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DESIGN AND COST OF NEAR-TERM OTEC PLANTS FOR THE PRODUCTION OF DESALINATED WATER AND ELECTRIC POWER

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by

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ABSTRACT

There currently is an increasing need for both potable water and power for many islands in the Pacific and Caribbean. The Ocean Thermal Energy Conversion (OTEC) technology fills these needs and is a viable option because of the unlimited supply of ocean thermal energy for the production of both desalinated water and electricity.

The OTEC plant design must be flexible to meet the product-mix demands that can be very different from site to site. This paper describes different OTEC plants that can supply various mixes of desalinated water and power -- the extremes being either all water and no power or no water and all power. The economics for these plants are also presented. The same flow rates and pipe sizes for both the warm and cold seawater streams are used for the different plant designs.

The OTEC plant designs are characterized as near-term because no major technical issues need to be resolved or demonstrated. The plant concepts are based on DOE-sponsored experiments dealing with power systems, advanced heat exchanger designs, corrosion and fouling of heat exchange surfaces, and flash evaporation and moisture removal from the vapor using multiple spouts. In addition, the mature multistage flash evaporator technology is incorporated into the plant designs where appropriate. For the supply and discharge warm and cold seawater streams, larger engineering and cost uncertainties do exist because the required pipe sizes are larger than the maximum currently deployed -- 40-inch high-density polyethylene pipe at Keahole Point in Hawaii.

INTRODUCTION

The ocean thermal energy conversion (OTEC) concept first proposed by D'Arsonal in 1880 uses the temperature difference between the warm surface seawater and the cold deep seawater as an energy source for a conventional Rankine cycle. The various options for the extraction of this virtually inexhaustible energy resource are the closed cycle, the open cycle, and various hybrid cycles. The early development effort and experiments focused on the open cycle. The components of the open-cycle OTEC are a flash evaporator that generated steam by extraction of the sensible heat from the warm seawater. The steam then flows through a low-pressure turbine to generate the electrical power. The spent vapor is finally condensed using a surface type if water production is desired or alternately using a direct-contact type.

Claude (1930) conducted experiments in 1928 on a simulated OTEC plant using the warm discharge water from a blast furnace of a steel plant and Meuse river water. In 1930, he succeeded with an open-cycle demonstration plant in the Matanzas Bay in Cuba. The power generation was much less than anticipated and was about 22 kW because of the relatively high temperature and excessive flow rate of the cold seawater.

In the 1940s, the French government with the help of Claude sponsored an open-cycle development program to design a 40 MW power plant near Abidjan (Ivory Coast). Many of the major design problems were addressed and solutions proposed; however, the project was shelved for political reasons and a hydro-electric plant was built instead. Massart (1974) presented an interesting overview of this effort. A major contribution of the research is the spout flash evaporator that was tested by Nisolle (1947) and is the proposed flash device for the current open-cycle (OC) OTEC effort in the United States.

The first demonstration of the OC-OTEC concept in the United States was performed in 1957 (see Howe, 1957, and Howe et al., 1970, 1978) as part of the Sea Water Conversion Program at the University of California, Berkeley. The experimental facility included a gas-fired heater to generate the warm water and a cooling tower to generate the cold, the water source being either the bay (20,000 ppm) or tap water with salt addition (1000 ppm). The turbine was a single-wheel, impulse-type machine obtained from a surplus airplane supercharger. This research concentrated on both the flash evaporator (annular nozzle flash type) and the impact of the noncondensables on the performance of a small surface condenser.

The U.S. OTEC program intensified in the 1970s concentrating on the development efforts for large (25 to 400 MWe) floating plants. The closed cycle was now considered a viable option because of experience gained in the control of heat-exchanger fouling. With the closed cycle, the working fluid is ammonia or an alternate refrigerant that is in a recirculating mode. The liquid refrigerant is converted to vapor in the evaporator, expanded through a turbine to generate the electrical power, condensed in a surface heat exchanger, and the condensate is pumped back to the evaporator to repeat the process. The heat source for the evaporator is the warm seawater and the heat sink for the condenser is cold seawater. Lockheed/State of Hawaii (Owens and Trimble, 1981) has demonstrated the closed-cycle OTEC concept.

Recently, the focus of the OTEC program has been shifted to small shore-based plants of one or a few megawatts for a potential island market. Many islands have a need for both power and water that can only be produced directly with the open- and hybrid-cycle plants. This paper describes the OTEC plant designs dedicated to the simultaneous production of power and water for small, shore-based plants for island markets. However, it is restricted to near-term plants where there are no major technical issues to be resolved or demonstrated which eliminates consideration of the open cycle because of the lacking technology base for the steam turbine. Many of the desired features of OC-OTEC are attainable with hybrid cycles. The first part describes a hybrid cycle that maximizes the power production, the conventional hybrid cycle, and the second describes alternate hybrid cycles that produce any desired mix of power and water. Recently the Japanese (Uehara et al., 1988) also decided to develop the technology base for hybrid-cycle plants.

CONVENTIONAL HYBRID CYCLE

This section of the paper will (1) present a brief description of the conventional hybrid cycle, (2) discuss the advantages and disadvantages, 3) describe a 1 MWe plant design, and (4) characterize the power and water mix that can be obtained from this plant.

