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OCEAN THERMAL ENERGY CONVERSION POWER SYSTEM DEVELOPMENT—I. PRELIMINARY DESIGN REPORT

Volume 3. Appendixes D, E, and F
Phase I—Final Report

December 18, 1978

Work Performed Under Contract No. EG-77-C-03-1568

Lockheed Missiles & Space Company, Inc.
Ocean Systems
Sunnyvale, California

U.S. Department of Energy

Solar Energy

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LMSC-D630248

OCEAN THERMAL ENERGY CONVERSION POWER SYSTEM DEVELOPMENT-I PRELIMINARY DESIGN REPORT

VOLUME III

APPENDIXES D, E, AND F

18 DECEMBER 1978

CONTRACT EG-77-C-03-1568

Prepared for

UNITED STATES DEPARTMENT OF ENERGY
DIVISION OF SOLAR ENERGY

by

OCEAN SYSTEMS
LOCKHEED MISSILES & SPACE COMPANY, INC.
SUNNYVALE, CALIFORNIA 94086

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FOREWORD

The Lockheed Missile and Space Company, Inc. (LMSC) together with corporate team members and subcontractors, has completed Task 2, Phase I, of the Power System Development (PSD-I) project. This report presents the results of an eight-month study performed for the United States Department of Energy, Division of Solar Energy, under contract EG-77-C-03-1568 and consists of:

- Conceptual design of a 50-MW(e) Commercial plant size Power Module (CPM)
- Preliminary design of a 10-MW(e) Modular Application Power System (MAPS)
- Preliminary design of Heat Exchanger Test Articles (HETA)

The corporate team members were:

- Lockheed Electronics Co., (LEC), Houston, TX
- Lockheed Shipbuilding and Construction Co., (LSCC), Seattle, WA

The subcontractors were:

- AIRCO Cryogenics, Newport Beach, CA
- Bechtel National Inc., (BNI), San Francisco, CA
- Foster Wheeler Energy Corporation (FWEC), Livingstone, NJ
- Rotoflow Corporation, Los Angeles, CA

This report presents a technical description of the closed cycle ammonia power system modules derived for a reference surface platform in an immersed configuration (with two variations) as well as an alternative power system incorporated into a submerged detachable module.

The complete report consists of the following volumes:

Volume I – Preliminary Design Report LMSC D-630248

Volume II – Appendixes:

- Appendix A. System Analysis and Performance (Computer Runs, Sensitivity Analysis, Individual Studies Etc.)
- Appendix B. System and Component Specs for TA and 10 MW Drawings
- Appendix C. General Arrangements Envelope, Flow, Piping, Subsystem, Etc.(Structure Analysis, Stability Analysis)

Volume III – Appendixes:

- Appendix D. System Equipment (Hardware Breakdown, Equipment Tables Maintenance and Support Equipment)
- Appendix E. Heat Exchangers Supporting Data
- Appendix F. Rotating Machinery

Volume IV – Appendix G. Ammonia Cycle and Auxiliaries

Volume V – Appendixes:

- Appendix H. Electrical Power
- Appendix I. Control and Instrumentation
- Appendix J. Module Assembly
- Appendix K. Cost Reporting Tables

APPENDIX D

SYSTEM EQUIPMENT

CONTENTS

APPENDIX D1 HAEDWARE BREAKDOWN STRUCTURE
POWER SYSTEM HARDWARE RESPONSIBILITIES

		Module Type 50 MW(e)			10 MW(e)		
		Submerged Unitized	Immersed Unitized	Immersed Segmented	Immersed Unitized	Immersed Segmented	Self - Contained
1.	HEAT EXCHANGERS						
1.1	Evaporator	L/1	L/1	L/1	L/1	L/1	L/1
1.2	Condenser	L/1	L/1	L/1	L/1	L/1	L/1
1.3	Mist Extractor (Note 1)	L/1	L/1	L/1	L/1	L/1	L/1
2.	SEAWATER SUBSYSTEMS (Pumps & ducts Connecting to Heat Exchanger)						
2.1	Warm Water (Note 2)	L/1	L/1	L/1 (L/2 or 3)	L/1 (L/2 or 3)	L/1 (L/2 or 3)	L/1 (L/2 or 3)
2.2	Cold Water	L/1	L/1	L/1 (L/2 or 3)	L/1 (L/2 or 3)	L/1 (L/2 or 3)	L/1 (L/2 or 3)
3.	TURBINE GENERATOR						
3.1	Turbine (With Speed Control and Lubrication	L/1	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)
3.2	Generator (With Cooling)	L/1	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)	L/1 (L/2)
4.	AMMONIA CYCLE						
4.1	Vapor Assembly (Piping & Valves)	L/1	L/1	L/1 & 2	L/1	L/1 & 2	L/1 & 2
4.2	Liquid Assembly						
4.2.1	Condensate Subassembly	L/1	L/1 (L/1 & 2)	L/1 & 2	L/1 (L/1 & 2)	L/1 & 2	L/1 & 2
4.2.2	Distribution Subassembly	L/1	L/1	L/1	L/1	L/1	L/1

KEY: Z/N
Z = Prime Responsibility
L = Lockheed Missiles & Space
U = Department of Energy
N = Relationship
Provided by Power System & Dedicated to Module
1=Integral with Module
2=Separable from Module
Provided by Platform
3=Dedicated to Module
4=Shared by Modules
5=Services the Module

NOTE 1 Internal to spherical shell evaporator
NOTE 2 Parenthetical listings are practical alternatives
NOTE 3 Vapor recovery, vent and purge, non condensible gas removal systems are integrated
NOTE 4 Modules which include testing as an objective

		Module Type					
		50 MW(e)			10 MW(e)		
		Submerged Unitized	Immersed Unitized	Immersed Segmented	Immersed Unitized	Immersed Segmented	Self - Contained
4.3	Ammonia Cycle Auxiliaries						
4.3.1	Storage and Fill System	L/5 (U/5)	L/5 (U/5)	L/5 (U/5)	L/5 (U/5)	L/5 (U/5)	L/1 (L/2)
4.3.2	Unloading System	L/1	L/1	L/1	L/1	L/1	L/1
4.3.3	Vapor Recovery System (Note 3)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/1
4.3.4	Vent and Purge System (Note 3)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/1
4.3.5	Noncondensible Gas Removal System (Note 3)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/1
4.3.6	Ammonia Cleanup System	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/4 (U/5)	L/1
4.3.7	Leak Detection System (to air)	L/1 (U/4)	L/1 (U/4)	L/1 (U/4)	L/1 (U/4)	L/1 (U/4)	L/1
4.3.8	Leak Detection System (to water)	L/1 (U/3)	L/1 (U/3)	L/1 (U/3)	L/1 (U/3)	L/1 (U/3)	L/1
4.3.9	Fluid Sampling System (Ammonia Contamination)	L/1	L/1	L/1	L/1	L/1	L/1
5.	CONTROL SYSTEM						
5.1	Console and Dispalys	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2
5.2	Data Processing	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/2 (U/3 or 4)	L/1 or 2
5.3	Control Instrumentation	L/1	L/1	L/1	L/1	L/1	L/1
5.4	Cabling and Connectors	L/1	L/1	L/1	L/1	L/1	L/1
5.5	Software	L/4	L/4	L/4	L/4	L/4	L/1 or 4
5.6	Communications	L/1 & 5	L/1 & 5)	L/1 & 5)	L/1 & 5)	L/1 & 5)	L/1
6.	AUXILIARY SYSTEMS						
6.1	Operational Support						
6.1.1	Power Distribution System	L/1	L/1	L/1	L/1	L/1	L/1
6.1.2	Auxiliary Power Generation (Standby Power)	L/5 (U/5)	L/5 (U/5	L/5 (U/5)	L/1 or 4) (U/5)	L/1 or 4) (U/5)	L/1
6.1.3	Startup Power	L/5 (U/5)	L/5 (U/5	L/5 (U/5)	L/1 or 4 (U/5)	L/1 or 4 (U/5)	L/1

		Module Type					
		50 MW(e)			10 MW(e)		
		Submerged Unitized	Immersed Unitized	Immersed Segmented	Immersed Unitized	Immersed Segmented	Self - Contained
6.2	Fouling Countermeasure						
6.2.1	Chemical	L/4 (U/4)	L/4 (U/4)	L/4 (U/4)	L/1 or 4 (U/4)	L/1 or 4 (U/4)	L/1
6.2.2	Mechanical	L/1	L/1	L/1	L/1	L/1	L/1
6.3	Corrosion Control	L/1 & 2 (L/4)	L/1 & 2 (L/4)	L/1 & 2 (L/4)	L/1 & 2 (L/4)	L/1 & 2 (L/4)	L/1 & 2
6.4	Safety and Emergency Equipment						
6.4.1	Sensors and Alarms	L/1	L/1	L/1	L/1	L/1	L/1
6.4.2	Ammonia Washdown & Deluge	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1
6.4.3	Fire Suppression	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1
6.4.4	Emergency Power and Lighting	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1
6.4.5	Damage Control	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1 & 4 (L/1 &U/4)	L/1
6.5	Personnel Safety						
6.5.1	Washdown	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1
6.5.2	Breathing Equipment	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1
6.5.3	Protective Clothing	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1
6.5.4	Safety Equipment	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1 & 4 (U/5)	L/1
6.6	Ancillary Subsystems						
6.6.1	Compressed Air	U/5	U/5	U/5	U/5 (L/4)	U/5 (L/2 or 4)	L/1

		Module Type					
		50 MW(e)			10 MW(e)		
		Submerged Unitized	Immersed Unitized	Immersed Segmented	Immersed Unitized	Immersed Segmented	Self - Contained
6.6.2	Hydraulic Power	U/4	U/4	U/4	U/4 (L/2 or 4)	U/4 (L/2 or 4)	L/1
6.6.3	Ballasting and Attitude Control	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4
6.6.4	Potable Water	U/5	U/5	U/5	U/5	U/5	U/5
6.6.5	Heating Ventilating and Air Conditioning	U/5	U/5	U/5	U/5 (L/1 or 2)	U/5 (L/1 or 2)	L/1
7.	STRUCTURE AND ARRANGEMENT						
7.1	Structure	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3
7.2	Personnel Access	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3	L/1 & 3
7.3	Material and Consumable Storage	L/1 & 5	L/1 & 5	L/1 & 5	L/1 & 5	L/1 & 5	L/1 & 5
8.	SPECIAL ANCILLARY EQUIPMENT	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4	L/1 & 4
9.	TEST INSTRUMENTATION (Note 4)	L/1 & 2 & 3 & 4	L/1 & 2 & 3 & 4	L/1 & 2 & 3 & 4	L/1 & 2 & 3 & 4	L/1 & 2 & 3 & 4	L/1 & 2 & 3 & 4
10.	SUPPLIES						
10.1	Ammonia (Anhydrous)	U/4 or 5	U/4 or 5	U/4 or 5	U/4 or 5	U/4 or 5	L/1

10-MW(e) HARDWARE LISTING

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Warm Seawater	Warm Seawater Pump	4	Axial Propeller	178,200 gpm at 6.4 ft and 150 rpm	Efficiency: 88% Suction Bell: 11.4 ft OD Shaft Diam: 6 in.; Material: 316L Stainless Steel		20.6-ft long impeller; 7.[illegible]5-ft diam.	58,300
	Warm Seawater Pump Motor	4	Submerged	565 hp, 1,160 rpm	Efficiency: 90% Three-Phase, 60 Cycle		3.7 ft long; 2.5 ft diam.	2 500
	Warm Seawater Pump Reduction Gear		Reduction	–	–		–	1 320
	Warm Seawater Pump Motor Cooling	–	–	–	Motor is Submerged With Cooling Fins		–	–
Cold Seawater System	Cold Seawater Pump	4	Axial Propeller	160,400 gpm at 10.17 ft and 150 rpm	Efficiency: 88% Suction Bell: 9.67 ft OD Shaft Diam: 6 in. Material: 316L Stainless Steel		17.4 ft long Impeller; 6.5 ft diam	37,000
	Cold Seawater Pump Motor	4	Submerged	725 hp, 1,160 rpm	Efficiency: 90% Three-Phase, 60 Cycle		4.1-ft long 2.5-ft diam.	2,800
	Cold Seawater Pump Reduction Gear		Reduction	–	–		–	2,460
	Cold Seawater Pump Motor Cooling		–	–	Motor is Submerged With Cooling Fins		–	–
Evaporator System	Evaporator	1	Spherical Shell, Horizontal Titanium Tube	51,400 Tubes; 1 in. Nom Diam; 0.028 in. Min. Wall; 0.025 in. High Fins (15 to 20 fpi)	S.W. Flow Rate: 1510 cfs S.W. Temp. In: 80°F S.W. Temp. Out: 75.17°F Design Pressure: 168 psi		39 ft 5.21 diam 26 ft 7 in. long Distance Between Tubesheet Faces 312.38 in.	Dry: 1,052,800 NH_3: 47,500 S.W: 768,000
Condenser System	Condenser Sump (Integral With Condenser)	1	Spherical Shell, Horizontal Titanium Tube	64,200 Tubes; 1 in. Nom. Diam; 0.028 in. Min. Wall; 0.025 in. High Fin (15 to 20 fpi)	S.W. Flow Rate: 1,312 cfs S.W. Temp. In: 40°F S.W. Temp. Out: 45.34°F Design Pressure: 168 psi		31 ft 5.21-in. diam. 26 ft 7 in. long Distance Between Tubesheet Faces 312.38 in.	Dry: 1,124,480 NH_3: 38,259 S.W: 676,100
Evaporator	Ammonia Distribution Pump	4	Single-Stage Vertical Centrifugal	5,240 gpm, 75-ft head	3 Operating 1 Standby	Evaporator Pump Room	3-ft diam. × 7-1/2 ft without motor	4,500 1 (ea.)
	Ammonia Distribution Pump Motor	4	Induction	70 Shaft hp, 1,160 rpm (max load)	3 Operating 1 Standby	Evaporator Pump Room		1,400
Condenser	Ammonia Condensate Pump	3	Three-Stage Vertical Centrifugal	5,240 gpm, 241 ft Head	2 Operating 1 Standby	Condensate Pump Room	3 ft diam × 11 ft without motor	5,500 1 (ea.)
	Ammonia Condensate Pump Motor	3	Induction	233 Shaft hp, 1,160 rpm (Payload)	2 Operating 1 Standby	Condensate Pump Room		2,000

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Ammonia Liquid Distribution System	Ammonia Filter	7	Startup Strainer	12-in. Line	Temporary Startup Strainers	Condensate and Distribu-tion Pump Inlets		
	Ammonia Valves	3	Butterfly	24 in.	Condensate Pump Suction Line	Condensate Pump Suction		640
		3	Butterfly-Hyd. Operated Stop-Check	12 in.	Condensate Pump Discharge Line	Condensate Pump Discharge		200
		2	Butterfly-Hyd. Operated	18 in.	Condensate Pump Discharge Manifold	Condensate Pump Discharge Manifold		400
		4	Butterfly	24 in.	Distribution Pump Suction Line	Distribution Pump Suction		640
		4	Butterfly Hyd. Operated Stop-Check	12 in.	Distribution Pump Discharge Line	Distribution Pump Discharge		200
Ammonia Vapor System	Ammonia Valves	2	Butterfly-Hyd. Operated	36 in.	Main Vapor Isolation Valve	Ammonia Vapor Line		1,900
		2	Butterfly-Hyd. Operated	36 in.	Turbine Throttle Valve	Turbine Inlet		1,900
		1	Butterfly	36 in.	Turbine Bypass	Turbine Inlet Manifold		1,380
		1	Butterfly Hyd. Operated	72 in.	Turbine Outlet	Turbine Outlet		3,600
Ammonia Conden-sate System	Ammonia Piping	1	Carbon Steel	36-in. Schedule 30 0.625-in. Wall	Condensate Pump Suction Manifold	Condensate Pump Suction	15 ft	3,600
		3	Carbon Steel	24-in. Schedule 20 0.375-in. Wall	Condensate Pump Inlet Piping	Condensate Pump Inlet	55 ft (total)	5,300
		3	Carbon Steel	12-in. Schedule 20 0.25-in. Wall	Condensate Pump Discharge Piping	Condensate Pump Outlet	15 ft	500
		1	Carbon Steel	18-in. Schedule 30 0.438-in. Wall	Condensate Pump Discharge Manifold	Condensate Pump Discharge Manifold	180 ft	15,200

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Ammonia Liquid Distribution System	Ammonia Piping	1	Carbon Steel	48-in. Schedule 30 0.75-in. Wall	Distribution Pump Suction Manifold	Distribution Pump Suction	15 ft	5,800
		4	Carbon Steel	24-in. Schedule 20 0.375-in. Wall	Distribution Pump Inlet Piping	Distribution Pump Inlet	55 ft	5,300
		4	Carbon Steel	12-in. Schedule 20 0.25-in. Wall	Distribution Pump Outlet Piping	Distribution Pump Outlet	12 ft	400
		1	Carbon Steel	16-in. Schedule 20 0.312-in. Wall	Distribution Pump Outlet Manifold	Distribution Pump Discharge Manifold	15 ft	800
Evaporator Irrigation System	Ammonia Piping	2	Carbon	14-in. Schedule 20 0.312-in. Wall	Evaporator Irrigation Distribution Header	Evaporator	27 ft	1,300
		2	Carbon Steel	10-in. Schedule 20 0.250-in. Wall	Evaporator Irrigation Distribution Header	Evaporator	8 ft	200
		8	Carbon Steel	8-in. Schedule 20 0.250-in. Wall	Evaporator Irrigation Distribution Line	Evaporator	90 ft	1,900
Ammonia Vapor System		2	Carbon Steel	36-in. Schedule 30 0.625-in. Wall	Evaporator Vapor Line to Turbine Inlet	Evaporator Outlet; Turbine Inlet	155 ft (total)	37,500
Diffuser		1	Carbon Steel	6 ft Inlet Diam 11 ft Outlet Diam 0.50-in. Wall	Reinforced With Stiffeners	Turbine Exhaust	28 ft	15,000
Ammonia Storage and Fill System	Ammonia Storage Tanks	4	Cylindrical	40,000 lb	Design Pressure: 250-psig		8.5 ft diam 25 ft high	Dry: 34,200 ea. Wet: 74,200 ea.
	Ammonia Fill Evaporator	1	Shell and Tube	1,600,000 Btu/hr	Heat Transfer Surface: 212 ft^2		1 ft 10 in. diam 9 ft long	400
	Fill Superheater	1	Shell and Tube	62,000 Btu/hr	Heat Transfer Surface: 44 ft^2		12 in. diam	100
Noncondensible Gas Removal System	Gas Removal Condenser	1	Shell and Tube	850,000 Btu/hr	Heat Transfer Surface: 116 ft^2 Tubes: Titanium Shell: Carbon Steel		1 ft 8 in. diam 4 ft long	200
	Ammonia Compressor	2	Reciprocating Pressure Crankcase	85 cfm	Compress. Ratio: 2:1 Disch. Pressure: 165 psig Motor hp: 5			230
	Seawater Pump	2	Centrifugal	600 gpm	Head: 100 ft Motor hp: 20			1,000

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Ammonia Cleanup System	Ammonia Evaporator	1	Shell and Tube Heat Exchanger	Heat Transfer Surface: 2,100 ft^2	Pressure (Shell): 250 psig Shell Side: Ammonia Tubeside: Seawater			3,000
	Compressor	1	Reciprocating Pressure Crankcase	1,200 cfm, 175 hp Motor	Inlet Pressure: 56 psia Disch. Pressure: 92 psia			4,200
	Ammonia Pumps	2	Centrifugal	2 gpm, 1/2 hp	Suct. Pressure: 56 psia Disch. Pressure: 92 psia			100
	Seawater Pumps	2	Centrifugal	3,000 gpm 125 hp Motor	Head: 100 ft			1,100
	Seawater Pump	1	Centrifugal	250 gpm 10 hp Motor	Head: 100 ft			200
	Ammonia Recovery Evaporator	1	Shell and Tube Heat Exchanger	Heat Transfer Surface: 120 ft^2	Pressure (Shell): 250 psig			200
	Distillation Column	1	Packed Column	Diam: 5 ft Length: 6 ft	Titanium Internals			1,000
	Ammonia Column Re-Boiler	1	Electrically Heated	100 kW	Titanium Internals			200
Ammonia Drain System	Ammonia Collection Tanks	2	Horizontal Cylindrical	Volume: 200 ft^3 Diam: 5 ft Length: 10 ft	Disch. Pressure: 250 psia			Dry: 3,000 Wet: 9,500
	Ammonia Transfer Pumps	4	Vertical Centrifugal Sump	150 gpm Motor hp: 10	Discharge Head: 100 ft			350
Ammonia Vapor Recovery Subsystem	Recovery Condenser	1	Shell and Tube	1,750,000 Btu/hr Diam: 16 in. Length: 6 ft	Heat Transfer Surface 115 ft^2			200
	Ammonia Recovery Compressors	2	Reciprocating Pressure Crankcases	75 cfm Motor hp: 40	Compression Ratio: 11:1 Disch. Pressure: 180 psig			1,000
Ammonia Vent and Purge System	Vacuum Pump	2	Rotating Cylinder	140 cfm Motor hp: 10	Vacuum Capacity: 1 psia			400
	Exhauster	2	Low Pressure Centrifugal	100,000 cfm Motor hp: 100	Suction Pressure: 3 in. H_2O			2,000
	Blower	2	Centrifugal	4,800 cfm	Disch. Pressure: 15 in. H_2O			500
Ammonia Vapor System	Ammonia Separator/ Demister				Integral to Evaporator	Evaporator		

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Leak Detection System	Infrared Ammonia Analyzer	10	Infrared	0 to 300 rpm				250
	Infrared Ammonia Analyzer	6	Infrared	0 to 3,000 rpm				200
	Combustible Gas Detection System	6	Combustible Gas					
Fluid Sampling System	Electrical Conductivity Cell and Converter	2						
	Thermal Conductivity Cell and Readout							
	Nitrogen Storage Cylinders	10	Cylinder	30,000 lb Capacity	Charge Pressure: 2,000 psig		2 ft diam × 40 ft	Dry: 20,000 Wet: 23,000
Ammonia Vapor System	Ammonia Turbine		Radial	778 lb/sec Inlet Pressure: 127.7 psia Outlet Pressure: 89.4 psia	Isentropic Enthalpy Head: 19.1 Btu/lb Efficiency: 99%		6 ft × 6 ft × 7 ft	40,000
Electrical System	Generator and Exciter		Two-Pole Brushless	Voltage: 12,800 Line-to-Line 3,600 rpm 3 pH, 60 Hz Efficiency: 98% 0.8 Power Factor	Rated 14,400 kW Voltage Supp < 15%			118,000
Rotating Machinery support equipment	Lube Oil Console	1		25 gpm at 8 to 10 psi 130 to 180 SSU at 100%	System Description is Attached		310 in. × 42 in. × 90 in.	15,000
Data Acquisition and Control System (DACS)		1	Modcomp II Processor					5,282
Computer Readout Terminal		2	Intelligent Systems Corp.	105 to 125 V, 60 Hz, 150 Watts, 19 in. Screen			22-1/2 in. × 19-3/8 in. × 17-1/2 in.	75 each
Remote Terminal Unit (RTU)		3				Machine Room (2) Machine Space (1)	24 in. × 21 in. × 72 in.	150 each

Service or Drive	Unit	No. Per Ship	Type	Capacity or Size	Remarks	Location	Dimensions	Weight (lb)
Chlorination System	Chlorine Generators Chloropac Units	2	Engelhard Chloropac	85 lb/hr Chlorine to Treat 680K gpm at 0.5 ppm	(Including Power Supply) 750 kW Max Capacity 150 kW in Combination With M.A.N. (Mechanical Brush Cleaning) Brush System Air Cooled Power Supply		Generator 215 in. × 75 in. × 20 in. Power Supply 60 in. × 60 in. × 60 in.	2,600 7,300
Fouling Counter-Measures	Evaporator Mechanical Cleaning System	1	M.A.N. Brush Cleaning	1-in. OD Brush and Cage	102,800 Cages 51,400 Brushes	Evaporator	1 in. diam. × 2-5/8 in.	600
	Condenser Mechanical Cleaning System	1	M.A.N. Brush Cleaning	1-in. OD Brush and Cage	128,000 Cages 64,000 Brushes	Condenser	1 in. diam × 2-5/8 in.	750
Cathodic Protection	Cathodic Protection System	132	Sacrificial Zinc Anode BTZ-27	5 yr Life 10 mA/ft^2	Replaced During Heat Exchanger Refurbishment	Heat Exchanger	69 in. long	3,464

APPENDIX D3

LIST OF SUPPORT AND MAINTENANCE EQUIPMENT, TOOLS AND SPARE PARTS

10-MW(e) Power System Module

1. Special Tools - None
2. Standard Tools - Common tools will be standard equipment
3. Chain Falls - 4 each, 5 ton capacity; 2 each, 10 ton capacity
4. Hoist System - 2 complete sets with 8 ton hoist, rails, lifting harness, slings, Bosun chair, etc.
5. Trolley System - 2 complete sets with 8 ton trolley, and rails
6. Jib Crane, 5 ton - 2 sets
7. Milling Cutter, air driven - for tube cutting/plugging.
8. Tube Plugging Set - Capacity to plug 7% of all tubes
9. Sensors, Cabling and Connectors - 10% of each type installed
10. Electronic Repair Parts for Major Components - Per vendor (TBD)
11. Turbine Power Cartridge - One (Radial Only)
12. L.O. Filters - 100% disposable type; 5% renewable type
13. Beam and Lifting Slings - Heat exchanger handling
14. Support Cradles - 2 - Shipping, testing and repair.
15. Valves, Under 3" Size - 5% of each type fitted
16. Gaskets - 50% of each type fitted.
17. Circuit Breakers - 1 for each type fitted over 250 amps; 4 for each type fitted under 250 amps.
18. Bearings - One complete set for each type and size motor and generator fitted (ABS)
19. Instrumentation and Test Gear - One 500V megger (ABS)
20. Valve Springs - One set for each governor, and relief valve (ABS)
21. Packing Gland Seals - One set for each packing gland type and site (ABS)
22. Thrust Pads - One set, complete with springs for each size thrust bearing (ABS)
23. Compressor Unloading Valves - One set for each type fitted (ABS)
24. Valve Control Elements - One set for each type fitted
25. Contact Relays - 10% of total fitted for each started type.
26. Fan Motors - One complete set for each type fitted (ABS)
27. Pump Rotating Assembly - One complete set for each type fitted, except large ammonia pumps

APPENDIX D3

LIST OF SUPPORT AND MAINTENANCE EQUIPMENT, TOOLS AND SPARE PARTS

0.2-MW(e) Test Articles

1. Special Tools - None
2. Standard Tools - Common tools will be standard equipment
3. Tube Plugging Set - Capacity to plug 7% of all tubes
4. Sensors, Cabling and Connectors - 10% of each type installed
5. Beam and Lifting Slings - For handling

Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1A1 - 13.8 kV - MEDIUM VOLTAGE SWITCHGEAR 13.8 kV - 580 MVA - Class 3Ø - Co H3 - (Vacuum Type) Marine Service Metal Clad Switchgear 1 - 1200 A Incoming Breaker with Auxiliary Compartment 1 - 1200 A Outgoing Breaker with an Auxiliary Compartment 1 - 1200 A Feeder Breaker	72"L x 109"W x 111"H	7750 lbs. 4750 lbs.	20,200 lbs.	Per Standards - CG-259 IEEE-45 & ABS Rules located in platform
1A2 - 4.16 kV - MEDIUM VOLTAGE SWITCHGEAR MAIN MODULE SWITCHGEAR 4.16 kV - 250 MVA - Class 3Ø - Co H3 - (Vacuum Type) Marine Service Metal Clad Switchgear 1 - 1200 A Incoming Breaker with Auxiliary Compartment 8 - 1200 A Feeder Breakers	216"L x 109"W x 111" H	7750 lbs. 4750 lbs.	31,750 lbs.	Per Standards - CG-259 IEEE-45 & ABS Rules located in unclassified * area Variable Speed Drive System
1A2 - 4.16 kV - MEDIUM VOLTAGE SWITCHGEAR CONSTANT SPEED DRIVE SYSTEM 4.16 kV - 250 MVA - Class 3Ø - Co H3 - (Vacuum Type) Marine Service Metal Clad Switchgear 1 - 1200 A - Incoming Breaker with Auxiliary Compartment 13 - 1200 A - Feeder Breakers	108"L x 109"W x 111"H	7750 lbs. 4750 lbs.	17,250 lbs.	Constant Speed Drive System
1A3 - 4.16 kV - MEDIUM VOLTAGE VARIABLE FREQUENCY SWITCHGEAR 4.16 kV - 250 MVA - Class 3Ø - Co H3 - (Vacuum Type) Marine Type Metal Clad Switchgear 1 - 1200 A Incoming Breaker with Auxiliary Compartment 4 - 1200 A Feeder Breakers	108"L x 109:W x 111"H	7750 lbs. 4750 lbs.	17,250 lbs.	Per Standards - CG-259 IEEE-45 ABS Rules located in unclassified area Variable Speed Drive System

Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1A4 - <u>4.16 kV - MEDIUM VOLTAGE VARIABLE FREQUENCY SWITCHGEAR</u> 4.16 kV - 250 MVA - Class 3Ø - Co Hz - (Vacuum Type) Marine Type Metal Clad Switchgear 1 - 1200 A Incoming Breaker with Auxiliary Compartment 4 - 1200 A Feeder Breakers	108"L x 109"W x 111"H	7750 lbs. 4750 lbs.	17,250 lbs.	As above.
1A5 - <u>4.16 kV - MEDIUM VOLTAGE VARIABLE FREQUENCY SWITCHGEAR</u> 4.16 kV - 250 MVA - Class 3Ø - Co Hz - (Vacuum Type) Marine Type Metal Clad Switchgear 1 - 1200 A Incoming Breaker with Auxiliary Compartment 3 - 1200 A Feeder Breakers	108"L x 109"W x 111"H	7750 lbs. 4750 lbs. 4150 lbs.	16,650 lbs.	As above.
1A6 - <u>EXTERNAL POWER SOURCE DISCONNECTING SWITCH</u> 600 A - 5 kV - Load Interrupter Switchgear	33"L x 70"W x 100"H	1200 lbs.	1200 lbs.	Platform Equipment Per Standards - CG -259 IEEE-45 ABS - Rules located in platform.
1B4 - <u>480 V - LOAD CENTER</u> 2000 A - 480 V - 3Ø - Co Hz - Load Center Marine Type Switchgear 1 - 2000 A Incoming Breaker 4 - 600 A Feeder Breakers 8 - 600 A Motor Feeder Breakers	102"L x 68"W x 90" H	2004 lbs. 1650 lbs.	6,950 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules located in unclassified area.

Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1B2 - 480 V - MOTOR CONTROL CENTER				
600 A - 22,000 A Symmetrical Marine Type Motor Control Center				
7 - NEMA Size - 1, Starters - FVNR 7 - NEMA Size - 2, Starters - FVNR 3 - NEMA Size - 3, Starters - FVNR 10 - NEMA Size - 1, Starters - FVR 2 - 30 kVA - Transformer 2 - 208Y/120V - NAB - Panels (Back to back arrangement)	120"L x 40"W x 90"H	1000 lbs.	6,000 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules located in unclassified area.
1C1 - MAIN PROCESS CONTROL PANEL				
(a) Main Control Console	108"L x 48"W x 72"H	1200 lbs.	2400 lbs.	Power Module Main Control Room Equipment
(b) Main Process Control Panel	108"L x 48"W x 90"H	1200 lbs.		
1D1 - 4160V - VARIABLE FREQUENCY PUMP DRIVE				
(a) 1 - 2500 HP - 4160V - 3Ø - Co Hz Adjustable Frequency Converter Only	180"L x 20"W x 90"H	44,000 lbs.	74,000 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules installed in (b) (a) (c) order length wise. Located in unclassified area.
(b) 1 - Step Down - Isolation Transformer 4.16kV - 480V - 3Ø Co Hz. Dry Type.	84"L x 55"W x 50"H	15,000 lbs.		
(c) 1 - Step Up - Isolation Transformer 480V - 4.16kV - 3Ø - Co Hz. Dry Type.	84"L x 55"W x 90"H	15,000 lbs.		
1D2 - 4160V - VARIABLE FREQUENCY PUMP DRIVE				
(a) 1 - 1600 HP - 4160V - 3Ø - Co Hz Adjustable Frequency Converter Only	126"L x I x 20"W x 90"H	25,000 lbs.	52,000 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules installed in (b) (a) (c) order length wise. Located in unclassified area.
(b) 1 - Step Down - Isolation Transformer 4.16 kV - 480V - 3Ø - Co Hz. Dry Type.	84"L x 55"W x 90"H	13,500 lbs.		
(c) 1 - Step Up - Isolation Transformer 480V - 4.16 kV - 3Ø - Co Hz. Dry Type.	84"L x 55"W x 90"H.	13,500 lbs.		

Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1D3 - <u>4160V - VARIABLE FREQUENCY PUMP DRIVE</u>				
(a) 1 - 750 Up - 4160V - 3Ø - Co Hz Adjustable Frequency Converter Only	80"L x 20"W x 90"H	15,000 lbs.	27,800 lbs.	Per Standards - CG-259 IEEE-45
(b) 1 - Step Down - Isolation Transformer 4.16 kV - 480V - 3Ø - Co Hz. Dry Type.	72"L x 55"W x 90"H	6400 lbs.		ABS - Rules Install in (b) (a) (c) order lengthwise. Located in unclassified area.
(c) 1 - Step Up - Isolation Transformer 490V - 4.16 kV - 3Ø - Co Hz. Dry Type.	72"L x 55"W x 90"H	6400 lbs		
1D4 - <u>480V - 160 HP - VARIABLE FREQUENCY PUMP DRIVE</u>				
(a) 1 - 160 HP - 480V - 3Ø - Co Hz Adjustable Frequency Converter Only	64"L x 20"W x 90"H	2750 lbs.	5,750 lbs.	Per Standards - CG-259 IEEE-45
(b) 8 - NEMA Size - 3 - Starters 20" Section - Arranged Back to Back	40"L x 40"W x 90"H	3000 lbs.		ABS - Rules Install lengthwise. Located in unclassified area.
1D5 - <u>125V - DC - EMERGENCY POWER SYSTEM</u>				
(a) 1 - 1000 AH - 125V - DC - Battery Cells and Marine Service Battery Racks	180"L x 180"W x 100"H Space	3000 lbs.	3,000 lbs.	Per Standards - CG-259 IEEE-45
(b) 1 - 150 A - Battery Charger 480V - 3Ø - Co Hz Input 125V - DC - Output	50"L x 25"W x 60"H	4800 lbs.	4,800 lbs.	ABS - Rules 124V - DC Power Supplies Power To:
(c) 1 - 15 KVA - Regulated Power Transformer 480V - AC - Input - 120V - AC Regulated Output	36"L x 30"W x 90"H	2200 lbs.	2,200 lbs.	o 125V - DC Control o 125V - DC Inverter System for 120V - AC Output.
(d) 1 - 15 KVA - 125V - DC/ 120V - AC Inverter Unit	55"L x 33"W x 90"H	2400 lbs.	2,400 lbs.	
1D6 - <u>250V - DC - EMERGENCY POWER SYSTEM</u>				
(a) 1 - 1000 AH - 250V - DC - Battery Cells and Marine Service Battery Racks	180"L x 180"W x 100"H Space	6000 lbs.	6,000 lbs.	Per Standards - IEEE-45 CG-259
(b) 1 - 100 A - Battery Charger 480V 3Ø - Co Hz - Input 250V DC - Output	30"L x 30"W x 90"H	1640 lbs.	1,640 lbs.	ABS - Rules 250V - DC - Power Supplies Power To: o Turbine Generator Lube/ Seal System

Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1X1 - UNIT AUXILIARY TRANSFORMERS				
4500/6000 kVA - AA/FA - 150°C 3 - 1500 kVA - 1Ø - Transformer Delta - Wye - Connection 13.8 kV - 4.16 kV - 3Ø - CO HZ Z - 6% - (4500 kVA - Base) Dry-Type Transformer Transformer Include:	90"L x 68"W x 96"H	10,000 lbs.	13,550 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules Located in Unclassified Area
2 - Incoming Line Sections - 2 x 36" Wide for 3Ø Connection	72"L x 68"W x 90"H	475 lbs.		
1X2 - 4.16 kV - GROUNDING TRANSFORMER				
25 kVA - 4.16 kV - 120/240V 1Ø - Dry Type Transformer and Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	Ground Leakage Detection Unit on 4.16 kV - System
1X3 - 13.8 kV - GROUNDING TRANSFORMER				
25 kVA - 13.8 kV - 120/240V 1Ø Dry Type Transformer and Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	Ground Fault Generator Protection. (Part of Generator Package.)
1X4 - 480V - LOAD CENTER TRANSFORMER				
1000/1333 kVA - AA/FA - 150°C 4.16 kV - 480 Y/277 - 3Ø - Co HZ Load Center Transformer	120"L x 68"W x 96"H	12,500 lbs.	12,500 lbs.	Load Center Transformer for 1E4
1X5 - GROUNDING TRANSFORMER				
5 kVA - 480V - 120/240V - 1Ø - Dry Type, Ground Current Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	480V - System Ground Leakage Detection Unit

Description	Dimensions	Unit Wt.	Tlt. Wt.	Remarks
1D9 - 4160 V 1000 HP Variable Frequency Drive Control				
(a) 1000 HP 4160 V 3Ø - 60 HZ Rectifier/Inverter Variable Frequency Control Unit	150"L x 20"W x 90"H	15,000 lbs	28,000 lbs	Per standards IEEE-45, CG-259 ABS rules. Install lengthwise (b), (a), (c) 1AV3. Located in unclassified area.
(b) 1000 KVA Step-Down Insulation Transformer. 4160 V-480 V - 3Ø - 60 Dry-Type	72"L x 55"W x 90"H	6,000 lbs		
(c) 1000 KVA Step-Up Insulation Transformer. 4160 V - 480 V - 3Ø - 60 HZ Dry-Type	72"L x 55"W x 90"H	6,500 lbs		
1D10 4160 V - 1000 HP Variable Frequency Drive Control	Same as Item 1D9	As above	32,750 lbs	Same as above except 1AV4 replaces 1AV3.
1D11 - 4160 V - 2500 HP Variable Frequency Drive Control				
(a) 2500 HP - 4160 V - 3Ø - 60 HZ Rectifier/Inverter - Variable Frequency Control Unit				
(b) 2500 kVA Step-Down Isolator Transformer 4160 V - 480 V, 3Ø - 60 HZ Dry-Type				
(c) 2500 kVA - Step-Up Isolation Transformer - 4160 V - 480 V - 3Ø - 60 HZ Dry-Type				
1D12 - 4160 V 1250 HP Variable Frequency Drive Control				Per standards IEEE-45, CG-259, ABS rules.
(a) 1250 HP 4160 V 3Ø - 60 HZ Rectifier/Inverter Variable Frequency Control Unit	150"L x 20"W x 90"H	15,000 lbs		Install lengthwise (b), (a), (c) and (d) order. Located in unclassified area.
(b) 1500 KVA Step-Down Insulation Transformer. 4160 V - 480 V - 3Ø - 60 HZ	72"L x 55"W x 90"H	6,500 lbs	32,750 lbs	

Description	Dimensions	Unit Wt.	Ttl. Wt.	Remarks
1D12 - 4160 V 1250 HP Variable Frequency Drive Control (Continued)				
(c) 1500 KVA Step-Up Insulation Transformer. 4160 V - 480 V - 3Ø - 60 HZ Dry-Type	72"L x 55"W x 90"H	6,500 lbs.		
(d) 1200 A - 250 MVA Class Vacuum-Type Marine Service Metal Clad Switchgear	36"L x 109"W x 111"H	4,750 lbs.		

