the company in the most economical service area to be the operator. For example, MSE is now building two 1250 MW nuclear plants at the Grand Gulf Nuclear Station in the MP&L area. The new electricity will be sold to the members of the System on some mutually agreed upon apportionment and at rates set by the Federal Energy Regulatory Commission (FERC). It has not yet been determined whether the allocation will be on an annual basis or on a one-time fixed block basis.

The operating companies are also entering into joint ownership arrangements with municipalities and cooperatives who have been their customers. This aids the financial condition of the operating company in that (a) they can build larger, more efficient plants, (b) they can meet their obligation to their customers (the cooperatives and the municipals) by selling plant capacity instead of electricity, and (c) the capacity owned by municipalities and cooperatives represents financing requirements which the System does not have to meet. Arkansas Power and Light holds a 60 percent interest and a rural electric cooperative and municipal electric company holds a 40 percent interest in two 700-MW coal plants at the White Bluff generating station. This same group has entered into a similar 60-40 sharing arrangement for the construction of two other 700-MW units near Newark, Arkansas.

In the case of OTEC plant additions, aspects of both exceptions (1) and (2) above would be present. Because of proximity to an OTEC Gulf site off New Orleans, LP&L would be the likely operator of the OTEC plant, and the transmission lines would run ashore into LP&L's service area. However, because (a) the plant would be baseloaded and very capital intensive, (b) it is in the System's interest to utilize new methods of electricity generation and (c) LP&L is not in financial health, MSE might own the plant.

The important point is that the System is developing unique methods of plant ownership which overcome some of the financing problems experienced by single operating companies. These methods should play an important role in supporting the integration of OTEC plants into the MSU System.

<u>Financing Methods</u>

Once ownership of the desired facility is determined, there are several methods which the owner can use to obtain the needed financing: short-term bank loans, bonds, preferred and common stock, leasing, and construction financing. The consolidated capitalization ratios for the MSU System in December 1978 were long-term debt, 59.9 percent; preferred stock, 7.8 percent; and common equity, 32.3 percent.

Short-term bank loans are used on an interim basis by all members of the System as required up to a charter-specified limit of 5 percent of debt. At some point, however, the short-term debt is converted to long-term financing. The most common form of long-term debt financing used by the System is first mortgage bonds. These may be in the name of any of the operating companies or Middle South Energy depending upon the ownership of the equipment. Under most circumstances, bond issues of utility holding companies are by SEC rule made publicly and competitively. A sizable exception was made in the case of MSE, the newly created generating subsidiary, when it issued bonds to finance its nuclear plants. Since it was a new company with no earnings, it required special treatment in its dealings with the financial community. Therefore, it was granted an exemption to the rule and sold privately $400 million in first mortgage bonds to 15 insurance companies. Once, however, its second plant begins operation, MSE will be required to place bonds competitively.

Preferred stock is currently issued by each operating company and will be issued by MSE when its two plants go on line. Common stock, however, can be issued publicly only in the name of the holding company. The holding company, in turn, buys all of the common stock of the operating companies and Middle South Energy.

All of the operating companies and MSE may engage in leasing,* which can be advantageous for several reasons.

1. The System can utilize assets with high rates of obsolescence without having to own them.

2. For an operating company in poor financial condition, leasing may be the only way to gain the services of a needed piece of capital equipment.

3. Because MSU has little taxable income, it may be advantageous for it to lease equipment from a company which is able to use the tax advantages (such as the Investment Tax Credit) of ownership. In return the implicit "interest" terms of the lease may be better than they would be with normal financing.

Reasons 1, 2, and 3 above for leasing are not relevant in the case of a utility in strong financial health which will have no trouble obtaining financing at rates lower** than that paid by most other firms. Therefore, it is unlikely that a leasing firm could offer an implicit rental cost which would be competitive. Moreover, a healthy utility will usually owe a sizable sum in taxes and, therefore, will be able to utilize all of the available tax preferences.

The MSU System does engage in leasing. For example, it holds simple leases on its nuclear fuel, and barges and tugs are leased by System Fuels. Currently underway is the study by MSU of the possibility of the leveraged leasing of local cars, coal handling equipment, and generating equipment.

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* Leasing may be a simple form wherein the "owner" of the equipment borrows all of the cost of the equipment. However, for the "owner" to be allowed tax privileges by the IRS, he must hold a minimum equity interest in the equipment.

** Long-term debt is the cheapest form of financing. Since utilities are able to use this method more than are other firms, their weighted cost of capital is lower This advantage is greater when the utility is healthy and has highly rated bonds. For example, the average utility costs of capital used by EPRI in their analyses are long-term debt, 8 percent; preferred stock, 8.5 percent; and common stock, 14 percent.

In summary, leasing is used by the System as an economical method of obtaining the services of certain assets, given the System's current financial position. This method, however, will become less competitive with other financing techniques as the System's financial health improves.

A new financing technique being considered by the System is a form of construction financing. Under this arrangement, a third party will set up a temporary trust for the duration of the project. Funding in the form of debt would come from large financial institutions such as banks on the basis of the value of the asset. The advantage to the System of such a technique is that mortgage financing is avoided.

It is conceivable that OTEC plants could be financed using any of the techniques described above. According to a representative of the System, once the technical performance and cost competitiveness of the OTEC plant is demonstrated, the System's financing of the plant could be done using conventional methods and sources.[1] The System is already dealing with the problems involved in financing capital-intensive projects. However, characteristics of OTEC may make possible the use of alternative methods which would overcome problems common to the financing of all major utility capital expenditures.

For example, a major problem faced by a regulated utility (such as MSU) is that carrying costs prior to plant operation cannot be collected at the time the expenses are incurred. Instead, these charges are capitalized, thus added to total capital expenditures on the plant and amortized over the life of the plant. This procedure can create severe cash flow problems, especially with capital intensive technologies having long construction times. Although capital intensive, OTEC plants have two characteristics which would ameliorate this problem to some extent. First of all, an OTEC plant is expected to have a relatively short build schedule, such as two to three years. Therefore, the time will be reduced prior to which financing costs can be recovered. Second, the OTEC plant, being portable, could be constructed by a non-regulated third party. The plant would be moved to the System's OTEC site and sold to the System just prior to operation. All acquisition costs, which would of course include financing cost during construction, could then be placed in the rate base immediately. The System would thereby reduce its cash flow problems associated with financing construction programs prior to plant operation.

Factors Affecting MSU's Financial Condition

The System must operate within a climate plagued by problems common to many utilities. These problems contribute to high costs, low earnings, and unsatisfactory debt coverage ratios, which in turn lead to even higher financing costs. Environmental expenditures have raised costs without producing more electricity. Inflation has had a great impact on the operating and capital costs of the System. In fact, the escalation of the prices of construction-related goods and services used by the utility has been greater than increases in the Consumer Price Index.[1] The cost of new money has doubled in recent years. Fuel price increases have adversely impacted the System, and finally, the public's lack of understanding of the utilities' economic environment has placed the utility in an adversary's position with respect to its customers. An MSU official commented that the public and regulatory response to the Three Mile Island mishap may result in an ultra-conservative backlash by the utility industry. That is, the industry may ask "why take risks to cut costs?" When the risk works out, the customers benefit from lower electricity prices while the stockholders can get no more than the regulated rate of return. On the other hand, if the risk doesn't work, the customers don't want to pay and the regulatory commission doesn't want to make them. There is no incentive, therefore, for the utility management and the stockholders to take risks. If this attitude does become widespread in the industry, then utilities may be even slower than anticipated in integrating OTEC.

The System also is at a disadvantage relative to the rest of the industry because of the regulatory climate (ranked below average by Value Line), which is particularly poor in Louisiana. The LP&L 1977 Annual Report refers to "the unfavorable regulatory climate" and the "unrealistic attitude of most members of the Commission." These particularly strong feelings come in the wake of a long series of hearings before the Louisiana Public Service Commission. Out of a $54 million rate increase requested, only $5 million was granted. On appeal to the district court, an additional $9 million in rates was granted. This lack of rate relief is responsible in part for LP&L's worsening debt coverage ratio (the ratio of net income before payment of interest and income taxes to interest obligations). Without sufficient rate relief to make the acquisition of outside financing possible, LP&L may be unable to complete its construction program on schedule.

A Scenario for the Integration of OTEC Plants by MSU

At least four criteria must be met before MSU completely integrates OTEC plants:

1. A plant of sufficient size (+500MW) must be demonstrated in the Gulf waters for a suitable period of time to prove technical performance including resource reliability, the performance of the cable, and the durability of the cold water pipe during storms.

2. Construction, operating and maintenance costs for commercial sized plants must be known within a reasonable range of certainty.

3. The OTEC plant must be competitive with other baseload options such as coal; and

4. There must be OTEC financing available at a cost competitive with financing costs for other types of generating capacity.

Table 3 outlines an OTEC integration scenario for MSU under which the government and MSU place a cost-shared commercial sized OTEC unit in MSU waters in 1995. Ownership of the plant could be turned over to MSU through a grant-to-loan conversion plan (1) once technical and economic characteristics of the plant are known and competitive on the subsidized basis (with the government's share decreasing for subsequent plants as plant costs decrease due to learning curve benefits and technological improvements) and (2) after the necessary lead time for inclusion in the generation and financial plan has passed.

Table 3. Scenario for OTEC introduction into MSU System

1987-1990	Design and construction of commercial OTEC under Government Purchase Order
1990	Start construction of several commercial units under Government Purchase Order
1995	Deliver first OTEC 100 to MSU under grant to loan conversion plan
2003-2005	Deliver additional units to MSU

For example, if it is assumed that the required demonstration period is three years, then the soonest the demonstration plant can be included in MSU's generation and financial plans is 1998. For the demonstration plant, inclusion in the generation plan is not critical since the System is so large it could absorb the additional electricity with little impact on the System. It is very important, however, that the demonstration unit be incorporated into the financial plan if ownership is to be transferred from the government to the utility.

An example is given here to show why the utility would not purchase the demonstration plant immediately after the required demonstration period (and before inclusion in the financial plan). It is assumed that the demonstration is a 100 MW unit that is proven technically by 1998. At that time the government offers the plant to the utility at a competitive cost of $200 million* (1978 dollars). If this financing requirement is compared to projected total MSU financing requirements in 1998 of $1.31 billion (1978 dollars),** then the OTEC purchase would increase their financing requirements by 15 percent in that year. Such an expenditure would be hard for the utility to justify both internally and with the regulatory commissions. It is unlikely that sufficient financing could be raised on such short notice, and even if it could, it is even more unlikely that the regulatory commission would allow the cost of capacity not required at that time by the System to be included in the rate base.[1] If, however, the grant to loan conversion could be delayed five years, then the System could possibly incorporate the OTEC plant into its financial plan and ownership could be transferred in 2002.

If MSU's projected load growth is perceived in 1998 (when the demonstration is completed) as sufficient and if the OTECs are competitive,* then MSU would place additional OTEC plants in its generation plan for the 2005 to 2008 period. The financing for these plants** could come from the utility's own sources (e.g., construction trusts) or it could come from federally guaranteed loans. However, even though federally guaranteed loans are easier for the utility to administer than are construction trusts, the former may represent more government control than the utility is willing to accept. At this time, one cannot predict which method will appear best to the utility in 1998.

*Based upon information in memo entitled "OTEC Contractors on Market Integration Issues" received from Department of Energy, 30 May 1979.

**Assumes an 8 percent nominal rate of growth of construction expenditures per year after 1978 and a 6 percent rate of general inflation. This estimate was made entirely for purposes of the example and is not sanctioned by the Middle South System.

Summary

The major conclusions of this paper can be summarized:

- Incentives designed to increase the availability of financing will not speed OTEC integration until the MSU System is assured of OTEC technical performance and costs.
- Once the System decides that OTEC performance and costs justify OTEC integration financing will be no more difficult to obtain for OTEC than for the rest of its construction program as long as there is sufficient lead time to incorporate OTEC into its generation and financial plan. Under these circumstances, financing incentives will be needed only to the extent necessary to make the first few OTEC plants cost-competitive with nuclear and coal plants.
- If sufficient lead time cannot be given, special financing arrangements may be required to induce the utility to purchase the demonstration plant once its reliability is proven.

*May require government subsidy.

**It is assumed that these plants are built under government purchase order. Therefore, ordinary first mortgage financing cannot be used since (1) the total is too large to be raised all in one year by this means and (2) since the plant is not being built by the utility there cannot be gradual financing because the utility has no equity until it takes possession. If the utility built the plants, the three to four year lead time would be adequate for conventional financing methods.

Acknowledgement

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ELECTRIC UTILITY SYSTEM PLANNING STUDIES FOR OTEC POWER INTEGRATION

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Abstract

is discussed.

~~This paper discusses~~ the plan developed to undertake a generation expansion study assuming availability of commercial OTEC plants as well as more conventional coal, oil, and nuclear options. Four major areas of study are discussed: Generation System, Electrical System, Licensing and Environmental. The approach to generation expansion is based on a system planning method using reliability criteria complemented with analyses of reserve requirements. System reliability calculations are based on frequency and duration methods. The following deliverables were requested in the plan: alternative generation expansion schemes, reliability evaluation, energy production costs, and OTEC break-even requirements. The electrical system portion is planned to depart from assumed interconnections of the added generation capacity at three optional sites. Determination of costs for various voltages, capacities, and maximum reliability cable configurations are principal items of this portion of the plan. The approach to licensing includes a study of Federal and local laws related to the licensing process of a stationary sea floating facility. The role of all regulatory agencies will be discussed and defined using information to be obtained through specific inquiries. The design of the environmental study was focused on the phenomena of the OTEC platform attraction on the pelagic organisms. This study, to be conducted by the Center for Energy and Environmental Research of the University of Puerto Rico, stems from the known fact that big platforms moored on high seas will attract great quantities of fish and pelagic organisms and will enhance their reproduction. ~~The nature and extent of this effect, frequently oversighted, is presented in the discussion as a very important issue of OTEC platforms siting bearing potential effect on project feasibility.~~

Introduction

The proposed study came out of the probable realization by DOE that OTEC technology has matured to the point where it was time to attempt its addition, at least in theory, into the generation mix of the electric power utilities. Accordingly, RFP No. ET-78-R-02-0019, "Electric Utility Planning Studies for OTEC Power Integration," was issued in the summer of 1978. As a result of this attempt, a valuable source of information for the electric utilities and for the advancement of the OTEC commercialization effort in DOE is expected to be developed.

The participation of Puerto Rico in OTEC promoting activities is not new. In 1967, engineers from the Puerto Rico Water Resources Authority (PRWRA) together with Researchers from the College of Engineering of the University of Puerto Rico (UPR) performed an engineering study for the location of an OTEC power plant a few miles to the east of the island. The work was not followed further in those years of plentiful oil at low prices, but it produced component sizes and identified the working fluid for a suitable thermodynamic OTEC cycle. In 1975 we formed an interagency committee to promote participation of Puerto Rico in solar energy development, and OTEC was among the priority items of that committee, which developed a good source of information on our potential sites and industrial capability to support an OTEC project. At about the same time, the National Science Foundation sponsored a site study that was undertaken by the Marine Science Laboratory of the College of Engineering of the UPR at a location a few miles south of Punta Tuna on the southeast coast of the island.

The creation in 1977 (sponsored by DOE) of the Center for Energy and Environmental Research (CEER) of the UPR added much to the ongoing OTEC effort. Since its beginning, the CEER has placed first priority in OTEC research and at present is participating in oceanography, ecology and biofouling studies at Punta Tuna and on laboratory studies of surfactants for the advanced FOAM OTEC concept. An OTEC interagency committee was created in 1977 and re-structured in 1978 by the Office of Energy of the Governor of Puerto Rico. This committee has been instrumental in developing a vigorous strategy and an integrated effort in Puerto Rico to promote advancement of OTEC technology with the objective of making it feasible in the shortest time possible.

The present system planning study is not unique. Florida Power Company will be working on similar or complementary subjects at the same time under another DOE contract addressing the Gulf-based utilities. In our case, the results will be more valuable for the islands.

General Plan

The plan is to conduct a standard generation expansion study emphasizing the results of a comparison of OTEC against the other alternatives, coal, oil, and nuclear power plants. Our baseline system is the integrated network of Puerto Rico (Fig. 1).

Unique aspects of OTEC will be identified using data readily available or to be requested

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from DOE. Such items as the mathematical model for the simulation of the vapor-turbine system and the generator and its excitation system are required for the electric system planning study. Other data required includes power cable cost and reliability, maintenance costs, operational costs, system availability, power consumption by auxiliaries and other aspects that may arise during the study.

Since the licensing of the OTEC facility is new to the existing laws and regulations, recommendations will be made on how to reduce the time and cost requirements for the licensing task. Three sites will be considered to establish the regulatory requirements (Fig.2). One of the sites will be few miles away from Punta Tuna on the southeast corner of the island; the second one will be south of Guayanilla on the south

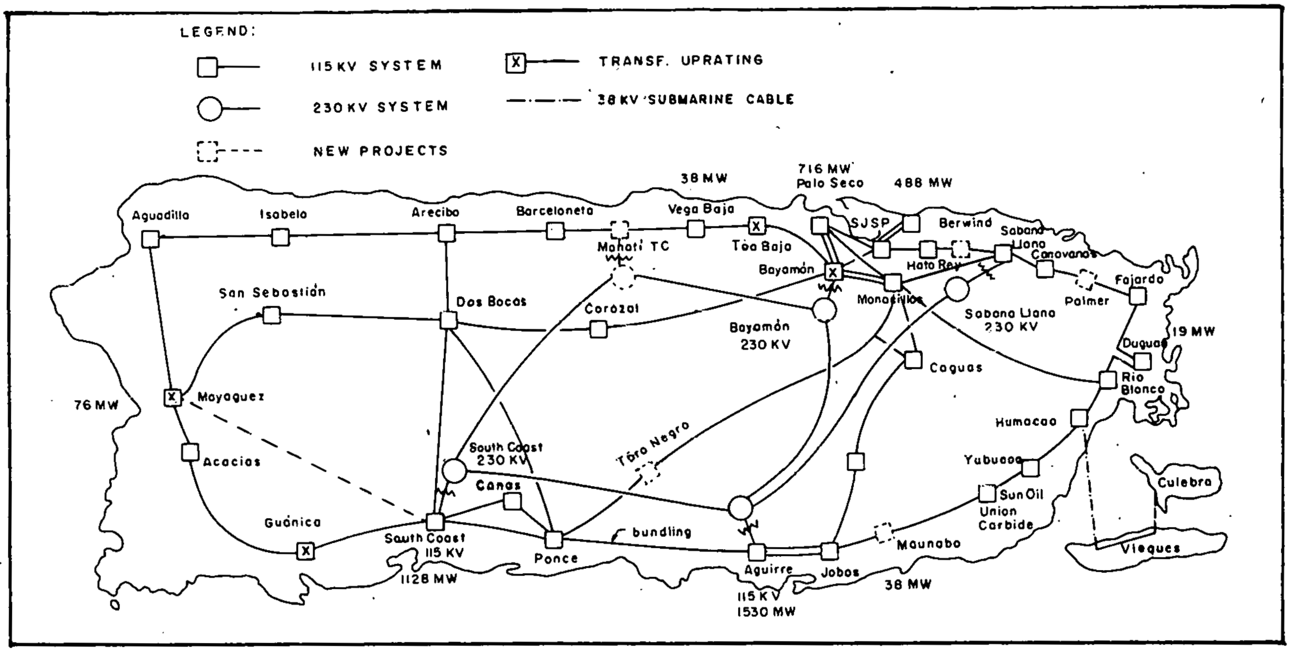

Fig. 1 Electrical System of Puerto Rico

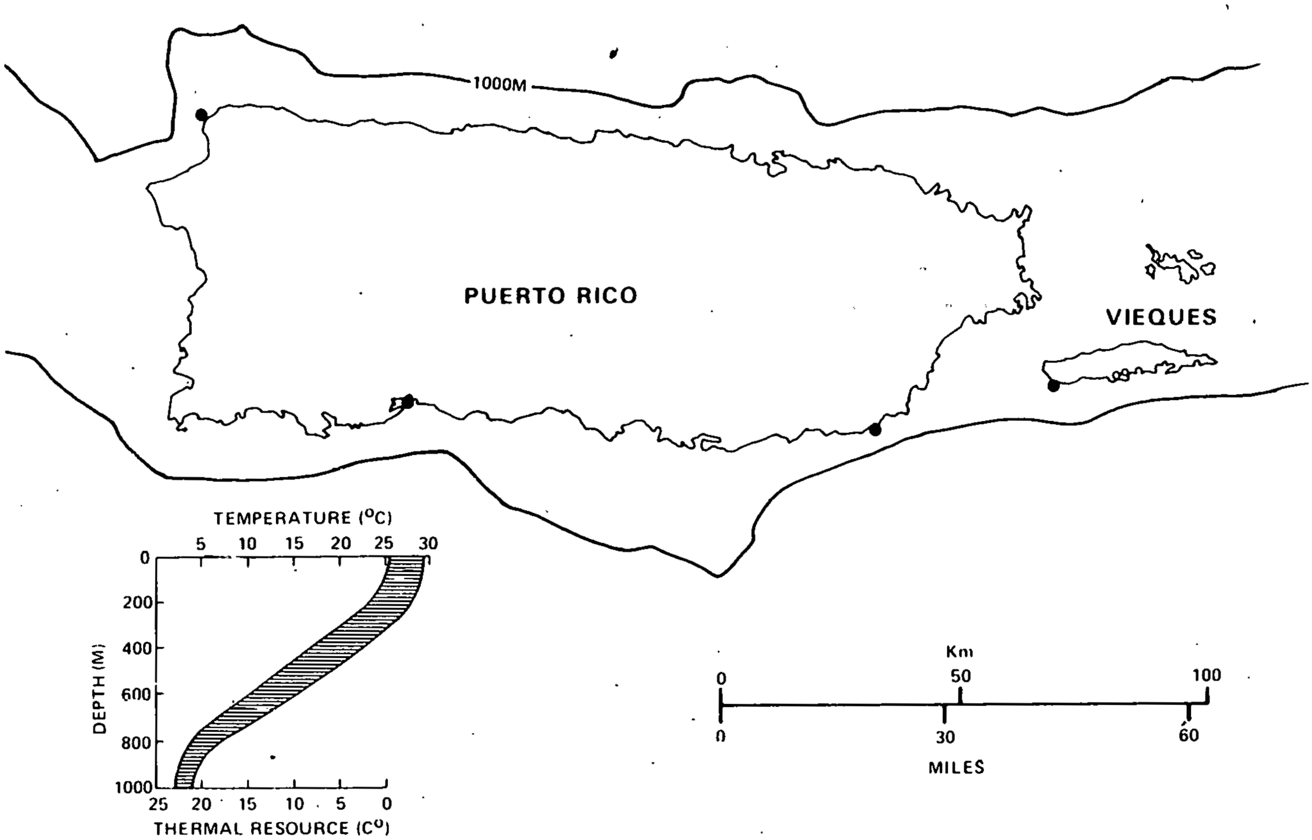

Fig. 2 Potential OTEC Sites

coast; and the third will be Punta Borinquen on the northwest corner. The study will address the impact of the maritime laws on the licensing procedure at the location most likely to be affected.

In performing these tasks, PRWRA will draw upon its internal resources, both manpower and computer programs, as well as the resources of a subcontractor, who will perform the environmental studies related to a platform off Punta Tuna. The Punta Tuna site was selected as the most likely for the construction of an OTEC facility. For Evaluating the economic aspects of generation alternatives, we will use the PROMOD III Production Costing Computer Model from Energy Management Associates, Inc.

The general approach is to develop economic and technical scenarios for OTEC development in the 1985 to 2000 time frame which, will be used to determine programs and incentives that will be recommended to increase the competitive position of OTEC. This approach consists of the following major steps:

- Update the present load demand models.
- Update the generation expansion pattern and power system delivery network assuming conventional power sources options.
- Determine what information is needed to evaluate OTEC for future generation capacity.
- Obtain from DOE or its subcontractors, or otherwise estimate OTEC evaluation parameters to complete an expansion and network study.
- Perform a sensitivity analysis on the OTEC evaluation parameter so that various scenarios for the successful deployment of OTEC in the 1985 to 2000 time period can be described, and recommend programs and incentives which could be implemented to give a stimulus to the successful deployment of OTEC for the electric power generation in the latter years of this century.

Generation System

General Description

This task was designed to evaluate OTEC plants in the overall generation system planning concept of an electric utility. It consists of three major sub-tasks: load forecasting, study of expansion alternatives, and economic evaluation of the alternatives.

Load forecasts. Load forecasts are performed periodically in PRWRA. They are performed by the Forecasting and Rate Studies Department applying econometric techniques to the econometric indicators of the island. These indicators are provided by the Puerto Rico Planning Board. Table 1 shows the results of a recent load forecast study. The average expected growth in peak demand is about 3.6% annually. This table provides the initial set of information for the study of expansion alternatives.