Cycle Description

A detailed flow diagram of this cycle is shown in Figure 1. Warm seawater is pumped from near the surface (10 to 15 m) into a vacuum flash chamber

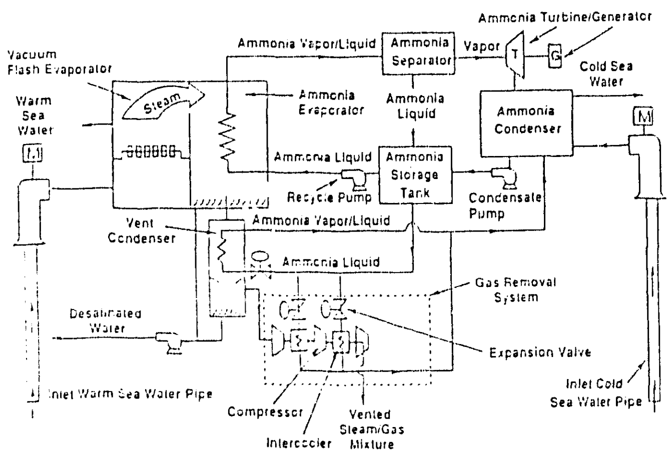

Figure 1 Schematic Flow Diagram for the Conventional Hybrid-Cycle Plant

where about 0.5 percent is converted (flashed) into steam. The steam is then condensed in an adjacent heat exchanger, the heat being transferred to an ammonia liquid stream to generate ammonia vapor. The ammonia portion of this hybrid cycle is similar to a typical, closed Rankine cycle except for the two ammonia liquid streams leaving the high-pressure storage tank. The first stream is used in the vent condenser following the ammonia evaporator to further reduce the steam content in the noncondensable gas or vent stream th must be discharged to the atmosphere in order to maintain a vacuum in the flash chamber. The second is used as the heat sinks for the intercoolers of the noncondensable gas removal system. Further information is presented on this cycle by Panchal and Bell (1987) and on the gas-removal system by Rabas et al. (1989).

Hybrid Cycle Advantages and Disadvantages

The major advantages when compared to the open cycle are (1) a state-of-the-art conventional ammonia turbine can be used, (2) the size and complexity of the gas-removal system is reduced, and (3) components can be easily scaled. The major disadvantage is that an additional heat exchanger is required, the steam condenser/ammonia evaporator. To substantiate these statements, a component cost comparison from earlier cycle studies is now presented.

Two comparison studies of the three cycle types were carried out during the early phase of the OTEC program: one by Gilbert/Commonwealth, Bartone (1978), and Westinghouse (1979). These studies were restricted to large, floating OTEC plants. All the heat exchangers were the shell-and-tube type with titanium and/or copper-base tube materials except for the flash evaporators used for the open and the hybrid cycles. Also, the more meaningful conclusions are restricted to cost comparisons for each study but not between studies because of the different design constraints.

Table 1 shows a cost comparison of only the major power-plant components. Note that the turbine/generator is a major cost contributor for the open cycle whereas the heat exchangers are the most dominant for the closed and hybrid cycles. A major cost contributor for the Gilbert/Commonwealth evaluations is the deaerator used for both the open and closed cycles. However, the Westinghouse study (1979) and later studies by Solar Energy Research Institute (Block and Valenzuela, 1985, Link and Shelpuk, 1987, and Parsons, Bharathan, and Althof, 1985) all concluded that pre-deaeration is not cost effective. A consistent finding of both comparisons is that the hybrid cycle is the most costly for these relatively large-capacity, floating-plant studies because of the heat exchanger costs.

Based on these findings, why is the hybrid cycle considered for small capacity, land-based OTEC plants for combined power and water production? The reasons are as follows:

1) Significant progress has been made with advanced heat-exchanger concepts such as plate-fin type (matrix) for phase-change applications (Panchal et al., 1981, and Panchal, 1990).

2) Experiments conducted at the Seacoast Test Facility (Thomas and Hillis, 1989, and Panchal et al., 1985) demonstrated the feasibility of aluminum as a heat-exchanger surface for OTEC conditions.

TABLE 1 Cost Comparison of Major Components

Study	Westinghouse	Gilbert/ Commonwealth
Net Power, MWe	100.	25.
Year	1979	1976
Costs, $/kW		
Closed		
Evaporator	481.	488.
Turbine/Generator	66.	124.
Condenser	534.	840.
Open		
Deaerator	0.	864.
Flash Evaporator	44.*	808.
Turbine/Generator	218.	1576.
Condenser	389.	112.
Gas Removal	23.	0.**
Hybrid		
Deaerator	0.	1640.
Evaporators, Flash+NH3	616.	1828.
Turbine/Generator	110.	160.
Condenser	699.	1160.
Gas Removal	19.	0.**

*Part of the flash-evaporator cost is in the platform
**Gas-removal cost included in deaerator cost

3) Biofouling studies also conducted at the Seacoast Test Facility demonstrated that the level can be controlled and design fouling resistances are about 1/5 of that used for the earlier studies.

4) The power-plant costs are a smaller percentage of the total plant cost because of the costly seawater systems required for land-based plants.