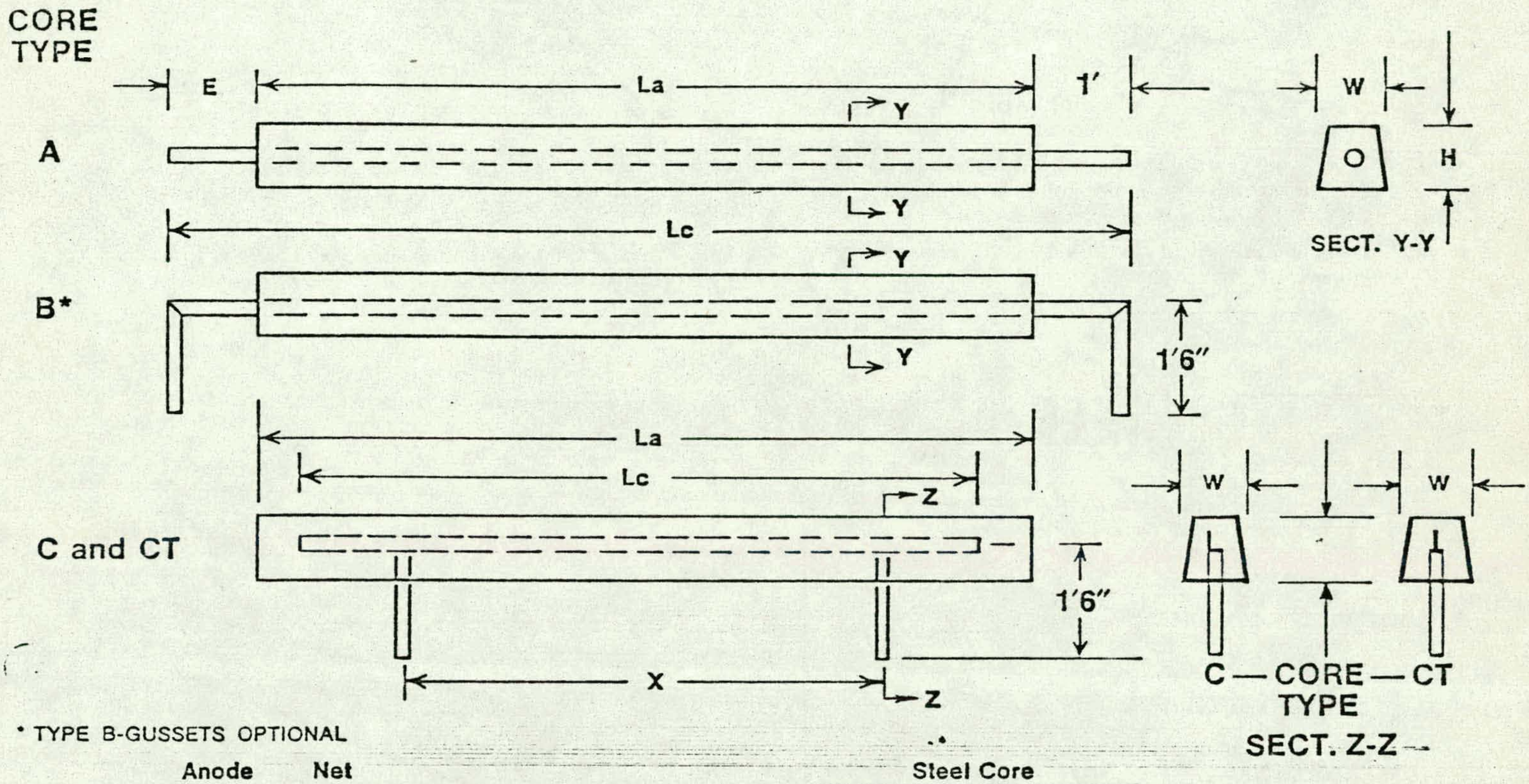
Description	Dimensions	Unit Wt. (lbs.)	Ttl. Wt. (lbs.)	Remarks
1X1 - <u>UNIT AUXILIARY TRANSFORMERS</u>				
4500/6000 kVA - AA/FA - 150°C 3 - 1500 kVA - 1Ø - Transformer Delta - Wye - Connection 13.8 kV - 4.16 kV - 3Ø - 60 HZ Z - 6% - (4500 kVA - Base) Dry-Type Transformer Transformer Include:	90"L x 68"W x 96"H	10,000 lbs.	13,550 lbs.	Per Standards - CG-259 IEEE-45 ABS - Rules Located in Unclassified Area
2 - Incoming Line Sections - 2 x 36" Wide for 3Ø Connection	72"L x 63"W x 90"H	475 lbs.		
1X2 - <u>4.16 kV - GROUNDING TRANSFORMER</u>				
25 kVA - 4.16 kV - 120/240V 1Ø - Dry Type Transformer and Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	Ground Leakage Detection Unit on 4.16 kV - System
1X3 - <u>13.8 kV - GROUNDING TRANSFORMER</u>				
25 kVA - 13.8 kV - 120/240V 1Ø Dry Type Transformer and Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	Ground Fault Generator Protection. (Part of Generator Package.)
1X4 - <u>480V - LOAD CENTER TRANSFORMER</u>				
1000/1333 kVA - AA/FA - 150°C 4.16 kV - 480 Y/277 - 3Ø - 60 HZ Load Center Transformer	120"L x 68"W x 96"H	12,500 lbs.	12,500 lbs.	Load Center Transformer for 1B4
1X5 - <u>GROUNDING TRANSFORMER</u>				
5 kVA - 480V - 120/240V - 1Ø - Dry Type, Ground Current Detection Unit	64"L x 40"W x 102"H	3,000 lbs.	3,000 lbs.	480V - System Ground Leakage Detection Unit

ALUMINUM OFFSHORE ANODES

ASARCO

Available in these anodes | **Galvalum**® or **Hi-Amp**™

Standard anode sizes listed. Other core arrangements and/or shorter anode lengths can be supplied in cross-sections shown.



* TYPE B-GUSSETS OPTIONAL

Anode No.	Net Al Wt.	WxH	La	Lc	X	Steel Core Type	Steel Core Schedule	
A 375	325	6½″x6½″	8′	10′	—	A	2″ schedule 80 pipe	
A 875	725	9½″x9½″	8′	10′	—	A	4″ schedule 80 pipe	
B 385	325	6½″x6½″	8′	10′	—	B	2″ schedule 80 pipe	
B 910	725	9½″x9½″	8′	10′	—	B	4″ schedule 80 pipe	
							Internal	**Legs-Pipe**
C 360	325	6½″x6″	8′	7′	5′	C	2″x2″x¼″ angle	2″ schedule 80
C 405	370	6½″x6½″	8′	7′	5′	C	2″x2″x¼″ angle	2″ schedule 80
CT 840	725	9½″x8½″	8′	7′	5′	CT	"T" 4″ D. x 5¼″ W. (ST4WF)	4″ schedule 80

Special anode sizes can be furnished in the cross-sections shown, from 4′ L to 8′ L.
The following nominal weights/ft. can be used for estimating:

Cross-section Core Type	6½″x6½″ 2″ schedule 80 pipe	6½″x6″ 2″x2″x¼″	9½″x9½″ 4″ schedule 80 pipe	9½″x8½″ "T" 4″ D. x 5¼″ W.
Net wt. Al lbs./ft.	40.6	40.6	90.5	90.5

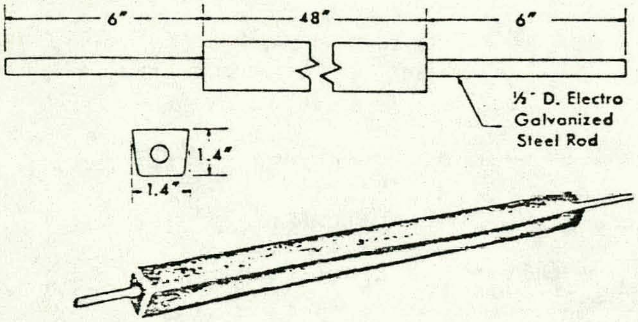
NOTE—DIMENSIONS & WEIGHTS ARE NOMINAL

AMERICAN SMELTING AND REFINING COMPANY
FEDERATED METALS DIVISION
Cathodic Protection Dept.
120 Broadway, New York, N.Y. 10005
212-732-9500

Bulletin No. L729-5-5M

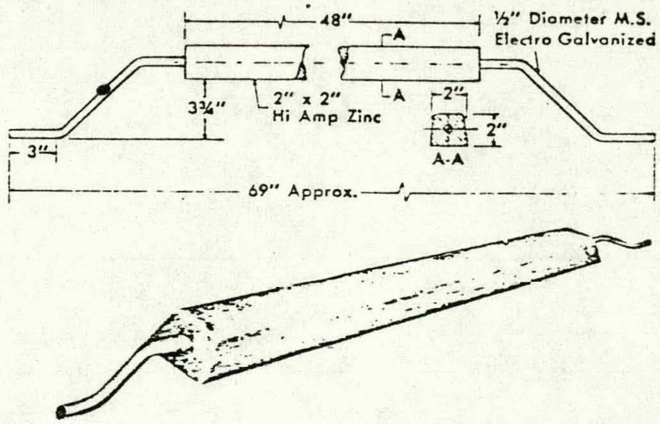
Printed In U.S.A.

BALLAST TANK ANODES



TZ-27: 1.4" x 1.4" x 48". Contains ½" diameter cast-in galvanized steel mounting core. Nominal weight, 27 lbs.; current rating, 1 amp-yr.

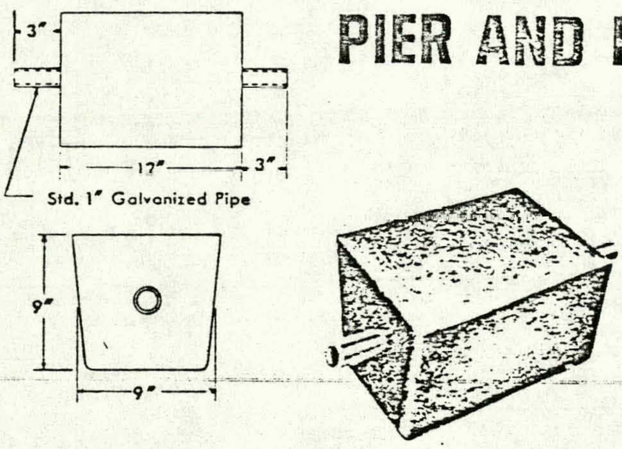
TZ-50: Also available, not shown, 2" x 2" x 48"; same core as TZ-27. Nominal weight, 50 lbs.; current rating, 2 amp-yrs.



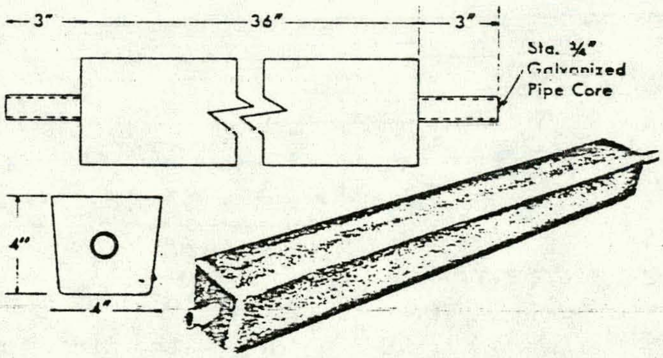
BTZ-50: 2" x 2" x 48". Contains ½" diameter cast-in galvanized steel mounting core. Nominal weight 50 lbs.; current rating, 2 amp-yrs.

BTZ-27: Also available, not shown. 1.4" x 1.4" x 48". Has nominal weight of 27 lbs. Core identical to BTZ-50. Current rating, 1 amp-yr.

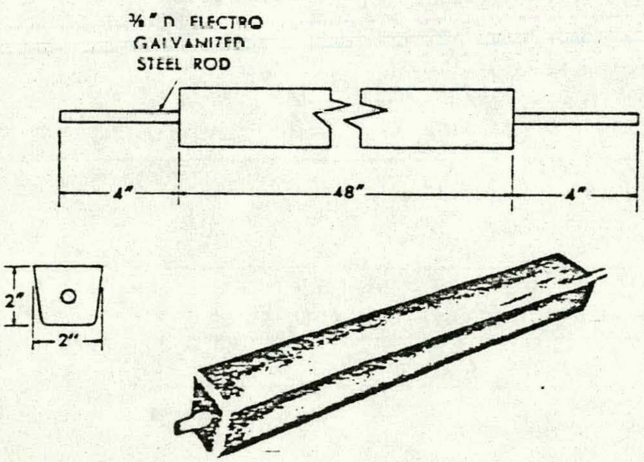
PIER AND PILING ANODES



PZ-250: 9" x 9" x 12". Contains 1" diameter cast-in galvanized pipe core extending 3" beyond each end; alternate size, 9" x 10" x 11". Nominal weight, 250 lbs.; current rating, 10 amp-yrs.

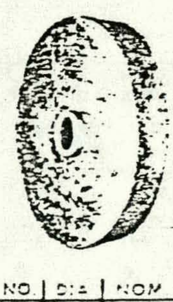


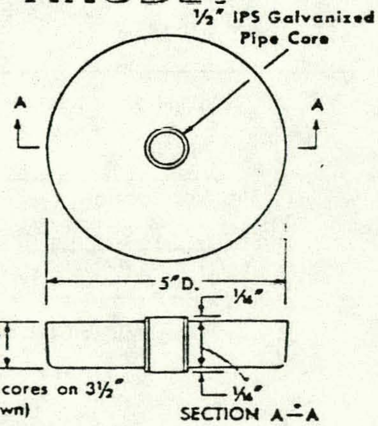
PZ-150: 4" x 4" x 36". Contains ¾" diameter cast-in galvanized steel pipe core extending 3" beyond ends. This anode can also be made with ⅜" diameter galvanized steel rod core. Nominal weight, 150 lbs.; current rating, 6 amp-yrs.



PZ-50: 2" x 2" x 48". Contains ⅜" cast-in galvanized steel rod extending 4" beyond each end. Nominal weight, 50 lbs.; current rating, 2 amp-yrs.

CONDENSER ANODES

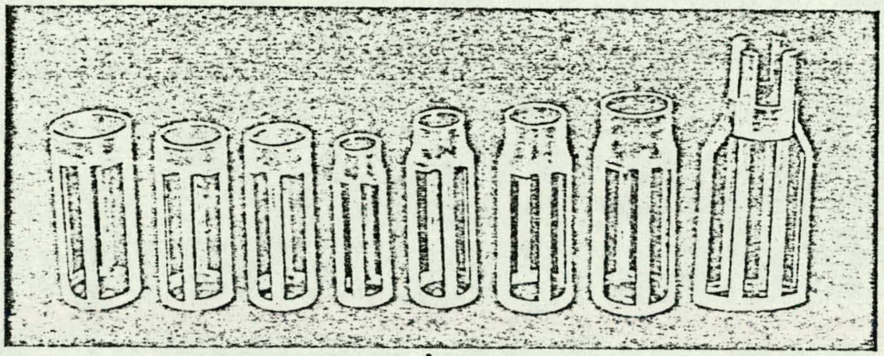




CAT. NO.	DIA.	NOM. WT.
CZ-5	5"	5 lbs.
CZ-6	6"	7 lbs.
CZ-9	9"	16 lbs.—two IPS cores on 3½" centers (not shown)
CZ-11	11"	24 lbs.—two IPS cores on 3½" centers (not shown)

CZ-5: Current rating ¼ amp-yr.; other sizes available on request.

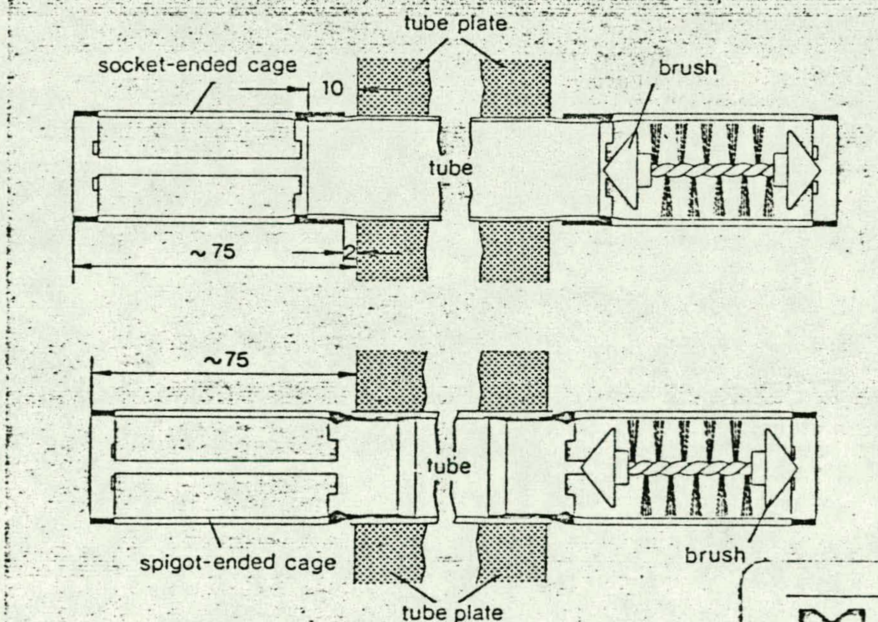
of tubes with bores of 13 mm and over. Cages and brushes are available for all common tube sizes, both metric and inch systems. The brush wire is titanium – to withstand seawater attack – and ensures a brush life of over 10,000 cycles. The plastic materials are heat resistant at temperatures over 100 °C and approved up to 70 °C.



7

Fig. 8: The cages are available for alternative methods of fastening:

1. with socket ends for fitting on a plain tube projection of about 10 mm length.
2. with spigot ends to fit a flush tube end which has been re-rolled by a special tube expander to give a calibrated bore and a bead about 10 mm from the end to register with a groove in the spigot.
3. Where tubes cannot be re-rolled the cages are permanently bonded in the tube end by a two-component cement.



8

The M.A.N. on-load cleaning system for shell-and-tube condensers and tubular heat exchangers has established a splendid record of dependability in numerous installations in Germany and abroad.

A million tubes are already being cleaned by this method.

A million tubes have been uprated for maximum heat transfer.

A million tubes have been given a new lease of life.

For further details please write or telephone to

M·A·N

AMERICAN M.A.N. CORPORATION

1114 Avenue of the Americas, New York, N.Y. 10036 (212) 221-3340

WEST COAST OFFICE

50 California Street, San Francisco, CA 94111 (415) 391-2935

October 25, 1978

Dr. W. H. Avery
Director of Ocean Energy Program
Applied Physics Laboratory
Johns Hopkins University
Johns Hopkins Road
Laurel, Maryland 20810

Dear Bill,

Attached are photomicrographs of cross sections from 0.032-inch Alclad 3004 sheet which has been exposed for 18 years to seawater, as a sheathing for cypress wood piling near Daytona Beach, Florida.

As the photographs show, the Alclad 3004 sheet has shown good resistance to corrosion during the 18-year exposure period. The deepest attack observed is only 3 mils (0.003 inches). The thin 7072 cladding alloy layer is still intact over 50 percent of the surface exposed to seawater.

The metallographic cross sections were taken from two six-inch squares of the Alclad 3004 sheathing. One square had been cut from the sheathing at a tide range level, i.e. immersed at high tide, exposed at low tide. The other square was removed from an area of continuous, total immersion. The squares were chemically cleaned to remove marine fouling, etc. There was no distinct pitting, only slight etching, when examined visually.

This performance seems typical for Alclad 3004 in surface seawater exposures. The marine fouling is not harmful. As far as the possibility that periodic cleaning might be a more severe condition, experience with aluminum cooking utensils showed this not to be the case. Utensils cleaned periodically with stainless steel wool or plastic brillo pads, had shallower pits than utensils which were not cleaned periodically.

In summary, it is our opinion that Alclad aluminum alloys such as Alclad 3003 and Alclad 3004 would provide

CENTER FOR TECHNOLOGY
POST OFFICE BOX 877 · PLEASANTON, CALIFORNIA 94566 · PHONE (415) 462-1122

satisfactory corrosion resistance in the heat exchanger systems for OTEC. We hope that they will be used in the construction of at least one of the prototype units so that directly applicable data can be collected.

Please write or call me if you have questions. At your request, I am sending a copy of this letter and my earlier letter of September 29th to Dr. Frank LaQue for his review and files.

Very truly yours,

Tom

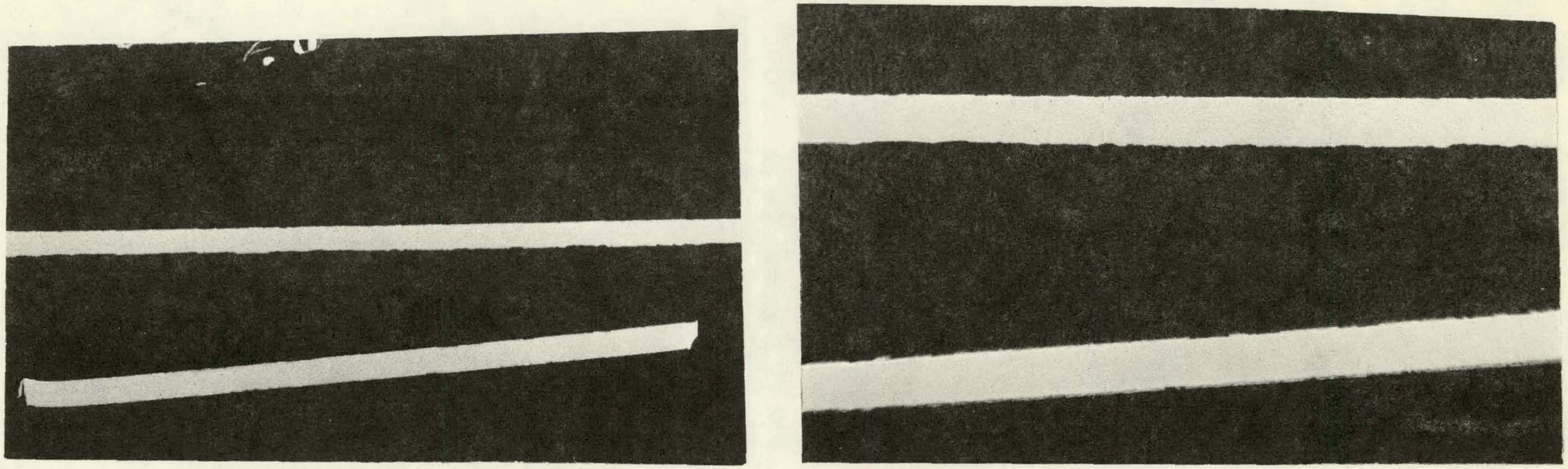
T. J. Summerson
Head, Corrosion Section

TJS/ahb
Attachments

cc: H. B. Lockwood, KACC

F. L. LaQue

bcc: J. Rynewicz, Lockheed Space & Missile
T. R. Pritchett, CFT 10



5X 10X

Figure 1

D6-3

0.032-Inch ALCLAD 3004 SHEATING FOR CYPRESS PILINGS AFTER 18 YEARS SEAWATER IMMERSION -- DAYTONA BEACH, FLORIDA

Total Immersion - Top Section: This sample comes from portion of sheathing that was continuously immersed in seawater. The upper surface was in contact with the wood piling. Corrosion is limited to the thin 7072 alloy cladding layer.

The lower surface was exposed to seawater and was covered with marine fouling. Deepest attack is approximately two times the cladding thickness, i.e. 3 mils. About 50% of the surface of the cross section exposed to seawater still has some cladding left.

Tide Range Immersion - Lower Section: Corrosion on both seawater side (lower surface) and wood piling side (upper surface) is confined to the cladding layer.

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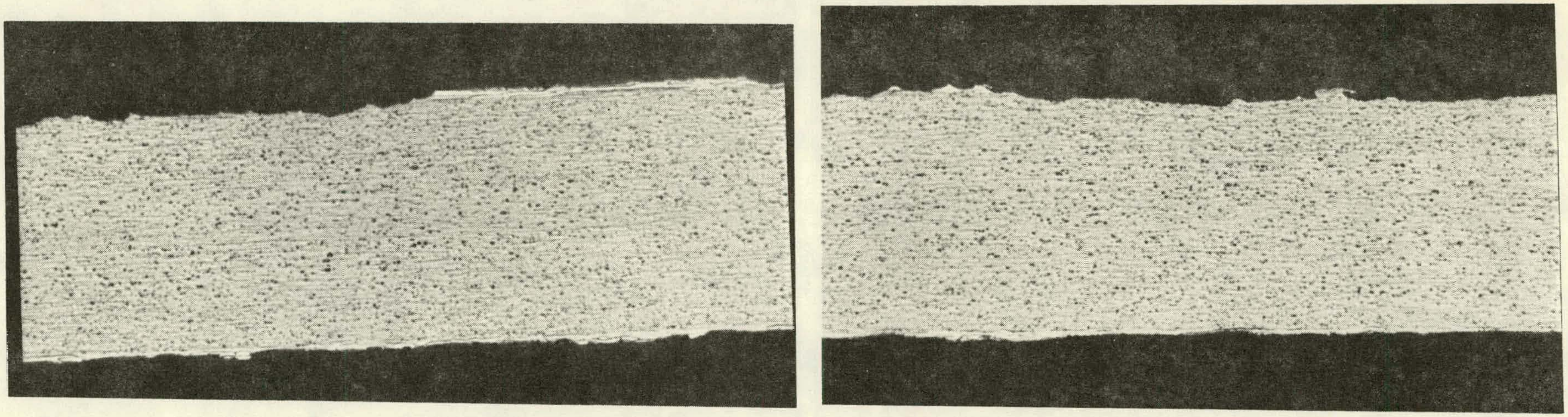
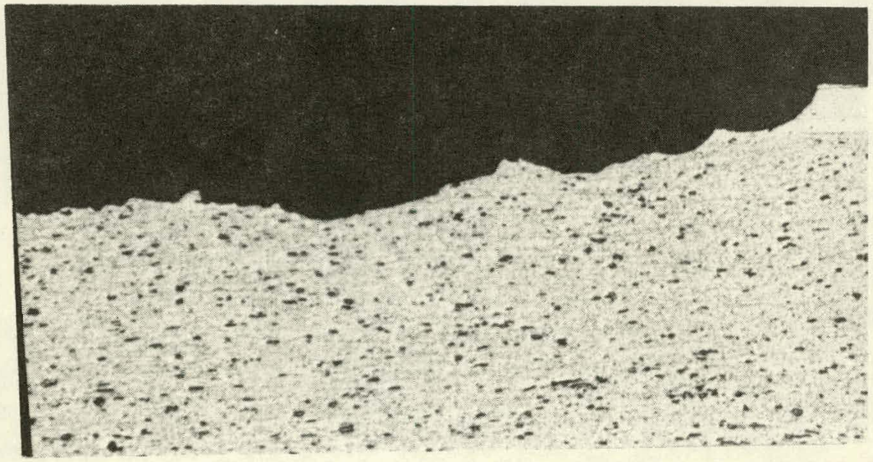


50X with HF - H_2SO_4 etch



7072 cladding layer, 1.2 mils

200X, etched

Figure 2

0.032 Inch ALCLAD 3004 SHEATHING FOR CYPRESS PILING AFTER 18 YEARS OF SEAWATER IMMERSION -- DAYTONA BEACH, FLORIDA

On the seawater surface (upper surface in photos), the deepest corrosion is only about 0.003 inches (3 mils). Much of the thin 7072 alloy cladding layer is still intact.

10/25/78
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SAN/1568-78/4
DISTRIBUTION CATEGORY UC-64

LMSC-D630737
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ATTACHMENT 1
TO
OCEAN THERMAL ENERGY CONVERSION POWER SYSTEM DEVELOPMENT-I PHASE I – FINAL REPORT

CONCEPTUAL DESIGN REPORT

LMSC-D566744 • 18 DECEMBER 1978
CONTRACT EG-77-C-03-1568

Prepared for

UNITED STATES DEPARTMENT OF ENERGY
DIVISION OF SOLAR ENERGY

by

OCEAN SYSTEMS
LOCKHEED MISSILES & SPACE COMPANY, INC.
SUNNYVALE, CALIFORNIA 94086

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OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM DEVELOPMENT

CONCEPTUAL DESIGN

Contract No.
EG-77-C-03-1568

30 JANUARY 1978 LMSC-D566744

Prepared for
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FOREWORD

The Lockheed Missiles & Space Company, Inc. (LMSC), together with corporate team members and subcontractors, has completed a Conceptual Design of a Power System for application to the OTEC 100-MW(e) Demonstration Plant. This report presents the results of the 5-month study performed for the United States Department of Energy under Contract EG77-C-03-1568.

This report presents a technical description of the configuration used for the reference surface platform/ship as well as a recommended power system incorporated into a submerged detachable module. The Appendixes provide additional details and related trade studies supporting these configurations.

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ILLUSTRATIONS

TABLES

SECTION 1

Section 1
INTRODUCTION

This report presents the results of a concept design study of an Ocean Thermal Energy Conversion (OTEC) power system module performed by Lockheed Missiles & Space Company, Inc. (LMSC) for the Department of Energy (DOE). The power module includes only those basic elements required for a power generating system; i.e., the heat exchangers, turbine generator, seawater pumps, ammonia plumbing and pumps, and the control system. Platform-related components are not included.

The major guidelines under which the study was conducted are:

1. The heat exchangers are horizontal shell and tube with heat transfer enhancement at LMSC's discretion.
2. Design temperature difference (ΔT) between warm and cold water is 40, $+4$ -11°F.
3. The power module size is to be a nominal 25-MW(e) net, but the objective is to determine the size of the power plant and its components which will minimize the cost of delivered electricity.
4. The power system is modularized such that a complete plant could grow to 100-MW(e) net by the addition of more modules.
5. Thirty year design life. (This is interpreted to mean periodic replacement of components if economically feasible.)
6. The warm seawater subsystem pressure drop is 10 ft of water; cold seawater subsystem pressure drop is 15 ft. These values may be varied if shown to be technically and economically feasible.
7. The working fluid is ammonia.
8. Biofouling heat transfer resistance is not to exceed 0.0005 $\frac{\text{hr-ft}^2\text{-}^{\circ}\text{F}}{\text{Btu}}$

9. Plant operability is 90 percent at a 60-percent confidence level.
10. Configurations to be evaluated include modules internal to the platform and detachable modules external to the platform.

In addition to the 25-MW(e) (nominal) power module, LMSC is to develop concepts for a 1-MW(e) (nominal) net scaled evaporator and condenser plus a 5-MW(e) (nominal) net proof-of-concept plant. The word nominal is used to indicate that LMSC is to determine the specific size of each of these two test articles.

Experimental data on heat transfer resistance due to biofouling, anticipated for use in component selection and sizing, were not generated in time to support this project. LMSC therefore used a conservative resistance coefficient of 0.0003 in the study. While the DOE experiments may show some variation from the performance projected herein, the experimental results are expected to have little impact on the selected concept or the cost estimates.

In the conduct of this study, LMSC has attempted to perform cost trades on the more costly components of the power module. Performance and cost data were acquired from component suppliers and specialists in the specific functional items that are parts of the construction, e.g., piping costs for the ammonia system were developed by Bechtel National, Incorporated (formerly Bechtel Corporation Research and Engineering). Interpolations and extrapolations on cost data were made by LMSC when the data were not included in supplier data.

Subcontractors and their area of study for this study are:

- Foster Wheeler Energy Corporation – heat exchanger design and construction
- Bechtel National Incorporated – ammonia cycle and auxiliaries
- General Electric Company – axial flow turbines
- Rotoflow Corporation – radial inflow turbines
- Lockheed Electronics Company – controls/data acquisition systems
- Lockheed Shipbuilding and Construction Company – module assembly

The trade studies conducted during the course of this project were initiated with the "best" data available. Therefore, various hardware or performance definitions were used as they evolved. The reader will observe minor inconsistencies in the configuration(s) of components, power modules, and performance characteristics. These inconsistencies, however, have negligible effect on the conclusions as presented herein.

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SECTION 2

Section 2
SUMMARY

This section presents the salient findings of the LMSC study. In capsule form, they are:

1. Heat exchangers were found to decrease in cost per unit of power with increasing size. For the 50-MW(e) size, heat exchanger unit kW costs were approximately 10 percent less than for the 25-MW(e) size.
2. Aluminum exchangers were found to be less expensive on a life-cycle basis than titanium or stainless steel.
3. Ammonia-side enhancement is slightly cost effective; seawater-side enhancement appears not to be cost effective.
4. Controllable-flow pumps are cost effective in sites where there is a significant ΔT variation. Multiple pumps reduce the length of the seawater path and improve reliability.
5. Radial in-flow turbines are lower in cost and provide a higher efficiency than the axial flow turbines.
6. Fully integrated power modules externally adaptable to spar-type platforms are least expensive, lend themselves most readily to scheduled maintenance, and require least capital for construction facilities.

The remainder of this section presents a summary of the trade studies conducted, a configuration description of a 25-MW(e) system, and finally a cost summary of the 25-MW(e) power module. The latter is presented rather than the 50-MW(e) to allow direct comparisons with concurrent studies being conducted by others.

2.1 TRADE STUDIES

Trade studies were conducted on heat exchangers, seawater pumps, turbine generators, power module construction, and ammonia plumbing as a function of heat exchanger size to determine minimum cost per kilowatt by capacity for the power module. The ΔT used in the basic trade studies was 36°F. Subsequently, the ΔT was increased to 40°F and the size maintained at 25-MW(e) to make the results comparable to other studies being conducted concurrently.

2.1.1 Heat Exchanger

The heat exchanger trade studies included:

- Size and number of heat exchangers required to provide the 25-MW(e) (nominal) capacity, but also included a set of 50-MW(e) heat exchangers
- Tube diameter from 1 to 3 in.
- Materials of construction, including aluminum, titanium, and stainless steel
- Depth of immersion from 1 atmosphere to 275 ft below sea level
- Cylindrical versus spherical shells

The results of these trade studies for a nominal 25-MW(e) set of spherical exchangers with plain tubes are summarized in Fig. 2-1. Major heat exchanger cost elements are labor and materials; other cost elements such as tooling and facilities are not included.