TABLE 1. PRWRA LOAD FORECAST

Fiscal Year	Generation (kwh in millions)	Year Peak (MW)	% Increase
1978-79	13,725.9	2,057.6	1.99
1979-80	14,239.6	2,133.7	3.70
1980-81	14,792.1	2,216.9	3.90
1981-82	15,310.9	2,294.5	3.50
1982-83	15,887.2	2,381.7	3.80
1983-84	16,538.6	2,479.3	4.10
1984-85	17,200.1	2,578.5	4.00
1985-86	17,853.7	2,676.5	3.80
1986-87	18,514.3	2,775.5	3.70
1987-88	19,162.3	2,872.7	3.50
1988-89	19,813.8	2,970.4	3.40
1989-90	20,467.7	3,068.4	3.30
1990-91	21,122.6	3,166.6	3.20
1991-92	21,777.4	3,264.7	3.10
1992-93	22,430.8	3,362.7	3.00

Study of expansion alternatives. Traditionally, PRWRA had relied on strict reliability criteria for determining the generating capacity needs. Since that criteria seemed to overbuild the system, it was decided to complement the reliability analysis with an analysis of the reserve requirements. The generating capacity on reserve is required to:

1. Permit maintenance to the generating units
2. Account for the forced outage of units
3. Account for the partial outages (units on limitation)
4. Provide an additional reserve for absorbing variations in any of the above items

Therefore, to determine the need for new generating capacity using this approach, all that has to be done is to add the reserve requirements to the peak demand and compare this figure with the installed generating capacity. Whenever the installed capacity is less than the peak demand plus reserve, new capacity must be provided. On Fig. 3 we present the generation capacity requirements to year 2000. Notice that a total of 1950 MW of generating capacity will have to be added. This additional capacity might be supplied with OTEC plants. Figure 4 presents the projected capacity additions to year 2000.

Using this required reserve methodology in conjunction with a reliability analysis, various generation schemes are developed and are evaluated in terms of production costs.

Economic evaluation of alternatives. For the production costs evaluation we will be using the PROMOD III production costing computer program of Energy Management Associates, Inc. PROMOD III is a widely accepted computer program for reliability analyses, generation planning and fuel budgeting. Some features of the program are:

- Reliability only option.

- Reliability equalization model which determines the capacity required (either as a purchase, sale, or deferral) to achieve an annual reliability objective.
- Generalized startup priority and unit minimum shutdown time model.
- Probabilistic multi-area model which recognizes transmission limitations and load diversity between areas.
- Hourly chronological reliability, marginal cost, and average cost model to perform time-of-day rate design studies.
- Explicit representation of energy limited resources such as hydro, solar, nuclear, and OTEC generation.
- Expected surplus energy and cost for each generating unit by subperiod.
- Load management extensions.

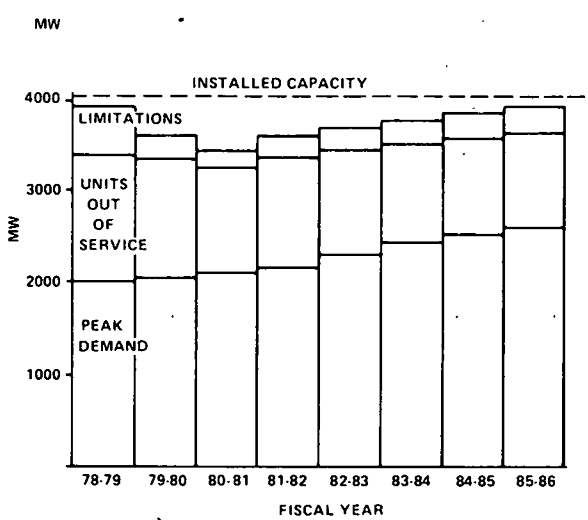

Fig. 3 PRWRA's Generation Capacity Requirements

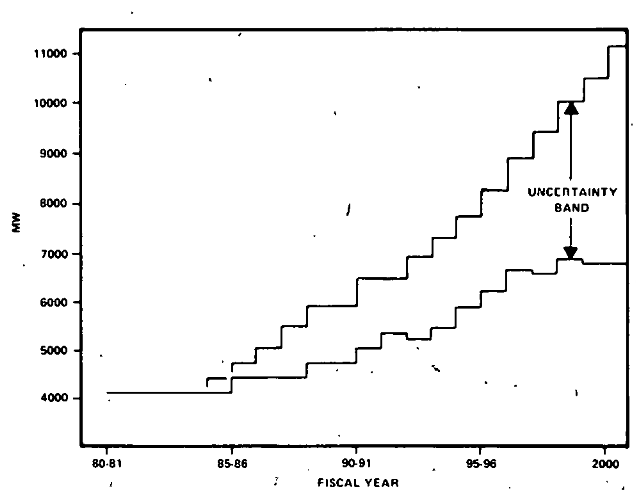

Fig. 4 Expected System Capacity to Year 2000

Electrical System

General Description

This task was designed to evaluate the effect of OTEC plants on the performance of the electrical system and to compare this case with more conventional generation plants like coal, oil, and nuclear. It consists of three subtasks: economic evaluation of cable alternatives, load flow analyses, and short circuit and stability studies. These studies require that the electrical system be divided by areas as shown in Fig. 5. Most of the OTEC plant parameters required for this task will be obtained from DOE and the dynamic simulation models being developed at the Massachusetts Institute of Technology.

Economic evaluation of cable alternatives. Preliminary designs of interconnections of the OTEC plant will be required at least at three potential sites. From these designs, estimates of alternative cable voltages and capacities are prepared. These include cable configurations to provide redundancy and desired reliability, and a sensitivity analysis to study assumed outage rate and repair time. Similar analyses will be performed for the conventional power plants. Finally, the OTEC case will be compared with these conventional cases.

Load flow analyses. Load flow analyses will provide the data required to evaluate the impact of the new plants on the projected system up to year 2000. Three base years will be studied according to the established plan. Necessary system improvements, preferred voltage levels for the interconnection of the new units, and economic impact will be determined. Once more, the plan was prepared to compare OTEC with other generation alternatives.

Short circuit and stability studies. These studies are performed to determine the behavior and characteristics of the modified electrical system (with the addition of the OTEC plant) under particular short circuit conditions. Two basic disturbances are analyzed: a short circuit or fault in a critical element of the system, and the trip-out of a generating unit or block. The short circuit causes angular displacement between the rotors (Fig. 6) of the generators and eventual voltage and power flow instability. At present, instability is not a problem in the electrical system of Puerto Rico because the critical clearing time of a 3-phase fault in the 230 KV transmission system is 0.20 seconds. We do not foresee particular problems with the addition of an OTEC plant, but the study will be carried over for all the alternatives in order to find out the differences among them.

The plan calls for a close look into the problem of generator trip-outs. This is one of the major problems of our electrical system at present because the latest additions are too large (450 MW units) in comparison with the peak demand (2000 MW peak demand). Large amounts of load shedding are thus required on each trip-out. This in turn requires large amounts of reserve generation from peaking units that have the highest generation costs. If technical reasons dictate modular construction for OTEC plants, then the probability of total plant trip-out will

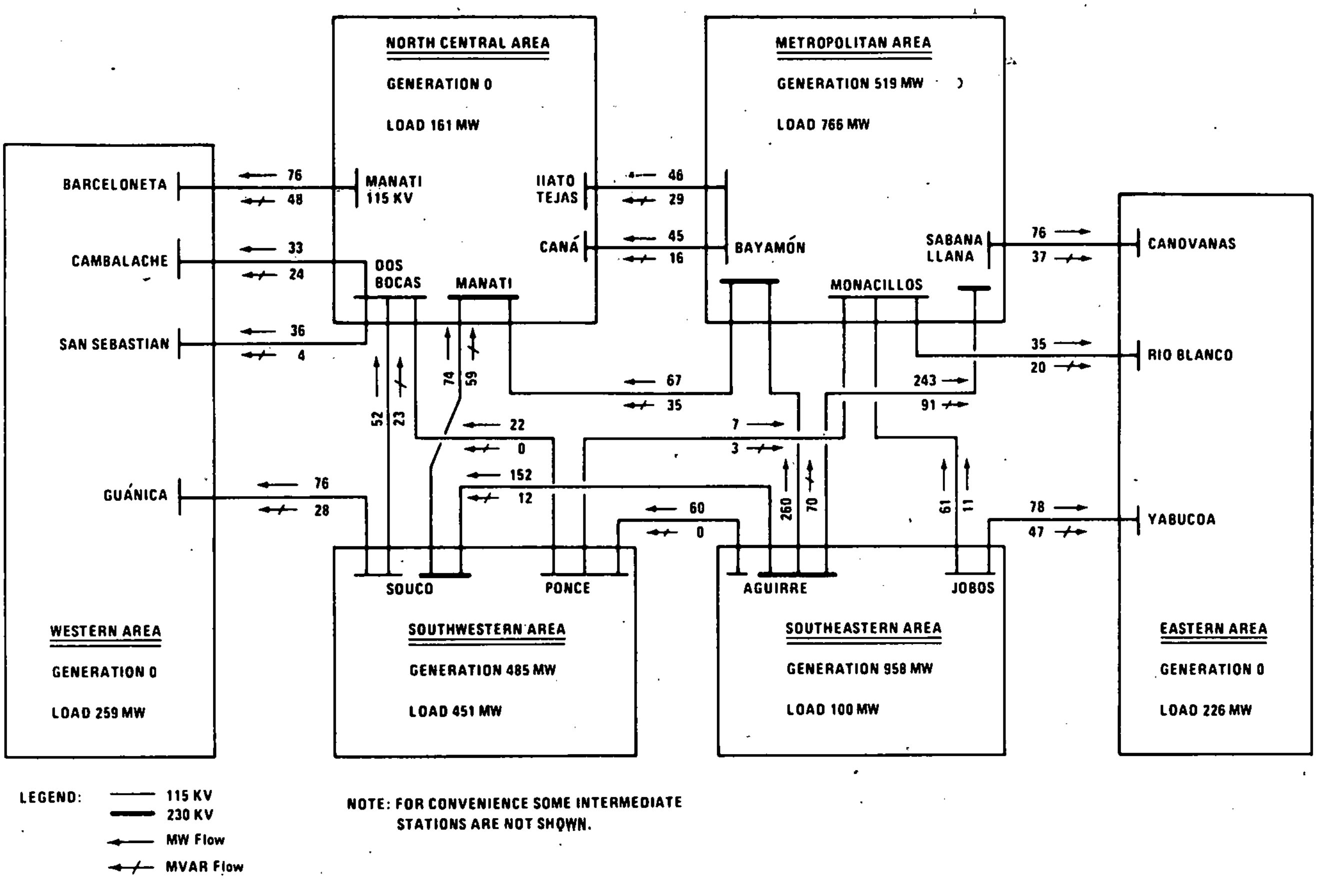

Fig. 5 Main Generation-Load Areas and Transmission Links - PRWRA System

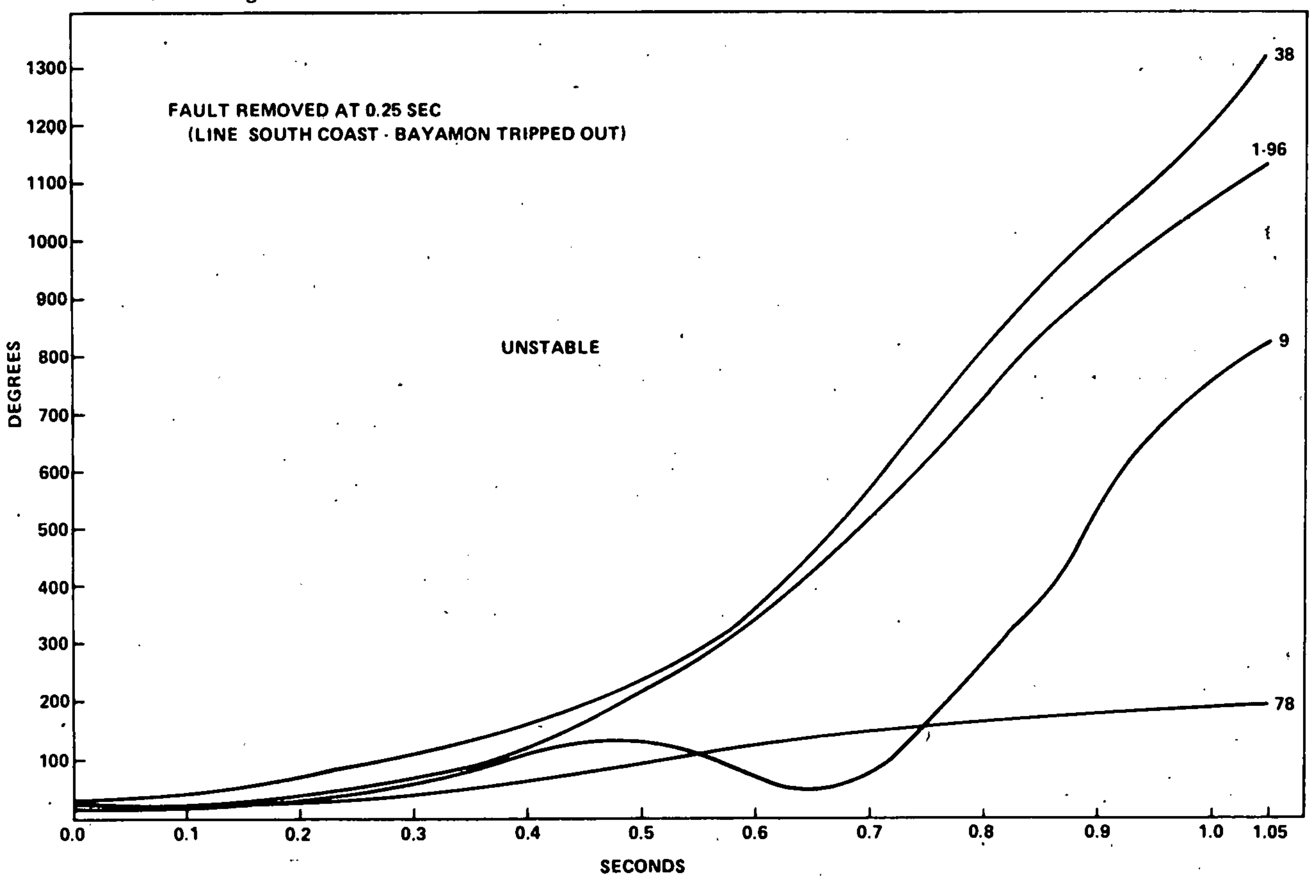

Fig. 6 Angular Displacement of the Generators Under Fault Condition (Simplified Example)

be small and load shedding and reserve capacity requirements will be greatly reduced.

Another use of short circuit analysis will be to determine the interrupting duty of circuit breakers near the OTEC location under short circuit of the plant's major electrical components. Checks will be made of the momentary duty of breakers while considering short circuits at the OTEC pump motors and also of the transient overvoltages on the submarine cable resulting from line-to-ground faults and switching surges.

Other phenomena like the cold start-up performance of the big OTEC plant motors will be studied. Given the electrical characteristics of plant transformers and motors as well as speed-torque curves, the starting current, voltage drop at motor terminals, and accelerating torque will be analyzed.

Licensing

The study of licensing issues from the standpoint of the electric utility will require simulation of an actual case of license application for an OTEC facility. Applicable regulations will be overviewed and inquiries to the regulatory agencies will be performed.

We have estimated that no less than nine (9) Federal and local agencies will have to be contacted to bring some light into these issues. These agencies are:

- Army Corps of Engineers
- Environmental Protection Agency
- Coast Guard
- Federal Aviation Administration
- Puerto Rico Environmental Quality Board
- Puerto Rico Department of Natural Resources
- Puerto Rico Planning Board
- U.S. Maritime Administration
- U.S. Navy

One important question to answer will be which one of these agencies will be willing and capable of assuming the role of lead agency. This will be the agency that will issue the environmental impact statement based on the responses of all concerned parties to our environmental report.

Some of the documents that we foresee that will be needed for an OTEC plant located within the three (3) mile limit include:

- Permit application for the installation and operation of facilities in navigable waters.

- National Pollutant Discharge Elimination System Permit for discharge of pollutants into navigable waters.

- Application for private aids to navigation.

- FAA application for a determination that certain structures will not be a hazard to air navigation.

- Environmental Report

To this list we will have to add the familiar list of documents that are normally required for ground installations (for the ground support facilities).

The final product of this task will include among other things a representative critical path matrix of the licensing events, a chart of the organizational structure recommended to undertake the licensing effort, an estimate of time and costs involved, and recommendations to reduce these time and costs.

Environmental

General Description

Studies and visual observations indicate that the presence of offshore floating or anchored structures tend to attract fish and other marine life. The congregation of marine life around an artificial structure serves many needs, such as protection and availability of food. Not only is marine life attracted to artificial structures, but this attraction is enhanced if there exists a light source at night. An offshore, moored OTEC platform will act, in the pelagic waters where it is found, as an artificial structure which can attract and cause to congregate various pelagic organisms. These organisms are normally found in low concentrations in pelagic waters, but an artificial structure is expected to enhance their concentration.

Information is not readily available as to the types and numbers of organisms that would congregate around an artificial structure located in very deep water such as an OTEC platform in the Caribbean. It is this question that this project attempts to answer for the benchmark OTEC site off Punta Tuna, Puerto Rico.

In Puerto Rico, most of the commercial fishing is done in shallow near-shore, shelf area because of the higher concentration of fish and the lower cost of harvest. The island shelf is limited, so there is only a small fishing area around the coast. The constant harvest of these fisheries leads to a depletion of the available fish and shellfish in these waters. The installation of an OTEC plant in the deep oceanic areas of Puerto Rico could be a great attracting force for the pelagic organisms. (This, in turn, could expand the "fishing grounds" available to the commercial fisherman).

Scope and Objectives

The scope of this program is to evaluate the changes in the oceanic pelagic population caused by the installation of an offshore OTEC structure situated southeast of Punta Tuna, Puerto Rico. The final product will be a report summarizing the data and recommending adequate actions to follow. The objectives are:

- Collect all available literature pertinent to the concentration of pelagic organisms around an artificial offshore structure in the Caribbean Sea South of Puerto Rico.

- Evaluate the change in the pelagic organism concentration and species composition with time at the Punta Tuna OTEC site during the presence of a mooring buoy alone, then a biofouling research vessel, and compare these results with a nearby control area.

Relation with Other Projects

This project relates very closely with the ongoing oceanographic program, "Measurements of Oceanic Variability Relatable to an "OTEC" installation at Punta Tuna, Puerto Rico", in that much of the hydrographic information needed in this proposed program shall be collected during the bimonthly cruises of the Oceanic Variability Program. Also, the Marine Ecology Division of CEER is to become involved in biological and ecological evaluation of the entire Punta Tuna area, beginning October 1979. Data from both these related projects shall supplement this study.

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FINANCING UNDER A TAX-EXEMPT SITUATION KEY TO OTEC COMMERCIALIZATION?*

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Abstract

Among all potential sites within U. S. flag jurisdiction, Puerto Rico poses very interesting characteristics in terms of providing tax incentives for OTEC commercialization. The island is not a state, thus not subject to federal tax. This special situation could work to the advantage of OTEC if exploited adequately. ~~This paper explores~~ site-specific incentive scenarios that could be constructed for investments in OTEC installations are explored. An analysis of required organizational structures to take advantage of both local as well as federal tax incentives is presented. The scenarios depicted include commonwealth - private - federal co-ownership possibilities.

Introduction

Past research related to OTEC Commercialization Issues, particularly in the areas of tax incentives and financing have explored a number of general alternate schemes involving private - federal possibilities. Very little detailed work has been done on site-specific issues which could improve investment possibilities.

Among all the potential sites within the U. S. flag jurisdiction, Puerto Rico, due to the legal relation to the U. S. as an unincorporated territory (Commonwealth), could provide very promising short-run, as well as long run attaction to potential investors. This special situation may be to the advantage of the initial commercialization decisions.

This paper first presents a background of the legal, tax, and jurisdiction issues that affect (both positive and negative) investment possibilities for the Puerto Rico Site.

Research is still underway as to possible required organizational structures that could maximize ROI given the present site-specific tax characteristics. It is expected that the results of our ongoing cash-flow/financial simulations will be able to prove that P. R. is the site that could provide the highest ROI for private investors in the initial OTEC commercial plants. This paper, thus, only explores the general scenarios, and points out the most promising combinations, based on the background presented.

The overriding assumption which sets the stage for the site-specific analysis in that any initial commercialization must provide two types of risk deflating incentives; a short-term, capitalization incentive (such as the Investment Tax Credit) and long term operating incentives (such as tax-free income). The scenarios under consideration also assume that in order to speed up commercialization very limited changes to present legal tax laws should be contemplated. Our preliminary results point to the fact that at least in the tax area this second assumption could prove valid. It also seems fairly clear at this point in our work that organizational structures, as well as incentive requirements will vary from initial commercialization stages, as investment uncertainties (and associated risks) are reduced.

The Legal Framework

A very brief description and history of the legal framework on which the relationship between the U. S. and Puerto Rico has been based is appropiate at this time, as a matter of background, to set the stage for the discussion on possible OTEC commercialization scenarios.

The Treaty of Paris signed in December 10, 1898 established the presence of the U. S. in Puerto Rico, when Spain ceded the territory. [1] The civil government was established by the 1900 Organic Act of Puerto Rico, known as the "Foraker Act. [2]

The Foraker Act created a local government with power to legislate on all civil and criminal matters under a governor appointed by the President of the United States. As this government was created by the United States Congress, who had power to amend or repeal the Act, Puerto Rico became a territory of the United States not organized[3] and not incorpored, [4] and as such was subject to the Federal powers exercised pursuant to the territorial clause of the Constitution of the

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Section 8 of the Foraker Act provided that: "The laws and ordinances of Puerto Rico now in force shall continue in full force and effect, except as altered, amended, or modified hereinafter, or as altered or modified by military orders and decrees in force when this Act shall take effect, and so far as the same are not inconsistent or in conflict with the statutory laws of the United States not locally inapplicable, or the provisions hereof, until altered, amended, or repealed by the legislative authority hereinafter provided for Puerto Rico or by Act of Congress of the United States." At that date was applicable to Puerto Rico the Spanish law of Ports of May 7, 1880, which had been extended to the island by Royal Order of February 5, 1868.[5]

The Spanish Law of Mines was also applicable to Puerto Rico, and very recently its applicability was reaffirmed by the United States Court of Appeals for the First Circuit.[6]

The Jones Act was enacted by the United States Congress on March 2, 1917.[7] The Jones Act granted United States citizenship to the people of Puerto Rico.[8] It also granted additional powers to the insular government. It created the Legislature of Puerto Rico although the Congress of the United States reserved the power and authority to annul any law.[9] The Jones Act granted Puerto Rico full power of local self determination, with an autonomy similar of the States and incorporated, territories, as extensive as those exercised by the state legislature.[10]

The political relation of Puerto Rico with the United States was restructured by the Federal Relations Act which made possible the creation of the Commonwealth of Puerto Rico.[11]

This Act fully recognized the principle of govenment by consent for the Commonwealth of Puerto Rico that was proclaimed on July 25, 1952.

From that date on, Puerto Rico became the first autonomous constitutional Commonwealth voluntarily associated with the United States.

Status and Legal Regime in Puerto Rico

Status of Commonwealth. Before the enactment of Public Law 600 the political status of Puerto Rico was unmistakably a colony of the United States. By the ratification of Treaty of Paris, Puerto Rico became a territory of the United States although not an organized territory in the technical sense of the word.[12] As such it was treated as an unincorporated territory and governed by the territorial clause in the Constitution of the United States.

"The Congress shall have power to dispose of and make all needful rules and regulations respecting the territory or other property belonging to the United States; and nothing in this Constitution shall be so construed as to prejudice any claims of the United States, or of any particular State."[13]

Under this clause the Congress had full power to make all necessary rules and regulations and authority to legislate for Puerto Rico beyond the limits fixed by the "Constitution" with respect to the States. After Public Law 600, the Puerto Rico Federal Relations Act, and the proclamation of the Constitution of the Commonwealth of Puerto Rico, the political status of Puerto Rico changed and became a Commonwealth. As such the new status is unique because it became the first self governing territory voluntarily associated to the United States by means of a compact, which as you may know, is an agreement comprising conventions between nations of sovereign states.[14] This implies that it is an obligation not unilaterally revocable by the convening parties. The Constitution of the Commonwealth of Puerto Rico organized the government of the Commonwealth of Puerto Rico in a republican form and established its legislative, judicial and executive branches with political authority extending to the Island of Puerto Rico and to the adjacent islands within its jurisdiction.[15]

The examination of the Federal Relations Act reveals the structure of the political and legal relations of Puerto Rico and the United States.

"The rights, privileges and Inmunities of citizens of the United States shall be respected, in Puerto Rico..."[16]

"...No export duties shall be levied or collected on exports from Puerto Rico, but taxes and assesments on property, income taxes, internal revenue and license fees, and royalties for franchises, privileges, and concessions may be imposed for the purposes of the insular and municipal governments, respectively, as may be provided and defined by the Legislature of Puerto Rico...."[17]

"...All citizens of Puerto Rico...and all natives of Puerto Rico who were temporarily absent from that island...are hereby declared and shall be deemed and held to be, citizens of the United States...."[18]

All property acquired in Puerto Rico by the United States under the cession in Spain the Treaty of Paris was placed under the control of the government of Puerto Rico, to be administered for the benefits of the people.[19] The harbor areas and navigable streams and bodies of water and submerged land around the island of Puerto Rico not reserved by the United States for public purposes was placed under the control of the government of

Puerto Rico.[20]

The statutory laws of the United States not locally inapplicable shall be enforceable in Puerto Rico as in the United States, except the internal revenue laws.[21] The Interstate Commerce Act does not apply to Puerto Rico.[22]

Constitution of the United States and Constitution of Puerto Rico. The applicability of the Constitution of the United States and of the Constitution of the Commonwealth of Puerto Rico to benefit a citizen is dependent upon the residence of the citizen As soon as Puerto Rico became a territory of the United States in 1898, the issue of what constitutional restrictions the United States Congress had in the treatment of its territories arose.[23] In 1901 the United States Supreme Court concluded that the Treaty of Paris did not provide for the incorporation of Puerto Rico to the United States territory but left Congress to decide Puerto Rico's status and as Congress had not acted to that date, Puerto Rico was an unincorporated territory. The United States Constitution applies fully to the incorporated territory but only the fundamental provisions of the Constitution apply to an unincorporated territory such as "the general prohibition....in favor of the liberty and property of the citizen which are an absolute denial of authority....to particular acts".[24]

Applicability of Federal Laws. The question of the applicability of Federal Statutes in Puerto Rico is a matter that brings about some uncertainty in some areas because of the drafting of Section 9 of the Federal Relations Act which states:

> "That the statutory laws of the United States not locally inapplicable, except as hereinbefore or hereinafter otherwise provided, shall have the same force and effect in Puerto Rico as in the United States, except the Internal-revenue laws: Provided, however, that hereafter all taxes collected under the internal-revenue laws of the United States on articles produced in Puerto Rico and transported to the United States or consumed in the Island shall be covered into the Treasury of Puerto Rico".