In summary, combined water and power land-based OTEC plants based on the conventional hybrid cycle become a viable option because of the potential reduction in the heat-exchanger costs and because a state-of-the-art ammonia turbine is used. Of course, the water production capability is the reason for the recent change from the closed cycle to the conventional hybrid cycle by the Japanese (Uehara et al., 1988).

TABLE 2 Cost Comparison of OTEC Cycles Based on the Current Technology Base, 1990 $

Subsystem	Present Technology (1990) Closed	Open*	Hybrid	Long Range Goal (2000) Closed	Open	Hybrid
Structure	400	800	900	300	600	800
Seawater System	6000	6000	6000	3700	3700	3700
Heat Exchangers	2850	1600	400	1600	1000	2400
Power System	850	2500	950	700	1500	780
Other Direct Costs	1500	1500	1500	1500	1500	1500
Subtotals	11600	12400	13350	7800	8300	9180
Site Development	40	40	40	40	40	40
Engineering Fees	170	170	170	170	170	170
Contingencies (10%)	1160	1240	1330	780	830	920
Project Management/Fee	200	200	200	200	200	200
Startup Costs	50	50	50	50	50	50
Total Installed	13220	14100	15140	9040	9590	10560

*Not in near-term category because of missing low-pressure steam turbine technology

Table 2 presents a cost comparison of the conventional hybrid-cycle plant with other OTEC plant types based on the 1990 technology base. A 10 MWe plant size was selected for the comparison because these values were available for both the closed and open cycles from a 1990 DOE study on the economic viability of OTEC. This table shows that the disparity between the total installed costs has substantially decreased because of the DOE-sponsored research described above and because the focus has shifted to shore-based plants with the more costly seawater systems. The most important point displayed by this table is that the conventional hybrid cycle is cost competitive with the other cycles. In addition, it can produce water whereas the closed cannot, and uses near-term technology whereas the open does not.

1 MWe Plant Design

A conceptual design of a 1 MWe hybrid plant considering all the components was completed. The 1 MWe capacity rather than 10 MWe was selected because the cold water pipe size is not a major departure from the current technology and the hybrid-cycle plants can be easily scaled to larger capacities. Figure 2 shows an elevation view including all the components. The plan area is 6400 ft^2 (595 m^2) and the height is 34 ft (10.4 m). Other noteworthy features of the design are the following:

1) The barometric principle to reduce the warm seawater pressure losses

2) A spout flash evaporator using design data obtained with the recent HMTSTA experiments at the Seacoast Test Facility

3) Minimal pressure losses for the steam flow path because of compact component integration

4) Brazed-aluminum matrix heat exchangers for the ammonia evaporator, the steam vent condenser, the ammonia condenser, and intercoolers of the noncondensable gas-removal system

5) Small turbine/generator on a skid installed under the ammonia evaporator along with the turbine/generator auxiliaries

6) A three unit (stage) noncondensable gas-removal system consisting of centrifugal compressors and two intermediate intercoolers

This design incorporates the latest heat-exchanger design improvements such as the use of aluminum, enhanced matrix heat exchangers to greatly increase the overall system efficiency and to reduce the heat-exchanger costs, the major objection to the hybrid cycle. However, there are some validating experiments still required for the heat-exchanger designing algorithms. This small, land-based OTEC hybrid-cycle plant is very different in concept from that considered in the previous Westinghouse and Gilbert/Commonwealth designs.

The appendix contains additional information on this 1 MWe design such as the external constraints and design assumptions, the system parameters (power, flow rates, and temperatures), and the component size details.