Using 5-MW(e) size exchangers to make up a 25-MW(e) module increases the cost per kW by approximately 40 percent over the single heat exchanger. Increasing the heat exchanger size to 50-MW(e) decreases the cost by about 10 percent.

At the 36°F ΔT, ammonia-side enhancement decreases the cost by 3 percent. Seawater-side enhancement alone was found to increase cost. Increasing the ΔT to 40°F results in a cost decrease of 14 percent.

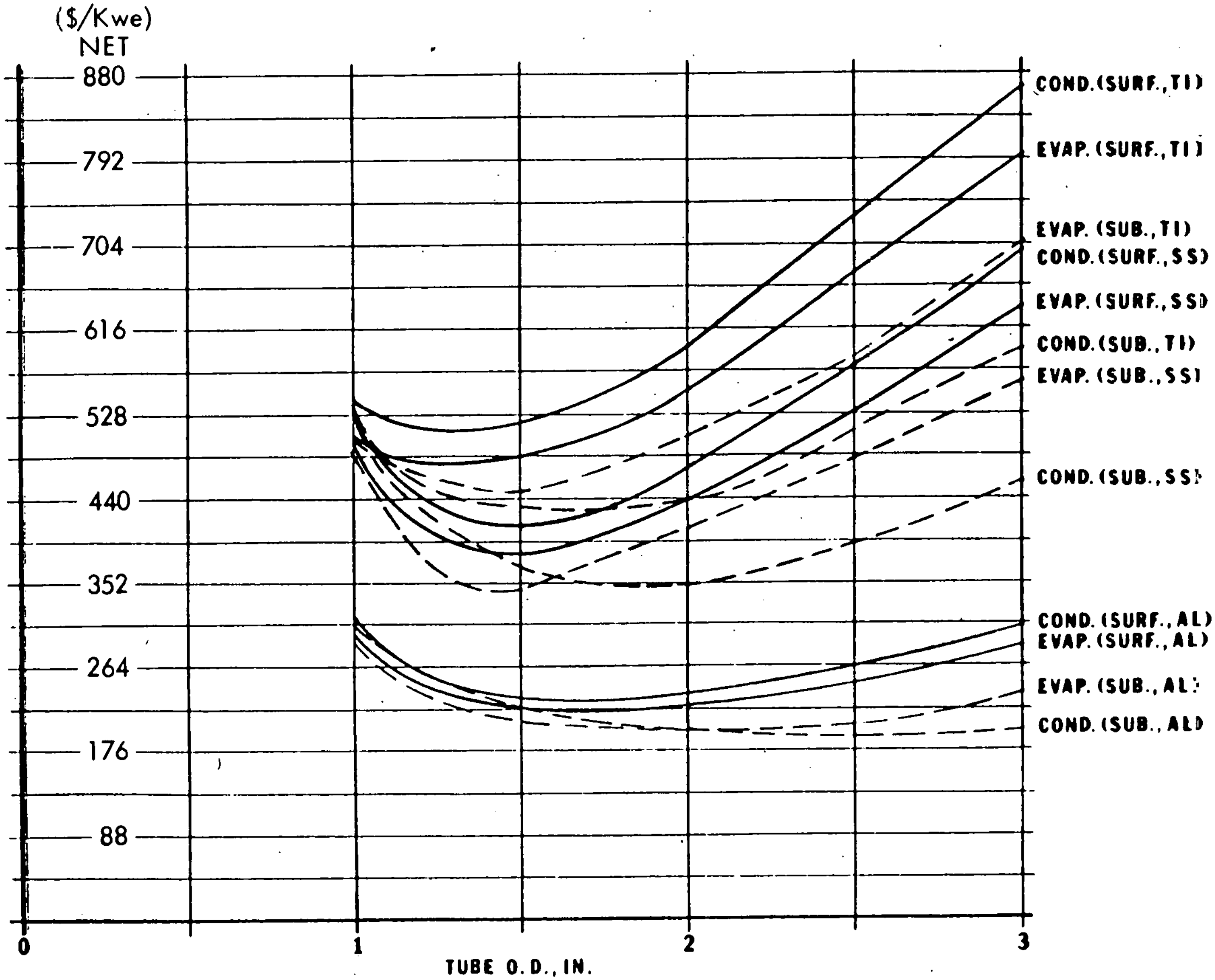

Fig. 2-1 25-MW(e) Evaporator/Condenser – Al/SS/Ti, Spherical

2.1.2 Seawater Pumps

Data for the seawater pumps are related to the module seawater flow requirements, system total pressure losses, and number of pumps installed. The performance data with pump failure in either or both systems results in module power degradations of 4 and 9 percent, respectively, with four pumps in each subsystem. With three pumps installed, these performance losses are increased to 9 and 18 percent, respectively, as shown in Fig. 2-2.

Information received from KSB, Hitachi, Mitsubishi, Allis-Chalmers, Worthington, and Pleuger indicate that there is little economy of scale in going from a pump sized for the four-pump installation to one sized for a three-pump installation. The physical installation is larger for the three-pump configuration, primarily in a 15-percent greater overall length. In addition, a longer transition section is required to provide uniform flow at the front face of the heat exchangers.

Pump performance requirements have been determined for three configurations — a baseline reference with the design requirement of 10 ft of head loss in the warm water subsystem and 15 ft in the cold water subsystem, an optimized surface module, and an optimized submerged module. The warm and cold seawater flow, head, and power requirements are presented in Table 2-1.

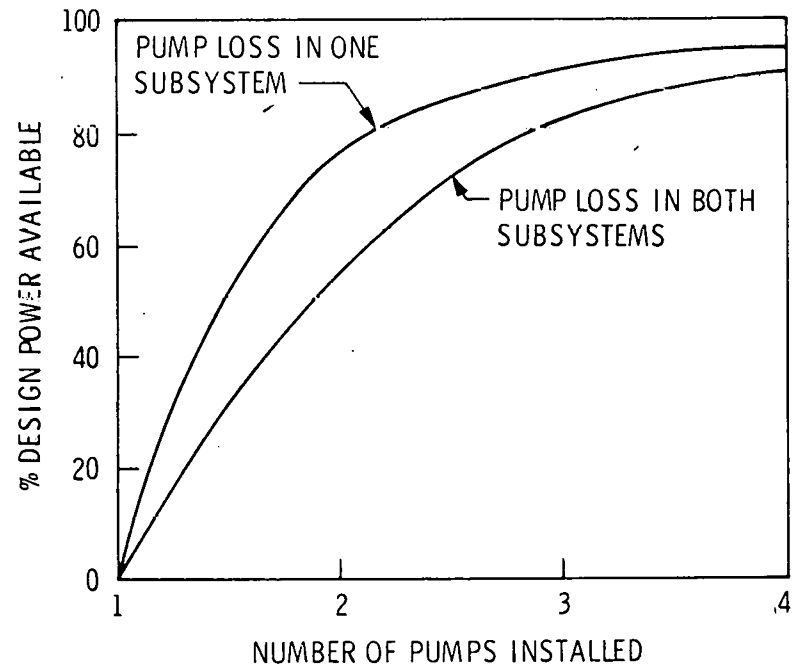

Fig. 2-2 Operational Capability With Seawater Pump Loss – Constant Flow Pumps

Table 2-1

SEAWATER PUMP FLOW RATES AND HEAD REQUIREMENTS

Item	Configuration		
	Baseline Reference	Optimized	
		Surface	Submerged
Warm Water Pumps			
Flow (cfs)	3,590	3,450	3,600
Head (ft)	10	8	7.6
Power Required (MW)	4.01	3.10	3.04
Cold Water Pumps			
Flow (cfs)	3,640	3,525	3,710
Head (ft)	15	9.5	8
Power Required (MW)	6.13	3.75	3.31

2.1.3 Turbine/Generator

Data for the turbines are related to the physical characteristics, thermodynamic performance, and cost. The physical dimensions of the axial-flow turbine installation are approximately 25-percent greater than the dual radial flow turbine installation. The weight of the radial turbine is approximately 75 percent that of the axial turbine in steel. An alternative aluminum assembly reduces this to approximately 30 percent. The heaviest component to be handled during maintenance is the 90,000-lb upper half of the turbine casing for the axial machine and the 32,000-lb generator rotor for the radial turbine installation.

The performance characteristics are considerably different in that the design efficiency of the radial turbine is 87 percent while that of the axial turbine is 77 percent. The effect of this difference on heat exchanger size and ammonia flow requirement is shown in table 2-2. While this comparison was done for an unenhanced tube configuration, the relative effect should be similar for the baseline configuration. The cost effect has not been evaluated in detail but is estimated to be 18 percent.

As part of this task, cost data were developed for a family of turbine sizes from 5-MW(e) net to 25-MW(e) net. The results, shown in Fig. 2-3, indicate that there is little economy of scale for the radial turbine beyond 25-MW(e) net. There is some economy of scale for the axial turbine, but the costs are more than double the costs for the radial turbine.

Table 2-2

COMPARATIVE TURBINE PERFORMANCE EFFECTS

	Radial	Axial	Percent Difference
Design Efficiency	0.87	0.77	13
Tube No. Required			
– Evaporator	78,014	81,915	5
– Condenser	77,924	81,820	5
Tube Lengths (ft)			
– Evaporator	42.5	50.1	18
– Condenser	45.7	54.2	18
Ammonia Flow (lb/sec)	1,960	2,250	15
Overall Length (ft)	22 (2 units)	27	23
Overall Height (ft)	11.25	14	24
Overall Width (ft)	11	14	27
Weight (lb)	150,000 (2)	200,000	33

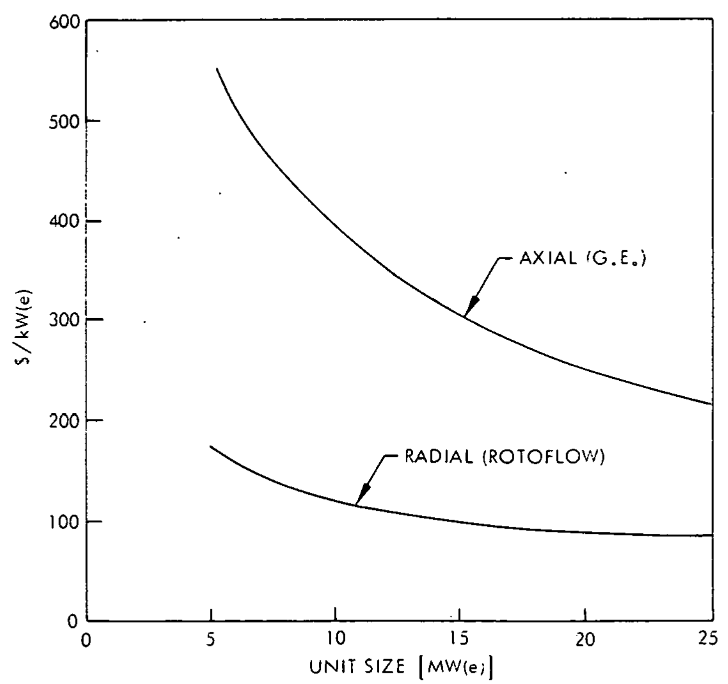

Fig. 2-3 Turbine-Generator Unit Cost vs. Size

2.1.4 Power Module Construction

This trade study evaluated various methods that could be applied in producing the completed end article. Considering the logistics problems, it was determined that the fabrication, assembly, installation, and final outfitting are best suited to the capabilities of a shipyard.

Configurations evaluated were

- Baseline Surface Platform (monolithic and multiple heat exchangers and turbines)
- Supertanker (twenty 5-MW(e), eight 12.5 MW(e), and four 25-MW(e) heat exchangers)
- Spar Power Module (ten 5-MW(e) dual modules, four 12.5-MW(e) dual modules, and four 25-MW(e) modules)

Installations costs, shown in Table 2-3, indicate that a 25-MW(e) power module for the submerged configuration is the least costly. As would be expected from the large number of components to be handled, the twenty 5-MW(e) installation is the most costly, and there is little difference between the supertanker and the baseline surface platform costs.

An estimated schedule for the supertanker and surface platform installation indicated that it would require approximately 3 years and 9 months to complete the installation and conduct predelivery tests. The schedule for the 25-MW(e) modules for a 100-MW(e) submerged platform is shown in Fig. 2-4. As indicated, the first module would be completed in 18 months, with subsequent modules requiring 6 months each for a total span of 3 years.

Table 2-3

100-MW(e) SYSTEM INSTALLATION COSTS

	Power Modules			Supertankers			Surface Platforms	
MW(e) System Size	5	12.5	25	5	12.5	25	Monolithic	Multiple
Condensers	270	200	160	250	220	170	190	230
Evaporators	270	200	160	250	220	170	190	230
Generators	100	80	60	150	120	80	100	120
Turbines	80	60	40	110	90	60	80	100
Seawater Pumps	600	400	250	470	390	320	340	320
Condensate Pumps, Distribution Pumps, Piping and Valves	3,450	2,530	2,000	6,650	5,700	4,650	5,300	6,400
Foundations (Install)	4,350	3,000	2,500	8,350	7,200	5,850	7,000	8,450
Foundations (Make)	6,100	4,200	3,500	11,950	10,250	8,480	9,800	11,800
TOTAL	15,220	10,670	8,670	28,180	24,190	19,780	23,000	27,650

NOTE: Costs in Thousands of Dollars

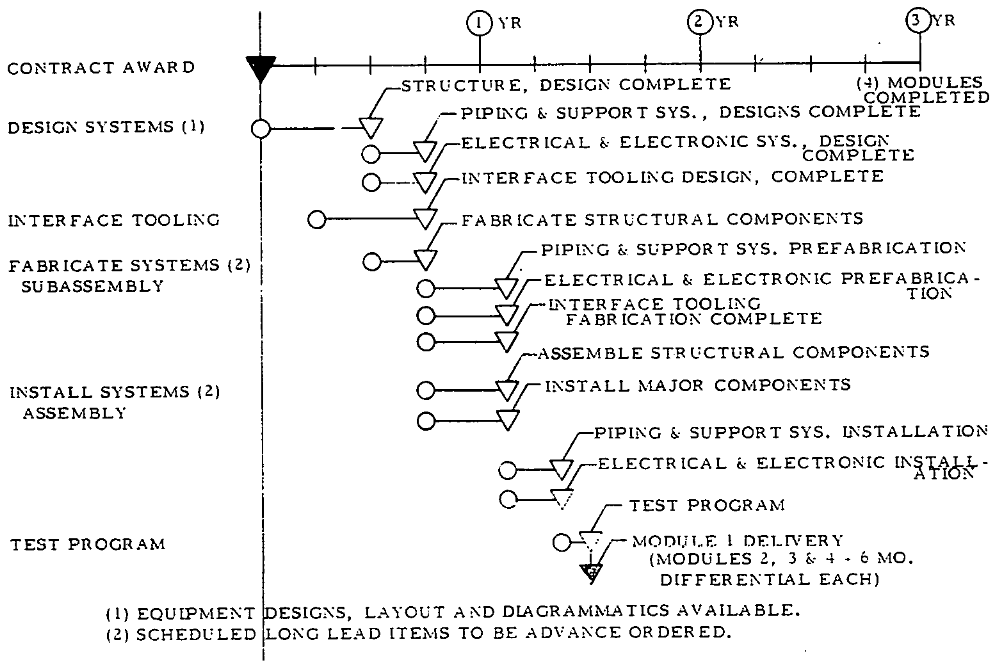

Fig. 2-4 Schedule for 25-MW(e) Modules for Spar Platform

2.1.5 Ammonia Plumbing

Ammonia vapor and condensate plumbing cost trade studies were conducted to evaluate the effect of component modularity on ammonia system costs. Three surface configurations (four 25-MW(e), eight 12.5-MW(e), and twenty 5-MW(e) heat exchangers) and two submerged configurations (four 25-MW(e) and twenty 5-MW(e) heat exchangers) were evaluated. Costs were obtained for plumbing required to maintain 60 fps in the vapor lines and 30 fps in the condensate lines at ammonia flow rates associated with design ΔT's of 32°, 36°, and 40°F.

The results shown in Fig. 2-5 indicate that the least costly arrangement is for the 25-MW(e) submerged configuration having four 25-MW(e) heat exchangers. For the ship arrangement, the four heat exchanger configuration is more costly than the eight heat exchanger arrangement because of the required pipe diameter and wall thickness. Special handling equipment is required for lines in excess of 90 in. in diameter. If higher flow velocities or dual flow piping were used, the four heat exchanger configuration would be less expensive.

The effects of vapor and condensate velocity changes were evaluated by determining the effect of ammonia system pressure loss on heat exchanger size to maintain 25 MW(e). The results, shown in Figs. 2-6 and 2-7, indicate that minimum heat exchanger and ammonia plumbing costs occur at a vapor velocity of 160 fps and a condensate velocity of approximately 25 fps.

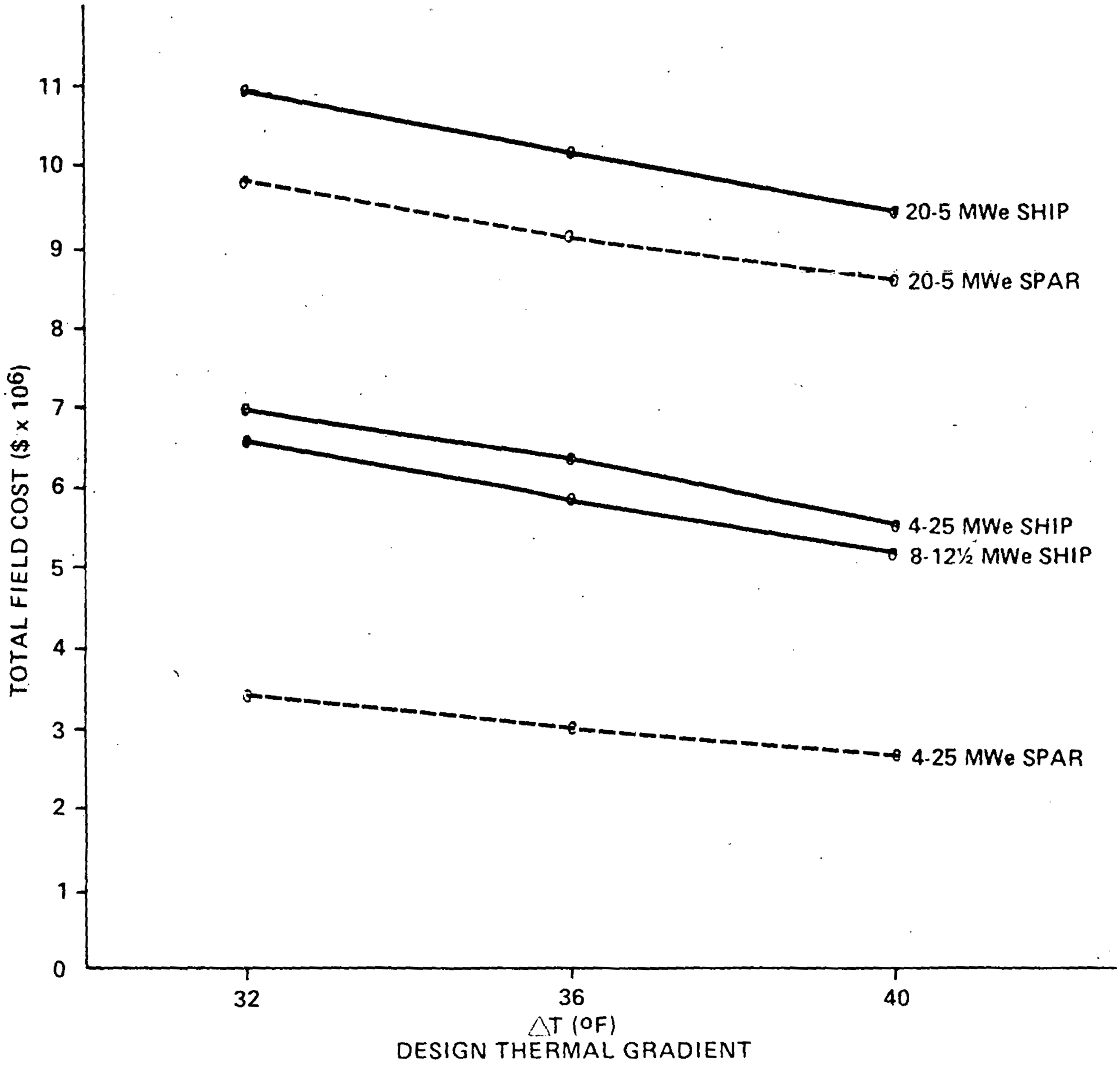

Fig. 2-5 Ammonia System Costs (Piping, Valves, and Pumps)

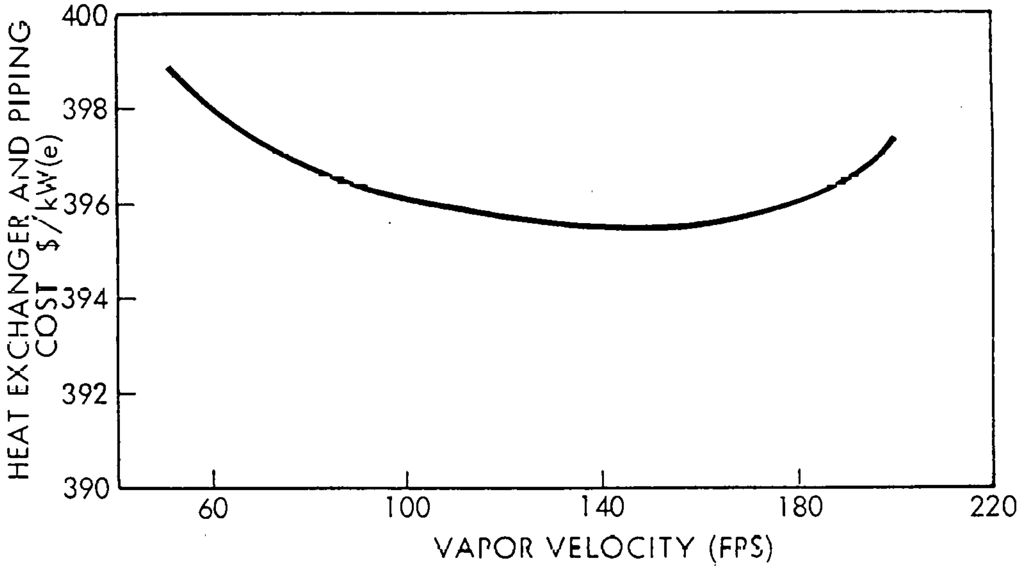

Fig. 2-6 Effoot of Vapor Volooity, Submorgod Configuration, Condensate Velocity = 30 fps

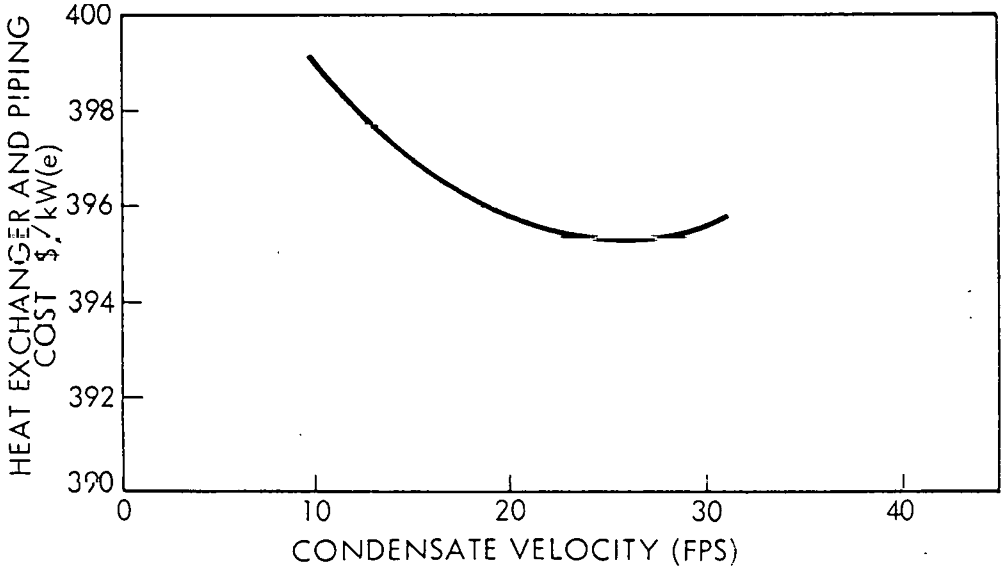

Fig. 2-7 Effect of Condensate Velocity, Submerged Configuration, Vapor Velocity = 160 fps

2.1.6 Power System Module Performance Size

There are indications of reducing the cost of energy as the size of the power module is increased beyond 25 MW(e). Our interpretation of the scope of work did not include investigations in all componentry areas beyond 25 MW(e) or a module beyond 25 MW(e). The two most significant areas of cost reduction are also the two highest cost areas in the power module. These are the heat exchangers and the seawater system. For the recommended power system module, the heat exchangers and seawater system account for 76 percent of the prototype power module costs and 67 percent of the first production unit.

It is therefore recommended that the first 3 to 4 weeks of Task 2 (the first part of the period called Requirements and Guidelines) be used to refine the power system module performance size in order to define the most cost-effective power module size. This approach would require close coordination with the OTEC Plant Configuration and Integration Study Contractors to ensure use of realistic criteria in the power module size evaluation.

2.2 SYSTEM CHARACTERISTICS

The following seven tables present the system physical and performance characteristics as requested by DOE. Each table summarizes three configurations: First, the power module for the "reference hull is a surface ship/platform" with a 10-ft head loss in the warm water circuit of the power module and a 15-ft head loss in the cold water circuit, as specified. Second, a power module for the surface ship/platform with the warm and cold water head losses optimized on the LMSC computer program. Third, a power module for a submerged, external power module adaptable to a spar type platform. In the former two cases the evaporator centerline is at 45 ft and the condenser centerline at 55 ft as shown in Fig. 2-8. In the latter case, they are at 150 ft and 275 ft, respectively (Fig. 2-9).

In Table A, net power is at the power module bus bar and does not include platform related losses for power conditioning and transmission.

Tables B through G are self-explanatory.

Table A

MODULE POWER

Item	Reference Baseline	Optimized	
		Surface	Submerged
Turbine Power	37.14	33.07	33.23
Generator Power [MW(e)]	36.59	32.58	32.73
Warm Water Pumping Power [MW(e)]	4.01	3.10	3.04
Cold Water Pumping Power [MW(e)]	6.13	3.75	3.31
Ammonia Feed Pumping Power [MW(e)]	0.81	0.61	1.27
Ammonia Distribution Pumping Power [MW(e)]	0.20	0.18	0.17
Controls, Hotel, Internal Power Loss, Etc. [MW(e)]	0.4	0.4	0.4
Chlorination Power [MW(e)] (a)	0.04 to 0.32	0.04 to 0.32	0.04 to 0.32
Mechanical Cleaning Power (b)			
Net Power Out [MW(e)]	25.00 to 24.72	24.50 to 24.22	24.50 to 24.22
Turbine Gross/Net Power	1.49 to 1.50	1.35 to 1.36	1.36 to 1.37

(a) 0.2 mg/l to 1.6 mg/l dosage

(b) System sized with added heat exchanger pressure loss. Cleaning system operated by flow reversal for 15 minutes once a day.

Table B

ROTATING MACHINERY (PER BASELINE MODULE OF 25-MW(e) (NET)

Item	Number	Type	RPM	Horsepower	Efficiency	Comments
Turbine and Diffuser	2	Radial Inflow	3,600	–	0.86	18.57 MW each
Generator	1	AC	3,600	–	0.985	
Warm Water Pump	4	Axial	100 ± 10%	–	0.86	
Warm Water Pump Motor	4	Submerged	870 ± 10%	1,500	0.90	
Cold Water Pump	4	Axial	100 ± 10%	–	0.86	
Cold Water Pump Motor	4	Submerged	870 ± 10%	2,250	0.90	
Ammonia Feed Pump	3	Centrifugal	–	–	0.90	
Ammonia Feed Pump Motor	3	–	1,200	600	0.90	
Ammonia Distribution Pump	4	Centrifugal	–	–	0.90	
Ammonia Distribution Pump Motor	4	–	1,800	75	0.90	

Table C

FLOW RATES AND CONDITIONS AT DESIGN POINT OPERATION

Item	Reference Baseline	Optimized	
		Surface	Submerged
TURBINES			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
INLET			
Pressure (psia)	136.9	136.9	137.5
Temperature (° F)	73.49	73.49	73.74
Velocity (fps)	150	150	150
Quality (%)	99	99	99
OUTLET			
Pressure (psia)	95.94	96	95.93
Temperature (° F)	53.84	53.87	53.83
Velocity (fps)	75	75	75
Quality (%)	96.5	96.6	96.5
PUMPS			
AMMONIA FEED			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
Pressure Rise (psia)	52.1	52.1	110
AMMONIA DISTRIBUTION			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
Pressure Rise (psia)	15	15	15
WARM SEAWATER			
Flow Rate (gpm) each	402,600	386,700	403,800
Pressure Rise (ft)	10	8	7.6
COLD SEAWATER			
Flow Rate (gpm) each	408,500	395,500	416,200
Pressure Rise (ft)	15	9.5	8

Table D

HEAT EXCHANGER DESIGN DATA – HORIZONTAL SHELL AND TUBE (EVAPORATOR)

Item	Reference Baseline	Optimized	
		Surface	Submerged
Number per Module	1	1	1
Thermal Rating (MBtu/hr)	3,970	3,540	3,508
Water Flow Rate (gpm)	1,610,350	1,546,900	1,615,300
Ammonia Flow Rate (lb/hr)	11,396,700	10,164,750	10,074,700
Design Pressure (psid)[(a)]	250/-	250/-	127/54
Operating Pressure (psia)	137	137	137
Overall Length (ft)	76	68	66
Water Enhancement Factor	1.0	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0	3.0
Total Number of Tubes	52,260	50,315	50,250
Tube OD (in.)	1.5	1.5	1.5
Tube Wall (in.)[(b)]	0.1025/0.090	0.1025/0.090	0.088/0.075
Tube Pitch (in.)	1.875	1.875	1.875
Tube Layout	Equilateral Triangle		
Tubesheet Diameter (ft)	43.5	43	43
Effective Tube Length (ft)	44	39.5	38
Actual Tube Length (ft)	46	41.5	40
Tubesheet Thickness (in.)	2.5	2.5	1.5
Second Tubesheet Thickness (in.) (if applicable)[(c)]	2.5	2.5	1.5
Number of Tube Supports	6	5	5
Tube Support Thickness (in.)	1	1	1
Tube Support Spacing (ft)	6.3	6.6	6.33
Shell OD (ft)	63.5	60	59
Shell Wall Thickness (in.)	3.2	3.2	2.0
Void Fraction	0.63	0.65	0.65
Tube Material	AL 5052	AL 5052	AL 5052
Tubesheet Material	AL 5083	AL 5083	AL 5083
Shell Material	AL 5083	AL 5083	AL 5083
Baffle Material	AL 5083	AL 5083	AL 5083
Cladding Material	–	–	–
Coating Materials[(d)]	TBD	TBD	TBD
Demister Size (ft × ft × ft)	26' d × 9" thick	25' d × 9" thick	25' d × 9" thick
Demister Design Pressure (psia)	(e)	(e)	(e)
Design Inlet Quality (%)	95	95	95
Design Outlet Quality (%)	99	99	99

(a) Burst/Collapse
(b) Mean/Minimum
(c) Annular Type
(d) Not defined at this time
(e) Mist Extractor inside evaporator shell

Table E

HEAT EXCHANGER DESIGN DATA – HORIZONTAL SHELL AND TUBE (CONDENSER)

Item	Reference Baseline	Optimized	
		Surface	Submerged
Number per Module	1	1	1
Thermal Rating (MBtu/hr)	3,845	3,430	3,400
Water Flow Rate (gpm)	1,633,900	1,582,200	1,664,600
Ammonia Flow Rate (lb/hr)	7,597,800	6,776,500	6,716,450
Design Pressure (psid)[a]	250/-	250/-	96/132
Operating Pressure (psia)	96	96	96
Overall Length (ft)	76	68	66
Water Enhancement Factor	1.0	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0	3.0
Total Number of Tubes	58,700	56,400	56,000
Tube OD (in.)	1.5	1.5	1.5
Tube Wall (in.)[b]	0.1025/0.090	0.1025/0.090	0.088/0.075
Tube Pitch (in.)	1.875	1.875	1.875
Tube Layout	Equilateral Triangle		
Tubesheet Diameter (ft)	43.5	43	43
Effective Tube Length (ft)	44	39.5	38
Actual Tube Length (ft)	46	41.5	40
Tubesheet Thickness (in.)	2.5	2.5	1.5
Second Tubesheet Thickness (in.) (if applicable)[c]	2.5	2.5	1.5
Number of Tube Supports	6	5	5
Tube Support Thickness (in.)	1	1	1
Tube Support Spacing (ft)	6.3	6.6	6.33
Shell OD (ft)	63.5	60	59
Shell Wall Thickness (in.)	3.2	3.2	2.0
Void Fraction	0.59	0.61	0.61
Tube Material	AL 5052	AL 5052	AL 5052
Tubesheet Material	AL 5083	AL 5083	AL 5083
Shell Material	AL 5083	AL 5083	AL 5083
Baffle Material	AL 5083	AL 5083	AL 5083
Cladding Material	–	–	–
Coating Materials[d]	TBD	TBD	TBD

(a) Burst/Collapse
(b) Mean/Minimum
(c) Annular Type
(d) Not defined at this time

Table F

HEAT EXCHANGER THERMAL DATA – EVAPORATOR

Item	Reference Baseline	Optimized	
		Surface	Submerged
Water Flow Rate (lb/hr)	824×10^6	792×10^6	827×10^6
Water Flow Per Tube (lb/sec)	4.38	4.37	4.57
Water Velocity Inside Tube (ft/sec)	7.54	7.54	7.53
Reynolds Number Inside Tube (-)	72496	72424	73970
Prandtl Number Inside Tube (-)	6.0	6.0	6.0
Inlet Water Temperature (°F)	83.88	83.88	83.88
Outlet Water Temperature (°F)	78.71	79.08	79.32
Average Water Temperature (°F)	81.29	81.48	81.55
LMTD (°F)	7.19	7.42	7.46
Average Tube Wall Surface Temperature (°F)	74.3	74.6	74.8
Average Shell Wall Surface Temperature (°F)	74.5	74.7	75
HW (Including Enhancement) (Btu/hr ft^2 °F)	1388	1388	1382
H Fouling (Btu/hr ft^2 °F)	3333	3333	3333
H Wall (Btu/hr ft^2 °F)	9709	9709	11235
H Ammonia (Btu/hr ft^2 °F)	3475	3475	3400
Uo (Btu/hr ft^2 °F)[(a)]	709	709	711
Ammonia Inlet Temperature (°F)	53.53	53.55	53.41
Ammonia Outlet Temperature (°F)	73.49	73.74	73.91
Vapor Quality (Outlet, %)	99	99	99
Maximum Vapor Velocity (Outlet, ft/sec)	7.0	7.1	7.3
Total Pressure Drop of Water (psid)	3.92	3.56	3.35
Total Pressure Drop of Ammonia (psid)[(b)]	0.30	0.30	0.32
Distribution Tubes/Heat Transfer Tubes	0.01	0.01	0.01
Maximum Feed Rate (lb/ft sec)	0.20	0.20	0.20
Minimum Feed Rate (lb/ft sec)	0.10	0.10	0.10
Design Feed Rate (lb/ft sec)	0.15	0.15	0.15
Recirculation Ratio (-)	0.5	0.5	0.5
Maximum Distribution System ΔP (psid)	15	15	15

(a) Based on tube ID
(b) Across tube bundle radius using Eqs. 6.13 and 6.13(b) of HEAT TRANSMISSION by McAdams.

Table G

HEAT EXCHANGER THERMAL DATA – CONDENSER

Item	Reference Daseline	Optimized	
		Surface	Submerged
Water Flow Rate (lb/hr)	841×10^6	814×10^6	856×10^6
Water Flow Per Tube (lb/sec)	3.98	4.0	4.25
Water Velocity Inside Tube (ft/sec)	6.862	6.821	6.9
Reynolds Number Inside Tube (-)	50723	50422	52147
Prandtl Number Inside Tube (-)	10.4	10.4	10.4
Inlet Water Temperature (° F)	43.88	43.88	43.88
Outlet Water Temperature (° F)	48.77	48.38	48.12
Average Water Temperature (° F)	46.32	46.35	46.00
LMTD (° F)	7.08	7.35	7.40
Average Tube Wall Surface Temperature (° F)	52.8	52.8	52.6
Average Shell Wall Surface Temperature (° F)	56.2	56.2	56
HW (Including Enhancement) (Btu/hr ft^2 ° F)	1086	1079	1084
H Fouling (Btu/hr ft^2 ° F)	3333	3333	3333
H Wall (Btu/hr ft^2 ° F)	9709	9709	11235
H Ammonia (Btu/hr ft^2 ° F)[(a)]	3475	3475	3400
Uo (Btu/hr ft^2 ° F)[(a)]	620	618	623
Ammonia Inlet Temperature (° F)	53.86	53.89	53.86
Ammonia Outlet Temperature (° F)	53.50	53.52	53.35
Vapor Quality (Inlet, %)	96.55	96.55	96.5
Maximum Vapor Velocity (Inlet, ft/sec)	9.0	9.2	9.6
Total Pressure Drop of Water (psid)	3.277	2.913	2.808
Total Pressure Drop of Ammonia (psid)[(b)]	0.646	0.672	0.705

(a) Based on tube ID
(b) Across tube bundle radius using Eqs. 6.13 and 6.13(b) of Heat Transmission by McAdams.