This clause is identical to Section 9 of the Jones Act (1917)[25] and almost the same as Section 14 of the Foraker Act, (1900).[26]

The phrase "not locally inapplicable" has provoked many uncertainties as to the applicability of many Federal laws. The phrase was most important in the Foraker and Jones Act but it became relatively unimportant as the Federal Relation Act was enacted. It has been interpreted that Section 9 exempts Puerto Rico from the application of Federal laws unless the United States Congress makes them expressly applicable to Puerto Rico.[27]

Even when a statute does not state clearly that it is applicable to Puerto Rico it has been held that the congressional intent and the character and the aim of the Act is the ratio "decidendi".

Federal taxes in Puerto Rico. Section 9 of the Federal Relations Act, quoted above, clearly provides that the internal revenue laws do not apply to Puerto Rico. For federal tax purposes, corporations which are organized under the laws of Puerto Rico are foreign corporations.

Section 9 applies to residents of Puerto Rico and corporations registered and operating under the provisions of the Law of Corporations of the Commonwealth of Puerto Rico, who shall pay taxes to the government of Puerto Rico unless exempt.[28]

If these induviduals or corporations do any business within the United States they will be subject to pay tax in the United States as foreign corporations at a rate of 30%,[29] and for such payments they will have a tax credit in Puerto Rico.

Doing Business in Puerto Rico

There are at least three basic ways by means of which business operations can be done in Puerto Rico resulting in tax savings:

1. The Western Hemisphere Trade Corporation
2. Puerto Rico Corporation
3. United States possession corporation

(1) A "Western Hemisphere Trade Corporation" means a domestic corporation all of whose business (other than incidental purchases) is done in any country in North, Central or South America or in the West Indies,....of which 95% or more of its gross income for three years is derived from sources without the United States and 90% or more of its gross income for the same period was derived from the active conduct trade or business.[30]

Before 1976 these corporations were allowed a deduction in computing taxable income amounting to a maximum of 14% less than the ordinary corporation rate.[31] The Tax Reform Act of 1976 provided for the phase out of the 14% differential, reducing it to 11% for 1976, 8% for 1977, 5% in 1979 and no deduction at all after 1980.

(2) A corporation organized under the laws of the Commonwealth of Puerto Rico that does not engage or do business within the United States does not pay taxes to the United States but if "engaged in trade or business within the United States",[32] it gross income is taxed at a rate of 30% as a foreign corporation connected with United States business.[33] Such corporation will be subject to pay taxes in Puerto Rico unless exempt.

(3) A United States possession corporation, hereinafter named a 936 corporation, is a United States domestic corporation which elects the application of Section 936 of Internal Revenue Code which "ab initio" provides:

"Allowance of Credit

(1) In general. Except as provided in paragraph (2), in the case of a domestic corporation which elects the application of this section, there shall be allowed as a credit against the tax imposed by this chapter an amount equal to the portion of the tax which is attributable to taxable income, from sources without the United States, from the active conduct of a trade or business within a possession of the United States, and from qualified possession source investment income, if the conditions of both subparagraph (A) and subparagraph (B) are satisfied:

(A) 3-year Period. If 80% or more of the gross income of such domestic corporation for the 3-year period immediately preceding the close of the taxable year (or for such part of such period immediately preceding the close of such taxable year as may be applicable) was derived from sources within a possession of the United States (determined without regard to Section 904(f); and

(B) Trade or business. If 50% or more of the gross income of such domestic corporation for such period or such part thereof was derived from the active conduct of a trade or business within a possession of the United States."[34]

This section permits a 100% tax exemption to a domestic corporation if all of its gross income is derived as provided in the section. For the purposes of this section:

"The term possession of the United States includes the Commonwealth of Puerto Rico but does not include Virgin Islands of the United States."

A 936 corporation will pay Federal Taxes for the dividends repatriated before 10 years have elapsed.

Corporations under Section 936 are not allowed credits against Section 56 minimum taxes, Section 531 accumulated tax on earning section 541, holding company tax nor section 1351 on recoveries of foreign expropiation laws. DISC (Domestic International Sales Corporation) or former DISC corporations as defined under Internal Revenue Code, Section 992 (a) are ineligible for the credit provided by Section 936.

Investment Tax Credit. The Internal Revenue Code allows an investment tax credit in certain depreciable properties. Although the investment tax credit does not apply to property which is used predominantly outside of the United States, an OTEC operation in Puerto Rico may qualify under some of the exemptions. The investment tax credit allowed by the Internal Revenue Code, Section 38 is applicable to: "Any property (other than a vessel or an aircraft) of a United States person which is used in international or territorial waters within the northern portion of the Wester Hemisphere for the purpose of exploring for, developing, removing, or transporting resource from ocean waters or deposits under such waters.

For purposes of Clause (X), the term "northern portion of the Western Hemisphere" means the area lying west of the 30th. meridian west of Greenwich, east of the International dateline, and north of the Equator, but not including any foreign country which is a country of South America.

The amount of the credit allowed by Section 38 is the sum of the regular percentage, the energy percentage, and the ESOP percentage adjusted for the credit carry over and carry backs.

The regular percentage is 10% and the energy percentage is also 10% with respect to the period beginning on October 1st. 1978 and ending on December 31st. 1982. The ESOP percentage is 1% with respect to the period beginning January 21, 1975, and ending on December 31, 1983 and an additional percentage not in excess of 1/2 of 1% with respect to the period beginning on January 1, 1977, and ending on December 31, 1983. As an OTEC operation removes the heat resource from the ocean water, it can be assumed that it is a qualified operation under IRC, Sec. 38 (2) (B) (x), and as such enjoys an investment tax credit slightly over 20% until Dec. 31, 1982 and of 10% after that date.

Puerto Rico Industrial Incentives Act of 1978

The 1946 Puerto Rico Industrial Incentives Act was amended by the 1978 Industrial Incentives Act of Puerto Rico, Law #26 of June 2, 1978.

The Act provides for the granting of partial tax exemption to qualifying manufacturing and service industries doing business in Puerto Rico. The exemption can be as high as 90% on the first five years and can extend up to twenty five years, depending upon the location within Puerto Rico of the exempted business. To enjoy its benefits an application has to be filed in the Tax Exemption Office, created by the Act, which is attached to the Office of the Governor of Puerto Rico.

Although OTEC is not specifically qualified by the Act, it can be understood to be included in the wide scope of the definition of a manufacture product. Considering that OTEC would contribute significantly to the relief of Puerto Rico's energy crisis and thereby to the industrial development as a whole, it can be expected, although discretionally of the Governor of Puerto Rico, that tax exemption would be granted to OTEC in Puerto Rico. The Act provides that a manufacturing exempted business will enjoy partial tax exemption covering its net income derived from operations in exempted activities, hereinafter refer as to Industrial Development Income, in the following percentages:

Years	Percentage
1 through 5	90%
6 through 10	75%
11 through 20	55%
21 through 25	50%

Additional benefits are provided by the Act as follows:

1. When the Industrial Development Income for any given year is less than $500, 000, the first $100, 000 can be excluded in computing the portion of the Industrial Development Income which is not exempted from taxation;

2. The Manufacturing Exempted Business can deduct from its Industrial Development Income 5% of its production payroll, up to an amount not in excess of 50% of its Industrial Development Income. For this purpose the production payroll will be considered to include salaries paid to personnel directly involved in the exempted activity.

This benefit is available to all Manufacturing Exempted Businesses except those that elect to choose the $100, 000 exclusion aforementioned.

In case of a group of corporations or partnerships exempted businesses which are controlled by 50% or more of the equity, the $100, 000 exclusion is applicable as a totality within the component group of controlled businesses or said group has the option to prorate the exclusion.

The Act also provides for the exemption of Commonwealth and municipal taxes on all properties utilized by the exemption business in the activity, hereinafter refer as to Industrial Development Property.

The tax exemption on the Industrial Development Property is the same percentagewise and in time as the exemption granted to the Industrial Development Income. The exempted business can exclude from the taxable value of the property, on a yearly basis, the first $100, 000 of the assesed value of its Industrial Development Property, and such exclusion is applicable within a component group of controlled businesses either as a totality or prorated, at the option of the group. Exempted businesses are totally exempt from license fees. excise or other municipal taxes for the duration of the exemption. The Act exempts from all property taxes the intangible personal properties of the nature of patent, production licence or trademark or tradename utilized in the activities exempted.

Distribution of Dividends

The distribution of dividends or profits by a orporation or partnership that is an exempted business, or by a corporation or partnership that is a member of an exempted business, shall be exempt from income tax in the same proportion, total or partial, in which the industrial development income is exempt from taxes in accordance with the Act if made from the industrial development income accumulated after the payment of the tax, if any, imposed by this Act of the Income Tax Act of 1954. as amended, but only if all of the following requirements are complied with:

1. Only an amount not to exceed 50% of the after tax net income for the year of distribution is paid.

2. Commencing not later than ninety (90) days after the date of filing of the corresponding income tax return for each taxable year, the exempted business shall place, invest and maintain, for a fixed term of not less than five (5) years, not less than fifty percent (50%) of its net industrial development income for each year after the payment of the taxes provided by law, in a qualified activity covered by the provisions of the Act or in the payment of the principal balance of any debt incurred by the exempted business for the acquisition of property devoted to industrial development and/or the acquisition of property to be devoted to industrial development. At the termination of the period of investment of such accumulated earnings as provided in this clause, the exempted business may continue to accumulate the same in accordance with law or distribute them at later dates subject to the credit provided by this subsection.

3. The remaining fifty percent (50% of the net income for each year after the payment of the other taxes provided by law) may be distrubuted subject to the credit provided in this subsection during the succeeding years, in annual amounts not greater than ten percent (10% of the total of such income for each of the succeeding years). The amounts that are available annually may be accumulated in accordance with law and distributed at subsequent dates.

The exempt industry has three (3) choices in dealing with repatriation of accumulated earnings. The first choice is to withdraw their earnings from Puerto Rico at any time and pay a tollgate tax of 10%. The second choice is to take 50% of any given year's earnings and invest them in the firm's own additional plants and equipment or repayments of principals of the firm's plants or equipment debt and/or in local designated investments, such as Puerto Rico Bonds, bank savings certificates and participation in constructions loans for a period of five years. They can repatriate the remaining 50% of the total profits at 10% per year for the next five years with a reduced tollgate charge of only 5%. Begining on the sixth year they can also withdraw the locally invested 50% earnings at a reduced tollgate charge of 5%. The third alternative is to liquidate their operations at the end of the tax exempt period and repatriate any part of their accumulated earnings that was invested in accordance with the second alternative above, at a tollgate of 4%.

Flexible Depreciation. The Act also provides that property acquired and in use by the tax payer that is operating under the benefits of the Industrial Incentives Act can not enjoy the benefits of flexible depreciation as long at it operates under the industrial incentive program.

The flexible depreciation provisions allows the tax payer to depreciate a qualifying property, in any year, totally, partially or not depreciated and provided that the depreciation deduction shall not exceed 50% of the net benefit determing without said deduction of the business or commercial activity in which the depreciated property is used. The Act also provides that once the benefits of the Industrial Incentive Act are terminated the business can apply in full the flexible depreciation provisions.

Taxing and the Issues of Coastal Waters Jurisdiction

At the present the United States recognizes 3 miles of territorial waters and maintains the policy that the breath of territorial sea is to be that claimed by the coastal states up to a maximum of three miles. Beyond the seaward limit of the territorial sea a special contiguous zone is recognized by international law.

Within the internal waters the states have exclusive and absolute jurisdiction with respect to all activities, the same in the upland. This is the location where an OTEC instalation is subject to minimal legal issues. Within the territorial sea the coastal states have an almost absolute jurisdiction and have full control of all activities. Within the territorial sea the control of the coastal state is almost the same within its territory.

Although no state can subject any part of the high sea to its sovereignity, any state may regulate such activities there if the installation has been registered, licensed or otherwise authorized by the States, or by the applicability of laws and regulations which are applicable to the citizens where ever they may be.

In 1953 the Congress enacted the Submerged Land Act which is applicable to all lands from the line of mean high tide and seaward to a line three geographical miles distant from the coast line of each state. This Act granted ". . . title and ownership of the land beneath navigable waters within the boundaries of the respective states and to the natural resources within such lands and waters, and the right and power to manage, administer, lease, develop and use the said lands and natural resources all in accordance with the applicable state law ..."

Although the Act intended to stop the continuing litigation, it provided that "nothing in this section is to be construed as questioning or any in manner prejudicing the existence of any state seaward boundary beyong three geographical miles if it was so provided by its constitution or laws prior to or at the time such state became member of the union, or if it has been heretofore approved by Congress."

The present position of the Government of Puerto Rico is that the Island had a previously adquired right (under Spanish rule) of three leagues (that is, 10.35 miles) and as such has the title to and right to exploit resources within these limits.

The position of the U.S. Federal Government is that, although P. R. was not specifically granted even a three mile jurisdiction, de-facto it has a right of domain to only its zone, but has no title to the resources.

For purposes of our financial analysis, it is assumed that the three mile limit theory will prevail. But in no way does this assumption imply that the authors feel that such a limit is or is not correct. Such decision is strictly up to the U.S. Courts or to Congress to define.

Potential Scenario

It is evident that an OTEC off-shore installation in Puerto Rico could take advantage of Section 936 of the IRS code as well as the local Industrial Incentives Act. But some questions remain unanswered:

1. Could U.S. investors also use the Investment Tax Credit?

2. If the plant is within the present "de-facto" territorial waters of Puerto Rico (3 mile limit) are there different tax considerations vis a vis being outside the limit.

3. Is lease-leaverage advantageous at this site?

4. Could the local utility undertake the investment in a commercial plant by itself?

5. How is ROI affected by using (or not using) combined tax incentives?

6. Finally, could the use of only a tax-incentive strategy assure initial commercialization.

Let us attempt to answer each question as best we can, with the limited analysis done to date.

The ITC Issue

At present, it seems that the only way an OTEC investment in Puerto Rico could avail itself of the Investment Tax Credit incentive is if a number of conditions are satisfied. The first, that the credit be, obviously, used against federal taxed income. Second, that the investment becomes qualified under Section X (38) as resource recovery investment. Third, that it forgoes the possibility of MARAD financing.

The ITC incentive could be a very critical short term attraction due to the high initial capitalization requirements of an OTEC plant. A trade-off analysis is required in the case of a plant-ship alternative vis a vis the moored platform due to the possible financing advantage thru MARAD.

Jurisdiction for Tax Incentives

The present territorial jurisdiction of Puerto

Rico is not clearly defined one. On the one hand the Island, due to Congressional oversight was not included in the definition of the state's right to a three mile limit. The Legislature of Puerto Rico ammended the Law of Mines in June, 1979 extending its jurisdiction on the coastal submerged lands up to a limit of 10.35 miles. Based on the available documentation it is evident that the Executive Branchof the **federal** government is willing only to amend the oversight and, thus is at present accepting a "de-facto" three mile limit. A third line of argument exists that defends a 200 mile limit for the Island. Our opinion is that in the long run the 10.35 limit will prevail, but after a long fight through the federal courts or thru Congressional action.

The situation will create additional uncertainties to an OTEC investor, particularly if the plant is located within the three to 10.35 undefined zone.

At present, investment outside the three mile zone could be disqualified for purpose of Section 936 tax incentive. Should the 10.35 zone prevail it could, on the other hand, imply disqualifying ITC incentives on federally taxed income.

Thus, any investment scenario proposed will have to consider the uncertainty of future qualification as a function of jurisdiction.

Leaverage Leasing

Leaveraged leasing seems to be an attractive mechanism for sindicated investments scenarios due to the tax advantages it provides in terms of pass-thrus of tax deductions. This type of financial instrument could be attractive under similar conditions as those presented for the ITC.

Utility Participation

Up to this point the discussion has ignored the participation of the prime potential uses of the OTEC electric output, the P. R. Water Resources Authority, and, in turn, has concentrated on the attractiveness of tax incentives for private investors. Based on the work done so far in terms of integration issues for commercialization it is quite plausible that an OTEC commercial installation cannot be financed by the local utility given the high capitalization requirements for it. Thus, the local participation must be based on scenarios of co-ownership or mixed investment sindicates.

Ongoing work at CTA is concerned with simulating different private-federal-local investment scenarios with particular emphasis on tax mechanisms. Although, based on the partial results that have been produced so far a tax-incentive strategy alone will not be sufficient without possible expenditure incentives for the initial commercialization stages.

Typical Scenario. An example of one of the most promising scenario class is what we are calling "hybrid organizations". A hybrid organization is one in which private-local-federal roles differ to type of OTEC plant system in order to take advantage of different tax situations and different jurisdictions. Within this class the following is an example:

1. Platform - classified as a vessel for purposes of MARAD financing, owned by a U. S. based corporation.

2. Power Plant, including Cold Water Pipe - owned by a different U. S. based corporation, which uses its investment for ITC purposes. Needless to say, the platform would be located outside the three mile limit.

   Power generated would be sold to a service or "qualified" manufacturing company based in Puerto Rico, who would own the power cable. This company will qualify for 936 and local incentives.

3. Shore Facility - could be a joint-ownership organization with possible local utility financing (or co-financing) purchasing the power from the private organizations.

Various other hybrid organizations are being analyzed with varying degrees of ownerships to determine which would be attractive both from a tax as well as from financing standpoint.

Conclusions

Although our integration analysis is still underway, partial results point to the fact that the Puerto Rico site could be very attractive from a private investors point of view because it allows for reduction of both short, as well as long-term risks thru existing tax incentives programs. But a tax incentive strategy alone will not be sufficient to attract investment without possible expenditure incentives (grants and/or subsidies).

The exploration of private-federal-local roles is important for the P. R. case given the fact that a sole financing strategy by the utility would not be feasible, at least within the next decade.

Research is still at a mid-point, thus actual cash flow pictures will be available, along with specific scenario recommendations, further down the line.

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27. Leibowitz, Arnold H. Moreno Ríos Vs. U.S., 256 2d 68, 74 (1st. Cir. 1958) Narcotic Drugs Import & Export Act); Nuchell Vs. Rubio, 139 F. Supp. 379 (D.P.R. 1956) (Fair Labor Standards Act), U.S. Vs. Ríos 140 F. Supp. 376, 381 (D.P.R. 1956).

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ANALYSIS OF PROSPECTS FOR OTEC COMMERCIALIZATION FOR BASELOAD POWER

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Abstract

Requirements for stimulating commercialization of Ocean Thermal Electric Conversion (OTEC) systems for offshore generation of baseload electric power are identified and analyzed.** Commercialization potential was determined in a model which projects competitive economics of regional power-producing systems and estimates market entry and rates of commercial buildup. The investigation resulted in major findings and conclusions as follows: (1) OTEC could make a significant contribution towards satisfying future baseload electric demand requirements in the southeast U.S.; (2) OTEC market performance could be even better if major design concepts were to evolve that resulted in lower costs or improved system performance; (3) development of first commercial-scale OTEC plants at favorable sites off U.S. equatorial islands--e.g., Hawaii and Puerto Rico--could serve as U.S. mainland market springboards; this anticipated result is based on such plants providing early manufacturing and operating experience that would push ahead OTEC power system introduction to the southeast U.S. market from the late 1990's by several years and stimulate rates of subsequent commercial buildup; (4) no fundamental technical obstacles were identified which would severely restrain achievement of OTEC commercialization--principal challenges center now on production and integration of long-life marine components which surpass the scale of present offshore structures.

Introduction

U.S. electric energy demand is projected to continue growing at substantial annual rates. Electrical demand is currently increasing about six percent per year and this growth is expected to decline to less than three percent per annum by 2020. Satisfying regional electrical demand one to two decades from now presents serious ramifications, especially should nuclear plant dilemmas not be resolved and should technological solutions for reducing mining and combustion problems associated with operation of coal-fueled power plants not be expedited.

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**Based principally on results of a study for the Department of Energy, Central Solar Technologies, Contract ET-78-C-01-2854 (January 1979).[1]

For OTEC to penetrate the baseload generating market it must possess advantages over nuclear technology, fluidized bed coal combustion and possibly certain advanced technologies such as biomass combustion throughout the Southeast. At today's energy prices, OTEC can not yet compete with conventional means of generating baseload power; however, when fuel prices escalate to several times current values (in today's dollars) and standardized/improved mass production techniques reduce OTEC capital costs substantially, the technology has promise of competing in the utility market. In addition to OTEC system capital costs, other important variables in the cost equation include cost of deployment as well as operating and maintenance costs. These costs are governed by site conditions including, specifically, the depth of the ocean at the proposed plant/complex site, ocean temperature differentials, length of power transmission cable from platform to shore, design environment (wave, wind and current conditions), and availability of fabrication facilities to handle construction of these large marine structures.

Any means by which OTEC's economic posture can be improved would be expected to permit such commercial systems to enter the market at an earlier date and would provide leverage for increasing the resulting penetration and benefits. Once OTEC installations are implemented and an industry is formed, effects of cumulative experience of the large ocean system construction and operation will be felt. Effects of production and operational buildup are well documented in new industries, and should such trends apply as well to OTEC, early system costs would be expected to decline substantially.

Approach

Market Simulation Model

The analytical tool utilized in OTEC market simulation studies was the SPURR model--System for Projecting the Utilization of Renewable Resources.[2] In the model, projections of yearly incremental baseload electrical demands, region-by-region, are successively satisfied by competing baseload generating technologies--either individual plants or a mixture of baseload facilities. The utility's decision to choose a certain type of technology to satisfy demand projections was modeled using the probability of energy cost as a primary figure of merit. This parameter aggregates all the financial components of a technology and allows a basic comparison of the cost merits among technologies competing for the same application.

Associated with each technology candidate are projections of expected ranges of energy costs, characterized by minimal, maximal and modal values. The probability that the energy costs fall between the two extremes is unity, and the cost having the highest frequency function, of course, corresponds with the modal. A triangular probability density distribution is formulated wherein the modal value is derived to force the area under the distribution to unity. In Figure 1 are shown the projected levelized busbar costs of three candidate baseload systems versus the probability distributions of these costs. In the particular future year illustrated (2010) (and averaged for the southeast regions), the probability distributions of costs for advanced nuclear power generation and the specified OTEC incentive scenario are nearly equivalent; on the other hand, the displacement of the probability distribution curve for an atmospheric fluidized-bed plant toward a slightly higher cost regime, would indicate that the advanced coal combustion plant probably competes less favorably in the specific future time frame. The probability that any baseload electrical generating technology will have a lesser energy cost than another is calculated and the market demands for energy are satisfied by the system with the greatest probability of least cost, within any constraints such as ocean thermal resource limitations.

OTEC Commercialization Stimuli

A number of Federal incentives--financial* as well as policy activities and measures--hold promise for spurring development, commercial introduction and accelerated growth of OTEC energy production.

Economic incentives. Willingness of potential OTEC developers to proceed will generally be based on (1) their perceptions of technical uncertainties and (2) time constants in realizing a payoff on their investments. Preferential application of certain economic incentives to OTEC could be effective in stimulating OTEC development and acceptance. Among the various types of possible tax incentives are: investment tax credits; partial defrayment of R&D expenses; tax deduction for income; tax deferral of taxable income; and deductions for uninsured business losses, accelerated depreciation and interest charges. The investment tax credits (ITC) customarily have the highest leverage of all the tax benefits. Especially with OTEC's high capital costs and relatively low O&M (no fuel costs), reductions in front-end outlays produce significant reductions in busbar costs of power.

Expenditure incentives, financed through authorized budget allocations, include various ship financing programs (should OTEC qualify as "vessels"), grants, loan guaranties, (to provide direct lower-risk funding at favorable interest rates), interest subsidies, guaranteed electrical rates, insurance against market losses, low-interest loans, guarantees of materials availability through strategic stockpiling, or outright government financing and ownership of OTEC plants (as is the case for some large, multi-purpose hydroelectric plants).

Institutional, legal and regulatory incentives. This class of policy options is a source of much concern to potential developers and users of any major new technology, and possible uncertainties could severely delay commercial development of OTEC were these generally nontechnical issues not addressed and resolved. They range from the currently challenged need for repetitive and detailed environmental impact statements to various OTEC classification, permitting and licensing requirements within the regulatory and legislative domain.

With moderate levels of OTEC developmental support, ocean thermal technology could prove to be a viable midterm complement to baseload energy supply in regions bordered by waters possessing adequate temperature differences.

Scenarios for Market Penetration Analysis

Three basic sets of future incentives were established as representative for modeling resultant impacts on OTEC market penetration and growth potential in the southeastern U.S. A fourth and very high-leverage scenario was also examined to determine its effects on subsequent mainland market penetration and growth--the so-called "Island Strategy."