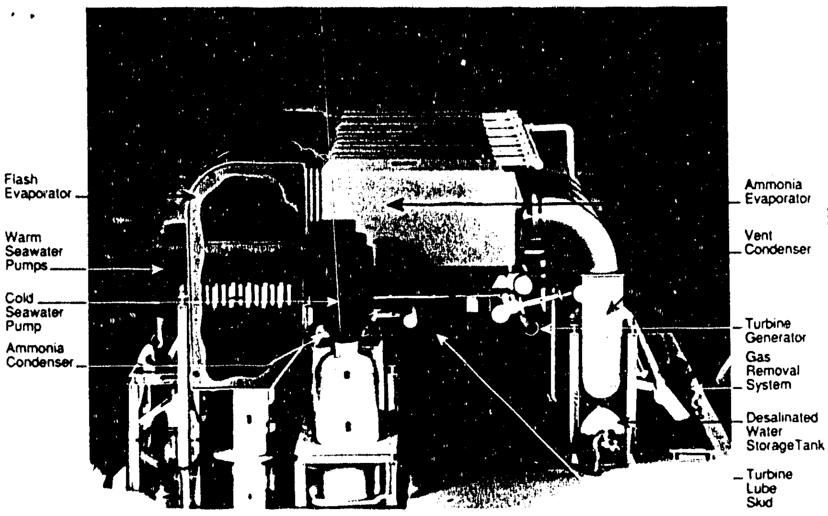

Figure 2 Model of a 1 MWe Conventional Hybrid-Cycle OTEC Power Plant

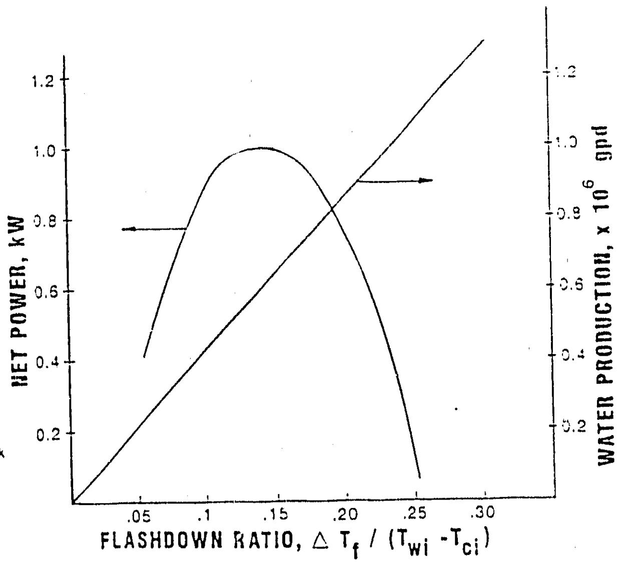

Figure 3 Variation of Net Power and Water Production with Flashdown for a Conventional Hybrid-OTEC Power Plant

Power/Water Mix for Conventional Hybrid Plants

This section of the paper will show that there is an optimum flashdown range that maximizes the power output. As a result, the water production is essentially fixed and determined by the flashdown that maximizes the power output. The variation of the net power and water production for 1 MWe plant design will now be determined with the specified seawater system and components. The assumption is made that the efficiencies of rotating equipment (turbine, compressors, motors) are not altered when operating at off-design conditions.

Figure 3 shows the variation of the net power and water production as a function of the flashdown in the flash evaporator. Note that there is a very sharp maximum of the net power output whereas the water production increases almost linearly with the flashdown. To operate in the 0.9 of maximum power range, the water production is limited from 450,000 to 770,000 gpd. These results show that there is a very limited mix of power and water production that can be obtained with this hybrid cycle.

A system parameter selected for this 1 MWe design was the ratio of the warm to cold seawater flow rates, this value being 1.50. An additional study was made to determine the maximum power and resulting water production rate as a function of the warm/cold seawater flow rate ratio. The cold water flow rate and pipe size was not altered; however, each result represents a completely new plant design. Although not considered, it is obvious that the plant cost will also increase with an increasing flow ratio because of the larger heat exchangers.

The results of this study are shown in Figure 4. Note that both the power and water production increase with the flow rate ratio but at a very modest rate. Also note that there is a knee in the power curve near 1.5, the value selected for the 1 MWe design. This study shows that the power and water production ratio cannot be significantly altered by changing the warm to cold seawater flow ratio.

The conventional hybrid cycle when optimized for power production is restricted to a narrow range of water production. If this power-water mix is acceptable for the selected site, this cycle is a viable option. However, if the need is for a different mix of power and water, alternate hybrid cycles should be considered. A similar finding was made by Westinghouse (1980) when studying the application of the open-cycle OTEC plant for water production with zero net power. The second part of this paper describes alternate hybrid cycles that are more suited for higher water to power ratios than those obtained with the conventional hybrid OTEC plant.

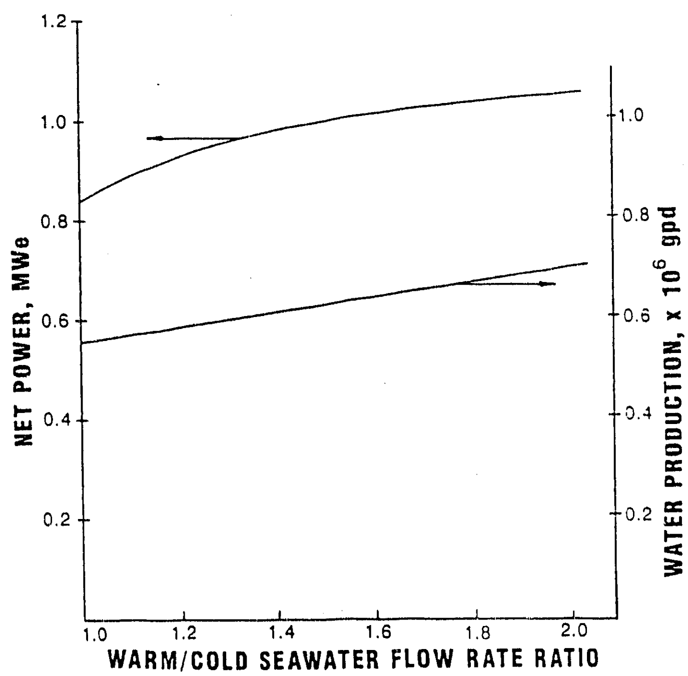

Figure 4 Variation of Net Power and Water Production with the Seawater Flow-Rate Ratio

HYBRID-CYCLE OTEC WATER PLANTS

A particular version of a hybrid cycle more suited for water production is shown schematically in Figure 5. Essentially, it is a combination of a multistage flash evaporator and a closed cycle or a conventional hybrid cycle OTEC power plant. They are coupled in a unique manner and the blending feature permits different quantites of power and desalinated water from the same equipment. The purpose of the multistage flash evaporator is water production and the purpose of the power plant is to generate power and to run the auxiliaries. Each of these components are now discussed.