2.3 SYSTEM DESCRIPTION

Figure 2-8 shows a 25-MW(e) power system module of a 100-MW(e) surface platform arrangement. The evaporator and condenser are 63.5 ft in diameter spherical aluminum shells with 1.50-in. OD aluminum tubes. Power system auxiliary components, other than the ammonia cycle pumps, are located in the space designated. For simplicity, the heat exchanger and turbine-generator foundations are not shown.

Warm surface water for the evaporator is drawn by four seawater pumps through a 45-ft diameter coated steel duct connected to the platform hull. The pipe centerline is 45 ft below the water surface to ensure that the inlet remains flooded in all operational sea states and a minimum positive head is maintained on the seawater pumps. Seawater discharge from the evaporator is directed through the bottom of the platform. Cold water supply to the condenser is drawn from the cold water pipe plenum located in the center of the platform. Cold water discharge from the condenser is also ducted through the bottom of the platform, with an extended duct to minimize potential for recirculation.

Four single-stage high flow, low head, axial flow pumps provide the water to each heat exchanger. The pumps are clustered into a 45-ft diameter to enhance uniform flow distribution to the heat exchangers with minimum duct lengths. The four pumps provide maximum reliability, as discussed in Section 2.1.

The ammonia distribution and condensate return pumps are located below the heat exchangers to provide adequate NPSH for the pumps. A spare pump is provided in each assembly to provide high subsystem reliability.

Figure 2-9 shows a 25-MW(e) module for a 100-MW(e) spar-type platform. The evaporator and condenser heat exchangers are attached to opposite ends of a dry machinery space. Seawater pumps, located upstream of the heat exchangers, draw seawater from the platform warm and cold water plenums when the module is mated to the platform. Access trunks lead from both the platform and the module lower pump room to the

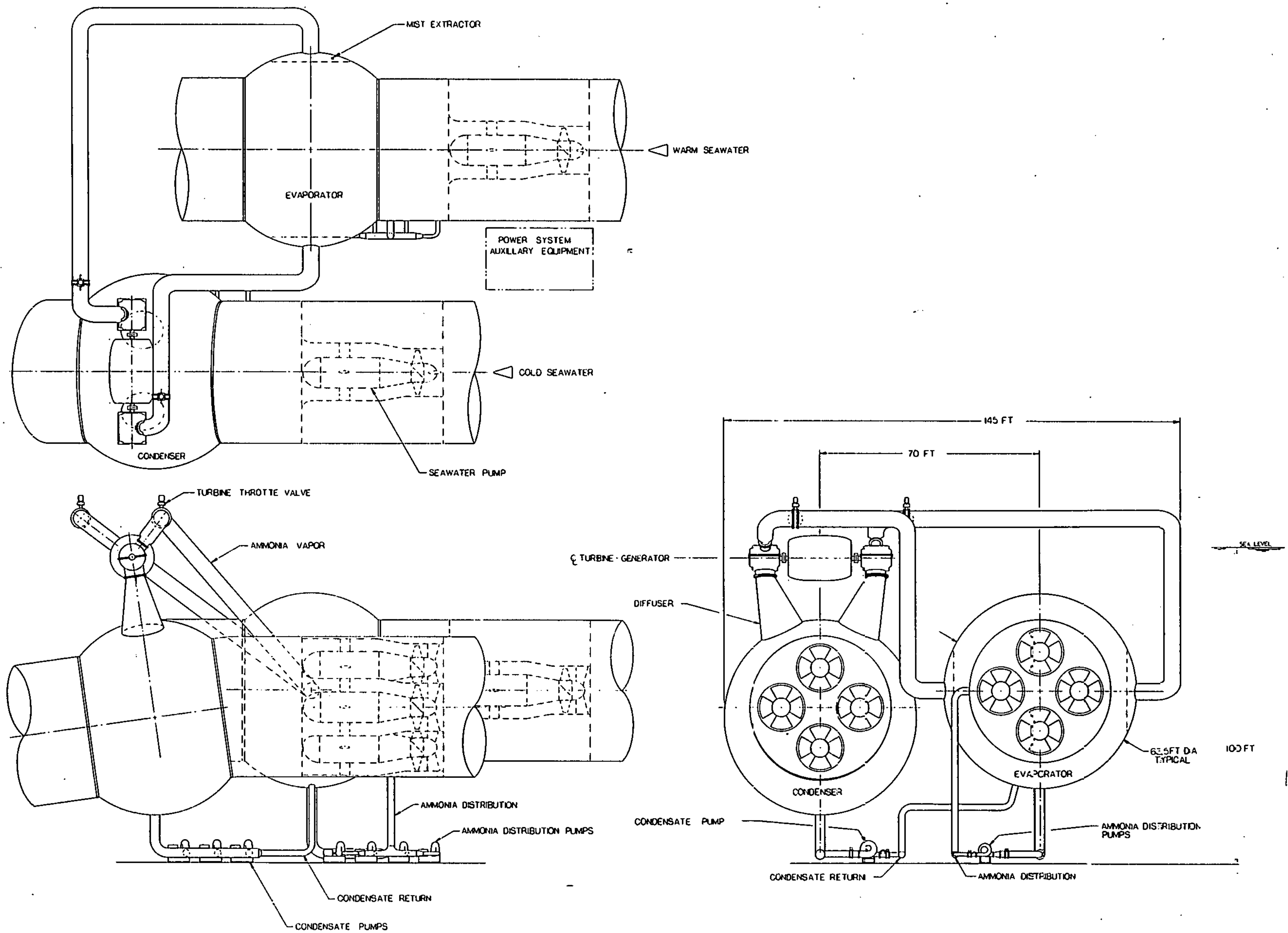

Fig. 2-8 25-MW(e) Power System Module for 100-MW(e) Surface Platform

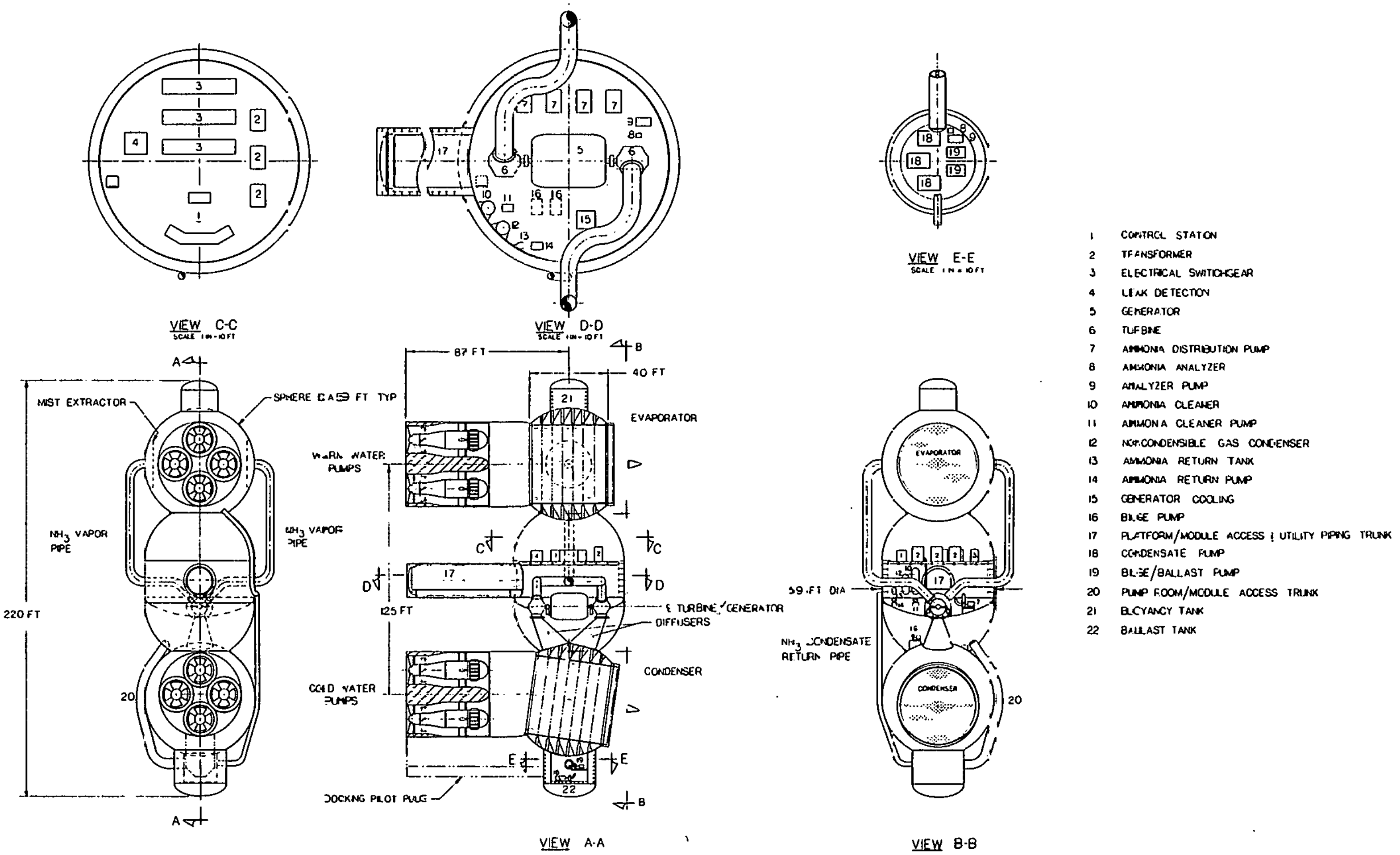

Fig. 2-9 25-MW(e) Power System Module for Spar-Type Platform center machinery space. The platform access trunk is large enough to allow passage of all piping and electrical cables between the platform and module as well as equipment for maintenance. A water-tight cover is fastened to the trunk entrance during transit. The pump room access trunk is a passageway large enough for maintenance access.

Tanks are attached to both ends of the module for additional buoyancy during transit. The bottom buoyancy tank also serves as a ballast tank to up-end the module for mating to the platform.

The heat exchangers are 59-ft diameter spherical aluminum shells with 1.5-in.-diameter aluminum tubes. An externally circumferentially grooved tube fabricated by an embossed process or integrally rolled fin with a smooth internal bore was selected as the most cost effective tube configuration. No seawater-side enhancement is presently included. Biofouling control is accomplished using chlorination and mechanical brushes daily. Chlorine is injected for 20-minute periods three times per day and the brushes (selected for cost and simplicity) are activated once a day for four passes.

The machinery space is a ring-stiffened cylindrical shell with hemispherical ends that intersect with the heat exchangers. The machinery space has three levels. The lower level, which is also the top of the condenser, provides space for the module bilge pumps and the upper termination of the lower pump room access trunk. The mid-level, a torispherically shaped deck, supports the turbine-generator and power system auxiliary equipment. The torispherical shape supports the machinery in membrane stress and does not require vertical supports. The upper level contains the control station, electrical equipment, and leak detection equipment.

Other equipment such as the nitrogen purge system, ammonia storage and fill, and crew accommodations are located on the platform.

2.4 AVAILABILITY

The power system module availability is a function of reliability and maintainability. For reliability, the critical component failure rate was estimated at just under two failures per year requiring 3.5 days repair each. This low failure rate is attributable to redundancy incorporated into the power cycle. Four critical areas have redundancy:

- Evaporator/condenser tubes
- Seawater system pumps
- Ammonia cycle condensate and distribution pumps
- Ammonia turbines

The heat exchangers have 7 percent excess tubes and all tubes have a 0.020-in. corrosion allowance to provide at least a 10-year life with not more than 5 percent of the tubes plugged. With failure of one seawater pump, the module power output drops to 95 percent rated; and with two pumps failed (one per heat exchanger), the power output drops to 90 percent of rated. The ammonia cycle condensate pump assembly and the distribution pump assembly each incorporate a spare pump. The turbine-generator assembly incorporates two turbines.

Seven levels of maintenance were established based on periodicity and activity duration. Level 1, fill and change, occurs weekly; Level 2, adjust and clean, occurs monthly; and to Level 6, overhaul and refurbish, which occurs every 36 months. Level 7 is special and provides for heat exchanger changeout every 10 years. Maintenance for the two power system module configurations with and without Level 7 ranged from 24.7 to 29.7 days per year average. Using a spare detachable module and performing Levels 6 and 7 maintenance at a shipyard reduces the yearly maintenance downtime to 12.1 days.

2.5 COST ANALYSIS

Tables H and I present power module and heat exchanger cost data for the reference baseline ship/platform and for the submerged installation. All costs are expressed in terms of 1977 dollars and an allowance of 30 percent of material and labor cost is included for contingency, engineering, and other changes as shown below.

STRUCTURE OF ALLOWANCE FACTOR

Contingency		0.10
Engineering	0.06	
Home Office	0.04	
Contractor's Fee	0.04	
Total Services		0.14
Owner's Cost and Integrator's Fee		0.04
Cost Multiplier = $1.10 \times 1.14 \times 1.04$ =		1.30

Mature industry is assumed for all components and construction. While a learning curve could have been applied, it is felt that most "learning curve reductions" will occur in the first several plants and no overall learning curve should be applied.

As seen in Table H, the heat exchangers and seawater pumps account for nearly two-thirds of the total power system costs. The turbine and generator account for approximately 10 percent of the costs. The three areas have the greatest potential for cost reductions based on potential size reductions for the heat exchangers and quantity purchases for the rotating machinery.

Table H

POWER SYSTEM COST SUMMARY

Item	Power System Costs ($/kW)			
	Prototype Unit		First Production Unit	
	Reference Baseline	Submerged	Reference Baseline	Submerged
Evaporator	375	272	327	225
Condenser	428	310	371	256
Demister	7	7	7	7
Turbine	83	74	62	55
Generator	57	51	50	45
Seawater System	260	260	200	200
Ammonia Feed Pumps	7	7	6	6
Ammonia Circulation Pumps	10	10	9	9
Ammonia Piping	20	14	17	12
Control Systems	17	17	15	15
Cleaning Systems	23	22	20	19
Other Auxiliary Systems	114	114	100	100
Chlorination	34	35	30	31
Equipment Installation	92	40	81	35
TOTAL	1,527	1,233	1,295	1,015

Table I

HEAT EXCHANGER COST SUMMARY
(EVAPORATOR AND CONDENSER)

Item	Prototype Unit Costs ($/kW)				First Production Unit Costs ($/kW)			
	Reference Baseline		Submerged		Reference Baseline		Submerged	
	Material	Labor	Material	Labor	Material	Labor	Material	Labor
Tubes	346 (a)	101	248 (b)	89	346 (a)	71	248 (b)	63
Tubesheets	19	42	11	26	19	25	11	16
Tube Support Plates	18	60	14	51	18	37	14	31
Shell (c)	57	123	31	75	57	94	31	36
Waterboxes	8	22	8	22	8	16	8	16
Ammonia Distribution (d)	7	2	7	2	7	2	7	2
Nozzles	1	1	1	1	1	1	1	1
Miscellaneous	1	2	1	2	1	2	1	2
TOTAL	457	353	321	268	457	248	321	167

(a) Tubes at $0.84/ft basic + $0.44/ft enhancement
(b) Tubes at $0.70/ft basic + $0.44/ft enhancement
(c) Includes final structural assembly
(d) Includes demisters

2.6 TEST ARTICLES

2.6.1 Guidelines

The major guidelines used for development of test article configurations and sizes are to provide, within reasonable cost, hardware representative of the 25-MW(e) power system which would yield data sufficiently accurate to evaluate the validity and relative merits of 25-MW(e) power system concepts. Further, the data are to provide confidence in (or upgrade) the analytical methods used for predicting performance and cost of the power system configuration.

2.6.2 1-MW(e) Heat Exchangers

No strong rationale exists to deviate from the 1-MW(e) size and, in fact, the use of OTEC-1 as the test platform for the 1-MW(e) heat exchangers is the strongest point in its favor. Configurations to satisfy the guidelines and scaling included variously shaped tube bundles, and additions of ammonia vapor and liquid droplets at various locations within the bundle.

The selected 1-MW(e) heat exchanger test articles are half-tube bundles divided vertically along the tube length. The division gives a half which is symmetrical about the longitudinal vertical plane. This approach provides all the necessary and sufficient data for verification of the liquid distribution, vapor flow, and thermal performance. In addition, the second half can be used for testing, simultaneously, a different tube or tube bundle configuration.

2.6.3 Scaled Power System

The scaled proof of concept power system is a performance-scaled power system. All componentry and subsystems would be configured for performance size in order to ensure component compatibility and operation as a system. Selected subsystems would not have redundant equipment in order to provide cost savings as the scaled power system is not a life test system. All operational features of the power system as well as the manufacturing feasibility and functional aspects (startup, shutdown, off-design operation control) would be demonstrated.

The recommended size of the scaled proof of concept power system is 10- to 12-MW(e) for the following reasons:

- Reasonable size to attract attention of potential users
- Reduce unknown-unknowns of scaling as the most cost effective size is in the range of 25- to 50-MW(e)
- Reduce $/kW capital investment by DOE
- Provide basis for validating commercial cost projections
- Provide reasonably sized component data and system data

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SECTION 3

Section 3
SYSTEM MODELING AND RESULTS

The basic objective of system modeling is to analytically establish the power system component characteristics in order to predict the power system performance at selected design and off-design operating conditions. LMSC has developed proprietary computer programs which define the power system configuration at particular design conditions and determine the off-design performance of the resultant power system. A brief description of these programs and the results of their application to the analytical efforts are presented in the section.

It is possible to use the programs to arrive at an "optimum" configuration for an assigned set of design parameters. This approach was taken to arrive at the configuration designed to provide 25 MW(e) (net) at the 40° F differential temperature specified in the Design Requirements. In addition, designs were optimized for differential temperatures of 32°, 36°, and 44° F. Application of the performance data for these various configurations to potential OTEC sites is discussed below, and the results of various trade studies are presented.

3.1 POWER SYSTEM ANALYTICAL MODELING

3.1.1 Approach

The analytical approach to the selection of an "optimum" configuration is an iterative procedure in that the costs associated with the "minimum unit power cost" system are dependent upon the criteria used to arrive at the configuration, and the criteria themselves may vary as the selection process continues. For a given set of criteria, such as warm and cold water temperatures, material, power system configuration and including platform related items such as cold water pipe length, an "optimum" configuration can be established.

The configuration optimization program developed by LMSC can be used to determine "minimum area" heat exchangers or "minimum cost" installation based on synthesized cost equations. As real data are developed on the manufacturing and component costs, the cost equations are updated and the configurations refined. For conceptual design purposes, the present equations are adequate to define optimization trends.

In order to provide the requisite "net power," the power system must be sized to provide the parasitic power requirements of the seawater and ammonia condensate pumps and other auxiliaries. It is, therefore, necessary to provide programmatic inputs that are platform related and site dependent. For example, the length and diameter of the cold water pipe and the seawater temperature profile are factors which affect the cold seawater pump head requirements. Similarly, the relative location of the heat exchangers is a factor in determining the ammonia condensate pump head requirements.

3.1.2 Optimization Program Output

The various items discussed above are exemplified in the sample Optimization Program output shown in Fig. 3-1. The first page presents comments concerning the particular run, provides data on the working fluid thermodynamic characteristics at various points in the cycle and the geometric configuration for the computation of pressure differences within the working fluid flow path. For the case shown, the plumbing is sized to provide a vapor velocity of 160 ft/sec and a condensate velocity of 30 ft/sec.

The various stations noted on the first sheet are defined as follows:

Station	Location
0	Mist Extractor Inlet (Evaporator Outlet)
1	Mist Extractor Outlet
2	Turbine Inlet
3	Turbine Outlet
4	Condenser Inlet
5	Condenser Outlet
6	Condensate Pump Inlet
7	Condensate Pump Outlet
8	Evaporator Inlet
9	Evaporator Outlet

```
OPTIMIZE ON AREA.  THE OPTIMIZATION VARIABLES ARE:
     T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
     MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
     SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
     SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
THIS RUN IS FOR THE  25-MW POWER SYSTEM DEVELOPMENT WORK
SERIES AT FORTY , 36, 32 -DEGREE DELTA TEMPERATURES    - ALSO 44 DEGREES
EXPONENTT  USED IN KAYS EQUATION IS 0.17
SHUPBOARD CASE - STANDARD BASELINE REFLUX IS NOW 0.5
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.
```

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0	73.74	137.487	.99000	624.66	.4656
01	73.74	137.487		[124.79] LIQUID AT AVERAGE DEMISTER PRESSURE	
1	73.74	137.487	.99000	624.66	.4656
2	73.49	136.898	.99008	624.66	.4637
3	53.87	95.999	.96553	608.01	.3378
4	53.89	96.036	.96552	608.01	.3379
45	53.70	95.699		[102.06] LIQUID AT AVERAGE CONDENSER PRESSURE	
5	53.52	95.364	.00000	101.85	38.8280
6	53.52	99.112	.00000	101.85	38.8280
7	53.55	161.294	.00000	102.18	38.8265
8	53.55	147.792	.00000	102.18	38.8265
89	73.80	137.641		[124.87] LIQUID AT AVERAGE BOILER PRESSURE	
9	73.74	137.491	.99000	624.66	.4656

```
GENERATOR EFF. = .985
DEMISTER BACKFLOW MASS RATE =      -.00 LB/SEC
RANKINE CYCLE EFF. =    .031236
CARNOT CYCLE EFF. =    .037215
EFFICIENCY RATIO =    .839323
```

*** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP (PSI)
1-2	5.672	160.001	1.286	162.000	.0	.203	.386	.000	.589
3-4	6.659	160.001	.933	24.000	-24.0	.019	.000	-.056	-.037
5-6	1.434	30.000	3.771	30.000	-16.0	.566	.000	-4.314	-3.748
7-8	1.434	30.000	3.771	84.000	40.0	1.586	1.131	10.785	13.502
9-0	5.672	160.001	1.286	3.000	.0	.004	.000	.000	.004

Fig. 3-1 Sample Output From Heat Exchanger Optimization Program

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR	
NUMBER OF TUBES	56500.	50000.	
TUBE PITCH	1.25	1.25	
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500	
HEAT EXCHANGER TUBE THICKNESS (IN)	.1025	.1025	
TUBE LENGTH (FT)	39.424	39.796	
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0	
SEA WATER TEMP. IN (F)	43.875	83.875	
SEA WATER TEMP. OUT (F)	48.382	79.076	
TUBE SEA WATER VELOCITY (FPS)	6.821	7.536	
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	419.425	512.960	
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000	
SEA WATER FLOW RATE (LB/SEC)	226085.04	219890.81	
SEA WATER FLOW RATE (CFS)	3525.07	3446.43	
SEA WATER REYNOLDS NO. THRU TUBES	50422.	72424.	
SEA WATER DENSITY (LBM/FT**3)	64.116	63.8[illegible]2	
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D.	618.	703.	BTU/HR-SQ FT
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS	1079.	1383.	USING KAYS EQ.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000	
WORKING FLUID HEAT TRANSFER COEFFICIENT	3000.	3000.	
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300	
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000103	.000103	
LOG MEAN TEMP. DIFFERENCE (F)	7.347	7.415	
NTU'S	.6135	.6472	
EFFECTIVENESS	.4585	.4765	
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000	
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.672	.301	
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000	
TUBE CORE DIAMETER (FT)	39.00	36.69	
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.4326	.4326	
TOTAL HEAT FLOW ACROSS (BTU/SEC)	952777.	983497.	
TOTAL HEAT FLOW ACROSS (MW)	1005.4199	1037.8375	
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	814952.	727271.	TOTAL =1542223.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	32.6	29.1	TOTAL = 61.7

Fig. 3-1 (Cont.)

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	881.	862.
FLOW RATE PER PUMP (GPM)	395541.	386717.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	607.6	513.0
PUMP HEAD (FT)	9.474	8.040

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

	CONDENSER	EVAPORATOR
PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75],	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75],	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.5310
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0251
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.265
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

	CONDENSER	EVAPORATOR
PUMP DIAMETER (FT)	9.26	9.26
PUMP SPEED (RPM)	132.2462	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	3.752	3.097

Fig. 3-1 (Cont.)

***** COLD WATER PIPE VARIABLES	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	4.523	10.177
C.W. PIPE REYNOLDS NO.	19519786.	29279679.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	4	
STATIC HEAD OF COLD WATER (PSF)	135.5	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	188.2	

***** TURBINE VARIABLES

GROSS TURBINE POWER OUTPUT OF 1 TURBINES	33.072	MW
NO. OF TURBINES	1	
FLOW RATE AT TURBINE EXHAUST / TURBINE	5572.96	CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	1882.36	LB/SEC
WET TURBINE EFFICIENCY	.876	
TURBINE WORK	16.649	BTU/LB
ISEN. TURBINE WORK	18.996	BTU/LB
AIRCO AXIAL FLOW		
SPECIFIC SPEED	91.570	RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710	DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420	FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.382	DIMENSIONLESS
DRY EFFICIENCY	.902	
FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION		
TURBINE SPEED	1645.	RPM
TURBINE DIAMETER	9.61	FT
REYNOLDS NUMBER	.372+09	
ROTOR TIP SPEED	827.81	FT/SEC
FLOW COEFFICIENT	.2288	DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.6941	DIMENSIONLESS
VELOCITY RATIO	.849	DIMENSIONLESS
SPOUTING VELOCITY	975.34	FT/SEC

***** CONDENSATE PUMP VARIABLES

NO. OF CONDENSATE PUMPS	2	
PUMP EFF.	.900	
PUMP MOTOR EFF.	.900	
PUMP WORK	.329	BTU/LB
ISEN. PUMP WORK	.296	BTU/LB
MASS FLOW RATE / PUMP	941.18	LB/SEC
VOLUME FLOW RATE / PUMP	24.2398	CFS
MP HEAD	62.182	PSI

Fig. 3-1 (Cont.)

The working fluid plumbing, shown on that sheet, is representative of the reference surface platform installation. The pipe lengths, bends, and head differentials are adjusted, as required, for the component orientation as influenced by the selected platform.

The second sheet presents data on the physical characteristics of the "optimized" heat exchangers, the seawater characteristics, including flow rates and temperatures, and the various heat transfer coefficients and thermal resistances which determine the overall heat transfer coefficient. The magnitude of the total heat flux across the heat exchangers is also shown.

The third sheet presents data on the seawater pump variables. The physical characteristics and speed data are based on averaging specific diameter and speed data provided by three manufacturers. These data are informative only to indicate the pump sizes required for OTEC power module application. As indicated, the parasitic power required for seawater pumping is based on a pump efficiency of 86 percent and a motor efficiency (including any reduction gearing) of 90 percent. As vendor data are received for actual pump configurations, these values are adjusted as required.

The cold water pipe data, presented on the fourth sheet, are included to show the pressure losses which are included in the determination of the cold seawater pump parasitic power. The program permits the incorporation of a constant diameter pipe or a segmented pipe of variable diameter sections.

The thermodynamic and physical characteristics of the turbine are also presented on this sheet. The turbine efficiency is based on synthesized data provided for the study conducted under Contract NSF/RANN/SE/GI-C937/FR/75/1. As the design is refined in the Preliminary Design task, the design point turbine configuration will be established and a guaranteed efficiency value used in the development of the final configuration.

The flow rate and pressure rise for the condensate pumps are also shown, together with the pump and motor efficiencies used in the determination of pump power requirements. These computations will be iterated, during the Preliminary Design task, based on the actual performance of selected pumps.

3.2 SITE THERMAL RESOURCES

Environmental data, presented in the OTEC Demonstration Plant Environmental Package, includes temperature and current data at five potential OTEC sites. The temperature data are presented in terms of temperature differences between the surface and various depths throughout the year. Figures 3-2 through 3-6 present the temperature differences between the surface and two depths (2,000 ft and 3,280 ft) for the candidate sites at New Orleans, Hawaii, Key West, Puerto Rico, and a section of the Atlantic Ocean off the coast of Brazil. It is apparent from these figures that the cyclic variation in available resource has a significant impact on the annual power output of a selected plant. For example, if a plant is sized to provide 25 MW(e) (net) at the 40° F ΔT, and operated at the New Orleans site, it will be capable of providing 25 MW(e) or more for less than 4 months. It is apparent, therefore, that an a priori selection of the design ΔT may not result in the most cost-effective configuration.

3.3 DESIGN/OFF-DESIGN PERFORMANCE

The off-design performance sensitivity of a selected design configuration can be determined by the two LMSC proprietary computer programs. One program maintains the design seawater and ammonia flow rates and determines the net power output as a function of the thermal resources. The second program varies the seawater and ammonia flow rates to maximize the net power output as a function of the thermal resources. This latter approach permits a trade study to determine the desirability of incorporating variable speed pumps to obtain the maximum net power at each operating condition.

The results of the off-design performance computations are shown in Fig. 3-7. These performance curves are based on the off-design characteristics of the radial in-flow turbine, and similar curves are in process for the axial flow turbine. The data shown in Fig. 3-7 are for configurations designed to provide 25 MW(e) (net) at 32°, 36°, 40°, and 44° F temperature differences over the ΔT range of 40° F $^{+4}_{-11}$ as specified in the Design Requirement.

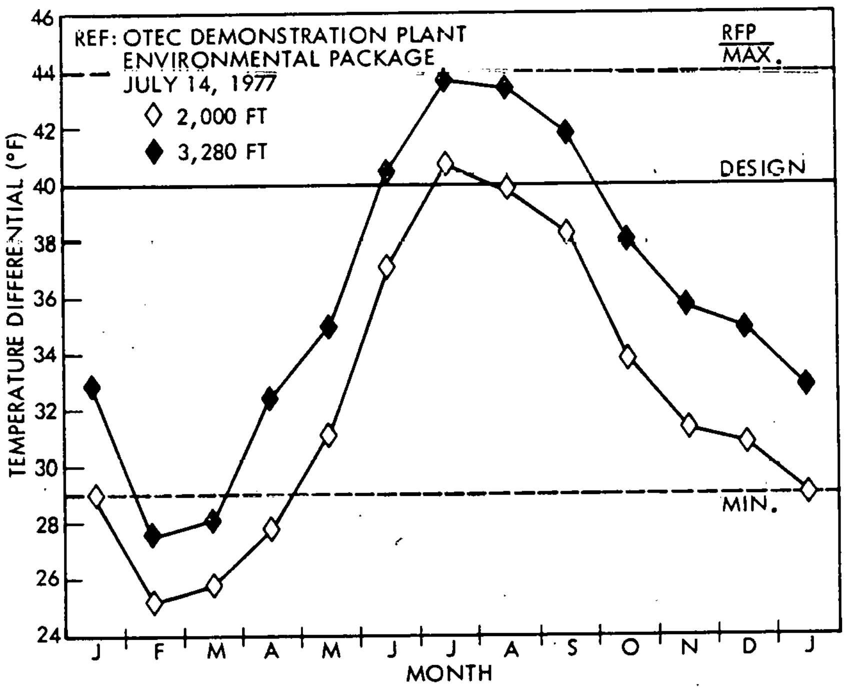


Fig. 3-2 Temperature Differentials Available at New Orleans Site

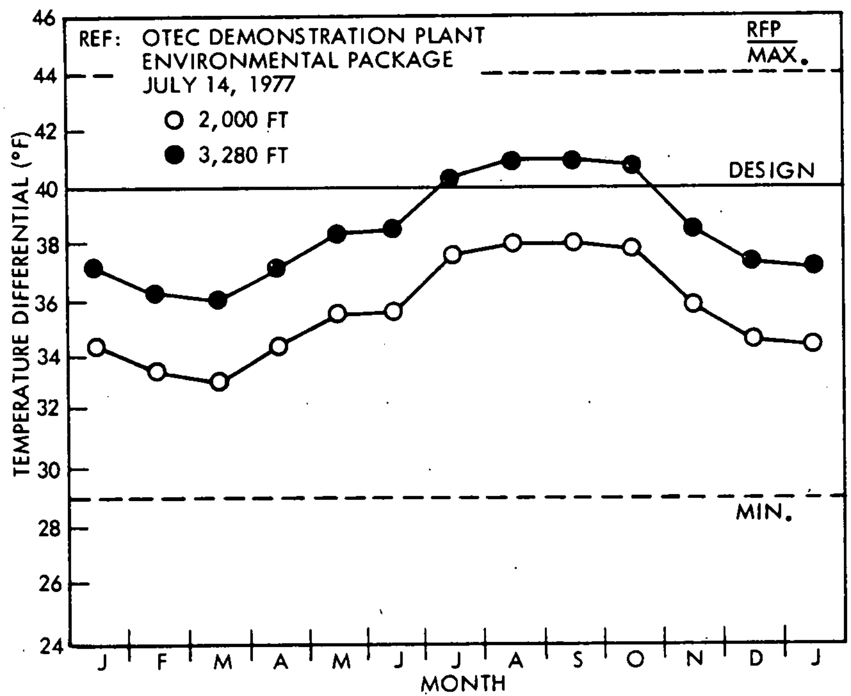


Fig. 3-3 Temperature Differentials Available at Hawaii Site

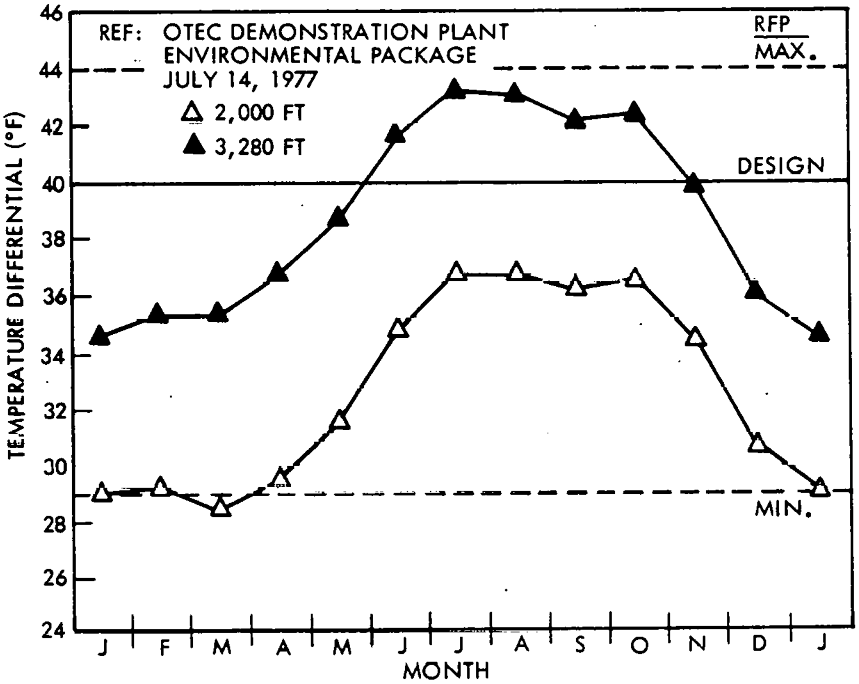


Fig. 3-4 Temperature Differentials Available at Key West Site

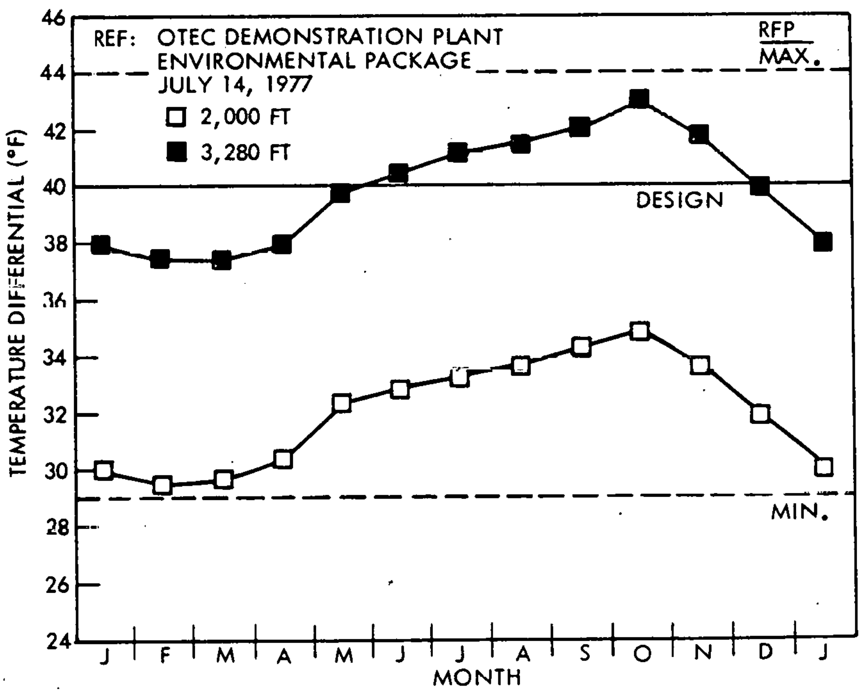


Fig. 3-5 Temperature Differentials Available at Puerto Rico Site

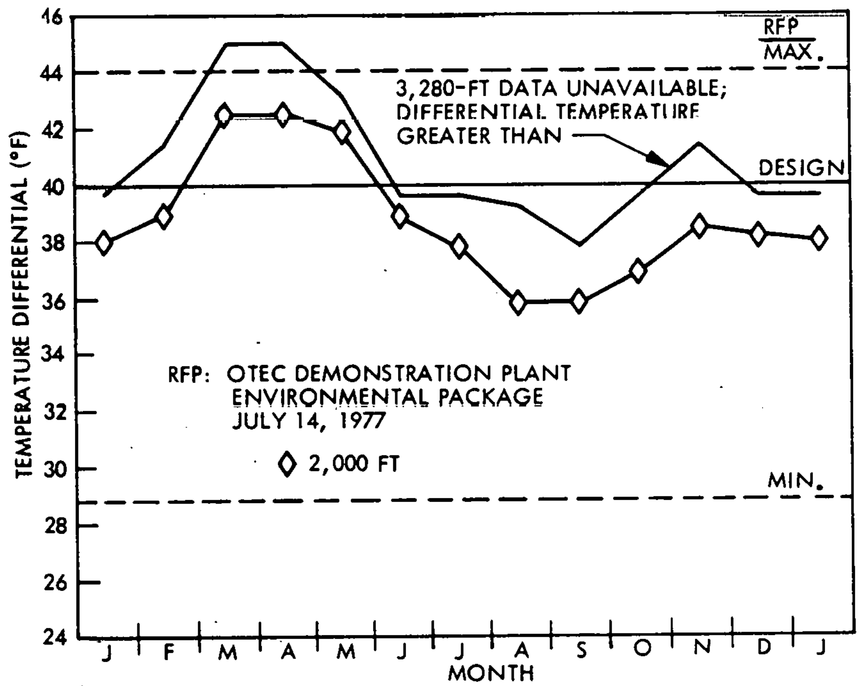


Fig. 3-6 Temperature Differentials Available at Brazil Site

Use of these curves is demonstrated in the following examples. Consider the temperature profiles shown for New Orleans for the 2,000- and 3,280-ft inlet depths. Assuming that the values shown are averages for the month, the total power output for each month can be determined. The annual power output of a Power System Module designed to provide 25 MW(e) (net) at a ΔT of 40° F is shown in Table 3-1 for inlet depths of 2,000 and 3,280 ft at the New Orleans site. These data are based on the optimized flow curve of Fig. 3-7. The relative annual power output for various design conditions is shown in Table 3-2. Using the 40° F ΔT design with the inlet at 2,000 ft as the baseline, the relative power outputs for the other design configurations are as shown. For example, increasing the cold water pipe length to 3,280 ft at the New Orleans site increases the power output by approximately 30 percent. Similarly, using the configuration designed for the 36° F ΔT, the power output is increased by approximately 30 percent with the 2,000-ft inlet, and approximately 70 percent with the cold water pipe inlet at 3,280 ft. Similar data are shown for other design conditions and the other potential sites.