Scenario 1, termed the "National Energy Plan (NEP)", contained the assumption that a 10-percent investment tax credit (ITC) above that normally allowed to industries and utilities be made available to OTEC developers.

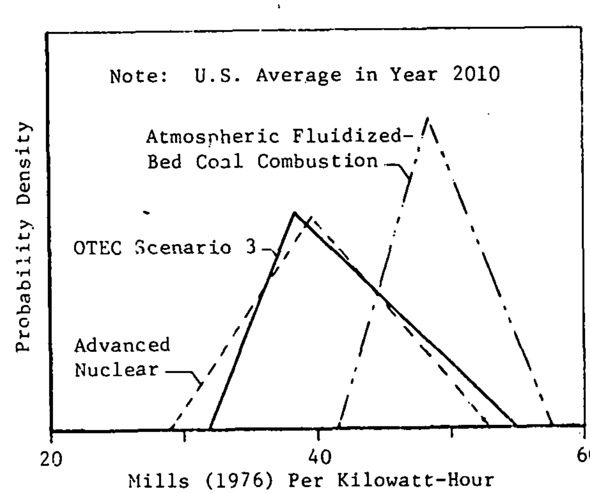

Fig. 1 Probability distribution of projected baseload energy costs for utility market allocation.

*In modeling of government incentives, ranges of investment tax credit subsidies were selected as surrogates for any type of financial incentives (e.g., loan guaranties, direct grants, cost-sharing, etc.) all impacting life cycle costs to corresponding degrees.

Scenario 2, the "NEP plus provision for Accelerated Market Acceptance (AMA)," in addition to the advantages of the previously described ITC, would make available important plant operator information and participatory experience from early (precommercial) operations of OTEC pilot plants. The result of this measure would be a relaxation of market penetration constraints (namely, an S-shaped market acceptance limiting curve in the SPURR model) to reduce the shortest time allowed to capture a given level of the incremental electric market by 50 percent.

Scenario 3, implementation of the "NEP plus the AMA plus a graduated ITC," adds to Scenario 2 an additional 10 percent ITC in the OTEC market entry year, such factor declining linearly to zero in year 2010. This combination of incentives exerted the greatest leverage in the commercialization of OTEC.

The minimal incentive under the island strategy (this supplemental incentive being assumed as implemented with each of the three basic scenarios), was provision for 100 MWe of OTEC island precommercial capacity. Such capacity, in addition to facilitating the assumed accelerated market acceptance levels judged achievable in Scenarios 2 and 3, would reduce engineering costs, through cumulative manufacturing experience applicable to all three scenarios.

The extremely favorable aspects of tropical island sitings[8] which could stimulate buildup of earliest OTEC capacity (with subsequent benefits to mainland market penetration) include: island OTEC plants would replace electrical generating units burning expensive imported oil obtained from unreliable sources; good thermal differences can be tapped at relatively short distances offshore, thus creating advantages such as good thermal efficiencies, short transmission cable runs, and operations in somewhat protected waters; OTEC modules would provide a potential for utilities to add small power increments (with any single addition supplying a relatively small fraction of their total installed capacity) as current oil-fired plants are phased out or as electrical demand rises; and local acceptability and favorable financial incentives appear prevalent.

OTEC Costs and Performance Characteristics

Ranges of engineering costs* and levels of expected plant performance were determined for 400 MWe plants (power delivered ashore) sited in the Gulf of Mexico and off the eastern coast of Florida. Table 1 conveys the characteristics of the generic ocean thermal plant.

Ranges of OTEC power system costs were determined from the work of the power system design contractors (Lockheed Missiles and Space Company (LMSC), TRW and Westinghouse Corporation). Costs for power systems of various module sizes, designs, and materials were summarized by MITRE.[3] A cost summary by Doty Associates,[4] which presented platform system cost estimates derived by the system contractors (Gibbs & Cox, Incorporated, LMSC and M. Rosenblatt & Sons) provided costing guidelines for the bulk of the OTEC system. Preliminary investigations of the platform-to-shore transmission system (the bottom cable, by the Pirelli Company,[5] and the riser cable, by the Simplex Wire and Cable

Table 1 Generic OTEC System

- RATED POWER OUTPUT - 400 MWe (NET POWER AT SHORE) AT DESIGN ΔT (40°F)
- REINFORCED CONCRETE HULL
- POWER SYSTEM - CLOSED CYCLE
  - AMMONIA WORKING FLUID
  - TITANIUM HEAT EXCHANGERS (NO REPLACEMENT OVER LIFETIME)
  - FOUR 100 MWe NET POWER MODULES/PLANT
- CELLULAR, RUBBERIZED NYLON COLD WATER PIPE
- A RANGE OF SITES
- STATIC MOORING SYSTEM
- PLANT AVAILABILITY IS 0.9

Company[6,7]) provided initial cost estimates for AC and DC transmission networks, tie points, and power conversion components.

The ranges of OTEC system costs, in turn, were updated to account for specific platform, power, and energy transfer systems** deployed at three potential southeast U.S. sites: (1) off Miami, (2) mid-Gulf (offshore New Orleans/Mobile/and Brownsville) and (3) off the Florida West Coast. Resultant costs in 1976 dollars per kilowatt for the first production OTEC systems were as follows:

Site	Transmission Distance	Engineering Cost
Miami	28 stat. mi.	$2550/kW
Mid-Gulf	93	2520
Florida W. Coast	161	2630

The expected, or modal, 1976 cost of the first production OTEC system was estimated to be about $2570 per kilowatt.

Variations in system costs about the modal value were next considered in the context of possible variations in system performance. The sensitivity of such interactions hinges on the fact that cost and performance together contribute to energy cost (mills per kilowatt-hour). Plant capacity factor--the ratio of the total energy expected to be delivered ashore per year to that which the plant would continuously produce and deliver ashore at its rated capacity--is the measure of performance. Total annual energy delivered ashore is influenced by time-variant ocean thermal temperature differences as well as by plant availability.

*Costs associated with construction and deployment of the OTEC system, assuming instantaneous system availability.

**Including effects of water depth; wind, ocean current, and wave conditions; transmission distance; and annual profiles of ocean thermal difference.

Factors which could create a more favorable cost-performance ratio include: improvements in conceptual designs and/or fabrication techniques for such major components as platforms, heat exchangers, cold water pipes, mooring subsystems, transmission subsystems; or, more favorable ocean thermal resources. Factors which could degrade cost-performance characteristics include: required hull shape complexities; extreme mooring environments; delays and cost overruns during construction; reduced turbine efficiencies due to lower-than-anticipated ocean temperature differences or biofouling; reduced longevities of seawater systems due to corrosion. Consideration of these factors resulted in a cost-performance range about the modal value estimated as +40 percent and -20 percent, or a range in 1976 dollars of $2056/kW to $3598/kW for the first 400 MWe unit produced and a range of from $1400/kW to $2450/kW for the ultimate plant. Effects of learning, i.e., the cumulative design improvements and production experience, were assumed beneficially to cause the engineering costs, modal, maximum, and minimum, to decrease as shown in Figure 2.

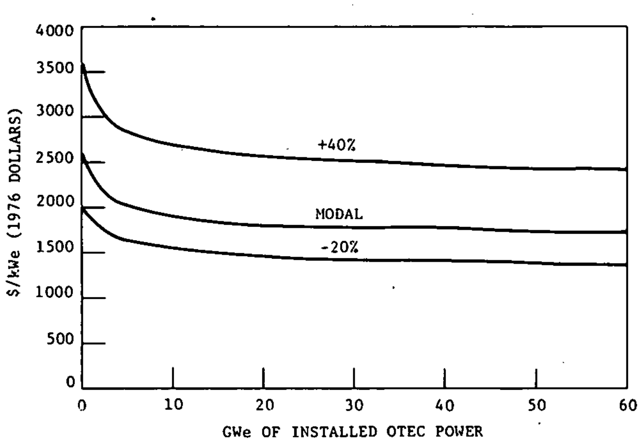

Fig. 2 Engineering cost reduction resulting from experience.

Results

Market Penetration and Impacts of Precommercial Capacity

The potential penetration of OTEC in the baseload electric market of the southeast United States has been estimated with the SPURR model for each of the three incentive scenarios just described. Table 2 shows the cumulative OTEC potential in the continental United States (CONUS) by the year 2020. Potential penetration under three levels of incentives and up to 400 MWe of island capacity are shown for two possible market entry dates.

Table 2 Potential OTEC CONUS Market Penetration by 2020 (GWe)

CONUS MARKET ENTRY DATE	INSTALLED OTEC ISLAND CAPACITY (GWe)						
	0.10	0.50	0.75	1.00	1.50	2.00	4.00
SCENARIO 1							
1995	28	28	28	45	45	----	----
2000	28	28	28	30	30	30	32
SCENARIO 2							
1995	56	58	62	87	89	----	----
2000	56	58	58	61	61	62	64
SCENARIO 3							
1995	107	113	115	115	116	----	----
2000	70	76	76	76	76	77	80

Market potential for OTEC is seen to increase with increasing levels of incentive from Scenario 1 through Scenario 3. Market potential essentially doubles from Scenario 1 to Scenario 2. Both have the same level of tax credit, but Scenario 2 incorporates added activities such as user education and hands-on experience programs to help reduce market resistance and develop accelerated market acceptance. Under Scenario 3, the addition of another incremental tax credit basically doubles market performance beyond that of Scenario 2.

Under Scenarios 1 and 2, island capacity is seen to have a great effect on ultimate market potential. Island capacity of between 750 and 1,000 MWe serves to increase the OTEC market potential by effectively lowering the capital cost of the first mainland plant through the effects of experience. At a lower initial cost, OTEC technology can enter the CONUS baseload market at an earlier date and then compete more vigorously with the other baseload alternatives. Under Scenarios 1 and 2 and low levels of island capacity, market potentials for OTEC remain virtually unchanged for CONUS market entry in the year 1995 or 2000. Beyond 750 MWe of island capacity, OTEC could have greater potential if it entered the market in 1995 rather than 2000. This is a result of an earlier competitive posture for OTEC and a longer period to capture a commensurately larger share of the market.

Expected OTEC Market Penetration

OTEC's U.S. baseload market potential within a given scenario is not only dependent on the level of island capacity installed prior to the CONUS market entry date but also on the actual entry date itself. If each market buildup potential under the three scenarios, as shown in Table 2, has a probability associated with it, the potentials can be aggregated and an expected value of OTEC market penetration estimated.

The probability* that OTEC will gain market entry in 1995 and 2000 was estimated. These values were combined with estimates of probability associated with market buildup potentials resulting from various levels of island capacity.

Table 3 presents the projected values of expected OTEC baseload market penetration by 2020 in terms of both capacity and equivalent oil energy (input to steam power plant) displaced per year for each of the three scenarios. The total baseload OTEC capacities in year 2020 (Table 3), resulted from multiplying estimated market penetration potentials by factors, as described, expressing expectations that OTEC will meet scheduled development within each scenario.[1]

Table 3 Expected OTEC Market Penetration by 2020

	Capacity in Place (GWe)	Energy Displaced (Quads/Yr)
Scenario 1	22	1.5
Scenario 2	46	3.2
Scenario 3	71	5.0

OTEC and the Competition

OTEC busbar energy costs compete favorably with those of other technologies. As shown in Table 4, OTEC busbar costs at U.S. island sites are projected to be below the range expected for oil-fired plants by 2000; (also, competitive OTEC costs should be considerably below those of oil-fired plants by 2020). Ultimate mainland OTEC plants are projected to provide power for about the same busbar costs as advanced nuclear power plants and for somewhat less than coal in 2020, depending on incentive scenario and geographical region.+

OTEC Net Benefits

If OTEC is successfully developed and penetrates the baseload electric market of the U.S., then electric power generated by OTEC systems could have associated net benefits when compared to other baseload alternatives. The net benefits to the nation can be estimated as the difference in the cost of energy generated by OTEC and that generated by other systems, less the Federal developmental expenditures required to bring OTEC to its expected level of market penetration.

In Table 5, OTEC systems are projected cumulatively to displace from 11 to 43 end-use quads of electricity (30 to 116 equivalent quads of primary energy) in the years from 2000 to 2020, depending on scenario. When the cost of this energy is compared to that of an expected mix of nuclear and coal technologies, this amount of OTEC energy could result in a net reduction of the

*Such probability hinges on judgmental expectations of achieving success by target dates versus possible programmatic slippages.

+The applicable southeastern U.S. census regions are the SATL (South Atlantic), ESCL (Eastern South Central), and WSCL (Western South Central).

Table 4 OTEC and the Competition

LOCATION	DATE	COMPETITION	COMPETITIVE COST OF ELECTRICITY (1976 Mills/kWh)	OTEC COST OF ELECTRICITY(1) (1976 Mills/kWh)		
				SCENARIO 1	SCENARIO 2	SCENARIO 3
U.S. ISLAND	2000	OIL	50.6 - 102.5	43.0 - 75.6	----	----
U.S. ISLAND	2020	OIL	69.3 - 143.4	29.3 - 51.4	----	----
S. ATLANTIC REGION	2020	NUCLEAR COAL	32.9 - 60.2 50.0 - 68.5	34.1 - 59.1	33.0 - 57.2	32.7 - 56.6
E. S. CENTRAL REGION	2020	NUCLEAR COAL	32.4 - 59.3 46.7 - 64.0	36.7 - 63.6	35.5 - 61.5	35.2 - 60.9
W. S. CENTRAL REGION	2020	NUCLEAR COAL	32.8 - 59.8 42.8 - 58.8	38.2 - 66.1	37.0 - 64.0	36.6 - 63.4

COAL HEAT RATE = 7,800 - 10,000 BTU/kWh
OIL HEAT RATE = 7,500 - 9,220 BTU/kWh

(1) 100 MWe of precommercial capacity

national electric bill (to both consumers and government) of from 6 billion dollars to 34 billion dollars by 2020, and an expected benefit-to-Federal cost ratio of from 2:1 to 3:1.

Table 5 OTEC Cumulative Net Benefits in the Southeast U.S. by Year 2020

SCENARIO	E(QUADS/YR)	CUMULATIVE E(QUADS) to 2020	EXPECTED(1) NET BENEFIT	FEDERAL COST(1)	$\Sigma\left(\frac{\text{BENEFIT}}{\text{FEDERAL COST}}\right)$
1	1.5	11	6	3	2:1
2	3.2	21	16	5	3:1
3	5.0	43	34	14	2:1

(1) Billions of 1976 dollars

In addition to the benefits expected from OTEC in the continental U.S. after 2000, OTEC could provide significant benefits before 2000 in U.S. islands where present capacity is oil-fired. A detailed analysis of the island energy market has not been performed, but a range of installed island capacity has been checked in preliminary evaluations to estimate a range for the level of benefits that may accrue from island OTEC deployment prior to 2000. Island net benefits (Table 6) range from 0.3 billion dollars to about 3.7 billion dollars through 2000 and have associated net benefit-to-Federal cost ratios of from 0.75:1 to about 2.6:1.

Table 6 OTEC Island Strategy Potential Net Benefits by Year 2000

ISLAND CAPACITY	CUMULATIVE ENERGY DISPLACED BY 2000	FEDERAL EXPENSE	NET BENEFIT	NET BENEFIT-TO-FEDERAL COST BY 2000
100 MWe	0.1 Q	$ .4 x 10^9	$ .3 x 10^9	0.75:1
500 MWE	0.3	.5	.7	1.4 :1
750 MWe	0.4	.6	.9	1.5 :1
1000 MWE	0.5	.7	1.2	1.7 :1
1500 MWe	0.7	.8	1.6	2.0 :1
2000 MWe	0.9	.9	2.1	2.3 :1
4000 MWe	1.6	1.4	3.7	2.6 :1

OTEC power @ 37 Mills/kWh; Oil power @ 70 Mills/kWh; assumed buildup schedule

OTEC technology has been projected to be capable of providing significant amounts of electrical energy to the southeastern U.S. and of providing substantial benefits to the nation about the turn of the century and thereafter. Several possible incentive mechanisms that could enhance OTEC performance have been identified. Two levels of financial incentives and an institutional incentive were modeled in an energy market simulation that projects the market performance of several baseload electricity-generating alternatives.

The greater the incentive level, the better OTEC technology has been seen to perform as measured by buildup of commercial capacity in CONUS. In addition to direct incentives, the plan known as the "Island Strategy" is seen to provide leverage for ultimate OTEC market penetration, particularly when executed in conjunction with Incentive Scenarios 1 and 2.

OTEC market performance has been estimated for the generic OTEC system described earlier. However, enhanced market penetration and increased net benefits can be realized if the system life cycle cost were to be reduced and/or the system output increased. Improvements in design concepts for both ocean systems and power systems or discovery of regions of high thermal gradient, for example, could help to further improve the commercial posture of OTEC baseload electric generating technology.

Conclusions

Principal findings of this investigation may be summarized as follows:

- OTEC has a potential to make a significant contribution toward satisfying future baseload electric demand requirements in the southeast U.S.; large-scale OTEC electric plants--design capacities of 400 MWe, power delivered ashore--should achieve Southeastern market entry about 1995.
- OTEC market performance could be even better if major concepts were to evolve that resulted in lower engineering costs or improved system performance.
- A market springboard effect may be predicated on establishment of a modest level of OTEC early precommercial capacity (about 750 MWe) off U.S. equatorial islands (i.e., "Island Strategy"); the anticipated effects would include inducement of early-to mid-1990 market entry in the southeast U.S. and accelerated commercial buildup of OTEC capacity in U.S. mainland electric grids.
- No fundamental technical obstacles which would restrain achievement of OTEC commercialization were identified; major remaining challenges center upon the design, integration, construction, and deployment of some of the world's largest ocean structures.

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DISCUSSION

Question: Does your model include regulatory risks?

R. Manley: To my knowledge nobody has been able directly to quantify regulatory risks. However, if regulatory burdens or delays can be related to financial risks, any resultant additional engineering and financing costs would comprise conventional input parameters of the SPURR model.

Question: Do you have any feeling for the obviously severe impact on the cost of electricity if a utility had to shut down a nuclear plant, like Three-Mile Island, for a long period, maybe permanently? Where would you plug that into your model?

R. Manley: Costs of nuclear plant shutdown could be plugged into the model by reducing the selected capacity factor of the plant. Additionally, we could assume that the plant might shut down prematurely, perhaps after five years of its useful life and then calculate energy costs based on the estimated shorter useful lifetime. Other expenses that will elevate busbar costs of power would be associated with required purchase of more expensive bulk replacement power from other utilities in a grid (plus possible wheeling charges), costs of generating power from marginal, standby units, and costs from incurred liabilities, cleanup operations, and possible plant decommissioning. Or, you could assign probabilities to various levels of risks and come up with resultant nuclear system costs. Increases in nuclear energy costs would make OTEC a good deal less expensive than this competing alternative for baseload power generation.

COMMERCIAL OCEAN THERMAL ENERGY CONVERSION (OTEC) PLANTS BY THE MID-1980'S

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Abstract

Ocean Thermal Energy Conversion (OTEC) promises to become a major new energy source which will be low in cost, environmentally clean, safe, renewable and reliable. The ocean area where the temperature difference exceeds 40°F (22°C) includes 22 million square nautical miles of tropical oceans, enough to provide 9 x 10^{13} kWh/year of OTEC busbar electric power, or 40 times that generated in the United States in 1977. Considerable progress in component development and pilot plant designs has been made in the past 18 months. Construction of an OTEC pilot-demonstration plantship of 40 MW_e(net) nominal size producing 43,000 tons/year of ammonia on board, and of a comparably-sized OTEC pilot plant delivering electric power ashore by undersea cable to an island such as Puerto Rico or Hawaii have been proposed by industry and local governments for deployment in 1983 or 1984. These schedules appear reasonable to implement. Successful operation of these pilot plants could be followed by deployment of commercial-size plants starting by 1987 if a national commitment of large funding is made to achieve an early abatement of U.S. dependence on foreign oil. Since our commercialization task was funded jointly by the Department of Energy and the U.S. Maritime Administration with directions to emphasize OTEC-produced ammonia, this paper addresses the OTEC ammonia option.

Introduction

The OTEC Resource

The OTEC resource for plantships cruising on the high seas near the equator is enormous as shown by Fig. 1[1a] and Table 1. The contours in Fig. 1 show that the areas where ocean temperature differences (ΔT's) are equal to or greater than 22°C (40°F) amount to some 22 million square nautical miles, enough to provide 9 x 10^{13} kWh/yr of OTEC busbar power. It should be noted that even higher annual average ΔT's can be obtained by cruising at 0.4 knot average speed within selected siting areas.[1b]

Basis for Negotiated Contracts

A survey of ship constructions since 1970 that have received Maritime Administration support[2] shows that with only one exception the contracts were negotiated with industry and were not placed in response to a government RFP. In these negotiated contracts a perceived need by industry led to a ship design that would fulfill the industry performance requirement at a cost and schedule that were judged to be acceptable for the commercial application. This method of contracting insures that industry gets what it wants at the price it is willing to pay; whereas the government competitive bid route requires general specifications to avoid preference to any potential bidder, and government restrictions on procedures, time scales, construction methods, construction sites, and materials. These restrictions inevitably lead to program delays, irrelevant requirements, diffuse objectives, and much higher cost.

It appears, therefore, that if the United States is to benefit from commercial OTEC ammonia plantships by the mid-1980's, industry must propose and the Government must be prepared to accept reasonable cost-sharing proposals to build and operate specific industry-based designs for the OTEC plantship. A baseline OTEC pilot plantship and workable components have now been designed[1c], and within the planned immediate OTEC program, will be tested. This is not necessarily an optimum design, but it is projected to be low in cost and competitive in performance. It is now time to let industry come forward with proposals for detailed design and construction plans tailored to their facilities--at their own expense. It is now time for the Government to help fund acceptable plans and get on with construction of pilot plants to demonstrate OTEC performance and attainable costs.

Within the commercialization schedule we propose, it is our opinion that the Government will get the most for its money, the interests of the American people will be served, design and construction will be effectively done under near-commercial practice, and the pressing need for energy will begin to be

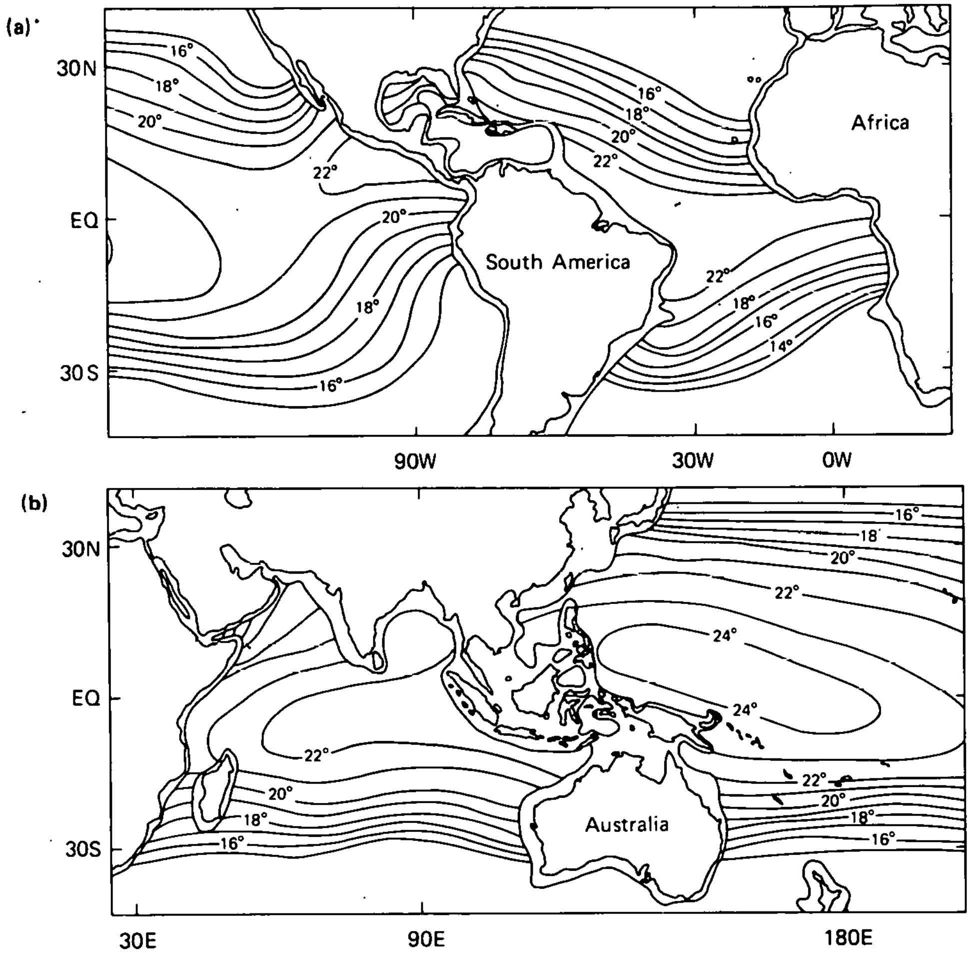

Fig. 1 The thermal resource for OTEC plantships. ΔT(°C) between surface and 1,000 meter depth.

Table 1 The thermal resource for OTEC plantships.

OCEAN AREA SUITABLE FOR OTEC PLANTSHIPS = 22 MILLION SQ. (NAVT.) MI. (ΔT GREATER THAN 40°F)
ESTIMATED MINIMUM OPERATING AREA PER 325 MW PLANTSHIP = 750 SQ. MI.
∴ TOTAL OTEC POWER GENERATION POTENTIAL IS
30,000 x 325 MW = 10,000 GW = 9×10^{13} KWH/YEAR
IN COMPARISON, THE U.S. TOTAL ELECTRIC POWER GENERATION IN 1977 WAS
2.2×10^{12} KWH OR 2.3% OF THE OTEC POTENTIAL

met--all within the next few years. We understand that the Department of Energy's five-year OTEC development plan will provide for cost-sharing initiatives, but as of this writing, details have not been released.