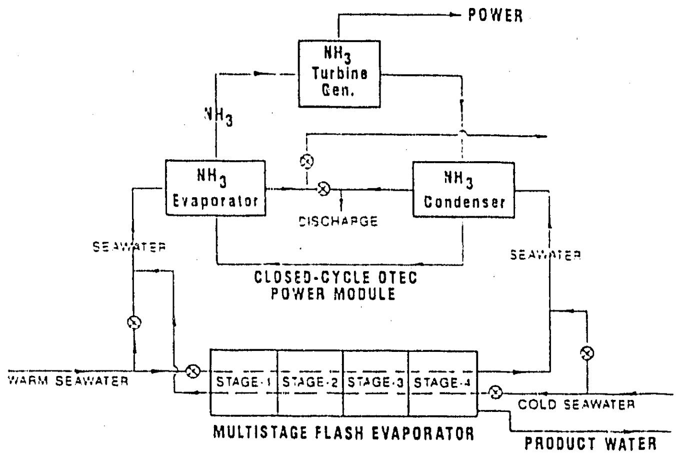

Figure 5 Schematic Flow Diagram for a Hybrid-Cycle OTEC Water Plant

Multistage Flash Evaporator

The most common system for the generation of desalinated water is the multistage flash evaporator. Each stage consists of a flash chamber where a small percentage of the warm seawater is flashed to steam. The steam then passes through a moisture-removal device and into a condenser where the vapor is condensed and a high-purity fresh water is obtained. The major components of each stage are a flash chamber, a moisture separator, and a steam condenser; the same as used in the conventional hybrid cycle.

Why use a multistage flash evaporator? The water production depends on the flow rate and the flashdown. For the conventional hybrid configuration, only about 15 percent of the total available temperature difference is used in this single-stage system (see Figure 3). If multistaging is employed as shown in Figure 5, about 65 percent of the available temperature drop is available for water production. For the same flashing liquid flow rate, about four to five times the water production is obtained. It is apparent, then, that the use of multistaging maximizes the water-production capability.

For this desalination plant, a power source is required to drive the compressors of the noncondensable gas removal system and the seawater pumps. Figure 5 shows that there is no predeaeration of the warm seawater, the reasoning being that it would not be cost effective as discovered for open-cycle studies already mentioned. As a result, the oxygen and nitrogen dissolved in the seawater is liberated in the first couple of stages and these gases must be continuously removed in order to maintain the vacuum in the flash chambers. The preferred seawater flow rate ratio for the multistage evaporator is unity and the flashing seawater flow rate is therefore equal to the cold seawater flow rate. As a result, the parasitic power requirement is almost the same as that for the 1 MWe conventional hybrid configuration (62 kW) if the same seawater system is used (5.82 ft pipe diameter and 57300 gpm). Even if predeaeration was determined to be cost effective, there is a parasitic power requirement to remove and pump the oxygen and hydrogen to the atmosphere. To reduce the warm seawater pumping power, the barometric principle is used or the evaporator is elevated by about 30 ft (10 m). The seawater pumping power will somewhat exceed that of the conventional hybrid configuration because of the additional pipe lengths and additional components. A power source is therefore needed to drive the compressors and pumps.

The use of the ocean temperature difference for the production of desalinated water with a multistage flash evaporator was first proposed by Saari (1978) and further amplified in a later publication (Saari, 1981). He suggested that an external power supply be used to drive the pumps and compressors. Also, he recommended that predeaeration be used and also incorporated hydraulic compression to further reduce the power demand of the gas-removal system. He and his company, Nord-Aqua of Finland, were also working with Japanese to further develop this desalination concept considering a wide range of waste-heat sources.

Power Plant

The closed- or conventional hybrid-cycle OTEC power system is proposed for generating the power and to power the multistage flash evaporator. The hybrid cycle is selected if the water production needs are of paramount importance. There is no need to describe the power plants. The new and important issue is how the power plant is coupled to the multistage flash evaporator.

Figure 5 shows that there is a switching of the cold and warm seawater streams after the evaporator or that they are connected in series if no mixing is used with the cold seawater stream -- the maximum water-production scenario. The cold seawater stream to the power plant is actually the cooled, warm seawater stream from the multistage flash evaporator which now is about 4°C to 6°C warmer than the initial cold seawater temperature. This exit temperature is a function of the number of stages, the condenser effectiveness values, and the sum of the boiling point elevation, the thermal nonequilibrium (related to the flashing effectiveness), and the separator thermal losses. More details on this calculation are presented in a previous study (Westinghouse, 1980). The warm seawater stream to the power plant is the heated, cold seawater that was heated in the condensers within the multistage flash evaporator. The temperature of this stream is about 4°C to 6°C lower than the inlet warm seawater. This stream can be mixed with a portion of the initial, warm seawater stream to further elevate the inlet temperature. Note that a major portion of the temperature difference is used a second time to drive the power plant.