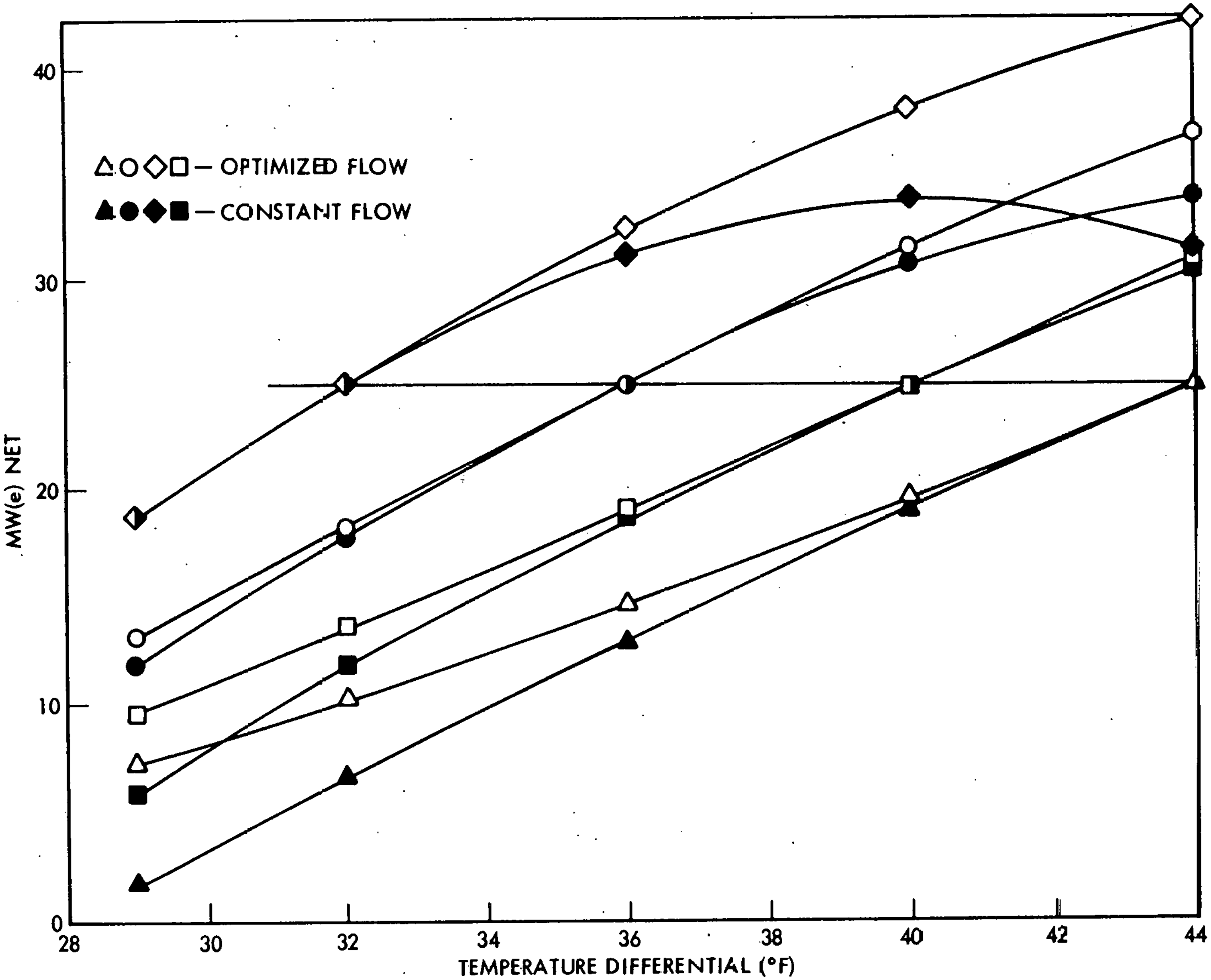


Fig. 3-7 Off-Design Performance of 25-MW(e) (Net) Module – Radial Turbine

Table 3-1

ANNUAL POWER OUTPUT, NEW ORLEANS (DESIGN ΔT 40° F)

Month	Power Output (MWh)	
	2,000 ft	3,280 ft
Jan	7,142	10,862
Feb	3,898	5,376
Mar	4,762	6,250
Apr	6,048	10,080
May	9,151	13,169
Jun	14,832	18,504
Jul	19,344	22,692
Aug	18,302	22,320
Sep	16,056	19,872
Oct	11,904	16,368
Nov	9,072	13,320
Dec	8,854	13,020
TOTAL	129,365	171,833

Table 3-2
RELATIVE ANNUAL POWER OUTPUT

Site (Inlet Depth)	Design ΔT			
	32° F	36° F	40° F	44° F
New Orleans (2,000 ft)	1.72	1.31	1.0	0.77
New Orleans (3,280 ft)	2.13	1.69	1.33	1.03
Hawaii (2,000 ft)	1.70	1.31	1.0	0.77
Hawaii (3,280 ft)	1.94	1.56	1.22	0.95
Key West (2,000 ft)	1.78	1.33	1.0	0.76
Key West (3,280 ft)	2.48	2.02	1.60	1.26
Puerto Rico (2,000 ft)	1.83	1.35	1.0	0.75
Puerto Rico (3,280 ft)	2.76	2.27	1.81	1.43
Brazil (2,000 ft)	1.57	1.27	1.0	0.78
Brazil (3,280 ft) [a]	>1.69	>1.4	>1.13	>0.9

(a) Temperature data incomplete at 3,280 ft (see Fig. 2-6); ratios are greater than values shown.

These data, together with the cost data developed during the course of the PSD program, are used to determine the configuration which provides the minimum cost power output. For example, the desirability of incorporating variable speed seawater and ammonia pumps can be evaluated by determining the relative power output from the curves of Fig. 3-7. Table 3-3 presents the annual power output for the configuration designed for the 40° ΔT and 2,000-ft cold water inlet with variable seawater and ammonia flow and with constant flow. The data shown for the temperature profile at New Orleans indicate that the use of variable flow pumps increases the power output by approximately 20 percent. Table 3-4 shows the performance gains that can be realized by optimizing the seawater and ammonia flows at the various sites and with the indicated design conditions. These data are all based on the off-design performance of the radial in-flow turbine. In some instances, the data were extrapolated and the performance at ΔT's that are more than 8° F from the design condition may not be exact.

3.4 SELF-SUSTAINING OPERATIONAL CAPABILITY

The data shown in Table 3-3 indicate that the ΔT available at the New Orleans site during the month of February is insufficient to provide any net power output for the constant flow configuration designed for the 40° F ΔT. Accordingly, the off-design computer programs were run at lower values of ΔT available with optimized flow rates. The results are shown in Fig. 3-8 for the configuration designed to provide 25 MW(e) at a ΔT of 40° F, and in Fig. 3-9 for the configuration having a design ΔT of 36° F. It is apparent from these results that it is desirable to incorporate variable flow seawater and ammonia condensate pumps in order to maximize the operational capabilities of the power system module and, consequently, the total plant.

3.5 SYSTEM EVALUATION CRITERIA

The development of the final cost curves for the determination of minimum cost of power over the life of the system requires the incorporation of all component costs, maintenance costs, and economic factors, as well as platform costs. Although the latter are outside the scope of this program, it is possible to develop "figures of merit" or evaluation criteria which establish trends leading to selection of an optimum configuration.

Table 3-3

ANNUAL POWER OUTPUT, NEW ORLEANS (DESIGN ΔT 40° F), COLD WATER INLET AT 2,000 FT

Month	Power Output (MWh)	
	Optimized Flow	Constant Flow
Jan	7,142	4,315
Feb	3,898	– (a)
Mar	4,762	360
Apr	6,048	2,736
May	9,151	7,291
Jun	14,832	14,616
Jul	19,344	19,344
Aug	18,302	18,302
Sep	16,056	15,984
Oct	11,904	11,011
Nov	9,072	7,344
Dec	8,854	6,845
TOTAL	129,365	108,148

(a) ΔT of 25.2° F indicates less than zero net power

Table 3-4

RATIO OF PERFORMANCE OPTIMIZED FLOW/FIXED FLOW

Site (Inlet Depth)	Design ΔT			
	32° F	36° F	40° F	44° F
New Orleans (2,000 ft)	1.08	1.07	1.20	1.41
New Orleans (3,280 ft)	1.11	1.04	1.07	1.17
Hawaii (2,000 ft)	1.04	1.01	1.04	1.16
Hawaii (3,280 ft)	1.09	1.01	1.01	1.05
Key West (2,000 ft)	1.02	1.03	1.16	1.45
Key West (3,280 ft)	1.14	1.03	1.01	1.06
Puerto Rico (2,000 ft)	1.01	1.03	1.16	1.55
Puerto Rico (3,280 ft)	1.15	1.03	1.01	1.03
Brazil (2,000 ft) [(a)]	1.11	1.02	1.01	1.05

(a) Temperature data incomplete at 3,280 ft (see Fig. 2-6)

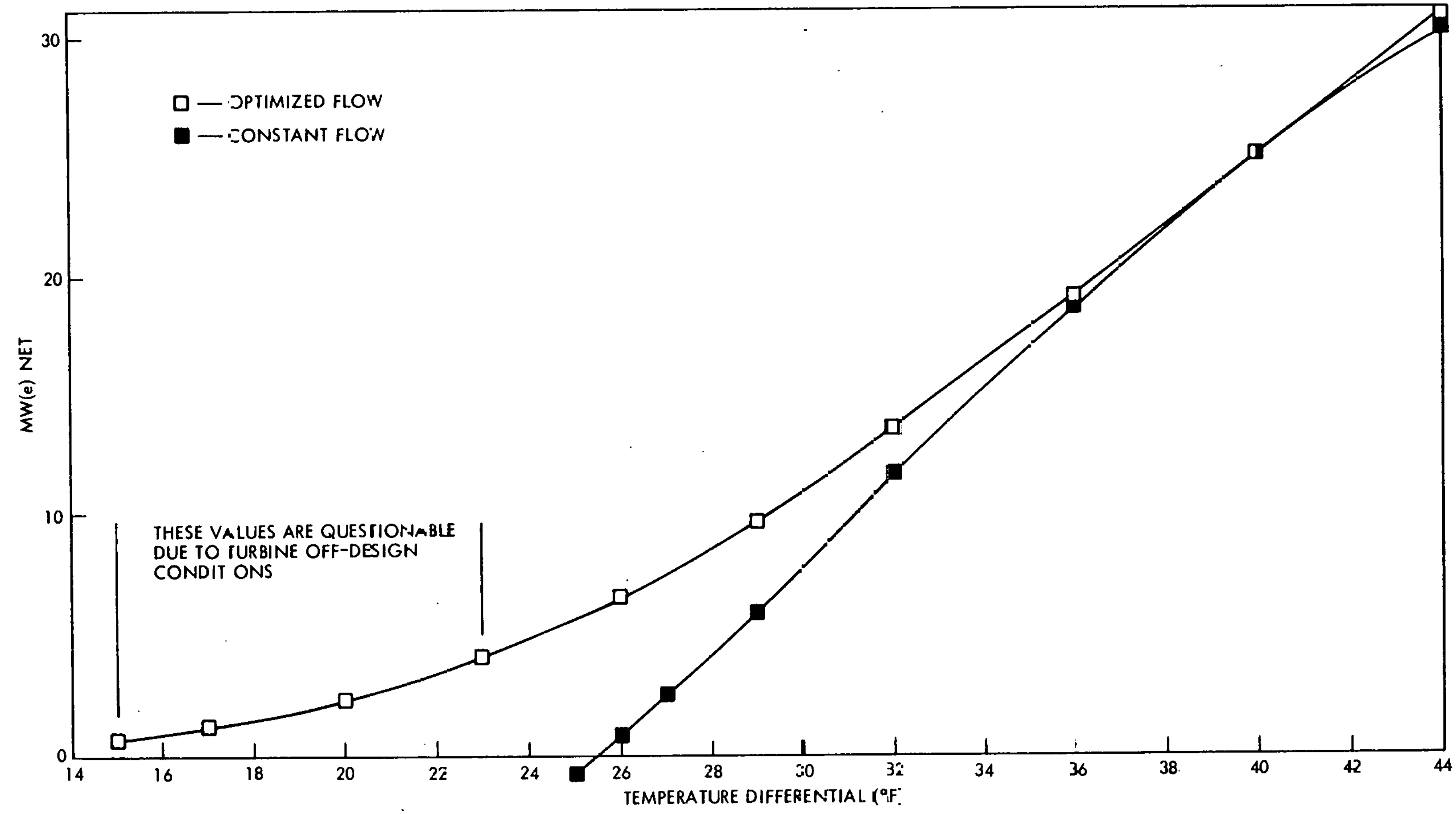


Fig. 3-8 Off-Design Performance for Radial Turbine – 25 MW(e) (Net) Module, Design ΔT = 40° F

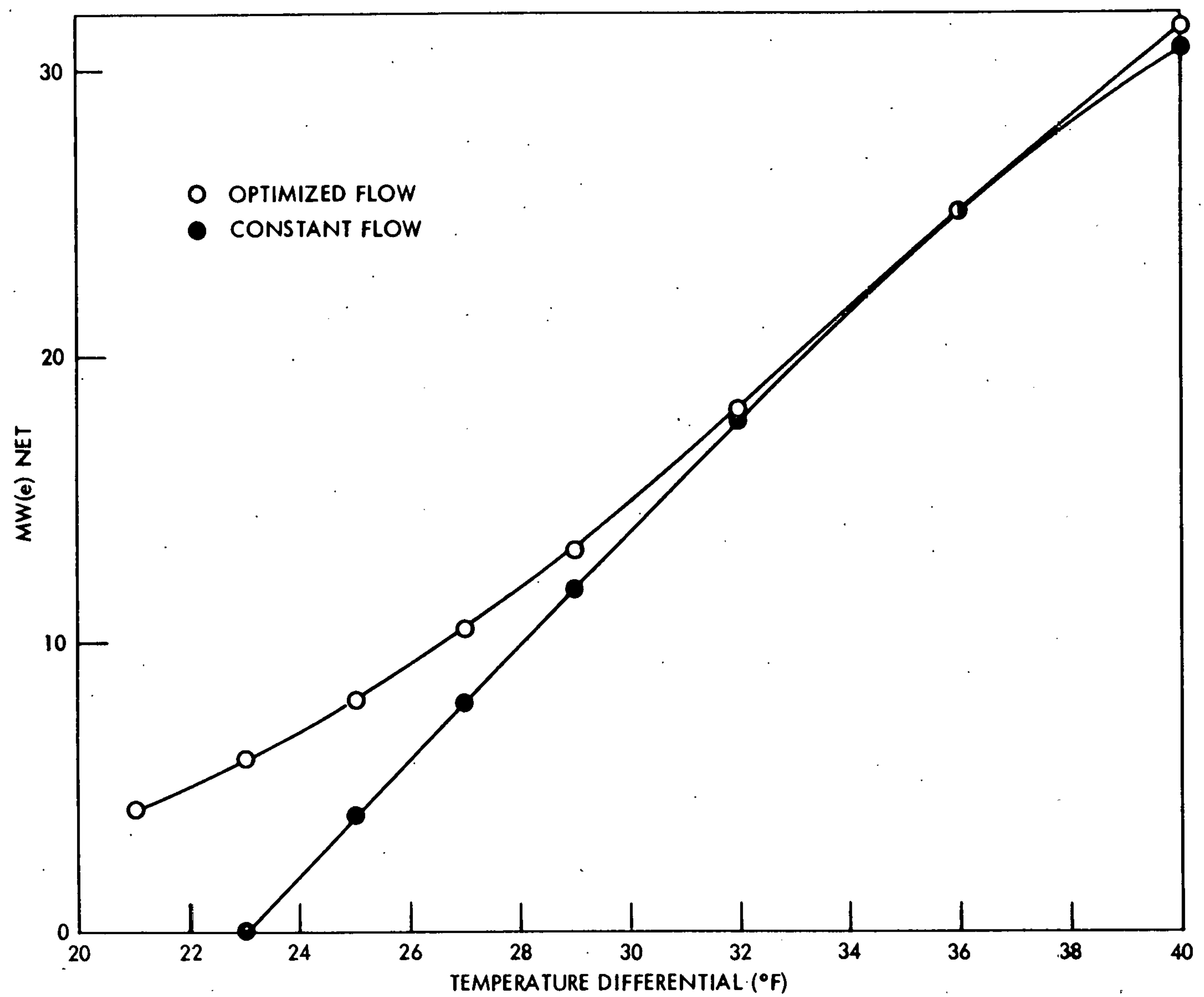


Fig. 3-9 Off-Design Performance for Radial Turbine – 25 MW(e) (Net) Module, Design ΔT = 36° F

One "criterion" evaluated is the cost of the major module components related to the annual power output as a function of the design ΔT. This parameter is site dependent as can be seen from the temperature profile data of Fig. 3-2 through 3-6 and the annual power data of Tables 3-1 and 3-2. Cost data have been provided for a series of heat exchangers designed for 32°, 36°, 40°, and 44° F ΔT using 1.5-in. OD aluminum tubes with 0.065-in.-thick walls. Data provided on the cost of families of turbine-generators and seawater pumps are used to establish costs for the design conditions. In the case of the turbine-generators, the units are sized for the maximum gross power output associated with the thermal resource profiles and the design ΔT. This "evaluation criterion" is shown in Fig. 3-10 for the New Orleans site for the two cold water inlet depths of 2,000 and 3,280 ft. From these curves, in which the lowest value is most desirable, it would appear that the lowest design ΔT is most desirable. However, the low ΔT requires larger heat exchangers and rotating machinery. The heat exchanger

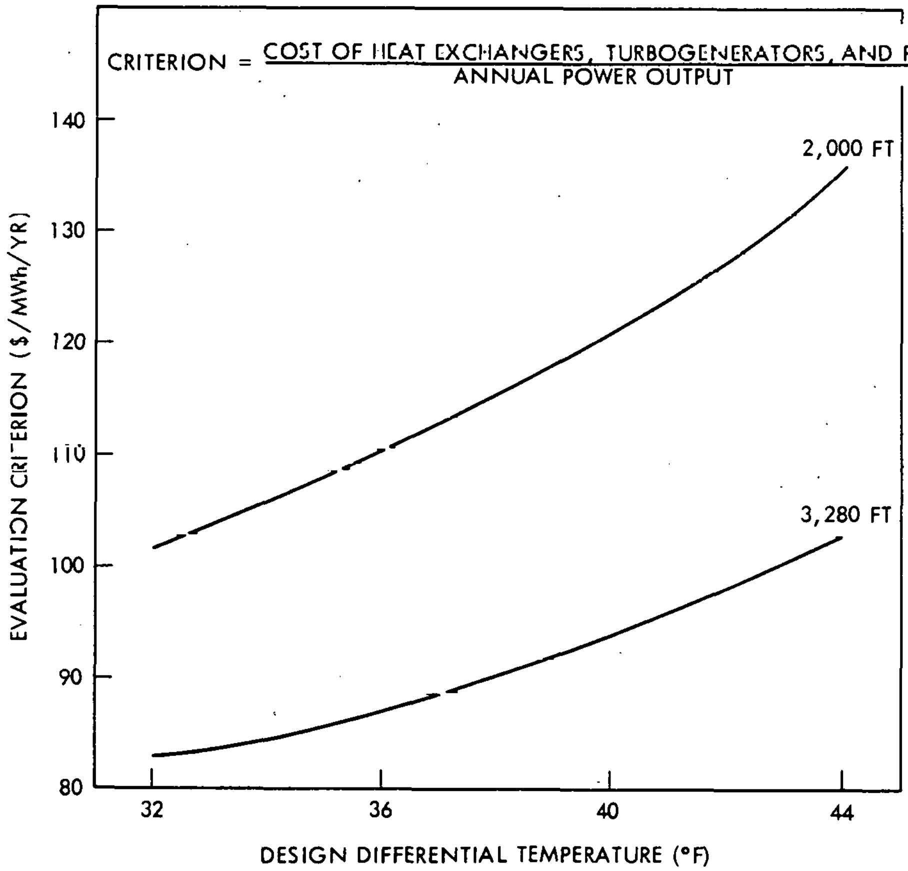


Fig. 3-10 Module Evaluation Criterion – New Orleans Site

volume has a major impact on the platform and should therefore be considered in the evaluation criterion. With cylindrical shells, the 40° F ΔT design volume is 100,000 ft^3. The incremental heat exchanger volumes from the 40° F design are as follows:

Design ΔT	Incremental Volume (ft^3)
32°	81,750
36°	32,100
44°	- 25,520

Assuming that the cost effect for integrating the additional volume ranges from \$20 to \$80 per cubic foot, depending on the type of platform and location of additional volume, the effect of incorporating this volume penalty into the "evaluation criterion" is as shown in Fig. 3-11. It is apparent that the volume effect and related cost penalty has a significant impact on the "optimum" installation.

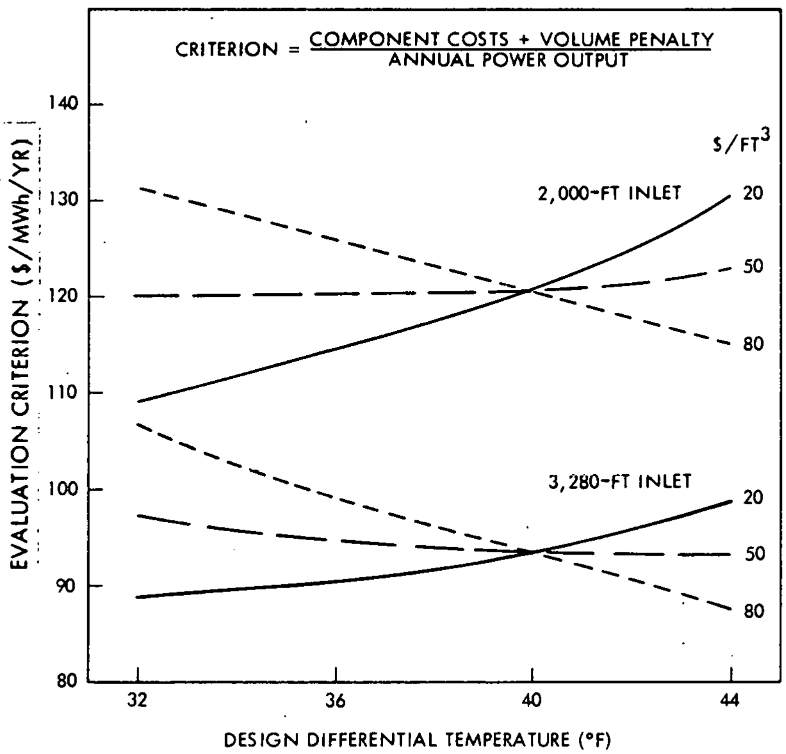


Fig. 3-11 Module Plus Volume Evaluation Criterion – New Orleans Site

From the results presented above, it is evident that a family of OTEC power plants must be designed in order to determine the most cost-effective installation for each site or annual available ΔT.

SECTION 4

Section 4
BASELINE POWER MODULE

This section presents physical and functional descriptions of the baseline power module and comparisons of this baseline with power module configurations optimized for two alternative platform installations. The baseline power module is configured to provide 25-MW(e) net power output under the following design requirements:

- Design point temperature difference of thermal resource is 40°F with a surface water temperature range of 70° to 85°F.
- A surface platform/ship is used as the basic reference hull.
- Total pressure drops for the warm and cold seawater systems are 10 and 15 ft of water, respectively.

In keeping with the requirement to address the "optimum power system design," two alternative configurations were developed, one for application to a surface vessel and the other for a submerged installation. Trade studies in the areas of heat exchanger design, materials, biofouling control, rotating equipment, and module subsystems are also included.

Tabulated data on performance and physical characteristics are presented to permit a rapid comparative assessment of the baseline and the optimized configurations.

4.1 LAYOUT DESCRIPTION

Two power system module arrangements were generated – the baseline or reference surface platform power system mounted, component by component, within a surface platform; and the recommended power system module which is a complete set of power generation componentry mounted in a submersible detachable module attached to a spar platform. These two configurations provide comparison and evaluation of internal and external modules.

4.1.1 Baseline Power Module

The 25-MW(e) power system module arrangement for a 100-MW(e) surface platform/ ship is shown in Fig. 4-1. For simplification, power system auxiliary equipment – other than ammonia cycle pumps – are not shown, but are located in the designated space. Foundations and minor mechanical equipment and componentry are also not shown.

The subsystems, assemblies, and subassemblies for the power system are tabulated in Appendix B, Hardware Breakdown Structure. A schematic of the power system is also in Appendix B. The estimated ammonia cycle volume is 195,000 ft^3, and 375,000 lb of ammonia are needed to fill the system for operation.

The aluminum, spherical-shell heat exchangers, 63.5 ft in diameter, are mounted on foundations attached directly to the platform hull structure. The evaporator centerline is 45 ft below the water surface to ensure a totally flooded water supply duct and heat exchanger during all operational sea states and a positive head at the seawater pump inlets. The condenser centerline is 53 ft below the water surface to minimize vapor piping lengths from the evaporator to the condenser, and to provide the inlet head height for the condensate pumps while staying within the 100-ft platform draft.

The evaporator contains 52,260 1.5-in. diameter 5052 alloy shell-side enhanced tubes welded to 2.5-in.-thick outer tubesheets. Flat annular inner tubesheets, of the same thickness, with the same OD as the outer tubesheet and a 28-ft ID, are spaced 2 ft inside

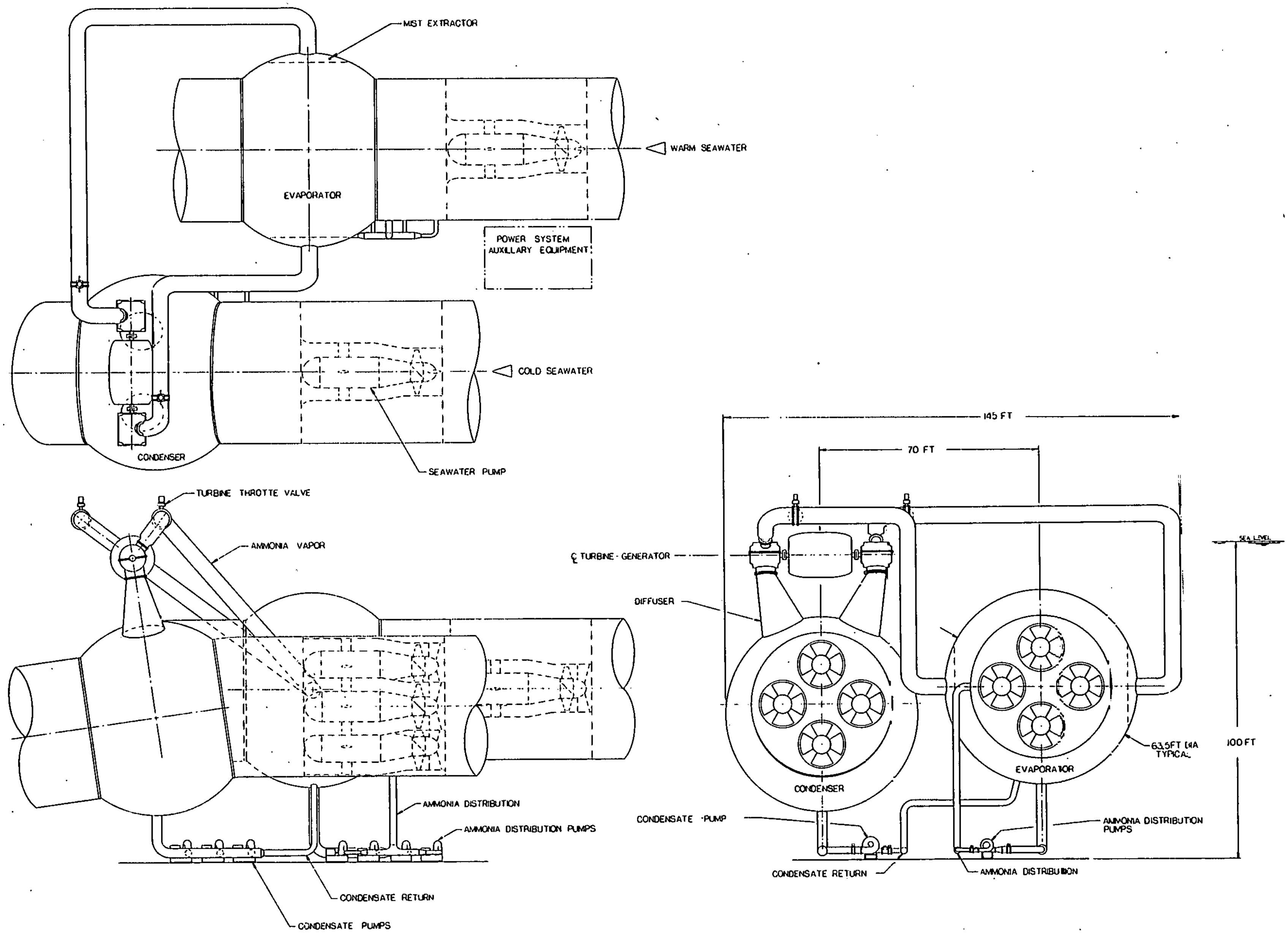


Fig. 4-1 25-MW(e) Surface Platform General Arrangement the outer tubesheet to take the shell bending loads. The condenser has 58,700 1.5-in. tubes of the same alloy and enhancement configuration as the evaporator. The tube-sheet structure is also the same. Approximately 50 percent of the tubes in each heat exchanger will penetrate the inner annular tubesheet.

The spherical shell provides internal volume for tube bundle support, pools of liquid that act as sumps (no external sump required), and the mist extractor. A 5-ft depth of liquid in the shell provides a volume of almost 2,500 cu ft. In addition, lugs are provided in the evaporator shell for mounting a 25-ft-diameter mist extractor over each vapor exit port. The mist extractor is 9 in. thick and has four to six liquid trays to drain off collected droplets and return the liquid to the sump. Details of the hook and vane mist extractor and its relative position within the evaporator are shown in Fig. 4-2. This type of mist extractor is in use in natural gas plants, chemical plants, and ammonia plants. The shells (3.2-in.-thick), tubesheets, baffles, and tubes for both the evaporator and condenser are of the same design to minimize the cost of manufacture and assembly. The condenser is canted at an angle of approximately 10 deg. During the course of the OTEC Heat Exchanger Design and Producibility Study, several references were found which indicated an improvement in condensate drainage with small angles of inclination. These references are included at the end of Section 4.1.

Carbon steel piping, coated on the outside, connect the ammonia vapor and liquid components. The 6-ft-diameter ammonia vapor pipe designed for 250 psia internal pressure conducts the ammonia vapor at a velocity of 160 fps. The liquid ammonia piping, coated on the outside, is 1.5-ft in diameter, and ammonia flows at 30 fps.

Three condensate pumps, each at 50-percent system flow capacity (12,500 gpm), provide the flow and pressurization of the liquid from the condenser to the sump in the evaporator. Four distribution pumps, each at 33-percent evaporator flow capacity (12,500 gpm), distribute the liquid ammonia through two manifolds over the top of the tube bundle and one manifold across the tube bundle at its horizontal mid plane. By

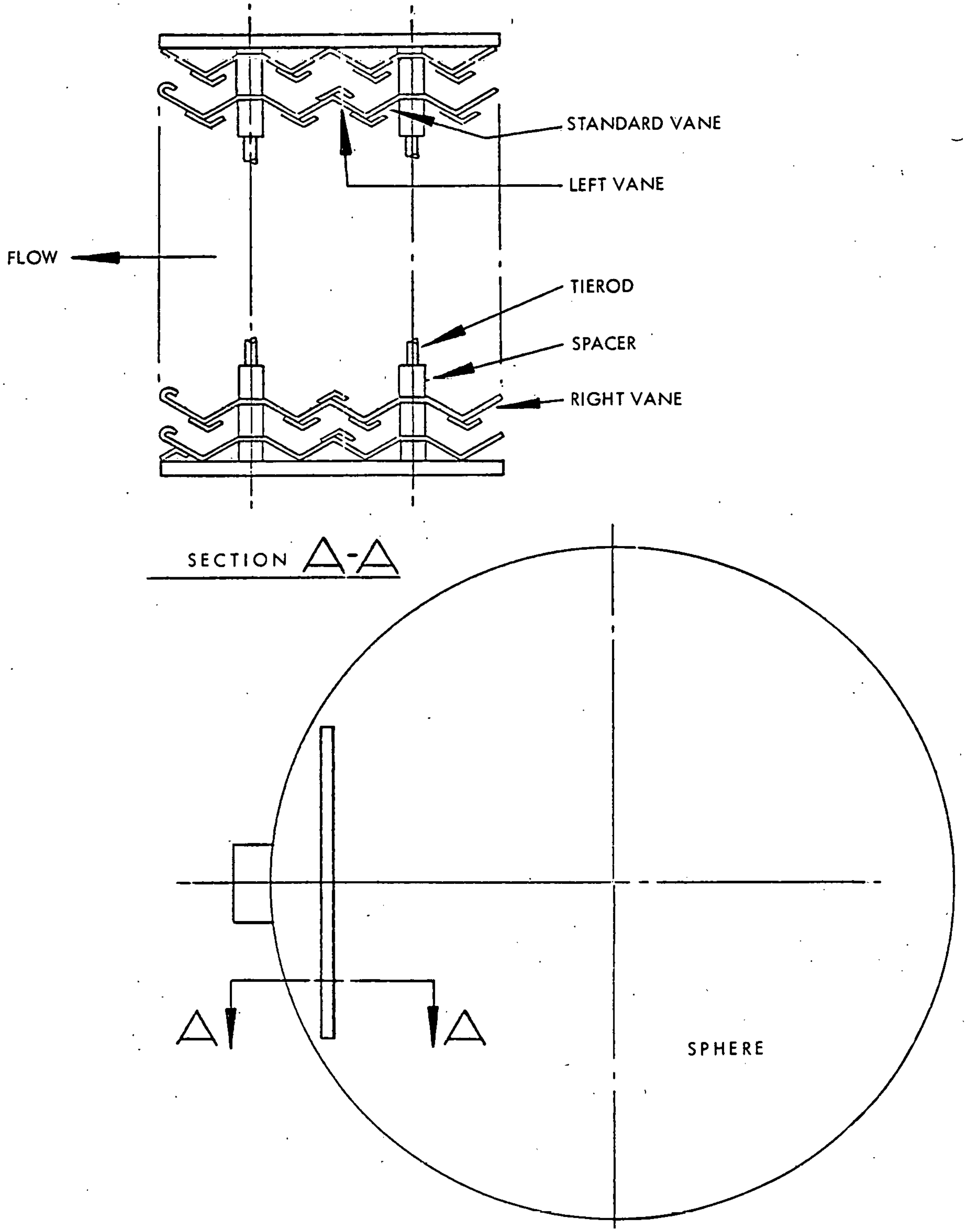


Fig. 4-2 Mist Extractor Details

providing a 50-percent reflux rate, all seven pumps have the same flow rate and therefore are the same except for the drive motor. In addition, by providing a spare pump in both the condensate and the distribution subsystems, the reliability of these subsystems is greatly improved.