Estimated Costs of a Pilot Plantship

Costs for construction and deployment of a 40-MW_e OTEC-ammonia pilot plantship (plantship is a term used for a self-propelled vessel with a process plant on board) have been estimated in FY 80 dollars using as a base the costs reported in Refs. 1c and 3. These costs were developed by the industry members of the preliminary engineering design team who were most qualified by prior experience and related commercial construction to make realistic calculations, as noted in Table 1-1 of Ref. 3. The costs were developed for a specific construction design and a specific deployment scenario. They are supported, where appropriate, with vendor quotes and independent consultant estimates. This does not mean that they accurately present the actual costs of construction by another builder; it does mean that they are representative of the magnitude of cost to build the OTEC-ammonia pilot plant.

Table 2 gives our construction and deployment cost estimates, in FY 80 dollars, for the 40-MW_e (net) pilot plantship and commercial OTEC-ammonia

Table 2 OTEC cost (price) data 1980 dollars.

	40-MW PILOT PLANT-SHIP				325-MW FIRST COMMERCIAL	325-MW 5th-6th COMMERCIAL
	ATLANTIC OR PACIFIC GRAZING ΔT 43°F	PUERTO RICO ΔT 40.3°F	HAWAII ΔT 38.5°F	GULF OF MEXICO ΔT 40°F	ATLANTIC OR PACIFIC GRAZING 43°F	ATLANTIC OR PACIFIC GRAZING 43°F
CONSTRUCTION AND DEPLOYMENT COST						
$ KW	3086	3851	4297	4451	1698	1208
TOTAL	$124M	$154M	$172M	$178M	$532M	$393M
ANNUAL OPERATIONS AND SUPPORT COST						
$/KW	98	60	65	60	14	13
TOTAL	$3.9M	$2.4M	$2.6M	$2.4M	$4.5M	$4.2M

plantships. No baseline design has been done for the commercial-sized ships. These costs are extrapolated from the pilot plantship[3] and other studies[4] and include estimates of economies from larger-sized components and modest learning curve experience They are consequently of lower confidence level than the 40-MW costs.

The estimated costs are shown first in dollars per kilowatt and then in millions of dollars. They depend strongly on both size and site. The self-propelled, cruising pilot plantship, which seeks the largest temperature differences in Atlantic or Pacific equatorial waters and produces ammonia on-board, is the lowest in cost; an off-shore U.S. Gulf of Mexico OTEC pilot plant cables the electricity to shore from a distance of 125 miles or more, and has the highest construction and deployment cost. In Puerto Rico and Hawaii the cable needs to be only a few kilometers in length and can carry A.C. power. Therefore, the transmission costs are much lower.

Because the costs of the baseline pilot plantship hull and the APL baseline-design heat exchangers are both substantial, additional fabrication cost-estimating work was undertaken in these two areas. We worked with qualified bidders not a part of the initial design team to get additional estimates. The additional hull estimate obtained was essentially the same cost as that provided by ABAM Engineers, Inc. The additional heat exchanger cost estimate was 15% or $1.2 million less than that obtained from The Trane Company. Subsequent design requirements review and test results, available after the cost estimates were made, indicate that further reductions in cost should be forthcoming with the next APL-design power system optimization. Costs of the electrolysis plant (based on the General Electric Company's solid polymer electrolyte (SPE) cells[1d]) and other process equipment are based on data provided by designers and vendors as given in Ref. 4 and updated to 1980 dollars.

Potential Cost-Sharing Team Makeups

A cost-sharing proposal for design and construction of the OTEC-ammonia plantship could take many forms. As one example, we have postulated the following team which we believe could be practical.

Consortium of Ammonia-Producer and Utility Companies

A consortium would be formed for the purpose of owning and operating the OTEC-ammonia pilot plantship, selling the ammonia product, and implementing a program for use of ammonia to produce electric power from hydrogen-chlorine or hydrogen-oxygen fuel cells. It would include a process plant design and construction firm such as Pullman Kellogg, Horst-Uhde, etc. Motivation for the ammonia-producer companies and utility companies is based on several factors including the decreasing availability of low-cost natural gas for feedstock and process heat, a very real concern that 25% to 50% of U.S. vital nitrogen fertilizer supplies will be imported in the 1990's unless something is done, need for environmentally acceptable, low-cost power, and profit potential. Table 3 shows projected cash flow for the design and construction period and three years operation of a 40-MW_e OTEC-ammonia pilot plant, equipped to produce 43,000 short tons per year of ammonia. Based on the assumptions made, a consortium of ammonia producers could be expected to provide about $24 million in equity and debt cash for cost-sharing. In addition, there is significant potential for cost-sharing through the pledged use of ammonia carrier ships, port facilities, storage and transport facilities, sales organizations and fuel cell research and development. The cash and cost-sharing potential from private companies for the uneconomical, first-of-a-kind 40 MW_e pilot plantship would be greatly increased for the first commercial-sized OTEC plantship to an estimated 50% or more.

State and City Government

The projected jobs impact of OTEC plantship construction is very significant for large depressed city areas where most major U.S. shipyards are located. Using a rough estimate that $63,000 in shipyard revenues indicates one shipyard worker-year, we estimate that the pilot plantship would provide employment for 1,000 workers for two years, and the 25-35 commercial OTEC-ammonia plantships that could be built in the 1980's and 1990's[4] would provide 150,000 to 210,000 worker-years of shipyard employment, corresponding to several tens of thousands of permanent new jobs in the second

Table 3 Cash flow projection for 40-MW_e OTEC-ammonia pilot plantship.

PLANT INVESTMENT: 124\overline{M}$		NH$_3$ SELLING PRICE, 1983 START-UP: $230/S. TON		
PRIVATE FUNDS: 75% DEBT AT 8%		7% PER YEAR INFLATION: PRICE AND COSTS ACCELERATED DEPRECIATION, 20% I.T.C.		
YEAR	PRIVATE CAPITAL OUTLAY	CASH OPERATING COSTS	SALES $230 + INFLATION	NET CASH FLOW
1980				
1981	2\overline{M}$	0.1		0.3
1982	10\overline{M}$	0.6		1.4
START-UP	12\overline{M}$	1.4		1.0
1984		4.1	9.9	4.4
1985		3.9	10.6	4.7
1986		3.7	11.3	5.1
–	–	–	–	–
–	–	–	–	–

PRIVATE FUNDS RECOVERED BY CASH FLOW: 4.3 YEARS
9-YEAR NET PROFIT/EQUITY COMPOUNDED: 16%

decade and beyond. To this number would be added the jobs generated by component, equipment and materials fabricators and support industries. Selected State and City Government officials have indicated that they are motivated by the potential for this new industrial employment within their borders. We estimate that State and City cost-sharing (which could include urban development grants) could be up to $30 million. We recommend that the Federal Government actively encourage this type of participation.

Federal Government

The national need to solve our energy problems combined with the economics and risks of the earliest OTEC plants indicate that the Federal Government will be the largest financial contributor to construction and deployment of the first OTEC pilot demonstration plantship. We estimate that the Federal contribution will need to be near $100 million in constant 1980 dollars. These funds should be included in the FY 81, FY 82 and FY 83 budgets. This would require cost-sharing proposals to be known to the Department of Energy by the end of 1979. It is our opinion that this early timing can be met.

Builder Team

There is little doubt that the builder team which does a successful design, construction and deployment of the OTEC-ammonia pilot plantship will be in a favored position for follow-on contracts to build many of the 25-35 projected plantships in the 1980's and 1990's. This profit potential should provide incentive for cost-sharing participation. We have made no estimate of the dollar potential this could involve. However, the more than $2 million of estimated private contributions to the "Mini-OTEC" commissioned in May 1979[1d] and the significant private contributions made by other potential builder companies during the past five years of OTEC development are strong indicators that some builder cost-sharing potential exists.

Design, Construction and Deployment Team

The builder team, as a minimum, will include one or more organizations with capabilities for hull construction, heat exchanger fabrication, cold water pipe construction and deployment, process plant construction, and installation and outfitting expertise. The number of companies in the builder team will be determined in part by the power system and platform designs selected, and in part by performance guarantee requirements. Although several of the OTEC heat exchanger designs[1e-g] show good promise and might be substituted for the ALL baseline design if more cost-effective or lower cost, we have postulated a design team based on the baseline plantship and heat exchanger designs. For these we have both cost and performance data.[1c,h]

In the postulated design team there are four principal builder companies: a marine concrete hull construction firm, a major shipyard for outfitting and equipment installation, a concrete construction company for cold water pipe fabrication and deployment, and a process plant design and construction firm. While it is not yet fully defined, it appears that the folded-tube heat exchanger, or an alternative design, can be fabricated at the shipyard or bought from one of several qualified suppliers--depending on costs and price. It is also not determined whether the team will be qualified to successfully perform the OTEC system integration tasks. If not, a large system integrator company should form or be added to the team. This matter could be clarified within the next few months, but can be expected to vary between different builder teams depending on the qualifications of the participants.

One critical element of the postulated builder team is that they have the confidence of the cost-sharing team. This is brought about by a combination of management, technical qualifications, demonstrated performance and perhaps subjective intangibles.

Technological Readiness

With expected progress, the technology will be ready to meet our projected schedule. Planned tests are: a 1/3-linear-scale test of a light-weight-concrete[1j] cold water pipe section with a flexible bearing pad connection; test of the 3-tube, full-scale core unit of the folded-tube heat exchanger[1h] as a condenser, tests for 9-12 months of an element of the heat exchanger and an acoustic cleaning unit[1i] in ocean water and tests of a 1/30th scale model of the pilot plantship in a wave model basin to verify the analyses of platform motion and seakeeping. It is assumed that DOE will fund these tests in a timely manner consistent with the schedule recommended herein.

The estimated \$70-\$100 million (1980 dollars) funding needed in the FY 81, FY 82, and FY 83 budgets as the Government's share of a 40-MW_e OTEC-ammonia pilot plant is in addition to OTEC funding for such off-shore OTEC pilot plants and OTEC research and development as are otherwise justified.

Recommended Government Incentives

Merchant Marine Act, 1936

We have assumed the Merchant Marine Act mortgage guarantee and subsidy programs (Title XI, O.D.S., C.D.S.) will be applicable where needed for OTEC plantships commencing with the first pilot demonstration plantship.

Title XI mortgage guarantee makes available long term financing for vessels at interest rates competitive with quality industrial and utility loans and removes the need for the lender to look at factors other than the return on his money. Several ammonia-producer companies have experience using Title XI financing. Our discussions with ammonia producers indicate that cost-sharing will be forthcoming only if Title XI is available. The conditions otherwise imposed by financial institutions and the educational program otherwise necessary to obtain commitments of funds, when no prior operating OTEC-ammonia ships exist, pose essentially insurmountable barriers. The Maritime Administration has not made it clear whether Title XI will or will not apply. Instead, they have reserved the decision for individual proposals and have made public their reservations against OTEC coverage. Therefore, we believe that an amendment to the Merchant Marine Act is needed to clearly include OTEC. We have assumed the amendment will be enacted consistent with our projected schedule.

We have discussed with officials of the Maritime Administration and other Government agencies possible elements of a Congressional bill which would facilitate Government support of OTEC commercialization. The following are possible items which could be covered in such a bill as qualifications for loan guarantees:

- OTEC vessels, designed to lessen dependence of the U.S. on foreign energy supplies, regardless of cargo status or operation between ports or a particular service, route or line;
- OTEC vessels engaged in production of energy, energy products and/or vessels transporting the products of American flag OTEC vessels; and
- A waiver of the economic criteria of Title XI for up to five OTEC pilot projects, if sufficient performance guarantee or payment is provided by DOE and/or private industry to reduce the risk of loss to a level the Secretary of Commerce determines to be reasonable taking into account U.S. energy needs and the benefits to the American merchant fleet.

You will recall that Table 2 showed larger operating costs for self-propelled "grazing" OTEC plantships in equatorial waters than were shown for moored plants. These higher costs stem from U.S. Coast Guard manning requirements developed for faster moving vessels, and by Federal requirements developed for all American crews. To compensate for these higher costs and allow American flag OTEC plantships to be competitive with foreign operated OTEC plants, we recommend the Merchant Marine Act be amended to include OTEC vessels under the Operating Differential Subsidy (O.D.S.), including the first pilot plantship. For similar reasons, we recommend consideration of Construction Differential Subsidy (C.D.S.) for commercial OTEC plantships.

Energy Tax Act, 1978

We have assumed that the Energy Tax Act will be amended to allow a 20% or 30% Investment Tax Credit (ITC) for OTEC plantships in their entirety. The purpose of the Act is to encourage the development of renewable energy sources which would reduce U.S. dependence on imported oil and natural gas and to conserve non-renewable domestic oil and gas supplies. Clearly this purpose would be served by early implementation of OTEC plantships. OTEC was excluded by the Act for reasons that are no longer valid; 25-35 commercial OTEC-ammonia plantships in the 1980's and 1990's would conserve 330 billion cubic feet of domestic natural gas per year for other uses, would support reasonable cost gasahol,* and would have a favorable balance of payments impact of about \$1 billion per year (1980 dollars) relative to importing LNG. OTEC electric power cabled to shore in the islands would reduce oil imports. OTEC power cabled ashore to Southeast U.S. utility grids or produced anywhere in the U.S. by dissociating ammonia to power fuel cells could have an even greater impact on U.S. energy supplies before the year 2000. We recommend that the accelerated Investment Tax Credit for OTEC be provided in a timely manner consistent with our proposed schedule. This would encourage and allow more favorable cost-sharing proposals, even for the first OTEC ammonia pilot plantship.

Concluding Remarks

In summary, we have discussed needed Government actions to ensure commercial operation of ocean thermal energy conversion plantships producing ammonia starting with deployment 1986-'87. The first requirement is support of a 40-MW OTEC ammonia cost-shared pilot plantship with demonstration detailed design starting in FY 80. OTEC plants cabled to shore off U.S. islands should be supported on the same time scale. We have recommended that construction contracts should be negotiated, based on receipt of specific

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*Via fertilizer (for corn and wheat fields) made from OTEC ammonia.

proposals to prove OTEC capability to meet a defined industrial need. OTEC costs estimates, based on industry experience, are attractive. We have postulated workable OTEC cost-sharing and builder teams.

The Government actions which appear necessary to make 1987 first commercial plants feasible are listed below. We believe they are reasonable and well justified by the progress on OTEC made to date and the great potential of the ocean thermal resource to solve U.S. energy problems.

The needed action items are:

1. Continuation of DOE plans for near-term tests.
2. Adequate funding for the OTEC ammonia pilot plantship of $2-10 million in FY 80 and $100 million in FY 81-83. This is in addition to other OTEC funding and is justified by the potential of OTEC to become a major supplier of U.S. energy needs.
3. DOE encouragement of cost-sharing construction.
4. Amendment of the Merchant Marine Act, 1936 to include OTEC plantships.
5. Amendment of the tax laws to provide up to 30% ITC for OTEC.

The above action items are believed by the authors to be a modest demand for the benefits expected: near-term, clean safe energy at low cost, plentiful low cost fertilizer for food and gasahol, and tens of thousands of permanent new jobs in depressed core city areas.

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DISCUSSION

Question: Do you assume a change in the Title XI tax structure to include OTEC plus a 25% investment tax credit?

E. Francis: The basis on which we calculated the amount of money that would be attainable from ammonia producer companies did specifically include Title XI financing, without which we think that OTEC commercialization will be very difficult. We also considered a 75% debt, 25% equity ratio without which we think it will be impossible to get cost sharing from the ammonia companies. It also included a 20% investment tax credit similar to that in the act for other solar equipment.

J. Bryan, Gifford-Hill & Co.: Everything we've heard as far as proposals on financing include some sort of government financing. Has any proposal been put forward to consider OTEC under a TVA authority or BPA authority?

E. Francis: I think that there was a study undertaken by DOE where Tefft, Kelly, and Motley made a proposal along those general lines. I don't know whether the thing was well received in DOE or what actual publication distribution was made of it. I don't have a copy of the study.

Question: You mentioned competitive prices. How does that compare? What are the prices in cents per kilowatt hour?

E. Francis: In the phraseology I am using — in competitive prices in 1980 dollars — we are using 40 to 55 miles/kWh, almost precisely the same numbers that are in an EPRI report that was published about a year ago, which, for a comparable time period and also 1980 dollars, ranged from something like 35 to 55 or 60 miles/kWh. The number that we used, if we're talking 1983 implementation of an OTEC-ammonia plant, to produce ammonia to sell on the fertilizer market in order to get the $24 million cost sharing, is a sales price FOB New Orleans of $230/ton. We use this only as a representative example. We played other games with a lot of other scenarios; the one that Jack Babbitt presented, to the Fertilizer Institute at their Greenbriar annual meeting in June 1979, was $190 for the 1983 implementation, but we also used a 30% investment tax credit.

*STATUS OF SOLID POLYMER ELECTROLYTE ELECTROCHEMICAL CELL TECHNOLOGY FOR ELECTROLYTIC HYDROGEN GENERATION AND FUEL CELL POWER GENERATION

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Abstract

The possible use of hydrogen as an energy vector is being considered for some of the possible Ocean Thermal Energy Conversion (OTEC) scenarios. This requires an efficient means of producing the hydrogen from electrical power on board the OTEC platform, and also for converting it back to electrical energy at the other end of the distribution network. Under a program sponsored in part by DOE and some of the electric and gas utilities, General Electric is developing a new solid polymer electrolyte (SPE) water electrolysis technology which offers a potential efficiency of 85-90% and capital costs in the range of $100/KW. This paper briefly describes the technology and the current status of the program relative to a planned 5-MW demonstration plant which is expected to be completed in 1983. The SPE technology has also been studied for use in a hydrogen/halogen regenerative fuel cell system for energy storage. A concept is presented for adapting this approach to a primary H_2/O_2 fuel cell by combining it with a catalytic process for converting HCl into H_2 and H_2O. For such a system, fuel cell efficiencies in the range of 70-80% may be feasible.

Introduction

The conversion of ocean thermal energy to electrical power represents a promising means for supplementing the nations energy from the ultimate renewable resource, solar heat. One of the major questions being addressed in considering the development of this resource, however, is that of the best way to utilize the resultant power in view of its relatively remote location from the power demand centers. One possibility that has been quite extensively studied by the Applied Physics Laboratory of Johns Hopkins University[1] is to use the power to produce ammonia on board the OTEC plant ship, which can then be used to supply part of the needs of the fertilizer producers around the world, and thus free up the natural gas and oil now used to produce the ammonia for this industry.

Alternatively, the ammonia might also be considered as an energy carrier in cases where the need for electric power may be overriding. It can be readily reformed to release the hydrogen for use in a fuel cell to provide electric power at, or convenient to the load centers.

In either case, the intermediate product is hydrogen, which can be produced from the electrical power by the electrolysis of water. The advanced solid polymer electrolyte water electrolysis technology under development at the General Electric Company is especially promising for this type of application because of its high conversion efficiency and small floor space requirements. It also offers a potentially significant savings in capital cost.

This same technology may also represent a promising means for generating electrical power from the reformed ammonia at the other end of the distribution network. The SPE cells have demonstrated excellent efficiencies and life as a hydrogen/oxygen fuel cell, and even greater efficiency should be possible by incorporating an intermediate H_2/Cl_2 couple.

Solid Polymer Electrolyte Electrolysis Technology

The solid polymer electrolyte is a solid plastic material which has ion exchange characteristics that make it highly conductive to hydrogen ions. The particular material that is used for the current electrolysis cells is an analogue of TFE teflon to which sulfonic acid groups have been linked. This plastic sheet is the only electrolyte required, there are no free acidic or caustic liquids, and the only liquid used in the system is distilled water.

A typical cell, shown schematically in Figure 1, employs a sheet of the polymeric material approximately 10 mils thick. A thin catalyst film is pressed on each face of this sheet to form the anode and cathode electrodes. Since the electrolyte is solid, the electrodes do not have to perform any structural or containment functions. Consequently, they are very simply designed for the sole function of providing sufficient catalytic activity to achieve desired performance levels.

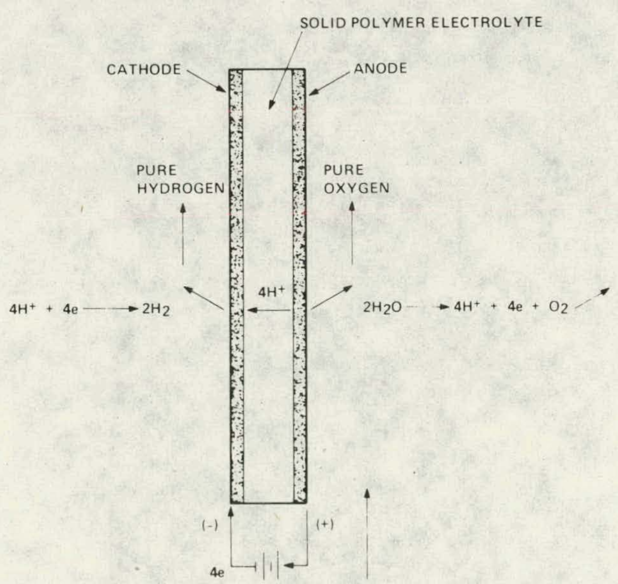

Fig. 1 SPE Electrolysis Cell Schematic

Water is supplied to the oxygen evolution electrode (anode) where it is electrochemically decomposed to provide oxygen, hydrogen ions and electrons. The hydrogen ions move through the sheet to the hydrogen evolving electrode (cathode) while the electrons pass through the external circuit. At the hydrogen electrode, the hydrogen ions and electrons recombine electrochemically to produce hydrogen gas.

An excess of water is supplied to the cell and recirculated to remove waste heat.

This technology was originally developed as a fuel cell power source for the Gemini spacecraft. However, it was adapted for use as an electrolyzer beginning in the early years of this decade. Typical current applications include a spacecraft regenerative life support system (Figure 2) and an oxygen generation system for nuclear submarines, the electrolysis stack for which is shown in Figure 3.

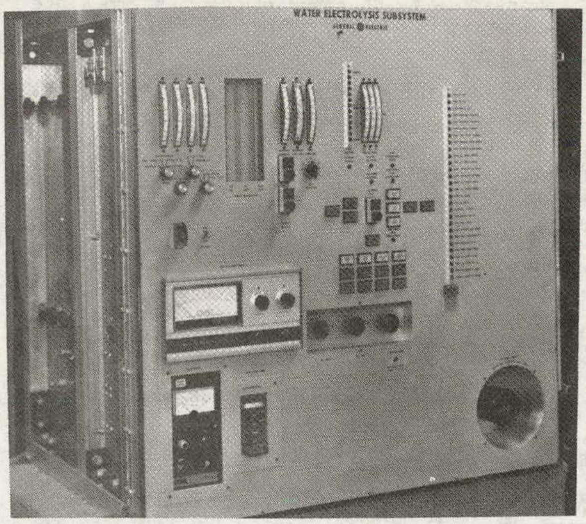

Fig. 2 Oxygen Life Support System for Manned Spacecraft and Space Station Applications

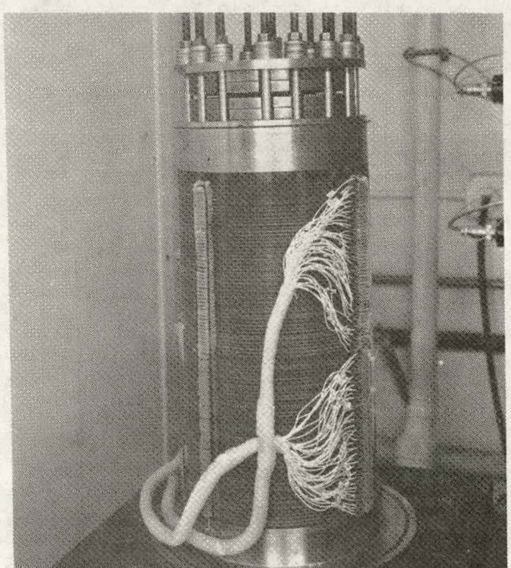

Fig. 3 100 Cell Electrolysis Module for Submarine Life Support

A small commercial unit for generation of pure hydrogen for gas chromatographs and other laboratory uses has been on the market since 1973.

In 1976 a program was initiated to develop a large-scale electrolysis design that will be suitable for bulk hydrogen generation for industrial and utility applications. This program is sponsored jointly by the U.S. Department of Energy, The Niagara Mohawk Power Company, the Empire State Electric Energy Research Corporation, the New York State Energy Research and Development Authority, the Gas Research Institute and the General Electric Company.

The initial phase of this program was a design study of a 58 MW (625,000 SCFH of H_2) system which would be suitable for a number of potential applications including energy storage, generating hydrogen for chemical and industrial feedstock or possibly as a supplement to natural gas in certain areas.

Figure 4 is a model of a typical energy storage plant based on the results of the study. The 58 MW water electrolysis system is shown cut away in the foreground, with power conditioning housed in the rear of the building. To the right are the metal hydride hydrogen storage cylinders. A typical module from a 26 MW air-breathing fuel cell installation for conversion back to electricity is shown to the rear of the building.