Another option is to couple the evaporator and power plant in a parallel arrangement as recommended in previous U.S. OTEC studies (Westinghouse, 1980, and Coffay and Rabas, 1981). Rey and Lauro (1981) described a similar hybrid water plant and suggested that an open-cycle power plant be used for the power generation and additional fresh-water production. Both studies clearly showed that OTEC hybrid water plants are cost-competitive with conventional multistage desalination plants and the latter demonstrated that the hybrid plant is very flexible with a broad range of water to power outputs.

Figure 6 taken from the work of Rey and Lauro (1981) illustrates the flexibility of the water-generating capacity of a hybrid water plant. Note that a single line is used to represent the fresh water and power production relationship for a single-purpose, open-cycle plant. For the hybrid water plant, a very broad fresh-water production range can be obtained for any desired power demand. Also shown is the cold-water pipe diameter for various power and water outputs. Any water-power mix above the single-purpose line can be obtained by varying the portions of the flows that go to the evaporator and open-cycle power system. A similar power-water production map developed for the series-coupled hybrid design would show a smaller cold seawater pipe diameter because of the series coupling.

The above arguments suggest that hybrid-cycle OTEC systems incorporating a multistage flash evaporator are capable of generating substantially more fresh water than a single-purpose, open- and hybrid- cycle system. Also, they can obtain a broad range of fresh-water production for any desired power requirement. The two previous economic studies (Coffay and Rabas, 1981, and Rey and Lauro, 1981) suggested that these systems are cost effective when compared to conventional desalination plants. Also, these studies did not consider all the recent heat exchanger advances from the U.S. OTEC program.

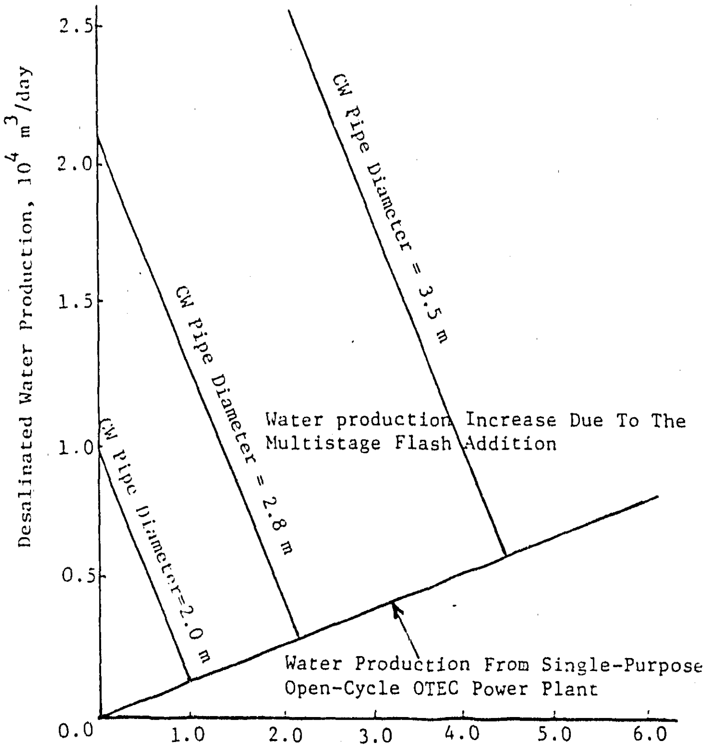

Figure 6 Water and Power Production Mix for a Open-Cycle OTEC Water Plant (Rey and Lauro,1981)

Table 3 presents a comparison of studies by Westinghouse (Westinghouse, 1980, and Coffay and Rabas, 1981), Alsthom-Atlantique (Rey and Lauro, 1981), and Nord Aqua Group (Saari, 1978, and Saari, 1981). Note that there appears to be consistent agreement between all three studies which were conducted independently at about the same time but in the U.S., France, and Finland. The capital and water costs of OTEC/MSF desalination plants are about twice the cost of conventional MSF and reverse osmosis desalination plants. Typical values are 5$/(gal/day) and 5$/Kgal for both plant types for the capital and water costs, respectively (Leitner, 1987).

TABLE 3 OTEC/MSF Hybrid Cycle Cost Comparison

Study	Westinghouse	Alsthom-Atlantique	Nord Aqua Group
Size	5.0 Mgal/day	2.64 Mgal/day	1.05 Mgal/day
Year	1981	1981	1980
Capital Cost	9.0$/(gal/day)	8.58$/(gal/day)	3.0$/(gal/day)*
Cost of Water	-	-	2.65$/Kgal* 5.30$/Kgal
Power Source	CC-OTEC	OC-OTEC	Grid
Plant Type	Floating	Floating	Floating
Location	St. Croix	-	St. Croix

*Without coldwater pipe and delivery system

There are some research issues that must first be resolved related to the flash evaporator that will be identified in the last section of this paper.

Multistage Flash Evaporator Design

Existing multistage flash evaporator technology can be used for scoping calculations of low-temperature flash unit for this OTEC water plant and to identify the areas requiring validating research. Table 4 below presents some of the important design details of the flash evaporator that were evaluated based on the same cold seawater flow rate and pipe size selected for the conventional hybrid design. Also, all the cold seawater passes through the flash evaporator, the maximum water-production scenario. The most important finding is that 2.5 mgd of fresh water is produced or about four times more than a conventional hybrid- or open-cycle, single-purpose OTEC plant with the same cold seawater flow rate and pipe size.