Two radial in-flow turbines, rated at 18.6 MW(e) each, drive one 37-MW(e) 13.8-kV generator. The turbines can operate together or independently. The electrical power from the generator is delivered to a platform bus for end item use, and to a transformer for power module usage. The 13.8-kV is controlled and transformed to 4,160 for seawater and condensate pumps, and 480 to 120 volts for other power system and platform purposes.

The seawater pumps are single-stage, axial flow with high volume, low head capacity (1000 cfs at 10 to 15 ft). Four pumps are mounted in a cluster to provide uniform distribution to the heat exchanger in a minimum duct length. All seawater ducting and pumps are steel, coated by corrosion resistant material compatibility with the aluminum heat exchangers. (See Appendix C for critical component physical characteristics).

As the biofouling mechanisms are not fully understood, the biofouling countermeasure system incorporates two means for combating the fouling. Chlorination is used to kill the macro- and micro-organisms, and mechanical brushes inside the tubes of the heat exchanger are used to remove any surface fouling material. The chlorination is injected for 20 minutes three times per day upstream of the seawater pumps. The specific dose rate is not known but is estimated to be between 0.2 mg/ℓ to 1.6 mg/ℓ. The brush system is operated twice a day (four brush trips per day) either at one time (15 minute cycle) or at two different times of the day (7.5 minutes per cycle).

A nitrogen purge system is used for transit purposes or for inerting the ammonia cycle prior to maintenance on components within the cycle. A noncondensible gas removal system is employed to remove the traces of nitrogen remaining after transit or after repair.

4.1.2 Submerged Detachable Power Module

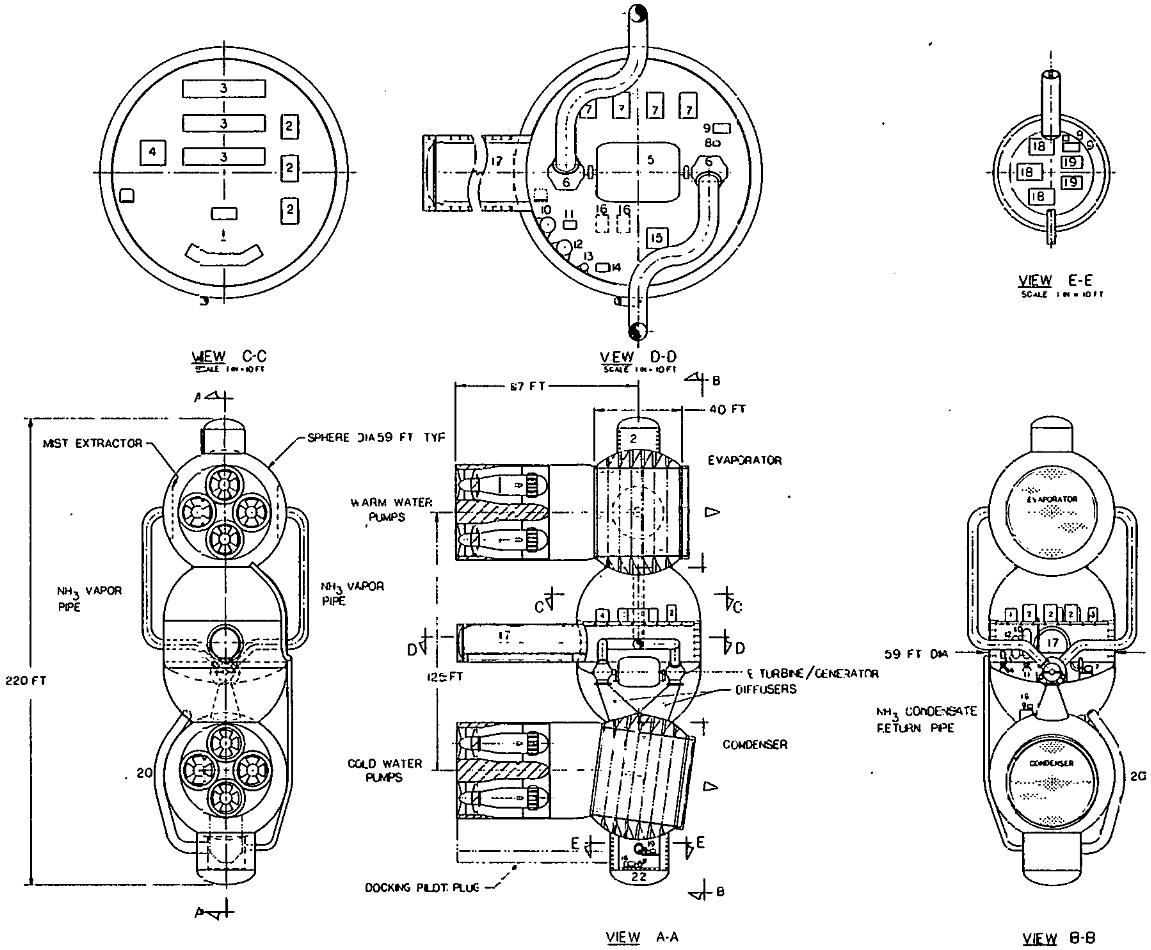
The 25-MW(e) power system module arrangement for a 100-MW(e) platform is shown in Fig. 4-3. The subsystem, assemblies, and subassemblies are tabulated in Appendix B, Hardware Breakdown Structure. The system schematic is given in Appendix B.

The submerged module, in terms of equipment, is very similar to the baseline or reference module. However, component and component mounting details are different. The remaining discussion pertains to these differences.

The major difference is the method of mounting and housing the power system equipment. The heat exchanger shells are exposed to seawater and the shells of the heat exchangers are connected by a dry machinery space. The platform payload weight and volume requirements are reduced to a minimum in the submerged configuration. Minimum module water ducting is required and the evaporator is located at the top of the module toward the warm water source; the condenser is placed toward the cold water source, minimizing pumping power requirements.

The estimated ammonia cycle volume is 175,000 ft^3, and 360,000 lb of ammonia are required to fill the system for operation.

The aluminum heat exchangers are 59 ft in diameter and the shells are 75-percent exposed to seawater. The evaporator centerline is at a 150-ft depth and the condenser is at a 275-ft depth. The evaporator contains 50,250 tubes 1.5 in. in diameter, and the condenser 56,000 tubes of the same diameter. The condenser is canted at an angle of approximately 10 deg to take advantage of the improved condensate drainage, previously noted, and to provide a downward component to the seawater exit flow. The heat exchanger structure is the same as the baseline except that the shells are reduced to 2 in. thick and the minimum tube wall is 0.075 in. versus 0.102 in. for the baseline. The pressure within the shell is balanced by the seawater hydrostatic head. Carbon steel pipes are coated externally, and special gaskets and anodes will be used at aluminum/steel joints in seawater to protect the aluminum against corrosion.



1 CONTROL STATION
2 TRANSFORMER
3 ELECTRICAL SWITCHGEAR
4 LEAK DETECTION
5 GENERATOR
6 TURBINE
7 AMMONIA DISTRIBUTION PUMP
8 AMMONIA ANALYZER
9 ANALYZER PUMP
10 AMMONIA CLEANER
11 AMMONIA CLEANER PUMP
12 NONCONDENSIBLE GAS CONDENSER
13 AMMONIA RETURN TANK
14 AMMONIA RETURN PUMP
15 GENERATOR COOLING
16 BILGE PUMP
17 PLATFORM/MODULE ACCESS & UTILITY PIPING TRUNK
18 CONDENSATE PUMP
19 BILGE/BALLAST PUMP
20 PUMP ROOM/MODULE ACCESS TRUNK
21 BUOYANCY TANK
22 BALLAST TANK

Fig. 4-3 25-MW(e) Module General Arrangement

The condensate pumps are housed in a dry chamber below the condenser. The distribution pumps are contained within the chamber between the heat exchangers. The seawater pumps are in their housings which attach to the platform and exit directly into 43-ft-diameter ducts approximately 22 ft long, and then into the heat exchangers which exhaust directly into the ocean.

The nitrogen purge system provides two additional functions in the submerged configuration. The nitrogen systems pressurize the ammonia cycle volume at dockside and maintain the pressure at 25 ± 5 psia until the start of module erection at the platform. Secondly, in the event of a tube leak that is to be repaired, nitrogen is used to pressurize the heat exchanger above the hydrostatic seawater pressure level; the bubbles provide an indication of which tube is leaking.

An access trunk from the platform is provided to the machinery compartment. This access trunk contains all piping, cables, and control lines from the platform to the power module. The trunk diameter is large enough to allow passage of the largest component part to be taken off the module for repair. The platform end of the access trunk has a water-tight cover to prevent seawater ingress whenever the module is not attached to the platform.

The machinery space has three levels. The top level contains the electrical equipment and the control station. The mid-level is a torispherically shaped deck mounting the turbines and generator and power system auxiliary equipment. The lower level, which is also the top of the condenser, mounts the module bilge pumps and the upper termination of the access trunk to the condensate pump room.

4.1.3 Alternative Configurations

The two foregoing power system configurations were selected for detail discussion as providing the comparison of internal versus external modules; however, three platform configurations were examined during the course of the study. The three are presented in Appendix B. These three platforms (a barge 370 ft beam × 700 ft length with components in a shirtsleeve environment, a ship 178 ft beam × 1100 ft long with heat exchangers and seawater pumps immersed in seawater, and a spar platform) were established as being representative of all five generic hull shapes considered as potential candidates for the Demonstration Plant (the five were ship type, submersible type, circular barge, and semisubmersible with two submerged hulls and spare type).

The two power system modules are not considered as having been optimized to size (net power output) or to plant cost of energy. They do represent the best configurations in terms of single versus multiple heat exchangers, heat exchanger material, fouling countermeasures, and reliability.

4.1.4 Power System Module Performance Size

There are indications of reducing the cost of energy as the size of the power module is increased beyond 25 MW(e). Our interpretation of the scope of work did not include investigations in all componentry areas beyond 25 MW(e). The two most significant areas of cost reduction are also the two highest cost areas in the power module. These are the heat exchangers and the seawater system. For the recommended power system module, the heat exchangers and seawater system account for 76 percent of the prototype power module costs and 67 percent of the first production unit.

Therefore it is recommended that the first 3 to 4 weeks of Task 2 (the first part of the period called Requirements and Guidelines) be used to refine the power system module performance size in order to define the most cost effective power module size. This approach would require close coordination with the OTEC Plant Configuration and Integration Study contractors to ensure use of realistic criteria in the power module size evaluation.

4.1.5 References

1. Sheynkman and Linetsky, "Hydrodynamics and Heat Transfer by Film Condensation of Stationary Stream on an Inclined Tube," Heat Transfer-Soviet Research, Vol. 1, No. 3, May 1969

2. Sheynkman, Linetsky, and Brodov, "Experimental Investigation of Hydrodynamics and Heat Transfer During Condensation of Water Vapor on a Slightly Inclined Tube," Fluid Mechanics-Soviet Research, Vol. 3, No. 1, Jan-Feb 1974

4.2 CYCLE PERFORMANCE

This subsection presents a description of the baseline configuration cycle performance and system size to provide the 25-MW(e) net power output. Comparisons are made between the baseline configuration and two optimized configurations – one surface and one submerged.

4.2.1 Cycle Optimization

The Rankine cycle used in the OTEC power system tends to optimize at a regular apportionment of the available temperature difference between the heat source and heat sink. This apportionment is 25 percent between the heat source and the working fluid, 50 percent across the energy extraction unit, and the remaining 25 percent is between the working fluid and the heat sink. The effect of absolute temperature on performance is minimal as evidenced by the results of four computer runs made at a ΔT of 40°F with heat source temperatures of 70°, 75°, 80°, and 85°F. The net power output and the temperature apportionment noted above varied less than 2 percent over the range noted. The performance data presented herein used a subsurface temperature profile generated for the NSF study, and warm water temperature was adjusted to provide the design ΔT of 40°F.

The proprietary optimization program, discussed in Section 3, computes the performance based on a given configuration of warm and cold seawater systems. Therefore, in order to arrive at a configuration which provides 25-MW(e) net with imposed warm and cold seawater pump head requirements of 10 and 15 ft, respectively, it is necessary to request an artificial net power output and correct for the pump power increment. The procedure is to run a series of artificial settings, make the corrections, plot the results, and select an artificial setting which provides the desired net power. Fortuitously, one of the initial series produced a cycle which, when corrected for the incremental pump power requirement, provides a net power output of 25 MW(e). The cycle state points, heat exchanger configuration, seawater and ammonia flow requirements, and turbine gross power output are included in the optimization data presented in Fig. 4-4.

4.2.2. Reference Performance

The procedure used to arrive at the net power output is to convert the turbine power to generator power, and then to correct for the seawater and ammonia pump power requirements; miscellaneous loads for controls, hotel, and internal power losses; and varying loads as required for cleaning. The results, shown in Table A, indicate that the reference configuration provides 24.72 to 25 MW(e) net, depending upon the chlorination power requirement. For comparison, Table A includes the results of optimization computer runs for the surface and submerged configurations. The differences in net power out are attributable to the fact that the sizing routine does not account for the miscellaneous loads and cleaning power requirements or for the reflux portion of the ammonia distribution system. The comparative data can be used to indicate that a 10-percent increase in the ratio of turbine gross/net power is required for the reference configuration relative to the optimized configuration. Resizing the optimized configurations has a negligible effect on this comparison.

OPTIMIZE ON AREA. THE OPTIMIZATION VARIABLES ARE;
T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
THIS RUN IS FOR THE 25-MW POWER SYSTEM DEVELOPMENT WORK
SERIES AT FORTY , 36, 32 -DEGREE DELTA TEMPERATURES. - ALSO 44 DEGREES
EXPONENTT USED IN KAYS EQUATION IS 0.17
SHIPBOARD CASE - STANDARD BASELINE REFLUX IS NOW 0.5
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0	73.73	137.474	.99000	624.66	.4656
01	73.73	137.474		[124.79] LIQUID AT AVERAGE DEMISTER PRESSURE	
1	73.73	137.474	.99000	624.66	.4656
2	73.49	136.898	.99008	624.66	.4637
3	53.84	95.937	.96549	607.98	.3376
4	53.86	95.975	.96548	607.98	.3377
45	53.68	95.652		[102.03] LIQUID AT AVERAGE CONDENSER PRESSURE	
5	53.50	95.330	.00000	101.83	38.8289
6	53.50	99.115	.00000	101.83	38.8289
7	53.53	161.172	.00000	102.16	38.8275
8	53.53	147.774	.00000	102.16	38.8275
89	73.80	137.626		[124.86] LIQUID AT AVERAGE BOILER PRESSURE	
9	73.73	137.477	.99000	624.66	.4656

GENERATOR EFF. = .985
DEMISTER BACKFLOW MASS RATE = -.00 LB/SEC
RANKINE CYCLE EFF. = .031291
CARNOT CYCLE EFF. = .037268
EFFICIENCY RATIO = .839635

***** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP (PSI)
1-2	6.006	160.001	1.286	162.000	.0	.189	.386	.000	.575
3-4	7.053	160.001	.933	24.000	-24.0	.018	.000	-.056	-.039
5-6	1.519	30.000	3.771	30.000	-16.0	.529	.000	-4.314	-3.785
7-8	1.519	30.000	3.771	84.000	40.0	1.481	1.131	10.785	13.397
9-0	6.006	160.001	1.286	3.000	.0	.004	.000	.000	.004

Fig. 4-4 Optimization Program Data for Reference Baseline Cycle (Sheet 1 of 4)

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR	
NUMBER OF TUBES	58000.	52000.	
TUBE PITCH	1.25	1.25	
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500	
HEAT EXCHANGER TUBE THICKNESS (IN)	.1025	.1025	
TUBE LENGTH (FT)	44.544	44.221	
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0	
SEA WATER TEMP. IN (F)	43.875	83.875	
SEA WATER TEMP. OUT (F)	48.768	78.706	
TUBE SEA WATER VELOCITY (FPS)	6.862	7.543	
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	471.869	563.705	
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000	
SEA WATER FLOW RATE (LB/SEC)	233472.90	228915.12	
SEA WATER FLOW RATE (CFS)	3640.26	3587.87	
SEA WATER REYNOLDS NO. THRU TUBES	50723.	72496.	
SEA WATER DENSITY (LBM/FT**3)	64.136	63.802	
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D.	620.	709.	BTU/HR-SQ FT
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS	1086.	1388.	USING KAYS EQ.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000	
WORKING FLUID HEAT TRANSFER COEFFICIENT	3000.	3000.	
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300	
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000103	.000103	
LOG MEAN TEMP. DIFFERENCE (F)	7.077	7.186	
NTU'S	.6915	.7192	
EFFECTIVENESS	.4992	.5129	
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000	
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.646	.297	
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000	
TUBE CORE DIAMETER (FT)	39.51	37.41	
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.4326	.4326	
TOTAL HEAT FLOW ACROSS (BTU/SEC)	1068231.	1102738.	
TOTAL HEAT FLOW ACROSS (MW)	1127.2534	1163.6661	
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	945227.	841303.	TOTAL =1786530.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	33.8	30.0	TOTAL = 63.8

Fig. 4-4 Optimization Program Data for Reference Baseline Cycle (Sheet 2 of 4)

***** 28.000 (MW) NET POWER OUTPUT ***** WORKING FLUID AMMONIA ***** RUN IDENTIFICATION 19 JAN 78 09:43:2

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	910.	897.
FLOW RATE PER PUMP (GPM)	408466.	402587.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	663.6	563.7
PUMP HEAD (FT)	10.346	8.835

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75].	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75].	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.5297
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0250
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.261
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

PUMP DIAMETER (FT)	9.20	9.20
PUMP SPEED (RPM)	139.0235	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	4.231	3.543

Fig. 4-4 Optimization Program Data for Reference Baseline Cycle (Sheet 3 of 4)

***** COLD WATER PIPE VARIABLES	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	4.671	10.510
C.W. PIPE REYNOLDS NO.	20157640.	30236461.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	4	
STATIC HEAD OF COLD WATER (PSF)	135.5	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	191.7	

***** TURBINE VARIABLES

GROSS TURBINE POWER OUTPUT OF 1 TURBINES	37.144	MW
NO. OF TURBINES	1	
FLOW RATE AT TURBINE EXHAUST / TURBINE	6252.01	CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	2110.50	LB/SEC
WET TURBINE EFFICIENCY	.876	
TURBINE WORK	16.678	BTU/LB
ISEN. TURBINE WORK	19.029	BTU/LB
AIRCO AXIAL FLOW		
SPECIFIC SPEED	91.570	RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710	DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420	FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.382	DIMENSIONLESS
DRY EFFICIENCY	.902	
FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION		
TURBINE SPEED	1555.	RPM
TURBINE DIAMETER	10.18	FT
REYNOLDS NUMBER	.394+09	
ROTOR TIP SPEED	828.54	FT/SEC
FLOW COEFFICIENT	.2288	DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.6941	DIMENSIONLESS
VELOCITY RATIO	.849	DIMENSIONLESS
SPOUTING VELOCITY	976.21	FT/SEC

***** CONDENSATE PUMP VARIABLES

NO. OF CONDENSATE PUMPS	2	
PUMP EFF.	.900	
PUMP MOTOR EFF.	.900	
PUMP WORK	.329	BTU/LB
ISEN. PUMP WORK	.296	BTU/LB
MASS FLOW RATE / PUMP	1055.25	LB/SEC
VOLUME FLOW RATE / PUMP	27.1769	CFS
PUMP HEAD	62.057	PSI

Fig. 4-4 Optimization Program Data for Reference Baseline Cycle (Sheet 4 of 4)

Table A
MODULE POWER

Item	Reference Baseline	Optimized	
		Surface	Submerged
Turbine Power	37.14	33.07	33.23
Generator Power [MW(e)]	36.59	32.58	32.73
Warm Water Pumping Power [MW(e)]	4.01	3.10	3.04
Cold Water Pumping Power [MW(e)]	6.13	3.75	3.31
Ammonia Feed Pumping Power [MW(e)]	0.81	0.61	1.27
Ammonia Distribution Pumping Power [MW(e)]	0.20	0.18	0.17
Controls, Hotel, Internal Power Loss, Etc. [MW(e)]	0.4	0.4	0.4
Chlorination Power [MW(e)] [(a)]	0.04 to 0.32	0.04 to 0.32	0.04 to 0.32
Mechanical Cleaning Power [(b)]			
Net Power Out [MW(e)]	25.00 to 24.72	24.50 to 24.22	24.50 to 24.22
Turbine Gross/Net Power	1.49 to 1.50	1.35 to 1.36	1.36 to 1.37

(a) 0.2 mg/l to 1.6 mg/l dosage

(b) System sized with added heat exchanger pressure loss. Cleaning system operated by flow reversal for 15 minutes once a day.

4.3 HEAT EXCHANGER

The conceptual design of the power system module heat exchangers was accomplished using material, cost, and performance trade studies along with supporting activities of manufacturing technologies and techniques to arrive at the final evaporator and condenser configurations. The first trade study, including evaluation of construction concepts and technologies, conducted by Foster Wheeler Energy Corporation, considered various heat exchanger configurations and materials to arrive at relative costs for the material procurement and manufacture of selected configurations. The second trade study was the performance and cost evaluation of heat transfer enhancement techniques. Foster Wheeler assisted in the second trade study to ensure fabricability and assembly of enhanced tube configurations.

4.3.1 Conceptual Design Conditions

The reference design for both heat exchangers (evaporator and condenser) is a horizontal shell and tube type design using state-of-the-art heat transfer techniques. The evaporator uses the irrigated thin-film concept for distributing the ammonia on the shell side of the unit. Although the reference design for the relative cost trade study incorporates a smooth heat transfer tube, several shell side enhancement techniques were evaluated for impact to the assembly process.

Conceptual Design Guidelines. The thermal and functional characteristics for the heat exchanger configurations based on the overall requirements of the OTEC power system are summarized in Table 4-1. The three sizes of heat exchangers correspond to a net electrical output of 5, 25, or 50 MW(e). The heat load is proportional to the electrical rating.

Three material combinations were specified. The first was an all-aluminum heat exchanger with alloy 5052 tubes and alloy 5083 material for the shell, tubesheets, support plates, and other parts. The second was commercially pure titanium tubing with a titanium cladding on carbon steel for the outer tubesheet and carbon steel for the

Table 4-1

CONCEPTUAL DESIGN GUIDELINES

HEAT EXCHANGER SIZES: 5, 25, 50 MW(e)

HEAT EXCHANGER MATERIAL

Tube	Tubesheets	Shell
5052 aluminum	5083 aluminum	5083 aluminum
CP titanium	Titanium clad steel	Carbon steel
AL-6X stainless	Stainless	Carbon steel

DESIGN REQUIREMENT	Evaporator	Condenser
Design Pressure – Shell		
Shell (Ammonia) Side (psia) (Surface)	265	265
Shell (Ammonia) Side (psia) (Immersed)	183	183
Differential Pressure, Burst (psid) (Submerged or Subsurface)	129	50
Differential Pressure, Collapse (psid) (Submerged or Subsurface)	65	143
Heat Load [MW(t)/MW(e)]	44.32	43.04
Seawater		
Flowrate [cfs/MW(e)]	260	252
Inlet Temperature (^{o}F)	80	40
Exhaust Temperature (^{o}F)	77.5	42.5
Ammonia		
Flowrate [lb/sec-MW(e)]	80	80
Saturation Temperature (^{o}F)	70	50
Tubing		
Aluminum, Min. Wall (in.)	0.065	0.065
Titanium and Stainless, Min. Wall (in.)	0.028	0.028

balance of the material. The third was type AL-6X stainless steel tubes with a solid stainless steel outer tubesheet and carbon steel for the balance of the material.

Three different design pressures were specified. The 265 psia corresponds to an uninsulated ammonia tank on the deck of a ship or platform which is heated to 115°F and is designated as a surface heat exchanger. If the heat exchanger is immersed in water but is still near the water surface, the hydrostatic head is low and the 183 psia corresponds to a maximum ammonia temperature of 85°F with a 10 percent factor on pressure. When the heat exchanger is submerged 100 ft or more below the surface, the hydrostatic head becomes significant. The two differential pressures in the chart correspond to the difference between water and ammonia pressure when the ammonia is (1) at 183 psia (85°F) and (2) at 15 psia. The condenser and evaporator values are different because the condenser is at a lower depth than the evaporator.

The tubing minimum wall thickness was based upon pitting corrosion requirements for the aluminum and manufacturing handling requirements for the titanium and stainless steel.

<u>Mechanical Design.</u> The conceptual design for the evaporator is shown in Fig. 4-5. The condenser design was similar to the evaporator except the ammonia liquid distribution system was omitted and the two hemispherical outlet nozzles were replaced by an inlet cone and an ammonia sump. The reference design for the heat exchangers has a double tubesheet welded to a rectangular cross-section transition ring. The double tubesheet provides adequate strength for minimum weight. The double tubesheet is stabilized by hollow stays located on approximate 24-in. centers. The array and size of the stays are such that ammonia distribution or seawater tubes can be inserted and secured. The stay wall thickness will be adequate for the structural loads. The shell is welded to the transition ring. The tube support plates are reinforced with a small ring which is supported from the shell with stays. The tubes are welded to the outer tubesheet; rolled to the inner tubesheet; and have a 1/32-in. clearance at the support plate holes. The support plate spacing is a function of the tube diameter.

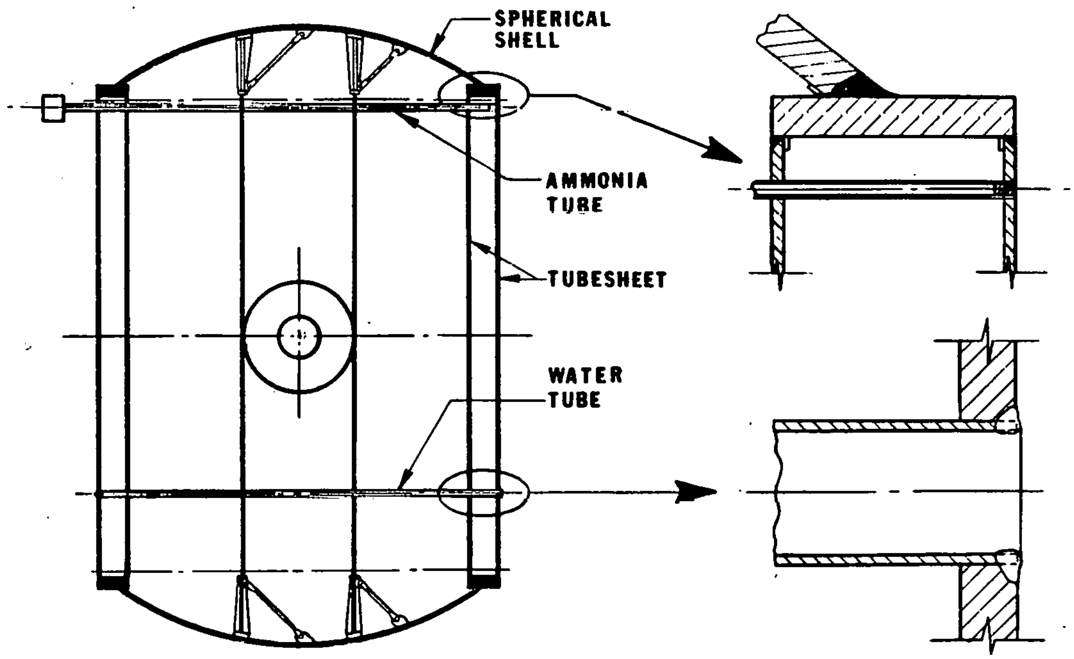


Fig. 4-5 Evaporator Cross-Section

With the double tubesheet design, the space between the two tubesheets can be used (1) for an ammonia liquid feed preheater at one or both ends of the evaporator, or (2) as a nitrogen pressurized seal to prevent ammonia outleakage or seawater inleakage. An alternative approach for preheating the liquid ammonia is to use an exterior manifold for the ammonia tubes. The double tubesheet scheme requires development during preliminary design to ensure the tube-to-tubesheet joint on the inner tubesheet provides a good seal and proper manifold tube stubs are provided for liquid distribution.

Thermal/Hydraulic Design. A family of heat exchangers was designed for the cost trade study. The flows and temperatures were as specified in Table 4-1. The tube-side heat transfer coefficient was calculated from the geometry and the water properties. The shell-side coefficient is assumed to be 1,000 Btu/hr-ft^2-^{o}F based on experimental work performed in an LMSC Independent Development Project. A tube-pitch-to-diameter ratio of 1.2 was selected as the minimum acceptable value. A single tube bundle diameter was selected for each MW size of heat exchangers because changing tube diameter (which varies the number of tubes) has a relatively small effect on water velocity. The surface area was calculated, and the active tube bundle length selected accordingly. This does not necessarily produce the optimum size heat exchanger for each design condition, but it simplified evaluation of relative costs. When the approximate size, material, and design conditions are selected, the most cost effective design can be found as part of the preliminary design.

A sensitivity analysis was performed for the 25-MW(e) evaporator with 1.5-in.-diameter aluminum tubes. For a 95-percent confidence level in performance, the surface margin should be 7.1 percent, which should be a typical allowance for all the OTEC tubed heat exchanger designs. This margin was not included in the cost trade studies because the costs in this trade study are on a relative basis.

Structural Analysis. A simplified finite element structural analysis using shell elements showed adequate margin for the configuration used in the cost trade study. More detailed analysis will be required for the design eventually selected using a finite element analysis with continuum elements. The cheaper shell element type of model can accurately predict wall thickness except at the junctions of two components. Therefore the tubesheet, ring, and shell thickness can be calculated; but the tubesheet-to-ring junction stresses cannot be accurately assessed at this time. Prior experience indicates that a satisfactory design can be developed during the preliminary design task. The model analyzed is shown in Fig. H-1 and Table H-1 of Appendix H. The two-dimensional model has about 150 elements which include the shell, the transition ring, two tubesheets, and the tubes. Thirteen cases were run with different materials and configurations for the 25-MW(e) size heat exchangers and 183 psia ammonia pressure. The results are tabulated in Table H-2 of Appendix H.

From these calculations it was concluded that a single 5- to 6-in.-thick aluminum tubesheet would be acceptable, provided that the shell junction is moved in line with the tubesheet. This approach would reduce the transition ring size and eliminate the stays between the tubesheets plus the 4 ft of inactive tube length between the two sets of tubesheets. However, the tubesheet material cost and drilling time would be increased, and the problem of welding tubesheet segments together after drilling would be more severe. Therefore, the single aluminum tubesheet is not recommended.

The double aluminum tubesheets must be 1-1/2 in. thick for the 183 psia pressure. Again, with the lower pressure of the submerged design, the thickness could be reduced. Since all the pressure loading is taken by the outer tubesheet, the shell must also be aligned with the outer tubesheet rather than having the offset which is shown in Fig. 4-5. The area between the two tubesheets can be used as an ammonia plenum in the evaporator or could contain a third fluid as a buffer between the water and ammonia. However, if this space is not needed, a large section of the inner tubesheet can be eliminated. Only the outer few rows of tubes are required to stiffen the outer tubesheet, and the inner tubesheet could be an annulus. This would save material and drilling time and would make the tubes active over their full length.

The double stainless steel tubesheet with 1-in. thickness is feasible based on the properties of Nitronic 50 type stainless, although the center of the inner tubesheet can be removed if desired. A single flat 2-1/2-in.-thick stainless tubesheet or a 1-3/4-in. -thick dished tubesheet are also acceptable. The extra cost of the dished tubesheet does not make this design cost effective. Similar results are expected for the titanium clad tubesheet.

4.3.2 Cost Trade Study

The cost trade study evaluated the major design variables which were considered for the heat exchangers. The trade study was performed in sufficient detail to establish the relative costs between the various concepts in order to select a candidate design for the preliminary design task.

<u>Design Variables</u>. Six major design variables were selected as listed below. In addition, several other variables were also evaluated on a more limited basis. The major variables were:

a. The type of unit which included the evaporator and condenser. The basic design was similar.
b. The location of the unit, which could be an uninsulated heat exchanger on the deck of a ship or platform, an immersed heat exchanger in the hold of a ship where low pressure seawater covers the unit, or a subsurface heat

exchanger where the seawater pressure offsets the ammonia pressure to a significant effect. The significance of the location is the pressure differential between the ammonia and seawater, which determines the tube and shell wall thicknesses.

c. Three sizes of heat exchangers were studied assuming a plant efficiency to convert the net electrical output to a required thermal rating as follows:

Net Electrical [MW(e)]	5	25	50
Evaporator [MW(t)]	221	1,108	2,216
Condenser [MW(t)]	215	1,076	2,152

d. The shape of the shell was either cylindrical with a transition cone at each end or spherical.

e. The three material combinations were all aluminum or stainless steel tubes and tubesheets in a carbon steel shell or titanium tubes with a titanium clad tubesheet in a carbon steel shell. Two types of carbon steel were used for the shell to evaluate a higher cost, high-strength material.

f. The five tube sizes included 1, 1-1/2, 2, 2-1/2, and 3 in. outer diameter tubes. The tube wall thicknesses varied depending on the tubing material and the pressure requirements.

The combinations of variables for the cost trade study are shown in Appendix H, Tables H-3 through H-9, which also summarize the cost for most cases.

In order to visualize the relative sizes of the heat exchangers, the first column from Tables H-3, H-4, and H-6 in Appendix H have been drawn to scale in Fig. 4-6. This figure also illustrates how the shell diameter varies which causes the spherical shell wall thickness to vary.

Other variables evaluated on a limited basis were the tube-pitch-to-diameter ratio and the hourly labor cost. Lockheed computer optimized configurations for differential temperatures of 32°, 36°, 40°, and 44°F were also costed.