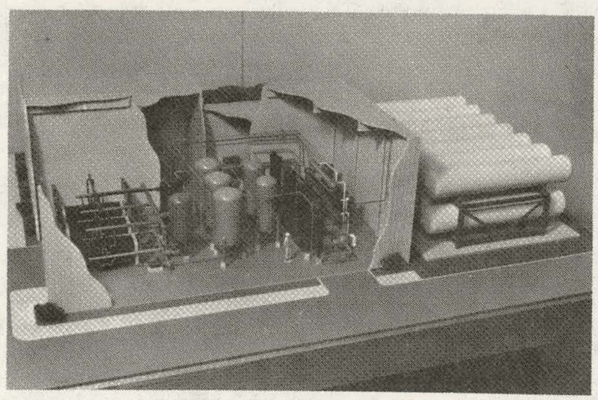

Fig. 4 58 MW-SPE Water Electrolysis Plant

On the basis of the study results, the goals for the electrolysis development program were established as follows:

Overall System Efficiency	85-90%
System Capital Cost (Battery Limits)	\$100/KW
Scale-Up	5 MW Demo System

Technology Development

The efficiency goal for this program requires a low cell operating voltage, and the cost goal requires a high operating current density as well as a low manufacturing cost. The key advantage of the SPE cells is the superior performance which makes the high current density possible at a low cell voltage. Figure 5 shows the current SPE electrolyzer

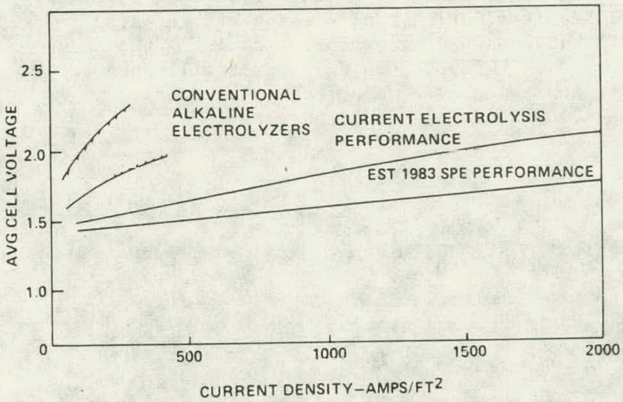

Fig. 5 Comparative Water Electrolysis Performance

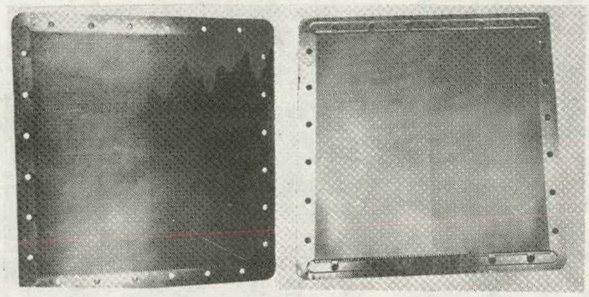

Fig. 6 Molded Current Collector 2 1/2 Ft^2 Active Area

performance compared with that of conventional alkaline units and also shows the improved performance which is expected to be achieved by the completion of this program. The design current density for most of the applications of this technology is around 1000 amps/ft^2.

The other objective of the program is to reduce the manufacturing cost of the SPE cells for commercial application by a factor of about 14:1 from the cost of those used in the space and submarine systems.

To date, the technology development program has resulted in significant progress toward these goals. Compared with a 1975 baseline technology of $202/KW, projected costs for the production electrolyzer based on currently identified materials and techniques is about $45/KW or 80% of the cost reduction required to meet the 1983 goal. The primary cost reductions have been in the areas of current collectors and catalytic electrodes.

Molded carbon current collectors are a major result of the technology development program. The function of the current collector is to:

- Provide the flow fields for the water and oxygen on the anode (oxygen) side and for the water and hydrogen on the cathode (hydrogen) side.
- Separate the oxygen and hydrogen sides.
- Provide for the conduction of electricity from one cell to the next.

The collector is molded from a mixture of carbon and resin which incorporates an in-situ formed titanium foil shield on the anode side to prevent corrosion. Small laboratory-sized collectors have accumulated over 12,000 hours of operational evaluation to date. Figure 6 shows large-sized molded collectors with 2-1/2 ft^2 active area.

A major advantage of the molded collector is the elimination of costly silicone rubber gaskets. The Solid Polymer Electrolyte itself acts as a gasket between the sealing surfaces of the collector. Gasketless sealing has been demonstrated up to 500 psi on laboratory-sized cells, and, in a previous prototype hardware program, up to 400 psi on a 120 cell stack.

Catalytic electrode development has involved the identification of a ternary oxygen evolution catalyst which offers both reduced cost and improved performance in comparison to the current state-of-the-art for aerospace systems.

Figure 7 shows the cost reduction and performance improvements of these new catalysts demonstrated in operational cells. Effort is continuing in both data base testing of currently identified catalyst systems, and identification of additional catalyst systems.

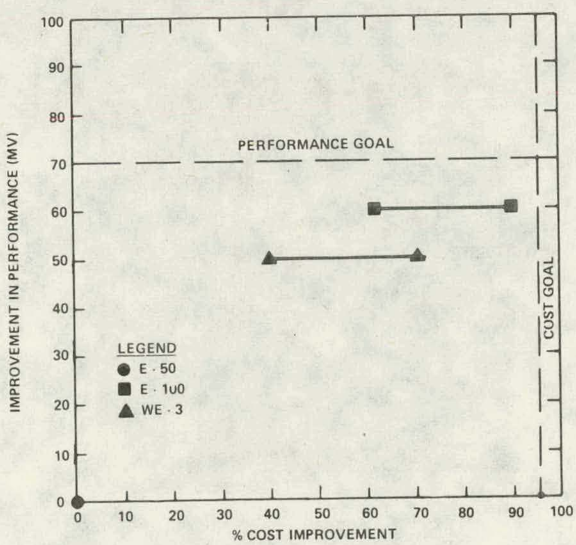

Fig. 7 Anode Catalyst Development

The catalyst identifications shown on this figure are merely G.E. designations. The catalyst compositions are considered proprietary.

The results of reductions in catalyst loading to effect a cost improvement with minimum impact on cell performance have also been encouraging. Techniques for reducing both anode and cathode loadings up to 93% have been identified and shown feasible.

High temperature operation (up to 300°F) offers advantages in operating efficiency and accounts for about one half of the improvement needed to meet the goal performance shown on Figure 6. Over 5000 hours have been demonstrated to date at 300°F using carbon collector components and Nafion® 120.

Alternative solid polymer electrolytes are also being evaluated to assess advantages in cost and performance. The Nafion® 120 is an extremely stable (and thus long-lived) material under water electrolysis operating conditions. Any alternative membrane considered viable must have equivalent life stability. A radiation-grafted trifluorostyrene was extensively evaluated, but demonstrated insufficient operational stability to be considered as a viable alternative. The search for other alternatives is continuing.

Scale-Up

The cell scale-up program is proceeding in parallel with the technology development program. The cell scale-up is planned in two steps: development of a 2 1/2 ft^2 active area cell, which started in 1977, followed by a 10 ft^2 cell development, which has been initiated this year.

Figure 8 shows one of the 2 1/2 ft^2 electrolyte and electrode assemblies which, in conjunction with a molded carbon current collector (shown in Figure 6) and suitable anode and cathode supports, form a bipolar cell. The cells are assembled between pneumatically loaded end plates to form an electrolysis module

Fig. 8 2 1/2 Ft^2 Active Area Membrane and Electrode Assembly

Figure 9 shows one of the many single and multicell modules of 2 1/2 ft^2 cells which have been tested during the last 6 months. Large cell performance, comparable to "baseline" laboratory cell performance, has been demonstrated as shown in Figure 10. Initial testing concentrated on sealing and fluid distribution, and present efforts are aimed at increasing the number of cells in the module. A 50 KW (500 SCFH H_2) module will soon be on test and assembly of a system to accomodate a 200 KW (2100 SCFH H_2) module has started.

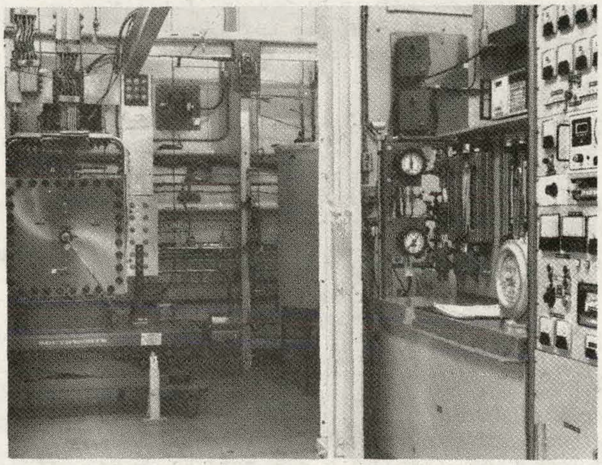

Fig. 9 2 1/2 Ft^2 Electrolyzer Cell Module

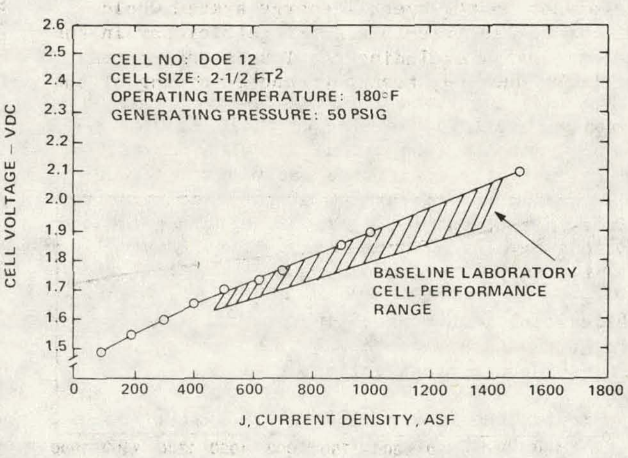

Fig. 10 2 1/2 Ft^2 Cell Test Performance

Future Plans

The planned program calls for operational evaluation of 50 KW, 200 KW and 500 KW systems leading to installation of a 5 MW demonstration system during 1983.

As technology improvements are identified and proven by the technology development effort, they will be incorporated into the hardware program.

As hardware development progresses, field installation of small prototype systems will be initiated.

Projected Production Cost

The calculated system cost from the 1975 58 MW system study was \$82/KW, broken down as follows:

Electrolysis Module	\$14/KW
Power Conversion and control	43
Ancillary components	16
Installation	9

At the present state of technology the estimated production cost of a 58 MW electrolysis module is approximately \$45/KW compared to the \$14/KW projection.

The program goal of \$100/KW allowed for some difficulties in achieving all of the calculated cost bogies, and, with today's technology, it appears that the capital cost for a 58 KW system could be approximately \$118/KW compared with the \$100/KW goal.

For smaller installations the cost per KW for the electrolysis system will increase. Figure 12 also shows the estimated cost as a function of the capacity of the system for lower capacity units.

SPE Fuel Cell Status

Current development effort on SPE hydrogen/oxygen fuel cells is directed toward future space power applications, and significant advancements in performance have been made as shown on Figure 11. Even so, however, the highest practical efficiency for a H_2/O_2 fuel cell is probably going to be in the range of 50%. Combining this with a 85-90% efficient electrolyzer in the overall energy system would result in an electric-to-electric efficiency in the range of 40-45%, excluding the losses associated with the production, transport and reforming of the ammonia.

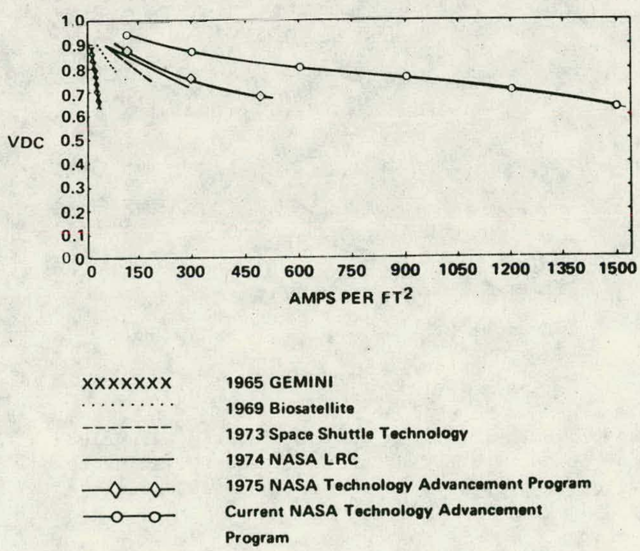

Fig. 11 Fuel Cell Performance History

During 1976-1978, G.E. conducted some studies and feasibility testing for the Brookhaven National Laboratory regarding the use of chlorine or bromine instead of oxygen in a regenerative fuel cell to achieve higher efficiencies in electrical energy storage applications. The results of this effort showed that electric-to-electric efficiencies in the range of 70-80% are possible, and that the technology currently exists to develop such a system.

The performance demonstrated in the preliminary feasibility tests is shown in Figure 12. Approximately 75% voltage efficiency was achieved at a current density of 300 amps/ft^2 with both H_2/Cl_2 and with H_2/Br_2. It is estimated that, with a moderate program for cell optimization, this level of efficiency could be extended out to 500-600 ASF as shown on Figure 13. The laboratory sized cells used for this testing were an adaptation of the design being developed for commercial HCl electrolysis applications under a privately funded development program. Such electrolysis cells have demonstrated thousands of hours of stable performance and life to date. Figure 14 shows a large scale module of this type of test.

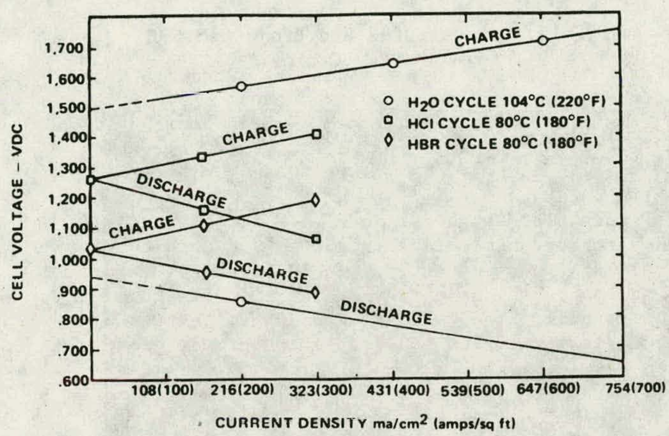

Fig. 12 Demonstrated SPE Regenerative Fuel Cell Performance

In order to take advantage of this higher efficiency in an OTEC energy system, however, it would be necessary to transport the chlorine (or bromine) to the load center, as well as the ammonia (hydrogen), and then transport the resulting acid back to the OTEC plant.

As a compromise, the possibility of combining a H_2Cl_2 fuel cell with the Kelchlor (or Deacon) process to recover the chlorine from HCl is being considered. Such a system, shown schematically in Figure 15, could theoretically approach the high efficiency of the H_2/Cl_2 fuel cell without having to transport the chlorine of HCl. In effect, the Kelchlor process would become a redox cycle in an overall H_2/O_2 fuel cell system. The reactions would be as follows:

Fuel Cell: $H_2 + Cl_2 \longrightarrow 2HCl$

Redox: $2HCl + 1/2\ O_2 \longrightarrow Cl_2 + H_2O$

Overall: $H_2 + 1/2\ O_2 \longrightarrow H_2O$

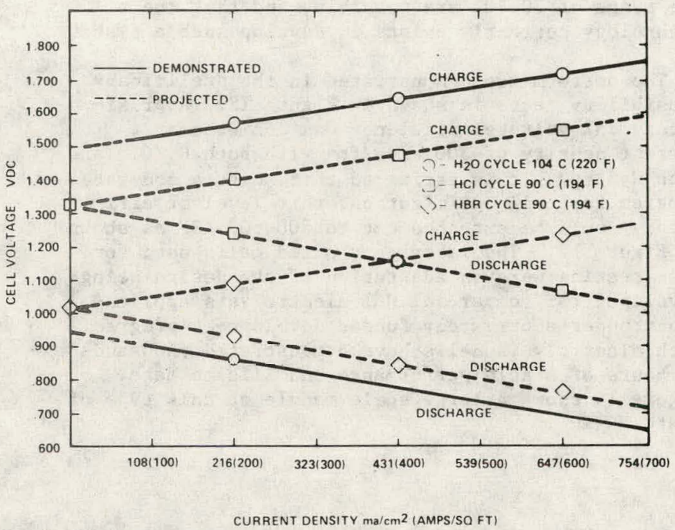

Fig. 13 Demonstrated and Projected SPE Regenerative Cycles

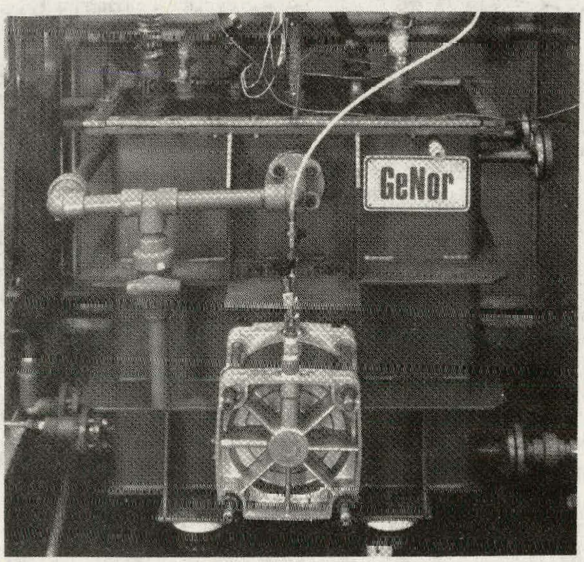

Fig. 14 Photo of GENOR Cell

The Kelchlor process is catalytic and under the proper conditions, proceeds spontaneously without any net energy input other than the requirement for pumps and other ancillaries. However, a preliminary analysis indicates that the energy required to concentrate the HCl from the level produced in the fuel cell to that required by the Kelchlor process may be sufficient to offset much of the gain resulting from the higher fuel cell efficiency. A more detailed study is required to fully assess the technical and economic potential for such a system.

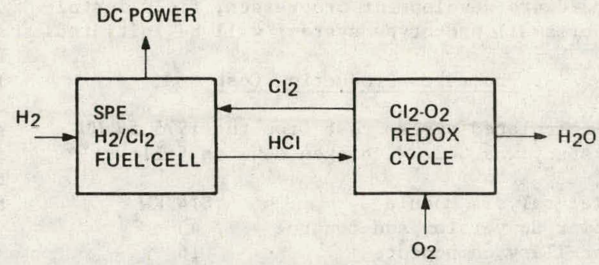

Fig. 15 $H_2/Cl_2/O_2$ Redox Fuel Cell Schematic

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LARGE ALKALINE ELECTROLYSIS SYSTEMS FOR OTEC

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Abstract

Tropical siting of Ocean Thermal Energy Conversion (OTEC) plants where the variations in winds, waves, and currents are small and where the available temperature differences are much higher essentially precludes the direct transport of electrical power to land. Of the alternate approaches for transporting OTEC energy to land, several involve the production of hydrogen, via water electrolysis. Teledyne Energy Systems is presently manufacturing two series of alkaline electrolysis systems specifically tailored for the specialty gas market. The largest unit in production produces 420 standard cubic feet per hour (50 lb per day) at pressures up to 100 psig. Special high pressure systems have been developed that generate gas directly at pressures up to 3000 psig, however these high pressure systems have proven to be very expensive. To meet the requirements of large users, Teledyne Energy Systems is now developing a commercial electrolysis system capable of producing several hundred pounds of hydrogen a day. Although this system is also being designed for specialty process gas applications, the system is large enough to satisfy OTEC applications up to several hundred megawatts. This paper discusses the optimization of large alkaline electrolytic hydrogen plants to minimize product cost, specifically the influence of operating pressure, since the transport of hydrogen or its products involves high pressures.

Introduction

Hydrogen has been produced commercially by the electrolysis of water since the turn of the century, shortly after it was first put to practice. Sales however have been confined for the most part to the small high-purity-gas market, due to the high cost of electricity relative to natural gas and oil. The only situations where large electrolysis plants could be justified, in the past, were those where the cost of electricity was exceptionally low as with hydroelectric installations. The market for electrolytic hydrogen has been expanding recently because of the relative rise in the costs of gas and oil with respect to electricity. Although this expansion is still taking place in the commodity gas market, it is probable that electrolytic hydrogen will begin to find application in selected energy storage and transportation schemes. The more immediate energy market possibilities are likely to occur in conjunction with the renewable resources, i.e., direct solar, wind, geothermal, and, of course, ocean thermal energy conversion.

The degree to which electrolytic hydrogen can be expected to make a meaningful contribution to future scenarios will depend to a great extent on how much generating costs can be reduced. A significant reduction will only be achieved through a combination of improved operating efficiency and a reduction in capital investment costs.

In the past, there has been little incentive on the part of industry to improve the efficiency of alkaline electrolysis systems, because of the artifically low cost of fossil fuels. At 60 to 70%, the process is already highly efficient by comparison to most other industrial processes. In 1973, however, with the enthusiasm for an economy based on hydrogen and a desire to scale up, Teledyne Energy Systems initiated a program to: (1) develop larger systems, and (2) improve system efficiencies.

Technology Scale-Up

The company funded effort to scale-up the commercial technology from the Kilowatt range (HS) to the Megawatt range (HP) involves an increase in active area of eight-fold, i.e., one square foot to 8 square feet.

A prototype version of the plant was tested early last year (Fig. 1). The prototype consists of a module with four full size cells and a complete set of subsystems. This first generation module utilized "state-of-art" HS concepts, i.e., asbestos separators and nickel screen electrodes. Total thickness of the four cells is about one-inch or one-quarter of an inch per cell. The massive looking positive endplate (with lifting lugs) is actually one-inch thick stainless steel plate stiffened by 12 inch "I" beams. This rigidity is necessary to insure the large seals have a uniform sealing load applied and to insure that the asbestos electrode separators are uniformly compressed. The negative endplate (hidden) is identical to the positive, except for the two fluid inlet and two fluid outlet connections. To the right of the module is the D.C. conversion equipment made up of a choke and a water cooled rectifier/SCR package for each of the three phases. The power supply is capable of delivering up to 2500 amperes at 10 volts D.C. To the left is the system control panel. All critical system functions are monitored and controlled from this location, including electrolytic temperatures, system pressure, differential pressure, pump loading, module current, module voltage, and gas contamination. Integral to this control and monitoring scheme are the system safety circuits for automatic shutdown in the event of a system abnormality. The electrolyte/gas separating tanks and handling plumbing is located above the module. These tanks run horizontally, to provide more fluid surface area for better gas bubble separation.

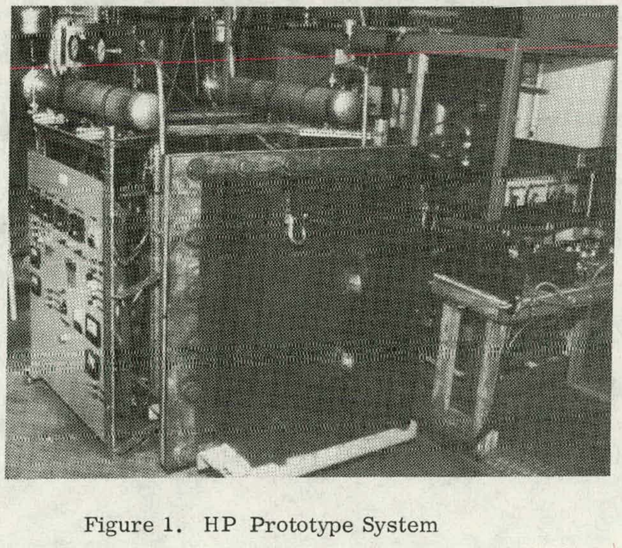

Figure 1. HP Prototype System

Earlier analyses indicated that for large low pressure plants, it was economically prudent to develop a cell with the largest cross sectional area possible, with a square configuration to minimize waste, since most cell materials are purchased as 48-inch wide sheet stock. As a result, a prototype cell was designed that was four feet square with an active area of approximately 8-1/4 square feet. Figure 2 shows one of the cell frames being manufactured. While these prototype cell frames were all individually machined out of glass filled polypropylene, it is intended to use molded frames in the final plant design; a necessary element in the design of a low cost, high production module. Figure 3 shows the module stack assembled and ready to be covered with the negative endplate. The larger center cutout area is the active cell area. The manifold ports, two for the hydrogen side inlet and outlet, and two for the oxygen side inlet and outlet, are at each corner.

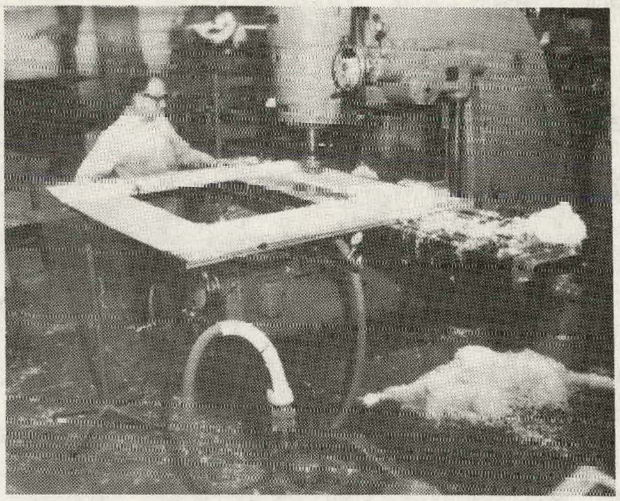

Figure 2. HP Prototype Cell Frame Manufacturing

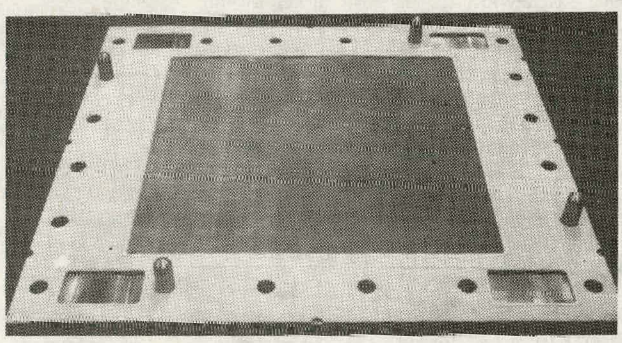

Figure 3. HP Prototype Module Assembly

The prototype plant was operated for about 50 hours during which the rig was shaken down and the first cell design was evaluated. The cell is now being re-designed to improve producibility and to correct for higher than expected current losses.