For the power plant, the inlet seawater flow rate and temperature are 86000.0 gpm and 72.00°F, respectively; the unused portion of the initial, warm seawater is blended with the total cold seawater flow stream. The temperature difference available for the water plant is now reduced, but only from 36.0 to 21.0°F.

TABLE 4 Some Multistage Flash Evaporator Design Details

Fresh-Water Production, gpd (m3/d)	2430000. (9230.)	
Dimensions, ft(m)		
Width	53.0 (16.2)	
Length	100.0 (30.4)	
Height	8.0 (2.5)	
Number of Stages	5.	
Seawater Flow Rates, gpm (l/s)		
Warm seawater	58300.0 (3615.)	
Cold seawater	58300.0 (3615.)	
Inlet Temperatures, °F(°C)		Outlet Temperatures, °F(°C)
Warm seawater	78.00 (25.55)	51.00 (10.55)
Cold seawater	42.00 (5.55)	69.00 (20.55)

The major areas requiring research deal with the flashing effectiveness of an open-channel flash evaporator and the thermal performance of the condensers. The open-channel is preferred in order to minimize the warm seawater pressure drop between the stages and to reduce the evaporator cost. The open-channel flash evaporator is used for all multistage flash distillation plants and was selected as the flash device for all multistage flash OTEC application described above. The separation between the stages is a vertical wall with an opening at the bottom to permit the seawater to flow into the next, lower-pressure stage. Table 5 shows the effectiveness and the thermal nonequilibrium based on current but extrapolated multistage flash distillation technology. These values are based on a brine level of 500 mm, a value commonly used by the desalination industry.

These values suggest the performance is comparable with that of a spout flash evaporator; however, there is no experimental data to validate the prediction method used to compute these values.

The other thermal losses for each stage are the boiling-point elevation and the moisture-separator loss. The former is about 0.3 C (M.W. Kellogg, 1975) and the latter is somewhat larger, (0.35 to 0.7°C), the larger values for the lower temperature stages.

The other area that requires further research is the prediction method for the surface condensers used within the multistage flash evaporator. The prediction methods for condensing steam in deep vacuums with large noncondensable concentrations are still being refined (Panchal and Bell, 1984). The prediction methods developed for multistage flash distillation that currently exist for higher pressure levels (Rabas, 1985, and Rabas and Mueller, 1986) are adequate to deal with the noncondensable gases but need to be validated for these low pressure levels. It is important to note that these surface condensers are almost identical to an OC-OTEC surface condenser and the resulting OTEC research is directly applicable. This would also suggest that plate-fin heat exchangers could be used for the evaporator condensers, a significant departure from the typical practice.

A standard shell-and-tube type was selected for the evaluation shown in Table 4; however, an enhanced rather than a plain tube was selected. This is another major departure from typical multistage flash practice because even today enhanced tubing is not an accepted practice for multistage flash desalination plants.

CONCLUSION

This paper demonstrates that hybrid-cycle OTEC plants may well be the most flexible and cost effective in obtaining any specific mix of electrical power and desalinated water. The major reasons for considering hybrid OTEC cycles are that they use state-of-the art ammonia turbines and produce desalinated water. When power is the desired commodity and water is the by-product, the most effective hybrid cycle consists of a flash evaporator, a steam condenser/ammonia evaporator, and a closed-cycle power system using the ammonia vapor generated in the steam condenser/ammonia evaporator. This cycle arrangement is called the conventional hybrid configuration. When the emphasis is on water production, and with no net power generation to the grid, the major components are a multistage flash evaporator with integral condensers and a closed-cycle or conventional hybrid-cycle power plant to generate the

TABLE 5 Effectiveness and Nonequilibrium Predictions for an OTEC Multistage Flash Evaporator

Flash Chamber Temp., °C	Flashdown °C	Effectiveness	Nonequilibrium °C
21.00	3.00	0.78	0.78
21.00	5.00	0.93	0.38
12.00	3.00	0.74	0.92
12.00	5.00	0.92	0.40

power to run the support equipment -- the seawater pumps, the gas-removal equipment, and the other auxiliaries. Any mix of power and water is obtained by varying the percentage of the total warm and cold seawater flow rates going to the multistage flash evaporator and the power plant.

The conceptual design of a 1 MWe conventional hybrid OTEC plant is presented displaying the integration of all the components. The plan area and the maximum height are 6400 ft^2 (595 m^2) and 34 ft (10.3 m), respectively. Brazed-aluminum plate-fin heat exchangers are used for the steam condenser/ ammonia evaporator and the ammonia condenser. It is shown that there is a very limited mix of power and fresh-water production that is possible with the conventional hybrid cycle.

A conceptual design of a hybrid-cycle water plant that maximizes the water production is described. The new component is a five-stage longflow flash evaporator. A portion of warm seawater flashes as it enters each of the five flash chambers and the vapor generated in each flows through a moisture removal device to a surface condenser that exists within each stage. The cold seawater flows in a countercurrent direction, it being the heat sink to condense the steam formed in each flash chamber. The criteria used for this design are the same as that used for conventional desalination plants, a mature technology of 30 years. However, there are some differences such as elevating the unit to reduce the warm seawater pumping power and the use of enhanced condenser tubes.