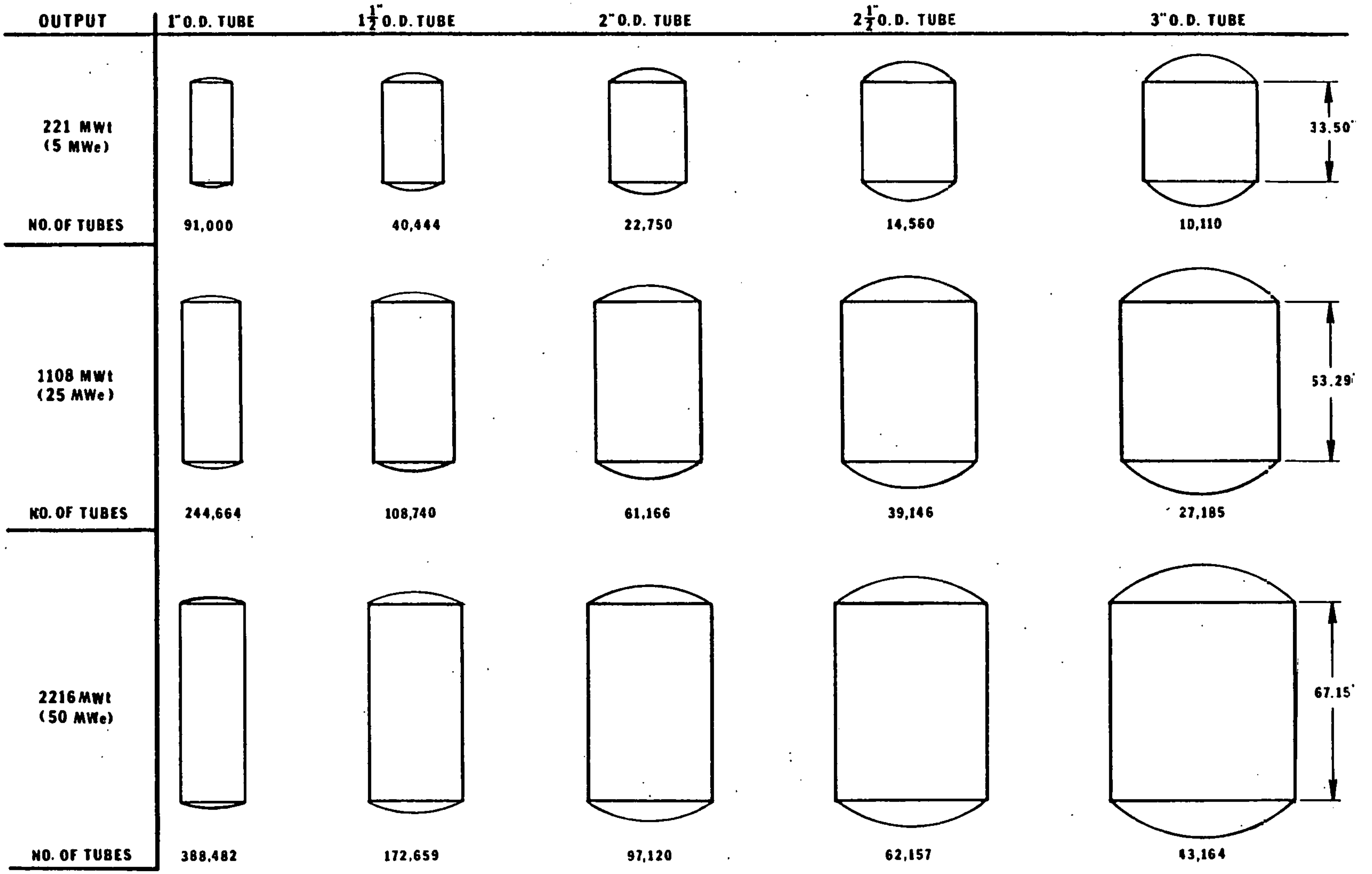


Fig. 4-6 Aluminum Spherical Surface Evaporator

Assumptions. The design conditions were provided by Lockheed, but Foster Wheeler had to make certain assumptions for the heat exchanger design and cost factors. Some of the design factors changed slightly as a result of the structural calculations discussed above, but the effect on the cost numbers is minimal.

The design was based on the design evaluated in Lockheed's OTEC Heat Exchanger Design and Producibility Study, LMSC-D507632. The tubesheets were the double tubesheet design with pipe spacer stays around the tubes on 24-in. centers. The tubesheets were each 1-in. thick for all materials. A solid stainless steel outer tubesheet and a carbon steel inner tubesheet were used with the stainless steel tubes. A 1/4-in. -thick titanium clad on carbon steel plate for the outer tubesheet and carbon steel for the inner tubesheet were used with the titanium tubes. The transition ring between the shell and tubesheets was 4 in. thick and 24 in. long and was made from rolled bar.

The tubes were welded to the outer tubesheet by an automatic tungsten inert gas (TIG) weld. The tubes were mechanically rolled into both tubesheets. All welding for the shell, tubesheets, support plates, transition ring, and their interfaces were assumed to be conventional techniques which were gas metal arc welding for the aluminum and submerged arc welding for the steel.

No external attachment structures were included because the interface with the rest of of the module was not defined. The estimate did include all weld wire and the mist extractor internals.

The tube walls were calculated based on pressure conditions and pitting allowance. Tube weights and costs were based on the minimum calculated wall thickness and did not include a manufacturing tolerance. The aluminum tubing was drawn and the stainless and titanium tubing were rolled and welded.

The costs were based on a mature industry, which means that a standard heat exchanger is being produced in sufficient quantity to minimize the amount of labor. However, it was also based on state-of-the-art fabrication techniques and did not take credit for more exotic methods which might reduce costs in very large quantity production.

The estimates included only material that would go into the heat exchangers. There was a material allowance on the plate for shop machining. The weights that are shown are required material weights and the finished heat exchanger would be somewhat lighter due to tube holes and other machining. The labor was for direct shop labor only but did include an allowance for the required inspections. Labor and overhead were priced at $15 per hour which represents the 1977 costs at Foster Wheeler, Panama City, facility if the shop is fully utilized. The costs did not include any allowance for tooling or engineering. The material costs were obtained from plate or tube vendors based on 1977 prices for large quantity orders and are summarized in Appendix H, Tables H-13 and H-14.

Results. The results of the cost trade study are summarized on Tables H-3 through H-9, Appendix H, as the cost per MW(t) for each configuration. The conversion from MW(t) to MW(e) is shown in the upper right corner of each sheet.

In order to better understand the cost factors, the dollar costs have been plotted in Fig. 4-7 for the aluminum evaporator. The labor costs are essentially the same for

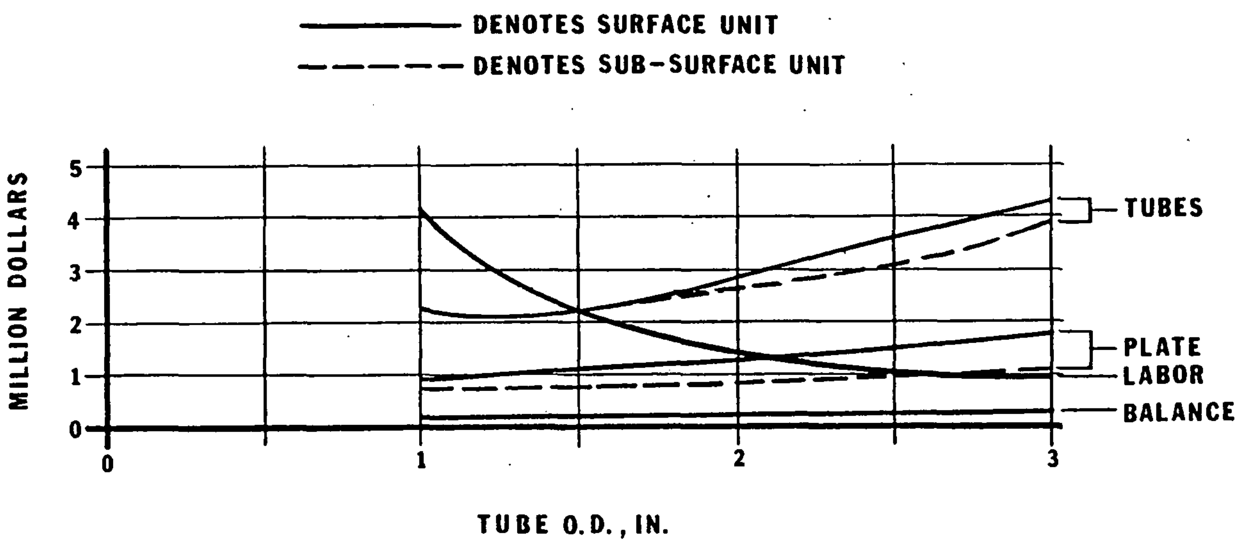


Fig. 4-7 25-MW(e) Evaporator — Aluminum, Spherical the immersed and submerged units and are shown as a single line. The labor is primarily affected by the number of tubes in the heat exchanger and is highest for the large number of small tubes. The tube costs for the submerged or subsurface evaporator are lower because the tube wall can be reduced as the water pressure increases and balances part of the ammonia pressure. The tube costs are similar for the 1- and 1-1/2-in. tubes because the wall thickness is set by a 0.065-in. wall minimum requirement to ensure adequate life with pitting, rather than by pressure requirements. The plate costs are also less for the subsurface evaporator due to a thinner shell. The "balance" includes the weld wire and the demister elements.

For the stainless steel and the titanium evaporators, the labor costs are slightly higher and the plate costs are lower or about the same as for the aluminum unit. However, the tube costs are significantly higher and become the controlling cost. This is shown in Appendix H, Figs. H-2 and H-3.

When the costs are added for each tube size and divided by the thermal capacity of the evaporator, Fig. 4-8 results. For the aluminum evaporator, about a 1-1/2-in. tube is optimum for the surface unit and about 2 in. for the subsurface unit. With the lower material cost for the subsurface evaporator, the minimum cost shifts toward a larger tube which has lower labor costs. The stainless steel and titanium evaporators optimize at or near the point where the wall thickness reaches the minimum value of 0.028 in. The higher strength HY-80 material was also considered for the subsurface units, but the shell design is dependent on the modulus of elasticity rather than the tensile strength and the A516 material is more economical.

The effect of evaporator size is shown in Appendix H, Fig. H-4. There is an economy of scale for the larger sizes.

The subsurface or submerged cylindrical shells require stiffening rings around the outside to resist the external collapse pressures when the heat exchangers are not operating with ammonia. While there is no significant difference between spherical and cylindrical shell costs, the spherical shell lends itself more readily to structural attachments to intersecting cylindrical structures and provides more internal volume between the shell and tube bundle.

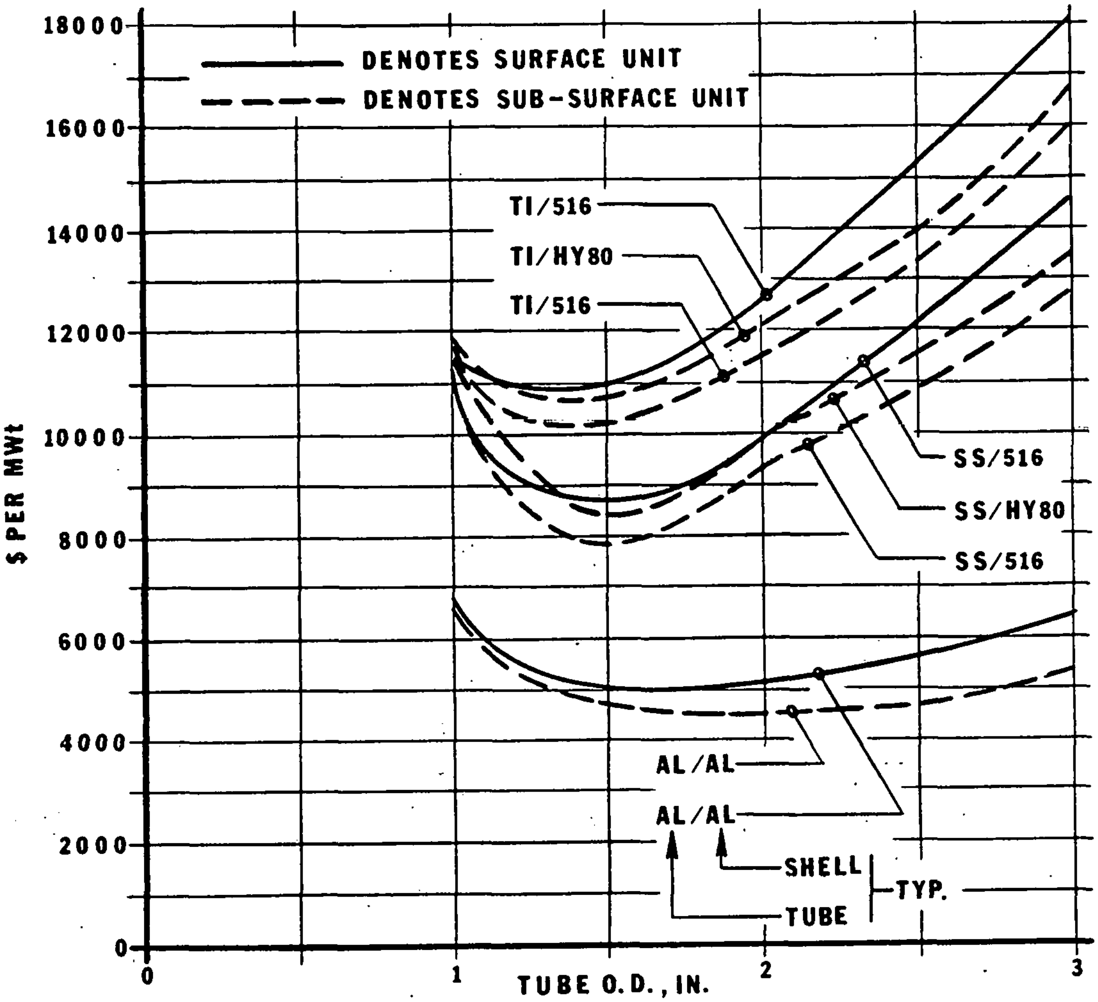


Fig. 4-8 25-MW(e) Evaporator – Al/SS/Ti, Spherical

The surface condensers are more expensive than the evaporators because of more required surface area. The submerged condensers are at a lower elevation than the evaporators and have correspondingly thinner shells and tubes, which again shifts the minimum cost toward the larger tube diameter.

The weights of the aluminum evaporators are plotted against tube diameter in Fig. 4-9. In some cases where the cost is a weak function of tube diameter, it may be advantageous to select the smaller tube diameter to save space and weight on the platform.

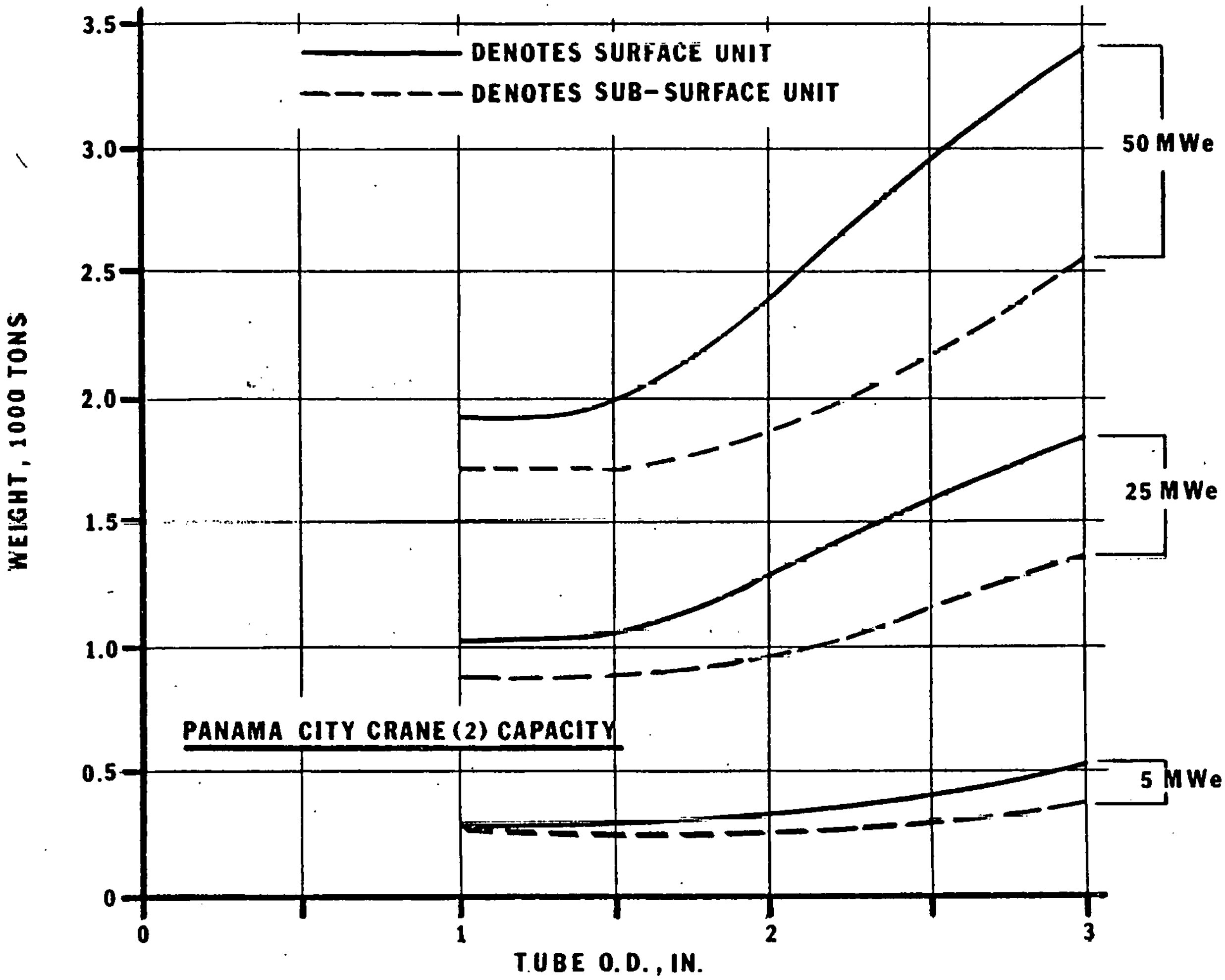


Fig. 4-9 Evaporator — Aluminum, Spherical

The tube-pitch-to-diameter ratio was 1.20 for the above studies. For the larger size evaporator, the vapor velocity at the outer row of tubes exceeds predicted values, which would cause carryover of moisture. It may be necessary to use a larger pitch-to-diameter ratio, which involves larger diameter tubesheets and support plates, a larger transition ring, and a larger shell diameter and thickness. For a 25 MW(e), aluminum, spherical evaporator with 1.5-in. tubes, the costs increase 1.3 percent for a 1.25 P/D and 3.4 percent for a 1.30 P/D over that of the 1.20 P/D.

Conclusions. The results of the cost trade study can be summarized by discussing the relative costs for the different variables. The relative costs are somewhat sensitive to the other variables, but the trends are consistent.

The cost per MW is least for the 50-MW(e) heat exchangers, increases 10 to 15 percent for the 25-MW(e) size, and increases 77 percent for the 5-MW(e) size. Thus, based on material and labor only, the largest units are more cost effective. If the additional tooling and facilities are considered, the 25-MW(e) (or slightly smaller) size may be more cost effective for a prototype. Costs per MW begin to increase at a more significant rate below 10 to 15 MW(e).

The submerged heat exchangers are the least expensive followed by a 5- to 6-percent cost increase for the immersed units and a 25-percent increase for the uninsulated surface units.

On a pound per MW basis, the 50-MW(e) heat exchangers are the lightest, and 25-MW(e) are 11-percent heavier, and the 5-MW(e) are 65- to 75-percent heavier. The aluminum heat exchangers are the lightest, with the titanium about half again as heavy, and the stainless steel about twice as heavy for the same duty.

The optimum tube diameter is sensitive to the design conditions and the material selection. The optimum sizes range from under 1-1/2 in. to 2-1/2 in. diameter. Smaller tubes always result in lighter heat exchangers if all other conditions are the same. In most cases, the smaller diameter tube designs were penalized on material cost by the restriction of minimum wall thickness for the tubing. Since the surface area is relatively constant among designs, tubing cost is approximately proportional to diameter (and thickness) until the minimum wall requirement is reached.

The high tensile strength carbon steel was not cost effective and is not recommended.

4.3.3 Construction Concepts

The construction concepts which are applicable to the OTEC heat exchangers will depend on the heat exchanger material and the module size. Since the selection of these variables was not made until the end of Task 1, alternative construction concepts were evaluated, but final selection of concepts must be made during Task 2 when the preliminary design is developed.

A manufacturing and inspection plan for the aluminum evaporator was prepared as a basis for the cost trade study. The fabrication methods used in this plan are conventional methods currently in use in the Foster Wheeler shops. Alternative methods that may require development and further evaluation are summarized in the cost analysis comparing various methods for performing the same operations as shown in Appendix H, Table H-17. These comparisons are for representative thicknesses of various materials. The main purpose of the analysis was to determine (within the confines of a particular operation and material) which would be the most economical choice. The figures should not be used for estimating since some of the costs are dependent on other related operations. The limitations in the comments column should also be considered. The production and development investment costs are a "ball-park" figure to indicate relative levels based on our best judgment.

When the alternative fabrication methods are compared to the methods used for the cost trade studies, there are some areas where further cost improvements may be possible. For example, if the electron beam welding can be used for the tubesheet and support plate seams, the cost can be reduced provided that the equipment investment can be justified.

The cost estimate for drilling holes was based on the single-spindle Lahr drill which is limited to 7-ft material widths. The Moline multi-spindle drill is about the same cost per hole, but it can handle larger sizes of plate material. The Lahr drill can drill a stacked set of plates that are 12 in. thick while the Moline is restricted to 4 in. thick. The individual spindles of the Moline are power limited and probably can drill only up to 1-1/2 or possibly 2-in.-diameter holes. Therefore, the eventual selection of the optimum method for drilling the tube holes is dependent on the design of the heat exchanger and the quantity to be produced. The table also shows the cost of drilling holes near the weld seams in the tubesheet and tube support plates after the welding operation. This cost was not included in the cost trade study. The number of these individually drilled holes is a function of the type of welding process that is used for the seams. Again, this illustrates that the cost factors are interrelated and are difficult to evaluate on a separate basis.

The reference tube-to-tubesheet joining procedure is weld and roll at outer tubesheet and roll at inner tubesheet. The GTAW welding is the least expensive process which has a proven record for heat exchanger application. Either the roller expansion or kinetic bonding can be combined with the GTAW welding for the same cost. The magneforming procedure has the potential for being less expensive (for the aluminum material), but additional development is required to demonstrate the reliability of the joint.

The initial OTEC heat exchangers would be built in Foster Wheeler's Panama City, Florida, facility. The main bay in this plant is 800 ft long by 100 ft wide with traveling overhead cranes of 30-, 250-, and 300-ton capacity. The main bay has a height of 50 ft under the hook. The cranes service a barge slip which is 150 ft by 70 ft with 12 ft of water at mean low tide. The adjacent Port of Panama City has a 32-ft-deep channel and can accommodate large oceangoing barges in excess of 200 ft by 400 ft.

The cost trade study has indicated that the largest size heat exchangers are the most economic. The limiting factor at Panama City is the lift capacity of 550 tons with the two larger cranes combined. This would correspond to about a 12.5-MW(e) aluminum heat exchanger, a 6-MW(e) titanium unit, or a 5-MW(e) stainless steel unit.

One method of making larger heat exchangers in the existing facilities would be to tube the units in the shipyard that is assembling the modules. The tubing represents about 50 percent of the aluminum weight, 40 percent of the stainless weight, and 30 percent of the titanium unit weight. The aluminum shell could then represent a 25-MW(e) heat exchanger and would have to be under 50 ft in diameter. However, the shipyard labor would probably be higher in both manhours and rate per hour. Also, the shipyard would need to furnish a housing to protect the tubesheet and tubes during the tube insertion and welding operations, as well as appropriate staging. Particularly with the aluminum material, the moisture must be carefully controlled to ensure sound welds.

Larger units can also be built at Panama City by building the shell and tubesheets in the shop and performing all the assembly work in an inflatable or other temporary building near the Port facilities. The final unit could be loaded on a barge by a sea-going crane. If a sufficient OTEC market develops, a larger plant can be constructed on Foster Wheeler land adjacent to the existing Panama City facility.

4.3.4 Construction Technology

The two most critical areas in the construction of the heat exchangers appear to be the joining of the tubesheet and support plate segments and the tube-to-tubesheet joint. Limited development programs were performed in these areas to provide guidelines for the preliminary design task.

Tube-to-Tubesheet Weld Development. Weld development tests were performed for the aluminum and titanium material. Since Foster Wheeler has produced many stainless steel heat exchangers, stainless steel development was not necessary at this time.

A 5083 aluminum tubesheet was drilled with fourteen 1.008-in.-diameter holes. The 6-in.-long type 5052 tubes were 1 in. OD by 0.065 in. wall. The tubes were cleaned to remove the oxide and installed in the tubesheet with about 1/16 in. extending beyond the tubesheet. The tube ends were flared mechanically, and the tubes welded with an automatic TIG torch without addition of filler material. Some of the tubes were expanded mechanically into the tubesheet, and some were left as welded. Tensile pull tests were made plus radiographic examination, dye penetrant inspection, sectioning, etching, and examination at 20× magnification.

Similar tests were made with 1 in. OD by 0.049 and 0.065 wall titanium tubes. A 1/8-in.-thick titanium sheet was bolted to a carbon steel plate for the tubesheet mockup, and the weld was made to the titanium plate.

In general, both the aluminum and titanium welds were acceptable based on normal ASME Code requirements. Test data are shown in Appendix H, Table H-16. Either material combination appears to be acceptable within the limits of the mechanical properties. There were no cracks in any welds before or after the rolling process.

Tube-to-Tubesheet Adhesive Tests. An alternative tube-to-tubesheet joining technique is adhesive bonding. A survey was conducted of adhesive manufacturers to determine what adhesives would be compatible with water and ammonia. Several candidates were recommended for water applications, but no ammonia data could be found. Seven candidate adhesives were selected for screening tests. These materials included Scotchweld 2214, Scotchweld 1838, Bondmaster M773, Bondmaster M777, Locktite 40, Permalok HM 128, and Eccobond 104.

Lap shear samples were made from 1/16-in.-thick aluminum sheet with about 1-sq-in. overlap area. Triplicate samples were made with each adhesive and pulled to establish a baseline. Shear stresses ranged from 270 to 1,650 psi. Another six samples of each were immersed in ammonia. After 30 days, all the specimens were inspected. The majority of the adhesives had suffered severe degradation from the ammonia. The two Scotchweld 2214 specimens that were intact broke with half the original load, and the Bondmaster M773 broke with one-third the load. It was concluded that none of these adhesives was suitable for ammonia, and testing was terminated.

Electron Beam Welding. Sections of the tubesheets and tube support plates are drilled first, and then the sections are welded together to form a larger perforated sheet. The joining method must not distort either the plate or the holes. It is desirable to drill the holes as close to the seam as possible to keep the tube bundle as compact as possible. Holes drilled near the seam after welding would be a factor of 100 more expensive compared to the predrilled holes.

One alternative to the gas or submerged arc welding, which is proposed for the shell, would be electron beam welding. Procedures have been developed by Leybold Heraeus to electron beam weld up to 1- or 1-1/4-in.-thick aluminum without using a vacuum. A 2-in.-thick plate could be welded from both faces.

Four 1-in.-thick 5083 aluminum plates were prepared for electron beam welding tests. Each plate had either 1-1/2-, 2-, 2-1/2-, or 3-in.-diameter holes drilled on a triangular pitch with a pitch-to-diameter ratio of 1.2 as shown on Fig. H-5 in Appendix H. The edges of the holes were 1 in. from one edge and 1/2 in. from the second edge. Selected hole diameters were measured, and welded blocks were used to establish distances over the seams.

After the welding parameters were established, the four plates were welded together to represent the intersecting weld on a tubesheet. The hole diameters were remeasured, and the blocks were measured to determine plate shrinkage. The data are tabulated in Appendix H, Table H-15.

4.3.5 Heat Transfer Enhancement

Of the four constitutents which contribute to the determination of an overall heat transfer coefficient, two are amenable to enhancement techniques. The fouling resistance and wall resistance are not, since the former is dependent upon the efficacy of the countermeasure system and the latter is dependent upon tube material and wall thickness.

The seawater and ammonia film resistances, which are the major factors in determining overall resistance, can be reduced through application of heat transfer enhancement techniques. Heat transfer enhancement reduces the required heat transfer area, thereby reducing heat exchanger size. However, enhancement adds to the cost of materials and labor, and increases producibility problems. A trade study was made to determine the desirability of using enhancement techniques. Figure 4-10 provides the relative influences of seawater and ammonia heat transfer coefficients on the overall heat transfer coefficient.

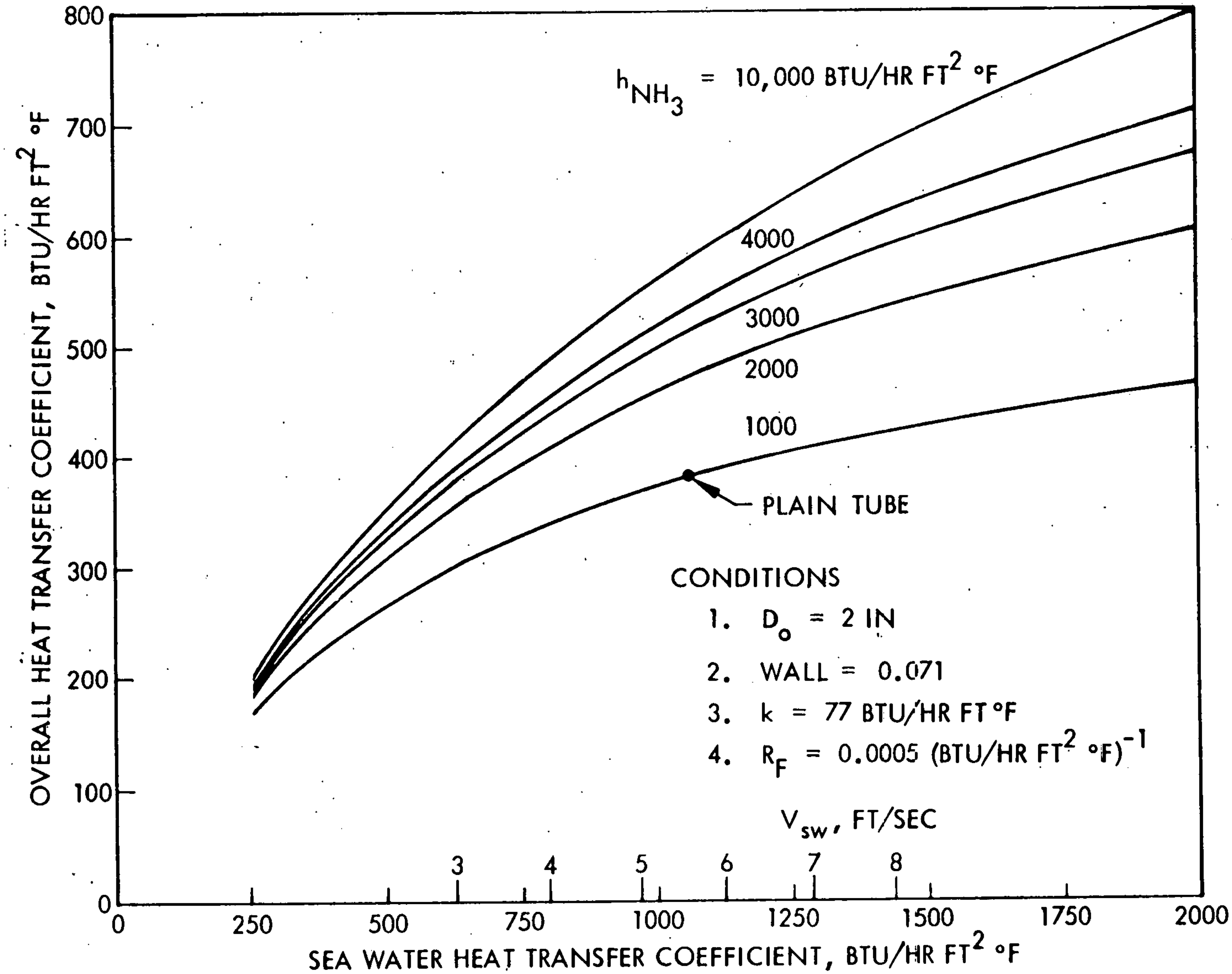


Fig. 4-10 Variation of Overall Heat Transfer Coefficient With Tube and Shell-Side Heat Transfer Coefficient

Potential enhancement techniques include turbulence promoters and extended surfaces on the seawater side, and enhanced surfaces on the working fluid side. The first was ruled out for several reasons. The turbulence promotors increased seawater pressure losses and parasitic power in proportion to their effectiveness, thereby partially cancelling any benefit. Surface contours which form deep crevices or sharp corners invite biofouling and corrosion and makes effectiveness of many fouling countermeasures difficult. The remaining seawater side candidates are the longitudinally, ridged type of surface or spirally wound tube and the ammonia side candidates are either the porous surface or circumferentially grooved tube. A matrix of the tube configurations considered is shown in Table 4-2 along with shell side heat transfer coefficients.

Table 4-2

MATRIX OF TUBE/SHELL-SIDE ENHANCEMENT CONDITIONS

		SHELL SIDE					
		EVAPORATOR			CONDENSER		
		PLAIN	GROOVED	LINDE	PLAIN	FLUTED	CORRUGATED
TUBE SIDE	PLAIN	H	H	H	H	V	H
	FLUTED		H	H		V	
	CORRUGATED			H			H

H = HORIZONTAL TUBING

V = VERTICAL TUBING

SHELL SIDE HEAT TRANSFER COEF (Btu/Hr Ft^2 °F)

EVAPORATOR		CONDENSER	
PLAIN	1000	PLAIN	700
GROOVED	3000	FLUTED	7000
LINDE	4000	CORRUGATED	1500

Typical results for plain tube evaporation heat transfer coefficients for a wide range of conditions are shown in Fig. 4-11. The column tube number refers to a flow rate which would be just sufficient to completely supply a vertical column of tubes of the number indicated. The experimental data indicate that over a large number of tubes an average heat transfer coefficient of approximately 1,000 Btu/hr-ft^2-^{o}F may be assumed for the gravity flow dominated case. The plain tube was used as the basis for comparing enhancement.

Tests have produced data on heat transfer coefficients for evaporation of ammonia from horizontal, circumferentially grooved tubes. Results shown in Fig. 4-11 provide a general comparison with the plain tube data obtained previously. Tube shell side enhancement techniques range from a 60-deg included angle thread with a pitch of eight threads per inch on a 2-in. aluminum tube to 30 grooves per inch with a depth of 0.025 in. The results indicate an increase in apparent heat transfer coefficient on the order of three over the plain tube heat transfer coefficient, and are comparable to results for the fluted vertical tube.