Improved Efficiency

Company funded efforts to reduce cell voltage and thus improve the process efficiency have been directed primarily toward the development of electrodes with greater catalytic ability and the raising of process temperatures from the present 80°C range to 125°C or above. Figure 4 (1) shows the 1979 performance projections. The upper curve shows the general performance level of the HS equipment currently being sold. The intermediate curve reflects the performance predicted for the first HP type plant, at 80°C, but with improved electrodes. Goals for the advanced high temperature system are shown as a band, where the upper boundary represents the expected performance at 125°C. Above 125°C; probably in the area of 150°C, it was and still is expected that corrosion problems will start to seriously influence the system cost and useful life, making higher temperatures less cost effective. (This upper limit is yet to be established.)

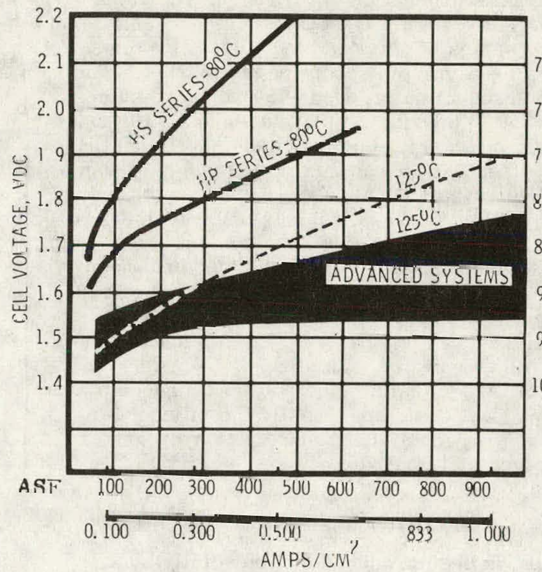

Figure 4. 1979 Performance Update

A key to successful operation above 100°C for extended periods is the identification of an electrode separator to take the place of asbestos. In 1975, Brookhaven National Laboratories (BNL), under the auspices of the Department of Energy, began to support the Teledyne effort to improve the efficiency of alkaline systems, by funding a modest program to evaluate several thermoplastic polymers as possible substitutes for asbestos. After an initial screening of five candidates, of which two showed promise, the program was re-oriented toward the building of an engineering test rig that would allow the evaluation of advanced separators and electrodes in a system representative of industrial equipment. As a result, the test rig shown in Fig. 5 was completed in 1976. Dubbed ARIES for Applied Research Industrial Electrolysis System, the system is a complete gas generating system, capable of operating automatically for extended periods, allowing studies of materials corrosion problems and cell voltage as a function of current density, temperature, pressure, and time. The system is designed to test a five cell bipolar stack at temperatures up to 150°C, current densities up to 2500 ma/cm^2, and pressures up to 100 psig.

To date, the rig has been used primarily to screen improved electrodes as mutually selected by BNL and TES. Electrode concepts tested so far have come from the U.S. (Teledyne), Great Brittain and Germany.

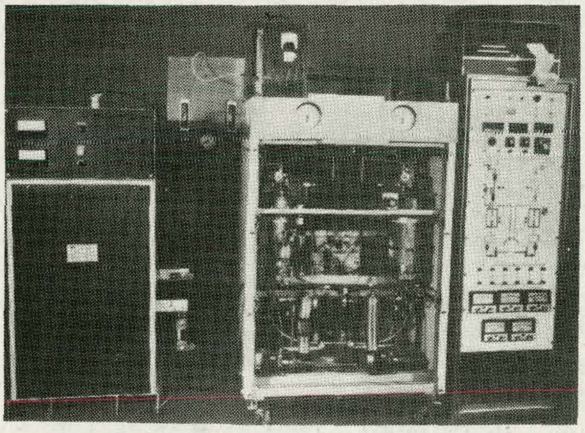

Figure 5. ARIES

The dashed curve shown in Fig. 4 illustrates the performance of the best anode and cathode combination tested to date. Cathode development has proceeded pretty much as expected. The best cathode tested to date is a proprietary Teledyne Electrolcatalyst-electrode, C-110. The best anode tested is cobalt nickel oxide on a nickel substrate by Tseung (2) however, none of the anodes evaluated so far have lived up to expectations. Even the cobalt nickel oxide anode does not appear to be cost effective, when compared to the performance of present commercial anodes.

This work will be continued for the balance of this fiscal year. Additional components will be evaluated as they are identified and a module made up with the best cathode and a teflon bonded nickel anode will be tested for up to 3000 hours at 125°C to determine stability. Company funded work will continue to be directed toward electrode optimization and separator development, with particular emphasis on the anode and separator.

Analytical Work

For the past several years, Teledyne has been developing and expanding a computer program to predict the cost of hydrogen for various electrolysis plant configurations. The original 1973 program (3) was relatively simple, written around state-of-the art equipment to examine the general effect of capital and operating costs on the cost of hydrogen. Figure 6 illustrates the trade-off which results. The first factor, capital, tends to push the design current density to the highest possible level, at the expense of lower operating efficiencies. The cost of electricity tends to push in the other direction. Basic alkaline cell materials historically cost less than SPE materials. Consequently, alkaline systems always optimize out at current density levels significantly lower than SPE systems, (500 ma/cm^2 versus 1000 ma/cm^2).

Since high pressures are generally desirable for storage, recent work has been oriented toward attempting to evaluate the cost impact of operating at higher pressures versus the cost of external compression. Unfortunately that portion of the program is still being developed and data is not available for presentation here.

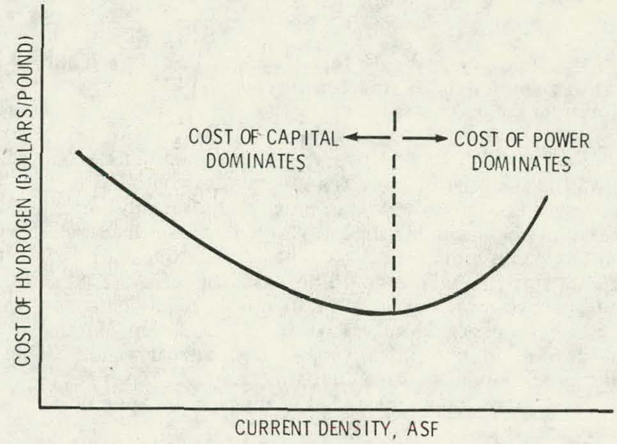

Figure 6. System Cost Optimization

As an additional task to be undertaken as a part of the BNL contract, the program will be further expanded to allow evaluation of the advanced separator and electrode options and the influence of higher temperature operations. This work will begin in October and should be completed by January of next year.

Projections for OTEC Plants

Although a hydrogen economy is not going to develop to the extent predicted by Gregory(4) in 1973, hydrogen will be used extensively in the future to compliment the other clean and versatile secondary energy carrier: electricity. First applications will probably be in conjunction with renewable resources, and deep water - tropical OTEC would be an excellent demonstration forum.

In late 1975, Teledyne Energy Systems provided the Applied Physics Laboratory (APL) with conceptual data on a 100 MWe hydrogen plant for a 1980 OTEC demonstration ship. The configuration suggested at that time consisted of six 16.13 MWe unit plants as shown in the first column of Table 1. In this case, each unit plant was made up of five 3.2 MWe building block modules, where a building block module is defined as the largest module that can practically be built. In 1975, this was considered to be a module with five hundred 8.25 square foot cells similar to that shown in Fig. 7. The modules seen here are each

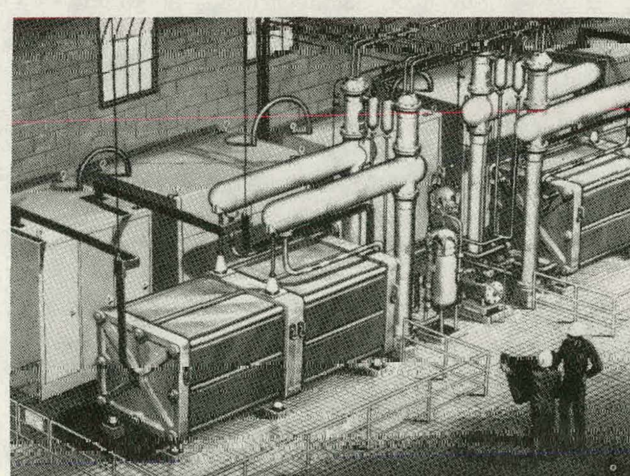

Figure 7. HP System (Artist's Concept)

about 4 x 4 feet square by 12 feet long. A 100 MWe plant with 30 of these modules and auxiliaries would occupy roughly 4000 square feet.

Recently, APL has proposed a 40 MWe demonstration ship for the 1983-4 time frame and Teledyne has again been asked to provide an input, and although some progress has been made since 1975, it is difficult to see how an advanced multimegawatt alkaline system with significant improvements could be ready by 1983/84, without a substantial increase in support. A modest program has been recently submitted to the DOE that calls for the development of an advanced high performance building block module within that time frame. However, the program could be accelerated to make a 40 MWe unit available.

Table 1 1975 OTEC Input

	Specification/Goals	
Item	1980	1983/84
Unit module H_2 capacity, SCFH	30,000	10,000 to 30,000
Number of cells per module	500	< 500
Active cell area, ft^2	8.25	> 8.25
Current density, ma/cm^2	500	~ 500
Voltage efficiency, %	85.6	90
Current efficiency, %	98.0	98
Overall system efficiency, %	83.8	88
Gas purity, %	98	98
Operating pressure, psig	100	100
Plant cost, $\$/KW_{in}$ (1975 \$)	142	100

The second column of Table 1 summarizes the current thinking of Teledyne with respect to size, cost and efficiency for an advanced multimegawatt building block module. Rated capacity will probably be between 52,000 and 156,000 SCFH, depending upon the final number of cells per module, cross-sectional area, and efficiency. This represents an input power level of 1 to 3 MWe per module. The module will probably have less than 500 cells, primarily because of handling and assembly limitations, and it would appear that the cell will optimize out at an active area substantially above the 8 to 9 square foot (1 square meter) size. The 8 to 9 square foot cell may be a suitable match for small applications such as wind or low head hydroelectric, however for multimegawatt applications such as OTEC or dedicated nuclear, a cell on the order of 50 square feet (5 square meters) will probably be more in order. System pressure should be at least 100 psig, however as module size increases the capital cost penalty of electrolytic pressurization increases. Finally, we believe that large multimegawatt electrolysis plants can be built for less than \$100 per kilowatt in (1975) quantity. This also requires the detailed definition of a new low cost cell concept now under consideration.

Alkaline electrolyzers offer a proven hydrogen generation technology that could be substantially improved both in terms of efficiency and capital costs. To date, however, there has been only token support for this attractive option. In fact, this year only about four man years of development will be supported by the U.S. Government and that will be divided between the government labs, universities, and industry. This is comparable to the level being supported by the Canadians and Japanese, but far less than the Europeans at 20 man years per year. If a large advanced electrolysis plant is to be available by 1990, the U.S. development effort must be accelerated, otherwise we will be buying European technology.

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INTEGRATION ISSUES OF OTEC TECHNOLOGY TO THE AMERICAN ALUMINUM INDUSTRY*

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Abstract

The technical and economic factors associated with utilization of OTEC energy by the aluminum industry are examined. Three scenarios are considered: (a) operation of an aluminum reduction plant on shore fed energy from the OTEC platform by cable or pipeline; (b) operation of a shore-based reduction plant powered with OTEC energy delivered by means of energy bridges; and (c) an OTEC/aluminum plantship operating at sea in a grazing mode.

The conventional Hall reduction process for converting alumina (Al_2O_3) to pure aluminum metal is considered as the baseline case. ~~Internal shorting of the Hall cell by the liquid metal as a result of cell motion is identified as a major problem area for floating applications. Preliminary indications are that a semisubmersible stabilized platform may provide the prerequisite stability.~~

Alternate reduction processes which may be less susceptible to cell motion in the floating environment are being examined. These include the aluminum chloride smelting process and the drained cathode Hall cell. ~~These processes may also require less energy for the reduction process and require less space on the floating OTEC/aluminum plantship.~~

One important technical issue for the floating reduction plant is the recovery of waste heat from the process cells and the use of this heat in a thermodynamic cycle to generate additional electrical energy, using water from the OTEC plant cold water pipe for condensation of the secondary working fluid.

The overall goal of the study is to determine the conditions under which aluminum can be produced competitively using OTEC-derived energy, and to provide an assessment of the risks associated with such a venture.

~~Introduction~~

Several authors have suggested that one potential method for economically using OTEC energy is in the production of aluminum metal from alumina.[1,2] The industry-average electrical energy consumption for the reduction of 1.89 lbm of alumina to one lbm of aluminum metal is about 8.0 kwhr of electrical energy. The non-communist world production of aluminum is about 11 million tons per year (1978) and is expected to grow at a rate of between 6 to 8% per year for the next decade. This projected increase in alumina reduction capacity will require between 1000 MW and 2000 MW of new electrical generation capacity per year if this growth rate is to be maintained. The use of OTEC energy to provide some of the additional capacity required for the aluminum industry could have a significant impact on the world's energy economy.

The objective of this study is to examine the technical and economic feasibility of using OTEC energy for the commercial production of aluminum metal.

Background

The production of aluminum from bauxite, its primary ore, involves several energy intensive steps. Most of the world's aluminum metal is produced by a combination of the Bayer process and the Hall process, although there are other processes being actively investigated. In the Bayer process, alumina (Al_2O_3) is chemically separated from bauxite, which is generally a mixture of aluminum, silicon, iron, and titanium oxides. In the Hall process, the alumina is dissolved in cryolite, then electrolytically reduced in the presence of carbon to form aluminum metal and carbon oxides, which are vented to the atmosphere.

Traditionally in the United States, the Bayer process and calcinization are carried out in regions close to the bauxite supply where low cost thermal energy (natural gas) is available, and most of the alumina is shipped to regions with low cost electrical energy (hydroelectric) where the Hall process is carried out. The electrical energy requirements of the reduction process suggest that it may be a candidate for OTEC commercialization.

OTEC Utilization Modes

There are three potential ways in which OTEC energy can be used by the aluminum industry where the most obvious application is powering a reduction plant. The first is to have the OTEC power plant moored at sea just off the coast in deep water, sending the energy to shore by means of cables or pipelines. This energy would then be

*This work is sponsored by the U. S. Department of Energy, Contract No. EX-78-C-02-5091.

used in a reduction plant or other facility at or near the shore line. The second method is to send the energy ashore by means of an "energy bridge."[3] In this case, some intermediate chemical such as hydrogen is produced on the OTEC platform using the generated electrical power and is then transported to a shore site where the chemical energy is transformed back into electrical energy. The third method is to put the reduction plant on a floating platform integrated with the OTEC power plant, and operate at sea in those regions with the highest thermodynamic potential (largest temperature difference). In the first two applications where the reduction plant is located on the shore, the OTEC plant must compete with other sources of electrical power which may be available.

The floating OTEC/aluminum plant has the greatest potential for achieving the lowest energy costs inasmuch as the capital cost of the OTEC power plant varies as $(\Delta T)^{-1.8}$.[4] Therefore, locating the OTEC/aluminum plants in those areas of the ocean which have the highest temperature differential should lead to the lowest energy costs. Such plants have been described in OTEC literature as "grazing" plants and preferred sites in the equatorial Atlantic and Pacific regions have been identified. In terms of the aluminum industry, the thermal resource available in these equatorial regions is very large. There are roughly 50 million square miles (120×10^6 km^2) of ocean surface where the depth is greater than 1000 m and the monthly average difference in temperature between the surface and the 1000 m depth is greater than 20°C.

For those options where the Hall process reduction plant is on the shore, it should be noted that the electrolytic processes use high amperage, low voltage, direct current; typically, 100,000 to 250,000 amperes at a few hundred volts. Therefore, any of the energy transportation systems which result in direct current at the voltage and impedance levels required by the reduction process should be more competitive in terms of capital cost and efficiency. This means that electrochemical systems such as fuel cells or batteries would be the preferred systems for powering the shore-based reduction plant with energy from the OTEC platform.

For the floating OTEC/aluminum plantship, it is anticipated that the electrical power would be generated as direct current, presumably using acyclic generators. It is also anticipated that low capital cost standby generation capacity, presumably gas turbine powered, would be provided to allow the reduction plant to be shut down in an orderly fashion in case of a forced outage of the OTEC power system, or for use in the event of natural phenomenon such as a storm which might reduce the OTEC generation capacity. Since the reduction plant operates continuously for 365 days per year, with a load factor which approaches unity, spare OTEC generating capacity must be provided to insure that name plate electrical capacity is available at all times, such as when the OTEC electrical generation equipment is down for routine repairs or maintenance or when heat transfer surfaces are being cleaned, etc. While it may be possible to have a short-term turndown of the power to the OTEC reduction plant, economics favor high utilization of the capital intensive facilities.

Candidate Reduction Processes

Previous studies[1,2] have assumed that operation of the OTEC/aluminum plantship would involve the reduction of alumina to aluminum using the conventional Hall cell. The cell consists of a carbon lined cathode and a consumable carbon anode. Alumina dissolved in cryolite is placed in the cavity formed by the cathode and an electrical current is passed from the cathode to the anode, producing carbon oxide gases at the anode surface which bubble off. The aluminum metal is deposited on the cathode and forms a pool of metal (called a pad) on the cathode where it collects until it is periodically removed. One of the major technical problems associated with the use of a conventional Hall cell in a floating plant environment is that platform motion may cause the molten metal pad to move so as to contact the anode, shorting the anode to the cathode. Once the short has formed, a magnetic pinch effect holds the liquid metal in contact with the anode. Continued operation with an anode-cathode short reduces the cell efficiency.

There are two options for overcoming this problem. The first is to stabilize either the platform or the cell, and the other is to use another reduction process which is more tolerant of motion.

Several methods of stabilizing the cells or the platform have been studied. Mechanical stabilization of the cells against rolling, pitching, and heaving motion would be an extremely difficult mechanical problem made doubly difficult because of the problems associated with providing flexible connectors for the fractional mega-ampere bus bars which connect each cell. There may be as many as 200 cells to be stabilized on a given plantship. There are several methods available for stabilizing the platform from pitch and roll. However, studies show that even if the cell or the platform could be stabilized against pitch and roll, the Hall cell would still be susceptible to shorting through heaving motion. It was found that dynamic methods for stabilizing floating platforms against pitch and roll generally result in increased heaving of the platform.

For this reason, one is led to the conclusion that other reduction processes may be more suitable for use in the OTEC plantship environment. Alternate processes which may be commercially available within the time frame that OTEC is commercialized are:

1. Hall cell with drained cathode.
2. Aluminum chloride smelting process.

The drained cathode Hall cell which has previously been described in the patent literature[5] is similar to the conventional Hall cell, with the exception that a special cathodic material wettable by molten aluminum covers the carbon-lined bottom of the cell. This means that a metal pad is not required, so that the molten metal can be drained from the area between the electrodes as it is produced. The absence of a metal pad implies that this type of cell should be much less susceptible to shorting as a result of motion, either horizontally or vertically.

In the aluminum chloride smelting process, alumina is converted to aluminum trichloride by reacting Al_2O_3 with Cl_2 in the presence of carbon,[6] releasing carbon oxides. A bipolar reduction cell is used to reduce the aluminum trichloride to chlorine gas and molten aluminum.[7] The chlorine is recovered and recycled. This process has three advantages for OTEC application: (a) the bipolar cells should be much more tolerant to platform motion; (b) they are reported to be more efficient in the use of energy; and (c) the bipolar nature of the electrodes means that they can be stacked one above the other to reduce the floor area required for the reduction process on the floating platform which is an important consideration for the OTEC/aluminum plantship application. It is contemplated that the conversion of alumina to aluminum trichloride would be accomplished on shore, that the $AlCl_3$ would be transported by ship to the OTEC/aluminum plantship for reduction, and that the chlorine would be recovered and transported back to the shore base.

OTEC/Aluminum Plantship System Configurations

As an example of the possible synergism between OTEC and aluminum production, the probable configuration of an integrated OTEC/aluminum plantship using the conventional Hall process is examined. As previously detailed, the conventional Hall process may not be suitable directly for use in a floating reduction plant because of the cell shorting problems due to motion. However, since the drained cathode cell technology is similar to the conventional Hall cell, it is worthwhile to examine the application of conventional technology to the OTEC environment to establish a cost and technology baseline which industry decision makers understand, and then present the other alternatives as perturbations to this baseline.

For this application, we will consider the aluminum production process divided into distinct phases shown in Figure 1. The floating plantship, which contains the OTEC power subsystem, is supported by a land base which provides the materials and personnel to support the floating reduction plant. Operations such as anode manufacture, cell refurbishment, etc., are carried out at the shore

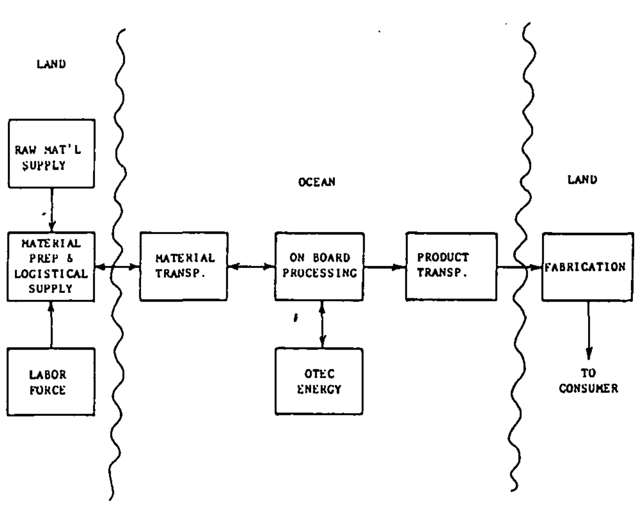

Fig. 1 OTEC/Aluminum Production Model

support base. The shore based functions include the provision of alumina, although the conversion of bauxite to alumina may or may not be done at the plantship support base. For the purpose of our studies, we have assumed that alumina was available from commercial sources. A transportation system brings the raw materials to the floating plantship and provides the required logistic support for the combined reduction plant and the OTEC power plant. Another transportation system is provided to take the product metal to shore locations for fabrication into commercial products. It is assumed that the product will be ingots of 99.5% pure metal. The cost of producing 99.5% pure metal in ingot form is generally the recognized benchmark for evaluation of new investment in capacity.

At this point, we have assumed that the OTEC power system and the reduction plant will be integrated into one hull. The reason for making this assumption is that the Hall reduction process is relatively inefficient. Only about 40% of the electrical energy is used in the reduction reaction and about 60% of the input energy appears as heat in the cell which must be dissipated to the environment. Since this heat is potentially available at temperatures of up to 300°C, it appears desirable to install energy recovery equipment on the Hall cells to recover this heat and to use it in a secondary thermodynamic cycle to generate additional electrical power. Assuming that 80% of the waste heat could be recovered, and that a Rankine cycle rejecting the heat to 4°C water brought up through the cold water pipe could achieve 65% efficiency, it is estimated that about 15% of the waste heat could be recovered as usable electricity. This may compensate for the additional OTEC generating capacity which must be provided to meet the 100% load factor requirement, but with lower cost generating equipment. The requirement for this secondary thermodynamic cycle to have the 4°C water from the cold water pipe provides a reason for the aluminum reduction plant and the OTEC power plant to be integrated on the same hull; however, it may be found that having the two systems on separate hulls may provide operational flexibility which would overcome the thermodynamic penalty of using the warm surface water to cool the condensers of the secondary thermodynamic cycle.

As a basis for analysis, we have assumed that the baseline plantship would have a production capacity of 70,000 metric tons per year of aluminum metal which would require about 120 MW_e of generating capacity. This size was chosen as representative of a single potline of 160 to 180 Hall cells, which is more or less the industry's building block for expansion. This annual capacity corresponds to an aluminum output mass flow of 2.2 kg/sec.

Based on this mass flow, a number of parametric studies have been performed varying the following parameters: (a) hull configuration; (b) reduction plant configuration; and (c) hull construction materials.

Parametric studies yet to be completed cover:

1. Hall cell anode current density variation.

2. The presence or absence of a heat recovery cycle.

3. Other candidate reduction processes.

4. Changes in the baseline plant mass flow or annual capacity.

Table 1 gives data on the baseline plantship where the hull configuration and reduction plant configuration were varied. In Concepts 1 and 2, the hull configuration is a rectangular barge; in Concepts 3 and 4, the hull configuration is a semisubmersible with the reduction plant on the platform; while Concept 5 is a circular barge design. Concepts 1 and 3 are for a single deck reduction plant, while Concepts 2, 4, and 5 are for a double deck reduction plant. For all cases, the reduction plant anode current density was assumed 7.0 kA/m^2. Under these assumptions, the hull cost is the major figure of merit on which to rate the various options since the cost of the OTEC power system and the reduction plant components are the same. Figure 2 shows an artist's conception of the single deck reduction plant on a semisubmersible hull, Concept 3.