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APPENDIX

This appendix contains the constraints and design assumptions, the system parameters such as the power details, flow rates and temperatures, and the component design details.

Constraints and Design Assumptions

To perform the computations necessary for the 1 MWe design, a plant site must be selected to determine the pipe lengths. A generic site was characterized with the following pipe lengths and diameters shown in Table 6:

TABLE 6 Seawater System Details

Seawater	Intake Length ft(m)	Dia. ft(m)	Discharge Length ft(m)	Dia. ft(m)
Warm	1050 (320)	6.12 (1.86)	2035 (620)	6.12 (1.86)
Cold	7345 (2239)	5.82 (1.77)	2035 (620)	5.82 (1.77)

Note that both the warm and cold seawater discharge pipes terminate at the thermocline or at about 300 to 400 ft (90 to 120 m) depth where the warm surface water ceases to mix. In addition, a gravity head loss of 1.40 psi (9.65 kPa) was selected along with a 1.5 dynamic entrance loss to the pipes and a pipe roughness of 0.0002 ft (0.061 mm).

Additional assumptions are required for the efficiencies of the rotating equipment. These values are 0.86 for the turbine, 0.97 for the generator, 0.87 for the motors, and 0.70 for the compressors used in the noncondensable gas removal system.

System Parameters

Table 7 lists the system parameters for this 1 MWe conventional, hybrid-cycle plant.

TABLE 7 System Parameters

Electrical Power, kW		
Gross	1492.	
Net	1002.	
Parasitic Losses:		
Warm Seawater	172.	
Cold Seawater	213.	
Ammonia	33.	
Gas Removal	62.	
Miscellaneous	10.	
Desalinated Water Production, gpd (m3/d)	627000.	(2373.)
Flow Rates, gpm (l/s)		
Warm Seawater	86000.	(5426.6)
Cold Seawater	57300.	(3615.6)
Ammonia	2240.	(141.3)
Flow Rates, lb/h (kg/s)		
Steam Condensed in NH3 Evaporator	2072000.	261.070
Steam Condensed in Vent Condenser	10240.	1.290
Steam Entering Gas Removal System	666.	0.084
Liberated Oxygen	286.	0.036
Liberated Nitrogen	481.	0.061
Temperatures, °F (°C)		
Warm Seawater In	78.00	(25.55)
Warm Seawater Out of Flash Evaporator	72.74	(22.63)
Cold Seawater In	42.00	(5.55)
Cold Seawater Out of NH3 Condenser	49.94	(9.97)
Saturated Steam In Flash Evaporator	71.26	(21.81)
Steam Saturation at NH3 Evaporator Inlet	70.06	(21.14)
Steam Out of NH3 Evaporator	67.82	(19.90)
Steam Out of Vent Condenser	53.89	(12.16)
Steam Entering Gas Removal System	53.89	(12.16)
Ammonia Evaporation	66.60	(19.22)
Ammonia Condensation	51.86	(11.03)

The assumption is made that there is no carbon dioxide release from the flashing process of the warm seawater. Also, the boiling-point elevation and the pressure loss of the moisture removal device account for the difference between the saturated steam in the flash evaporator and the inlet saturated steam temperature to the ammonia evaporator. The difference between the warm seawater outlet temperature and the steam temperature in the flash evaporator is commonly called the thermal nonequilibrium loss and is directly related to the flashing effectiveness.

Component Design Criteria and Geometry Details

Table 8 lists the design criteria and resulting geometry details of all the major components.

TABLE 8 Design Criteria and Geometry Details

Flash Evaporator		
Design Criteria		
Flashing Effectiveness	0.85	
Desorption Effectiveness	1.00	
Velocity in Spouts	6.00 f/s	(1.83 m/s)
Spout Pressure Drop	1.40 psi	(9.65 kPa)
Separator Pressure Drop	0.015 psi	(0.10 kPa)
Design Details		
Spout Diameter	5.00 in	(127 mm)
Spout Length	20.00 in	(508 mm)
Spout Number	234.00	
Separator Area	515.90 ft^2	(48 m^2)
Steam Condenser/Ammonia Evaporator		
Design Criteria		
Steam Condensed	0.95	
Steam Velocity	100.00 ft/s	(30.50 m/s)
Design Details		
Length	20.00 ft	(6.10 m)
Height	20.00 ft	(6.10 m)
Width	17.80 ft	(5.43 m)
Vent Condenser		
Design Criteria		
Steam Condensed	0.885	
Steam Velocity	60.00 ft/s	(18.29 m/s)
Design Details		
Length	10.00 ft	(3.05 m)
Height	4.00 ft	(1.22 m)
Width	17.80 ft	(5.43 m)
Ammonia Condenser		
Design Criteria		
Water Velocity	4.00 ft/s	(1.22 m/s)
Design Details		
Length	48.00 ft	(14.63 m)
Cross Section	55.7 ft^2	(5.17 m^2)
Gas Removal System		
Compressors		
Number	3	
Type	Centrifugal	
Average Efficiency	0.70	
Intercoolers		
Number	2	
Type	Matrix	
Pressure Drop	2%	
Approach Temp.	2.0	
Geometry	See Rabas et al. (1989)	

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