For the plant layouts examined, one immediate conclusion is that, when using the Hall process, the reduction plant characteristics completely dominate the physical size of the hull, with the OTEC power system representing only a minor fraction of the hull space. When comparing the single deck versus the double deck configurations, the double deck configuration shows about a 25% lower cost. However, plant operators, fearful of the consequences of cell bottom "burn through," prefer the single deck.

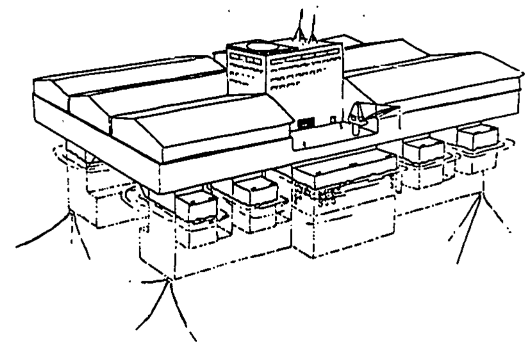

Fig. 2 Single Deck Reduction Plant on Semisubmersible Hull

Reinforced concrete hulls appear to have a cost advantage over steel hulls, in addition to a potential freedom from the problems of circulating currents when the magnetic fields associated with the fractional mega-ampere currents in the Hall process are changed. This will require care in construction to insure that the steel reinforcing bars in the appropriate sections of the concrete hulls do not provide continuous paths for circulating currents.

From Table 1 it may be seen that the semisubmersible hull design results in almost a factor of two increase in the pitch, heave, and roll period of the platform. This increase in period, when compared with the design hurricane wave spectra

Table 1 Summary Characteristics of 70,000 MT/Yr OTEC/Aluminum Plantships Using Hall Process Reduction Cells with Anode Current Density of 7.0 kA/m^2

	CONCEPT #1	CONCEPT #2	CONCEPT #3	CONCEPT #4	CONCEPT #5
Principal Dimensions	Single Dk. Barge	Doubld Dk. Barge	Single Dk. Semisub	Double Dk. Semisub	Circular Barge
Length (ft)	1,000	714	900	480	Diameter =
Breadth (ft)	440	440	440	440	430 ft
Depth to Mn Dk (ft)	90	90	212	212	
Transv. Lower Hull (LxBxH)	None	None	(1)-200x440x82	(1)-200x440x82	Height =
Longl. Lower Hulls (LxBxH)	None	None	(4)-300x100x52	(4)-140x100x52	182 ft to
Vertical Columns (LxBxH) (LxBxH)	None	None	(2)-200x100x100 (8)-75x75x130	(2)-200x100x100 (4)-70x75x130	storage deck
Operating Conditions					
Draft (ft)	56.0	56.0	132.0	132.0	132.0
Displacement (lt)	706,306	436,096	506,971	357,028	547,687
Fixed Ballast (cy)	65,185	None	None	None	
Vert. Center of Gravity (ft)	28.65	44.06	87.56	97.07	111.83
Vert. Center of Buoyancy (ft)	29.05	28.49	53.51	54.93	66.0
Transv. Metacentric Height(ft)	307.84	303.93	53.58	46.39	38.95
Longl. Metacentric Height (ft)	1,930.0	811.2	194.97	37.09	
Transv. Gyradius (ft)	194.0	176.0	152.0	147.0	118.0
Longl. Gyradius (ft)	328.0	214.0	201.0	88.0	
Roll Period (sec)	12.25	11.2	23.0	24.0	21.0
Pitch Period (sec)	8.27	8.33	16.0	16.0	
Heave Period (sec)	8.18	8.0	16.0	15.8	13.0
Stability					
Required Transv. Metacentric Height(ft)		2.16		2.05	1.35
Required Longl. Metacentric Height (ft)		1.71		1.99	
Dynamical Stability Transv.		204 Ft-Deg		6.1	A+B/B+C = 8.4
Dynamical Stability Longl.		209 Ft-Deg		4.96	
Hull Cost					
All Steel (Millions)	288.3	203.9	279	215.9	
Concrete/Steel (Millions)	229.6	186.6	244.3	195.8	230.63

for deep water, which peaks with a period of about 13 to 14 sec, predicts a marked degree of platform stabilization for the semisubmersible when compared to the rectangular barge hull configuration. The circular platform periods are between the rectangular barge platform and the semisubmersible platform. We may note that the projected cost differential between the rectangular barge hull and the semisubmersible hull is about 5%. It may be possible that this small difference in hull cost may be made up by the continued production of a plant on the stabilized platform, if it can operate at full capacity during any weather. The greater stability of the semisubmersible platform will favor safety in the molten metal operations such as casting which occur after the reduction process.

Inasmuch as the cost of the hull to support the floating reduction plant and the OTEC power system is a major element of the overall system cost, it appears appropriate to utilize techniques which can reduce the physical size of the reduction plant. These include increasing the anode current density, which will reduce the number of cells required for a given annual production, and hence the weight and volume of the reduction plant (but at the expense of energy efficiency), or using other processes such as the aluminum chloride smelting process, which due to the nature of the bipolar cells can be arranged to require substantially less floor area, and consequently a much smaller sized hull.

Conclusions

The OTEC thermal resource is substantial in terms of the aluminum industries' future requirements. The grazing OTEC/aluminum plantship offers the lowest potential energy costs. The conventional Hall reduction process has several technical and economic factors which mitigate against its use in the floating OTEC/aluminum plantship. Preliminary investigations suggest that other reduction processes such as the aluminum chloride smelting process which are more energy efficient and which can be made much more compact and less susceptible to difficulties due to platform motion are more favorable for the floating OTEC/reduction plant.

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DISCUSSION

J. Coyle, Navy: Speaking as a metallurgist, I gather you don't do the calcining on the OTEC platform, you do that on the beach as part of the processing before the stuff is sent out. Since that uses natural gas, perhaps it's not as large a fraction of the total energy requirement. Maybe there isn't too much of an economy, but perhaps one could obtain gases, or at least enough for calcining on board. One could presumably rotate your muds back to the beach in the same ships used to carry the bauxite to the platform.

M. Jones: That is one option. The calcining energy is about 3% of the energy that is required for the project, and it could be done on the plantship. The thing that mitigates against it is that it's done in kilns that are several hundred feet long, and when you're paying $650/ft^2 for real estate at sea, that doesn't seem to be the kind of apparatus you want to have.

OTEC POWER FOR OCEAN MINERALS

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Abstract

The similarity between large floating platforms in the deep ocean for recovery of metallic nodules and large ocean plants for Ocean Thermal Energy Conversion leads to the possibility of combining the two into a single floating structure. Not only does ocean mining require a huge vessel far from shore, it also requires power in substantial blocks, both for recovery and for partial processing of nodules. In fact, since the by-products of processing include toxic wastes, and since these wastes are preferably disposed of far from sources of potable drinking water, there is good reason to consider the economic and ecological feasibility of complete processing at sea. By this means, the volume of materials that must be transported to or from the plant could be minimized and the wastes could be returned to remote deep-ocean seabeds, whence the nodules came. The increased power demands would there be met by OTEC, if a suitable double-purpose vessel can be devised. ~~The following paragraphs present~~ such a concept is presented, and ~~recommend~~ a program of further study is recommended.

Introduction

Rising interest in the commercial development of deep ocean deposits of manganese nodules, and the improvement of techniques for recovering them and extracting and refining their several metallic components has led to technical studies of processing, as well as the legal and marine aspects of harvesting them.

Recent studies have indicated several constraints on manganese nodule processing. These constraints fall in four categories:

1. Availability of low cost electrical power;
2. Availability of desalinized water;
3. Transporting vast amounts of nodules from remote areas with less than 3% of the nodules' weight being converted into finished products;
4. Disposal of toxic solid and liquid wastes.

The effect of these constraints depends upon the location of processing, the amount of preprocessing at sea, and the method of processing.

Some preprocessing at sea may be economically attractive. The nodules would be transported to land sites in the form of intermediate products, such as:

- pregnant liquors
- impure, mixed metal precipitates

The production of highly concentrated metallic phase products at sea has been assumed to be not practical, based upon the expectation of excessive ship motions, expensive power generation and lack of fresh water.

Of the several methods developed by the four major mining companies for extracting metals from seabed nodules, all have one fatal drawback. They leave behind effluents that are toxic. The reason is simple. The metals are so inextricably bound into the nodule matrix that powerful chemical solvents must be used, which after precipitation of the iron, nickel, copper and manganese, must be, in part, wasted.

The disposal of a mixture of solid and liquid toxic waters virtually eliminates plant sites on the continental land masses because of the certain danger of contaminating groundwater. Water looms as the final limit to the population of the human race on planet Earth, and groundwater is the largest water resource. It must be protected.

Lined reservoirs or impounding basins have been considered as a means of preventing toxic fluids, leached out of solid wastes, from mixing with groundwater. There are plastic liners today that are almost completely impervious and insoluble to the fluid effluents from nodule processing. But none of them can be proven safe against perforation. ("Pond Liners, Anyone"), by Ralph W. Woodley, The Military Engineer, Vol. 70, No. 458, November - December 1978. And it is a characteristic of laminar seepage through porous media that flow can be kept small only if openings are zero. The smallest rip or cut or imperfect joint or seal between sheets or mylar, or other liner, would destroy the effectiveness of the seepage barrier.

Accordingly, there is only one sector of the earth suitable for disposal of this kind of effluent, and that is the oceans whence the nodules came. The recycling must be done in two components - the fluids by dilution and mixing with the vast volumes of sea water, and the solids by deposition directly on the sea-bed, their original environment, where leaching gradients are infinitely smaller than on land.

This paper is directed toward the need for a maximum recycling at sea, using natural ocean thermal gradients to power the nodule collection and processing, and minimizing the environmental changes that would result from harvesting a vast metallic resource.

Nodule Processing

Previous studies, "Descriptions of Manganese Nodule Processing Activities for Environmental Studies", U.S. Department of Commerce, NOAA Office of Marine Minerals, August 1977 have shown that substantial blocks of power are needed for harvesting and processing of nodules. For example, a four-metal plant (manganese, copper, nickel and cobalt), collecting a million tons of nodules a year would require 70 to 100 megawatts of electrical power just for preprocessing. It would generate over 1,200,000 tons of wastes, half liquid.

To furnish this power, there are only two practical fuel alternatives: fossil fuel, transported to the plant by tanker (oil or gasoline) or bulk carrier (coal); or OTEC, ocean thermal energy conversion, making use of the natural characteristics of the water column in the areas of the ocean where the nodules exist. Depending on the process used, even preprocessing would require transportation and transfer to the plant of up to 2700 tons of fuel oil per day.

The preprocessing alternatives so far studied have been predicated on the assumption that plant motions at sea would preclude direct production of marketable metals. The penalties of this assumption include rehandling of nodules, raffinates, pregnant liquor, reagents and water, as well as fuel, some of them toxic and dangerous (chlorine, ammonia, propane, butane and sulfuric acid). Splitting the sequence of operations, some at sea, some on land, then some at sea, eliminates energy saving and multiplies cleaning operations in the process. The quantities of materials to be transported and re-handled are very large, any delay in any shipment being a breakdown in the smooth sequence of flow. Weather and varying sea states, as well as Murphy's Law for ship and material handling, would surely produce many such breakdowns.

All of these uncertainties would be eliminated if the processing were accomplished on a stable ocean platform, whose large size, small waterplane and specific design would reduce downtime to intervals during the most severe storms. Ship motions would be within acceptable tolerance for mixer-settler units, particularly if coalescers are used. Pitch, roll and yaw can be reduced to very small angles. Surge and sway would be tolerable, and heave is not critical to such operations.

This type of design would be enhanced by the increase of mass accompanying large quantities of circulating water inherent in the OTEC power cycle.

Similarities With OTEC

Increasing government support for ocean thermal energy conversion (OTEC) leads to the potential opportunity of combining these two technologies. To study the feasibility of plants that might perform both functions, one must keep in mind the similarities and differences between recovery of thermal as compared with deep ocean mineral resources.

The common characteristics shared by the two concepts are striking:

1. Both need very deep ocean sites - the mining plant because that is where the nodules are found, and the OTEC because sufficient temperature differential between upper and lower water strata occurs only in deep water;

2. Both are most practical only at tropical or subtropical ocean sites, which are those ocean areas that have both warm surface water, cold bottom water and nodules;

3. Both are generally remote from major centers of population and require sea transportation to carry their output to consumers;

4. Because of the remote, deep-ocean sites exposed to large waves, both types of plants face the economic burden of very high capital costs which can only be amortized over a number of years of operations;

5. Operating and maintenance crews must function far from supply bases;

6. Energy conservation is key to both plants - one because it must produce more energy than it consumes, and the other, because its use of energy must subtract from the net value of its products;

7. Freedom from motion at sea is highly desirable for both plants, so that very large size would be an advantage in ironing out the irregular nature of a floating support.

By combining the needs of both types of plants, anet benefit can be achieved with respect to most of these common characteristics and requirements. Conceptual design of a hypothetical floating plant is a process in which knowledge of existing or past floating designs is channeled into the needs and purposes of a new function. This is innovation tempered by experience. Intuitive judgement is derived from technology transfer, applying learning about special purpose floating craft other than ships, which are designed only for movement from shore to shore, whereas plants for these purposes must be designed for slow but constant movement far from shore.

A Dual Purpose Concept

One conceptual design is portrayed in the following sketches. However, further study is needed so that information will be available for presentation to industry and to government agencies, setting forth technical data and cost estimates for the concept, both construction operating and maintenance costs, along with estimates of input and output of materials.

The concept envisages a large semi-submersible, probably catamaran vessel. A long, deep cold water pipe at the bow would feed probably cold water to one side of the catamaran, which would contain a number of condensers of an ammonia OTEC system. A parallel intake of surface water would feed the evaporators of the OTEC system, all pumps being in line and below sea surface in order to assure low-head operation. The separate discharges of the two circulating systems would force water astern, providing propulsion force. However, the cold water discharge, curved downward, would return the slightly

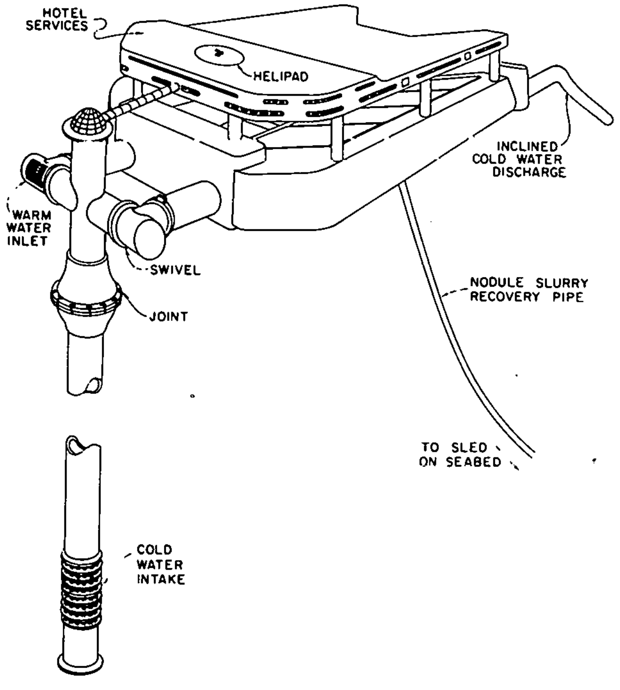

Fig. 1 OTEC mining vessel.

warmed water to elevations in the sea's thermocline near the natural depths corresponding to the temperature, and the warm water discharge, curved downward much less, and equipped with a rudder for steering, would return the slightly colled surface water to depths matching its respective temperature. This design would thus minimize environmental effects of the system.

The dimensions of the structure, and the small water plane of the semi-submersible in normal operating position, would provide good stability and low response to heave, pitch and roll in any sea. Relative differential heave between the cold water pipe and the plant should be small. The cold water pipe would be of honeycomb structure and would be ballasted to provide small vertical forces at the swivels in a calm sea. The water plane of the pipe would be about 2,820 feet^2, and the plant 3,760 feet^2.

Plant Operation

With the entire circulating water system below sea level, the head against which either large pump must work to move water at around seven feet/second will be at all times limited to the fraction losses in the pipes and in the heat exchangers. This will minimize starting and eliminate siphoning problems.

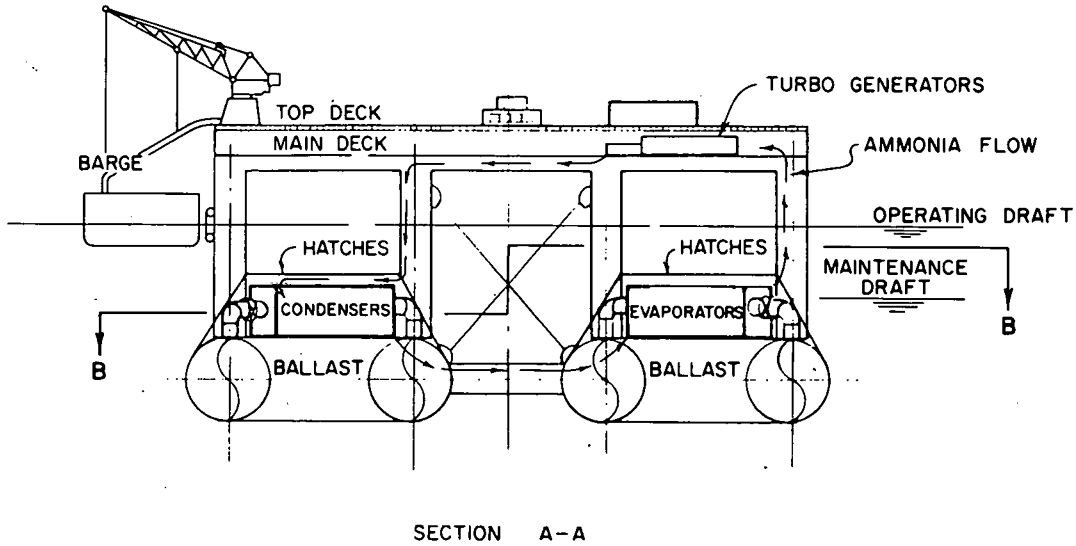

Fig. 2 Cross-section of hull.

Accessibility for removal and replacement of each heat exchanger is provided in the concept by reducing the draft in favorable weather by deballasting to a draft that exposes the Access Deck. Hatches in it would permit access to each heat exchange unit, which would be lifted to the Access Deck, and loaded onto a maintenance barge, at the same time replaced with a spare unit.

The swiveling catamaran configuration allows considerable pitching with respect to the angle of inclination of the cold water pipe, which may incline as the vessel moves ahead in response to the jet propulsion from the discharges of cold and warm water. Rolling would cause bending of the articulated, rubber-jointed section of the pipe just below the swivels.

Such a configuration would tend to move forward to pinwheel around the cold water pipe, because of the latter's large resistance to horizontal movement. However, a hydraulic lift nodule recovery pipe, dragged from the stern of the vessel, would move the center of resistance abaft the center of propulsion and assure a relatively straight course, controlled by the rudder as desired. Such a course would be programmed so that the nodule recovery sled, dragged along the seabed would traverse those areas where nodules of suitable size and density are known to exist. Hydraulic recovery moves the nodules up the pipe, after screens on the sled eliminate silt and clay.

Arrangement

The plant would provide sufficient hull and deck space for processing facilities such as tanks, piping, generators, turbines, hotel quarters, transshipment berths, cranes and related equipment for a virtually self-sufficient operation. The input would come almost entirely from the sea and the output would be partially or fully processed materials of use in commercial applications, minimizing both the quantities to be transshipped and the land-based processing and disposal of superfluous end-products. On the deck there will be nearly three acres of surface, with a like amount available on the top deck for nodule processing and separation of metals. A barge berth on the port side provides transportation of metal to mainland markets. The waste would be returned to the sea floor, nearest whence they originated, thus reducing environmental effects to a minimum, particularly those effects that might be detrimental to mankind. The dimensions of the plant, along with its semi-submersible design will so reduce response to wave action as to provide a stable platform for the nodule processing.

Recommended Program

The closed cycle OTEC mining plant could be investigated by the steps outlined below.

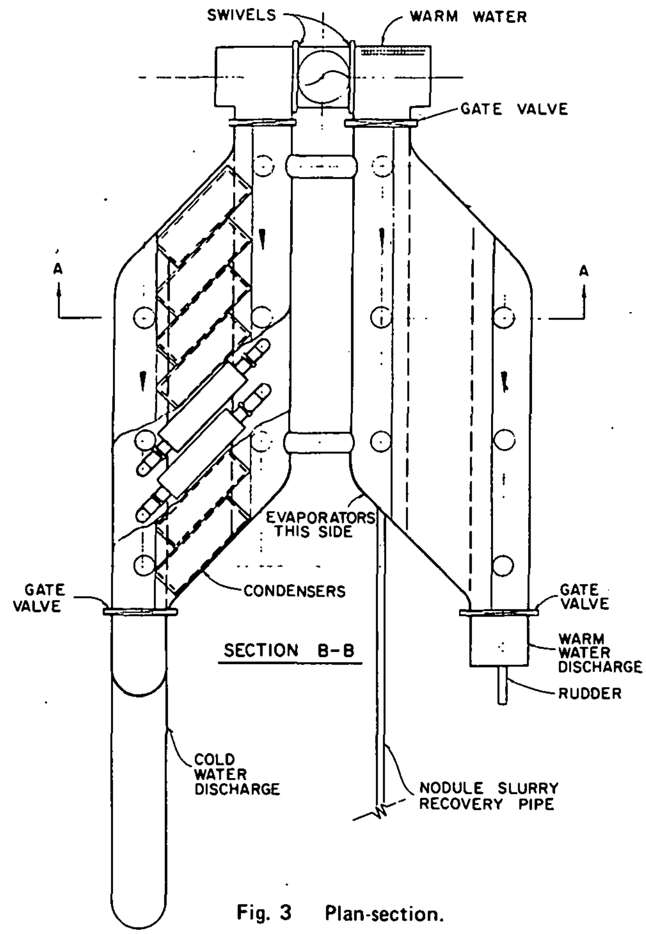

Fig. 3 Plan-section.

Preliminary Design For Vessel

1. Compile existing information on construction cost, operating and maintenance costs of nodule mining systems with land based processing; assume impervious ponding of toxic waste discharge at a remote land site;
2. Determine major criteria which vessel must satisfy; determine OTEC equipment requirement; assume a cold water inlet and discharge configuration; assume a warm water inlet and discharge configuration; determine mining and metalurgical equipment requirement; determine preliminary vessel structural requirements and vessel weight, carrying capacity and draft during operating and maintenance modes.
3. Determine vessel characteristics and performance against parameters such as surface waves of various sea states, surface current, subsurface current and wind; determine vessel motion in three specific sea states and determine probably maximum operational sea state and expected operational hours per year at a given sea site;
4. Determine construction, operating and maintenance costs. Preliminary study indicates that such a plant could be built in conventional shipyards and other existing sites, and that the operation is feasible. The economics however, combining the capital costs of OTEC with mining vessels, need further study. The inherent advantages of reduced environmental effects and enhanced applicability of the OTEC principle should result in benefits.

Definition of Industrial Application

Mining: Analyze the feasibility of the vessel as a vehicle for nodule mining; determine probable propulsion, drag and capacity.

Metalurgical: Select a metalurgical process and demonstrate the feasibility of the vessel as a practical processing plant platform.

Ammonia Production: Review the feasibility and practicality of including ammonia production facilities on board the vessel.

Other Sea-Site Industrial Applications: Review the feasibility of utilizing the vessel for industrial applications, other than mining or in combination with mining, such as fertilizer production, hydrogen production of fish processing.

Economic and Environmental Analysis

Compare the economics of sea-site plants to land-based plants; nature and quantities of effluents; probable costs and environmental effects; siting constraints.

Consideration of Open Cycle Plant

Review differences required in plant configuration for open-cycle plant; show schematic comparison of fluid flow requirements and a probable modification of external structure; evaluate effect of these differences on plant cost. Describe the benefits and uses of fresh water condensate discharge, its application to nodule processing, dilution of effluents and crew requirements.

Concluding Remarks

The program outlined above would yield a potential solution for the numerous economic and environmental factors that affect commercial viability of ocean mining. At the same time, a practical use may be found for the conversion of ocean thermal energy at its very source - in the deep tropical oceans, far from land.

DISCUSSION

Question: Have you extended your thinking to the floating city concept? If an OTEC plant were part of a floating city, producing energy-intensive products on the site — ammonia or hydrogen or whatever else they can make from OTEC electricity — how would this affect the OTEC electricity economics?

E. Harlow: Basically you are expounding upon John Craven's idea of the floating city and matching with OTEC. Roughly the superstructure excluding the cold-water pipe of an OTEC is approximately 10 to 15% of the cost of an OTEC. If you deferred the cost of the superstructure of the city away from the power production, and charge it as real estate to people living on the floating city, you could by this bookkeeping reduce the power costs by about 15% of the OTEC capital costs that I have shown. You would have further savings, compared to the OTEC cable-to-shore option, by not having a transmission cable to worry about as a capital cost. There would also be a saving of power because there would not be energy losses due to that transmission system.

Question: Are you proposing to use any new electrical processes that have somehow been adapted to your particular mining situation?

E. Harlow: No, this concept has been developed with the idea of using present technology.. But, of course, each of the mining companies and combinations of companies that are studying this subject have some information that they do not divulge to the public. This concept is based on what is known about those processes as derived from publications available from NOAA.

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