The influence of the evaporation heat transfer coefficient on the overall heat transfer coefficient for two conditions of biofouling is shown in Fig. 4-12 as a function of seawater heat transfer coefficient. Heat exchanger optimization studies generally indicate an optimum seawater velocity on the order of 5.5 to 6 ft/sec. This, along with the experimentally determined evaporation heat transfer coefficient of 1,000 Btu/hr-ft^2-^{o}F, establishes a plain tube baseline overall heat transfer coefficient of approximately 367 Btu/hr-ft^2-^{o}F for the maximum allowable biofouling condition. The effect of enhanced shell and shell-and-tube-side heat transfer on the overall heat transfer coefficient is indicated by Fig. 4-12. Shell-side enhancement, in general, produces an improvement in the overall heat transfer coefficient at the expense of greater tube cost per unit surface area. For seawater side enhancement, an increase in friction factor accompanies the increase in heat transfer coefficient along with greater difficulty in incorporating most methods of biofouling control. Data available from tests on extended surface enhancement for liquids flowing in tubes indicate a one-to-one increase in friction factor with heat transfer coefficient. This has the deleterious effect of

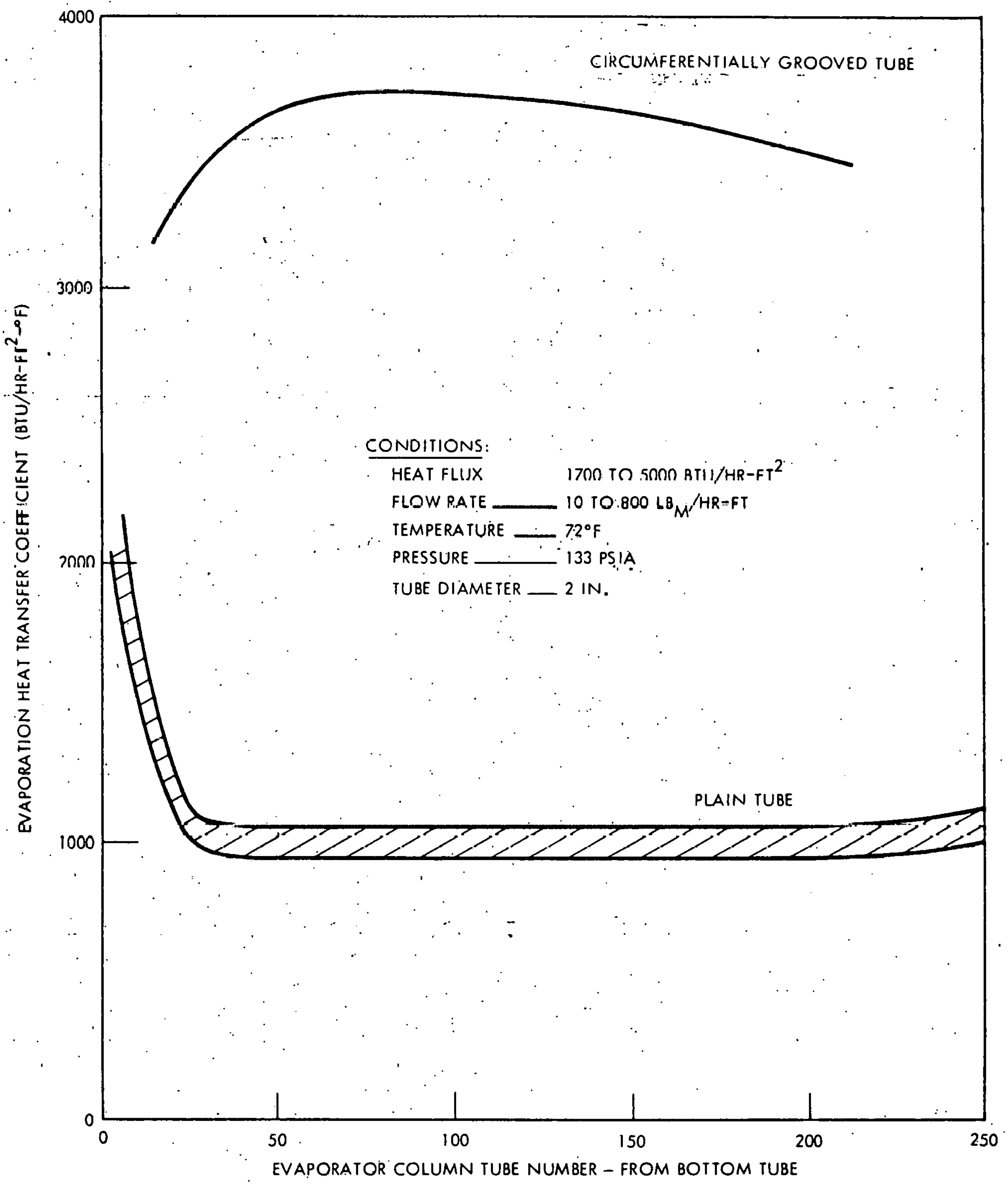


Fig. 4-11 Experimental Local Ammonia Evaporation Heat Transfer Coefficient as a Function of Column Tube Number for Horizontal Tubes

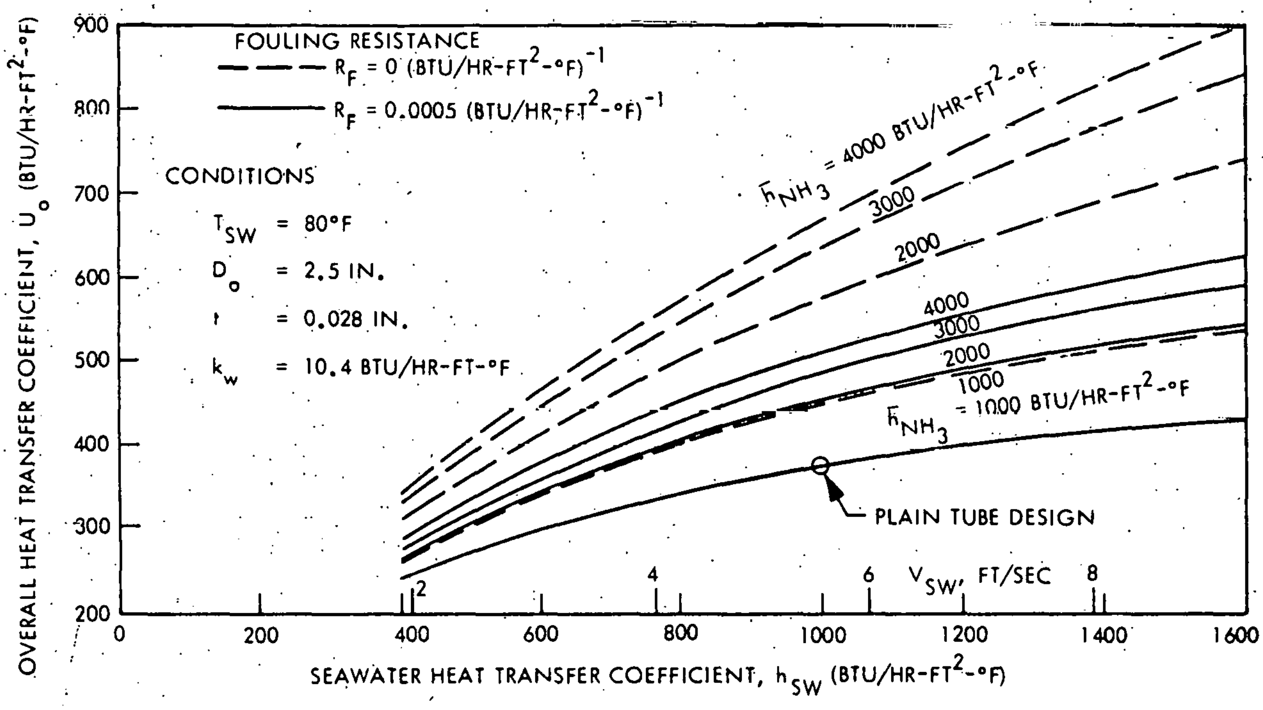


Fig. 4-12 Overall Heat Transfer Coefficient vs. Seawater Heat Transfer Coefficient With Ammonia Shell Side Heat Transfer Coefficient as a Parameter

increasing the pumping power in almost the same ratio as the increase in friction factor, less any overall reduction in tube length due to an improvement in heat exchanger performance. The reduction in tube length is offset somewhat by the need to produce more power to supply the increased seawater pumping power requirement.

The influence of shell-side enhancement tube cost is shown in Fig. 4-13 which shows the relationship of the ratio of enhanced tube heat exchanger to unenhanced tube heat exchanger costs to enhanced to unenhanced tube cost ratio. The figure indicates, for the conditions specified, that if the shell side heat transfer coefficient were doubled and the cost of the enhanced tube were less than 50 percent over an unenhanced tube, then a heat exchanger cost reduction could be realized. Figure 4-14 indicates the same cost type reduction for tube-side enhancement and includes $\eta\,[(h'/h)/(f'/f)]$ — the ratio of enhanced coefficient divided by unenhanced coefficient divided by the ratio of enhanced frictional factor by unenhanced frictional factor).

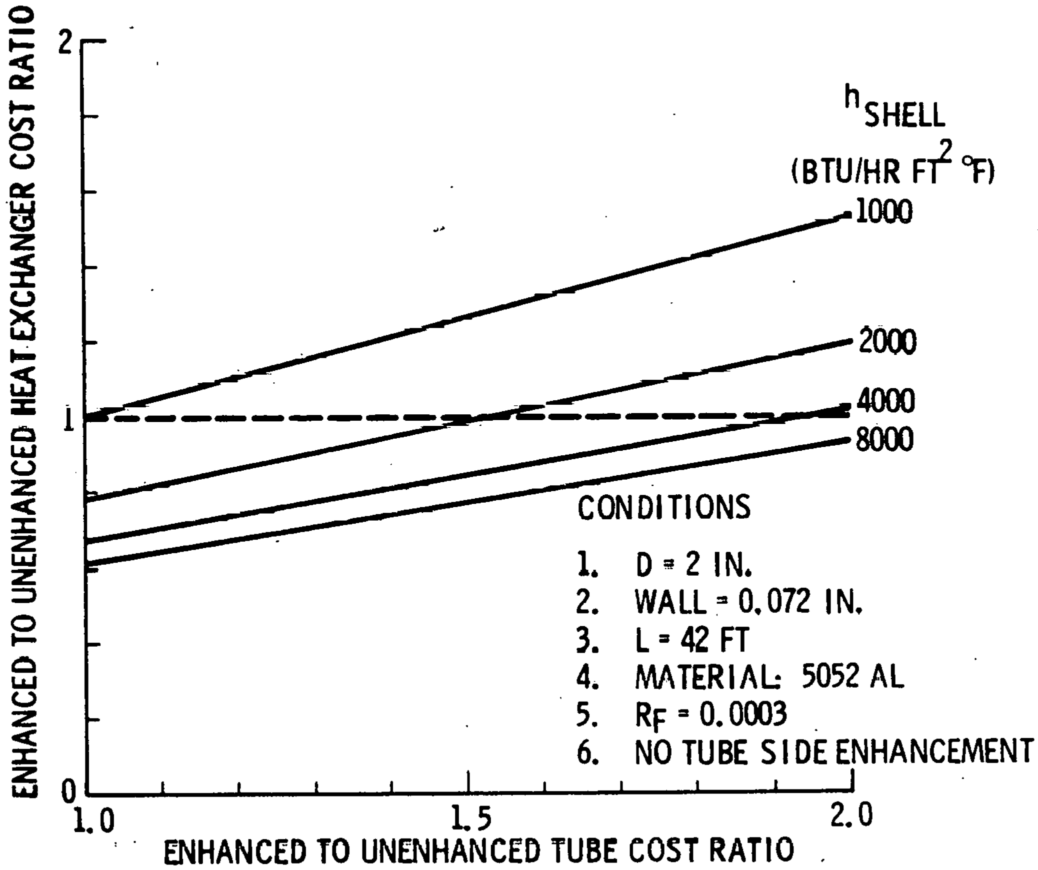


Fig. 4-13 Influence of Shell-Side Enhancement Tube Cost on 25-MW(e) Heat Exchanger Cost – Aluminum

The matrix of tubes shown in Table 4-2 was evaluated on a cost/producibility/performance basis. The trade studies were based on enhanced tube heat transfer coefficient and related cost and performance data supplied by manufacturers and independent research organizations.

For the Linde surface the following cost data were received:

"1. The incremental price above the bare tube titanium substrate price varies approximately 30 to 40 percent in the 1-1/2- to 2-in.-diameter range of interest (for very large quantities as required by OTEC). This is considerably reduced from May 21, 1976 report.

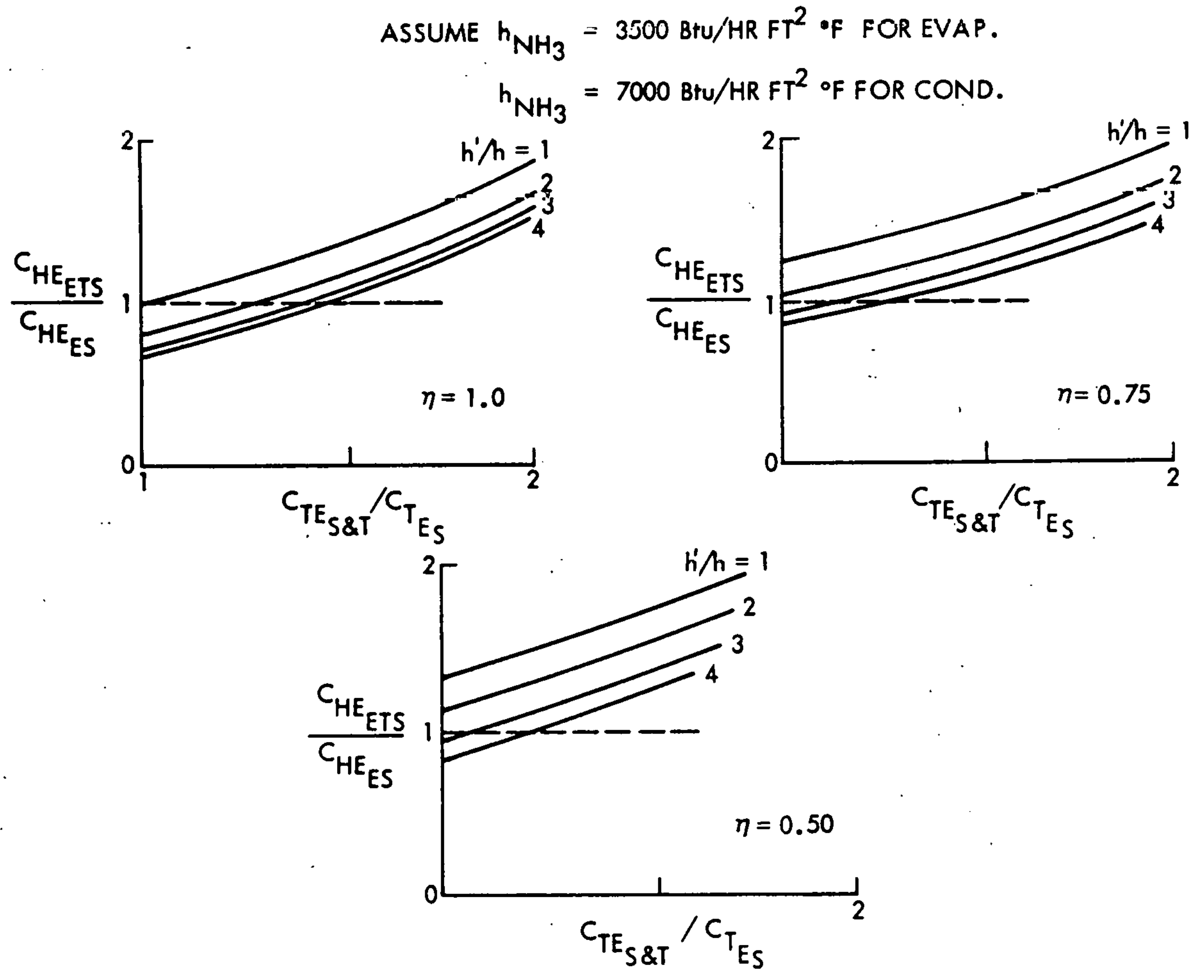


Fig. 4-14 Cost Relationship Effect of Tubeside Enhancement with Shell-Side Enhancement

2. The incremental price for aluminum tubing is approximately 90 percent of the dollar difference for the titanium tube in (1).
3. Base titanium tube wall thickness which will meet code requirements (0.035 and 0.049 inches, respectively).
4. As discussed, the performance of High Flux on titanium and aluminum substrates has been tested and reported to ERDA (see Figure 1A-2 attached). This data has been generated in 1977. It should be pointed out that the high flux coating process variables have been modified considerably since your stainless steel tubes were processed."

Cost interpretation of the above data indicates that the Linde high flux surface is not cost effective for either the titanium or aluminum tubes.

Discussions with Alcoa and Reynolds metals indicate the circumferentially grooved tubes can be produced in the 1-1/2- to 2-in. diameters at cost-effective prices. Southwest Alloy has a rolled integral fin process which can produce aluminum or bi-metallic finned tubes in the 1-1/2- to 2-in.-diameter sizes at cost-competitive prices. Verbal data from the above noted vendors indicate that tube enhancement costs will range from $0.25 to $0.50/ft, depending on tube diameter, material, and the specific manufacturing process used. This information will be further refined and documented during preliminary design.

The aluminum circumferentially grooved tube fabricated by an embossed process or an integrally rolled fin with a smooth internal bore is the most cost-effective tube determined. No seawater-side enhancement technique has been included in the recommended heat exchanger configuration.

4.3.6 Performance Considerations

The thermal performance capability of the candidate heat exchanger materials was based on overall heat transfer of the tube. Using the material characteristics and representative tube wall thicknesses, thermal resistances were established. By selection of seawater and ammonia film heat transfer coefficients and fouling factors, the overall heat transfer coefficient is established. There results are plotted in Fig. 4-15 against the fouling resistance. The tube material characteristics are tabulated below.

Material	Thermal Conductivity (Btu/hr-ft-^{o}F)	Tube Wall Thickness (in.)
Aluminum 5052 (0)	77	0.088 (mean)
Copper Nickel 90-10	26	0.049
Titanium CP	9.5	0.034
Stainless AL-6X	8	0.028

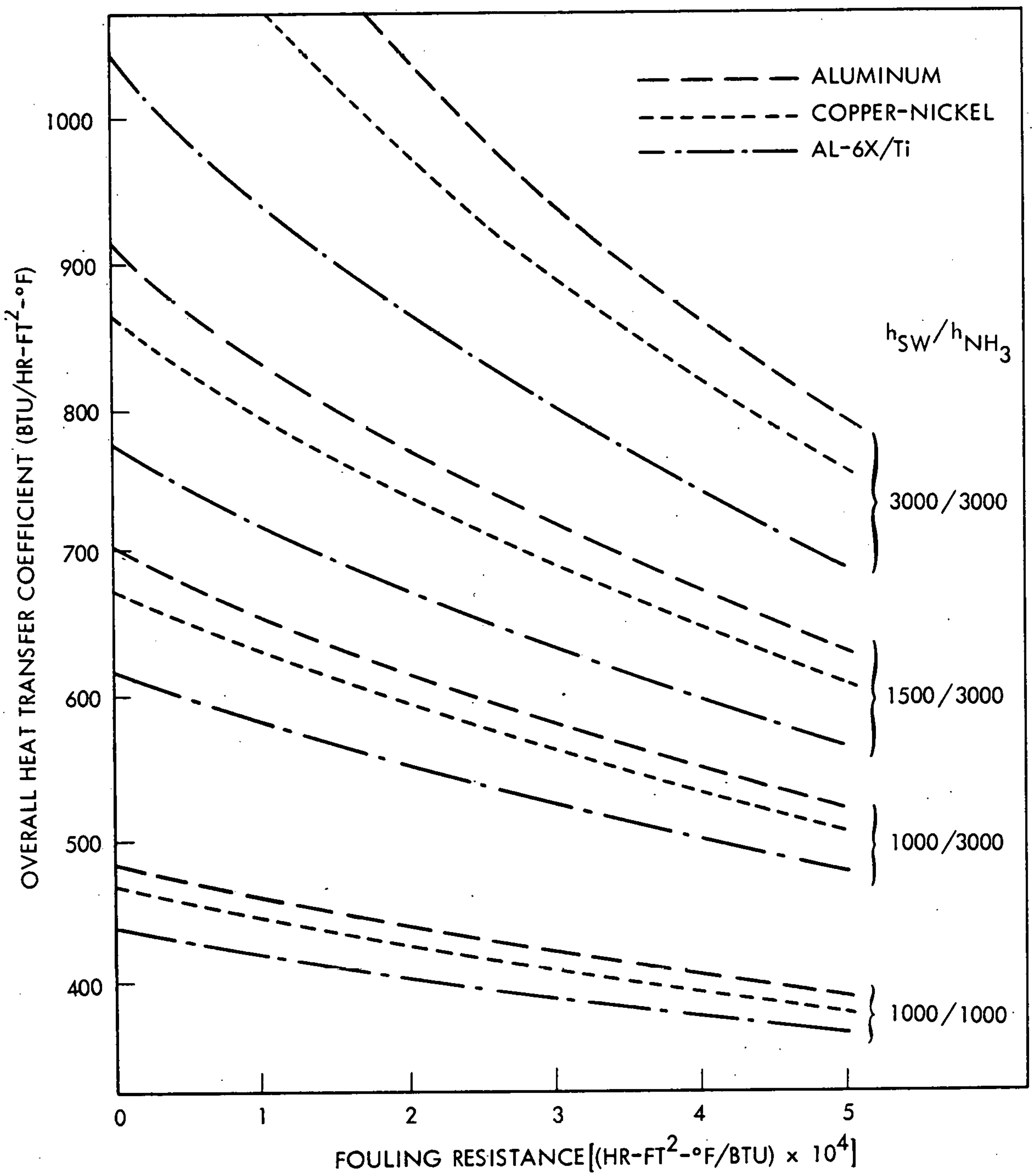


Fig. 4-15 Enhancement/Material Effects on Overall Heat Transfer

The tube material thermal resistance is seen to play as important a part as the fouling resistance. At a film coefficient of 1000 for seawater and 3000 for ammonia, aluminum tubes with a fouling resistance factor of 0.0004 are the performance equivalent of stainless steel or titanium tube at a fouling resistance factor of 0.0002. It is noted that a 50-percent increase in the seawater film coefficient from 1000 to 1500 with a 3000 ammonia side at a fouling factor of 0.0003 raises the overall coefficient from approximately 580 to 715 — an increase of 23 percent. For this reason, several tube-side enhancement techniques are still under consideration and will be defined early in preliminary design.

4.3.7 Physical Configuration

Various heat exchanger configurations were established to evaluate physical characteristics such as shell diameter, number of tubes, tube length and bundle diameter, dry weight and operating weight. These physical characteristics were used in evaluating effects of thermal resource ΔT's, platform influences, and manufacturing and handling.

A tabulation of six computer optimized designs for 25-MW(e) net, $36^{\circ}F$ ΔT, submerged heat exchangers is presented in Table 4-3. Weight reduction in the order of 25 percent are found in going from plain to enhanced tubes.

In addition, the volumes are reduced approximately 25 percent. The benefit of weight and size reduction is not only realized in the cost of the heat exchanger but also in the cost and size of the manufacturing facility, heat exchanger handling, and the platform. Using platform partials of $10 to $100 per pound payload or per cubic foot payload, enhancement provides savings of several million dollars. The weight of water in the tubes of the heat exchanger is reduced by one-third in going to enhanced tubes over plain tubes. The material weight effects are also substantial when comparing aluminum to stainless steel or titanium.

Table 4-3

HEAT EXCHANGER PHYSICAL CHARACTERISTICS – 36°FΔT, SUBMERGED, 25-MW(e) NET

Item	Plain Tube						Enhanced Tube					
	Aluminum		Stainless		Titanium		Aluminum		Stainless		Titanium	
	Evap	Cond	Evap	Cond	Evap	Cond	Evap	Cond	Evap	Cond	Evap	Cond
Number of Tubes	78,014	77,924	79,023	82,633	73,903	82,491	62,411	62,411	63,027	66,497	63,054	66,403
Tube Length (ft)	42.0	45.0	45.2	46.3	45.2	46.3	38.6	41.8	41.2	42.3	41.2	42.3
Tube Bundle Diam (ft)	44.0	44.0	45.2	46.3	45.2	46.3	41.0	41.0	41.2	42.3	41.2	42.3
Tubesheet Diam (ft)	46.0	46.0	47.2	48.3	47.2	48.3	43.0	43.0	43.2	44.3	43.2	44.3
Shell OD (ft)	63.8	66.0	67.0	68.4	67.0	68.4	59.4	61.8	61.9	62.9	61.3	62.8
Shell Vol. (ft^3)	117,700	132,400	138,000	147,000	138,000	147,000	94,000	107,500	104,700	113,800	107,900	113,300
Shell Thick.(in.)	2.0	2.0	1.0	1.0	1.0	1.0	2.0	2.0	1.0	1.0	1.0	1.0
Shell Weight (lb $\times 10^6$)	0.261	0.287	0.421	0.422	0.422	0.440	0.224	0.250	0.354	0.372	0.358	0.372
Tubesheet/Support Wt (lb $\times 10^6$)	0.107	0.109	0.240	0.253	0.241	0.227	0.106	0.107	0.215	0.227	0.216	0.223
Tube Weight (lb $\times 10^6$)	1.468	1.556	2.454	2.718	1.790	1.900	1.163	1.243	2.031	2.174	1.420	1.513
Misc Weight[(a)] (lb $\times 10^6$)	0.037	0.055	0.091	0.095	0.084	0.086	0.012	0.048	0.077	0.087	0.072	0.075
Total Dry Weight (lb $\times 10^6$)	1.873	2.007	3.206	3.488	2.537	2.653	1.505	1.648	2.677	2.860	2.066	2.193
Ammonia Weight (lb $\times 10^6$)	0.177	0.140	0.189	0.147	0.186	0.147	0.160	0.123	0.171	0.129	0.171	0.129
Water in Tubes Weight (lb $\times 10^6$)	2.424	2.571	2.797	3.120	2.879	3.074	1.615	1.730	2.081	2.246	2.054	2.211
Displacement (lb $\times 10^6$)	7.533	8.474	8.832	9.408	8.832	9.408	6.016	6.880	6.700	7.283	6.905	7.283

(a) 10 percent of shell plus tubesheet/support weight plus 1 percent tube weight.

The water weight in the aluminum heat exchanger is approximately one-third greater than the dry weight of the heat exchanger in the plain tube but reduces to less than 10 percent in the enhanced case while for stainless steel the water weight is approximately 90 percent the dry weight in the unenhanced case and drop to about 65 to 75 percent in the enhanced case. The change in operating (wet) weight is also substantial.

A tabulation of the physical characteristics of submerged and surface aluminum heat exchangers optimized at a ΔT of $40^{o}F$ is presented in Table 4-4. The main differences occur in tube wall thickness and shell thickness. The surface configuration has a minimum tube wall of 0.090 in. and the submerged configuration is 0.075 in. Both incorporate a 0.020-in. corrosion allowance. The shell thickness is 2.0 and 3.2 in. thick for the submerged and surface, respectively. The dry weights of the surface heat exchangers are 13 to 14 percent heavier than the submerged units and, when including liquids, are about one-third heavier. The major distinction, however, is that in the surface units the total weight must be supported by the platform while in the submerged configuration the heat exchangers provide positive buoyancy.

4.3.8 Tube Arrangement

The arrangement of the heat exchanger tubes can take any one of four patterns when viewed from the open end of a tube. These are, as shown in Table 4-5, triangular or square pitch with major/minor axes oriented vertically or horizontally. A brief evaluation of these arrangements was conducted using the four parameters given in the Table, and as indicated the triangular pitch with the minor axis vertically was selected.

4.3.9 Heat Exchanger Summary

The characteristics of the selected horizontal shell and tube heat exchanger with circumferentially grooved heat transfer enhancement is summarized in Tables D, E, F, G, and 4-4.

Table D

HEAT EXCHANGER DESIGN DATA – HORIZONTAL SHELL AND TUBE (EVAPORATOR)

Item	Reference Baseline	Optimized	
		Surface	Submerged
Number per Module	1	1	1
Thermal Rating (MBtu/hr)	3,970	3,540	3,508
Water Flow Rate (gpm)	1,610,350	1,546,900	1,615,300
Ammonia Flow Rate (lb/hr)	11,396,700	10,164,750	10,074,700
Design Pressure (psid)[(a)]	250/-	250/-	127/54
Operating Pressure (psia)	137	137	137
Overall Length (ft)	76	68	66
Water Enhancement Factor	1.0	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0	3.0
Total Number of Tubes	52,260	50,315	50,250
Tube OD (in.)	1.5	1.5	1.5
Tube Wall (in.)[(b)]	0.1025/0.090	0.1025/0.090	0.088/0.075
Tube Pitch (in.)	1.875	1.875	1.875
Tube Layout	Equilateral Triangle		
Tubesheet Diameter (ft)	43.5	43	43
Effective Tube Length (ft)	44	39.5	38
Actual Tube Length (ft)	46	41.5	40
Tubesheet Thickness (in.)	2.5	2.5	1.5
Second Tubesheet Thickness (in.) (if applicable)[(c)]	2.5	2.5	1.5
Number of Tube Supports	6	5	5
Tube Support Thickness (in.)	1	1	1
Tube Support Spacing (ft)	6.3	6.6	6.33
Shell OD (ft)	63.5	60	59
Shell Wall Thickness (in.)	3.2	3.2	2.0
Void Fraction	0.63	0.65	0.65
Tube Material	AL 5052	AL 5052	AL 5052
Tubesheet Material	AL 5083	AL 5083	AL 5083
Shell Material	AL 5083	AL 5083	AL 5083
Baffle Material	AL 5083	AL 5083	AL 5083
Cladding Material	–	–	–
Coating Materials[(d)]	TBD	TBD	TBD
Demister Size (ft × ft × ft)	26' d × 9" thick	25' d × 9" thick	25' d × 9" thick
Demister Design Pressure (psia)	(e)	(e)	(e)
Design Inlet Quality (%)	95	95	95
Design Outlet Quality (%)	99	99	99

(a) Burst/Collapse
(b) Mean/Minimum
(c) Annular Type
(d) Not defined at this time
(e) Mist Extractor inside evaporator shell

Table E

HEAT EXCHANGER DESIGN DATA – HORIZONTAL SHELL AND TUBE (CONDENSER)

Item	Reference Baseline	Optimized	
		Surface	Submerged
Number per Module	1	1	1
Thermal Rating (MBtu/hr)	3,845	3,430	3,400
Water Flow Rate (gpm)	1,633,900	1,582,200	1,664,600
Ammonia Flow Rate (lb/hr)	7,597,800	6,776,600	6,716,460
Design Pressure (psid)[(a)]	250/-	250/-	96/132
Operating Pressure (psia)	96	96	96
Overall Length (ft)	76	68	66
Water Enhancement Factor	1.0	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0	3.0
Total Number of Tubes	58,700	56,400	56,000
Tube OD (in.)	1.5	1.5	1.5
Tube Wall (in.)[(b)]	0.1025/0.090	0.1025/0.090	0.088/0.075
Tube Pitch (in.)	1.875	1.875	1.875
Tube Layout	Equilateral Triangle		
Tubesheet Diameter (ft)	43.5	43	43
Effective Tube Length (ft)	44	39.5	38
Actual Tube Length (ft)	46	41.5	40
Tubesheet Thickness (in.)	2.5	2.5	1.5
Second Tubesheet Thickness (in.) (if applicable)[(c)]	2.5	2.5	1.5
Number of Tube Supports	6	5	5
Tube Support Thickness (in.)	1	1	1
Tube Support Spacing (ft)	6.3	6.6	6.33
Shell OD (ft)	63.5	60	59
Shell Wall Thickness (in.)	3.2	3.2	2.0
Void Fraction	0.59	0.61	0.61
Tube Material	AL 5052	AL 5052	AL 5052
Tubesheet Material	AL 5083	AL 5083	AL 5083
Shell Material	AL 5083	AL 5083	AL 5083
Baffle Material	AL 5083	AL 5083	AL 5083
Cladding Material	–	–	–
Coating Materials[(d)]	TBD	TBD	TBD

(a) Burst/Collapse
(b) Mean/Minimum
(c) Annular Type
(d) Not defined at this time

Table F

HEAT EXCHANGER THERMAL DATA – EVAPORATOR

Item	Reference Baseline	Optimized	
		Surface	Submerged
Water Flow Rate (lb/hr)	824×10^6	792×10^6	827×10^6
Water Flow Per Tube (lb/sec)	4.38	4.37	4.57
Water Velocity Inside Tube (ft/sec)	7.54	7.54	7.53
Reynolds Number Inside Tube (-)	72496	72424	73970
Prandtl Number Inside Tube (-)	6.0	6.0	6.0
Inlet Water Temperature (° F)	83.88	83.88	83.88
Outlet Water Temperature (° F)	78.71	79.08	79.32
Average Water Temperature (° F)	81.29	81.48	81.55
LMTD (° F)	7.19	7.42	7.46
Average Tube Wall Surface Temperature (° F)	74.3	74.5	74.8
Average Shell Wall Surface Temperature (° F)	74.5	74.7	75
HW (Including Enhancement) (Btu/hr ft^2 ° F)	1388	1388	1382
H Fouling (Btu/hr ft^2 ° F)	3333	3333	3333
H Wall (Btu/hr ft^2 ° F)	9709	9709	11235
H Ammonia (Btu/hr ft^2 ° F)	3475	3475	3400
Uo (Btu/hr ft^2 ° F)(a)	709	709	711
Ammonia Inlet Temperature (° F)	53.53	53.55	53.41
Ammonia Outlet Temperature (° F)	73.49	73.74	73.91
Vapor Quality (Outlet, %)	99	99	99
Maximum Vapor Velocity (Outlet, ft/sec)	7.0	7.1	7.3
Total Pressure Drop of Water (psid)	3.92	3.56	3.35
Total Pressure Drop of Ammonia (psid)(b)	0.30	0.30	0.32
Distribution Tubes/Heat-Transfer Tubes	0.01	0.01	0.01
Maximum Feed Rate (lb/ft sec)	0.20	0.20	0.20
Minimum Feed Rate (lb/ft sec)	0.10	0.10	0.10
Design Feed Rate (lb/ft sec)	0.15	0.15	0.15
Recirculation Ratio (-)	0.5	0.5	0.5
Maximum Distribution System ΔP (psid)	15	15	15

(a) Based on tube ID
(b) Across tube bundle radius using Eqs. 6.13 and 6.13(b) of HEAT TRANSMISSION by McAdams.

Table G

HEAT EXCHANGER THERMAL DATA – CONDENSER

Item	Reference Baseline	Optimized	
		Surface	Submerged
Water Flow Rate (lb/hr)	841×10^6	814×10^6	856×10^6
Water Flow Per Tube (lb/sec)	3.98	4.0	4.25
Water Velocity Inside Tube (ft/sec)	6.862	6.821	6.9
Reynolds Number Inside Tube (-)	50723	50422	52147
Prandtl Number Inside Tube (-)	10.4	10.4	10.4
Inlet Water Temperature (°F)	43.88	43.88	43.88
Outlet Water Temperature (°F)	48.77	48.38	48.12
Average Water Temperature (°F)	46.32	46.35	46.00
LMTD (°F)	7.08	7.35	7.40
Average Tube Wall Surface Temperature (°F)	52.8	52.8	52.6
Average Shell Wall Surface Temperature (°F)	56.2	56.2	56
HW (Including Enhancement) (Btu/hr ft^2 °F)	1086	1079	1084
H Fouling (Btu/hr ft^2 °F)	3333	3333	3333
H Wall (Btu/hr ft^2 °F)	9709	9709	11235
H Ammonia (Btu/hr ft^2 °F)[a]	3475	3475	3400
Uo (Btu/hr ft^2 °F)[a]	620	618	623
Ammonia Inlet Temperature (°F)	53.86	53.89	53.86
Ammonia Outlet Temperature (°F)	53.50	53.52	53.35
Vapor Quality (Inlet, %)	96.55	96.55	96.5
Maximum Vapor Velocity (Inlet, ft/sec)	9.0	9.2	9.6
Total Pressure Drop of Water (psid)	3.277	2.913	2.808
Total Pressure Drop of Ammonia (psid)[b]	0.646	0.672	0.705

(a) Based on tube ID
(b) Across tube bundle radius using Eqs. 6.13 and 6.13(b) of Heat Transmission by McAdams.

Table 4-4

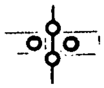
SELECTED HEAT EXCHANGER PHYSICAL PARAMETERS – ΔT = 40°F, 25 MW(e)

Item	Submerged		Surface	
	Evap	Cond	Evap	Cond
Tube Length (ft)	40	40	41.5	41.5
Number of Tubes	50,250	56,000	50,315	56,400
Tube Bundle Diam (ft)	41.0	41.0	41.0	41.0
Tubesheet Diam (ft)	43.0	43.0	43.0	43.0
Shell OD (ft)	59.0	59.0	60.0	60.0
Shell Volume (ft^3)	92,604	92,604	98,626	98,626
Shell Thick (in.)	2.0	2.0	3.2	3.2
Shell Weight (lb)	210,000	210,000	365,700	365,700
Tubesheet/Support Weight (lb)	131,000	131,000	158,000	158,000
Tube Weight (lb × 10^6)	0.913	1.018	1.129	1.258
Misc. Weight[(a)] (lb)	43,600	44,600	63,800	65,100
Total Dry Weight (lb × 10^6)	1.301	1.407	1.718	1.849
Ammonia Weight (lb × 10^6)	0.156	0.112	0.160	0.116
Water in Tubes (lb × 10^6)	1.186	1.321	1.209	1.345
Displacement (lb × 10^6)	5.927	5.927	6.312	6.312
Buoyancy (lb × 10^6)	3.284	3.087	–	–

(a) 10 percent of shell plus tubesheet/support weight plus 1 percent of tube weight

Table 4-5

TUBE ARRANGEMENTS

ARRANGEMENT	VAPOR-LIQUID INTERACTION	PLATFORM PITCH-ROLL SENSITIVITY	TUBE PACKING DENSITY	LIQUID DISTRIBUTION SYSTEM	RANKING
TRIANGULAR PITCH 	D	D	A	D	4
TRIANGULAR PITCH 	C	A	A	A	1
SQUARE PITCH 	B	B	C	B	3
SQUARE PITCH 	A	C	C	C	2

RATING:
A = BEST

4.4 MATERIAL CONSIDERATIONS

The purpose of this section is to describe the rationale that resulted in the determination that aluminum is the optimum material for OTEC heat exchangers. This determination was derived through the evaluation of candidate materials characteristics in relation to the performance and cost requirements of the power system module. The primary material characteristic considerations discussed in this section are mechanical and thermal properties, corrosion resistance, biofouling, wear resistance, and life-cycle costs.

4.4.1 Material Candidates

Candidate materials include aluminum alloys 3003, 3004 alclad, and 5052; copper-nickel 90-10; stainless steel alloy AL-6X; and commercially pure titanium. These materials have been used in various seawater heat exchanger applications with fair to excellent performance depending on the operating conditions. All the material candidates have good producibility in that they have good to excellent welding, forming, and machining characteristics using conventional production methods.

4.4.2 Mechanical/Thermal Properties

The tensile strength and modulus of elasticity of aluminum and copper nickel are low but are acceptable in relation to OTEC design requirements. For example, a 0.050-in. wall aluminum tube will meet the strength requirements.

AL-6X and titanium have the highest strength and 1.5 to 3 times the modulus of elasticity of aluminum, which permits the use of thinner wall (0.029-in.) tubing. The thinner wall compensates for the lower thermal conductivity. Power plant studies have shown that even though copper nickel has a higher thermal conductivity than stainless steel, the development of its oxide film degrades the copper-nickel heat transfer capability to values close to stainless steel and titanium which have an oxide film that is less than 0.0005-in. thick. Table 4-6 summarizes the key mechanical and thermal properties of the materials.

Table 4-6

MATERIAL CHARACTERISTICS

Material	Yield Strength (psi)	Ultimate Strength (psi)	E (psi $\times 10^6$)	Density (lb/ft^3)	Thermal Conductivity (Btu/Hr-ft-°F)
Aluminum 5052 (0)	13,000	28,000	10	170	77
Alclad 3003 (0)	6,000	16,000	10	170	110
Titanium CP	40,000	50,000	15	281	9.5
Stainless AL-6X	45,000	93,000	29	500	8.0
Copper-Nickel 90-10	15,000	40,000	18	558	26

4.1.3 Corrosion Resistance

The corrosion resistance of the candidate materials was evaluated from the standpoint of resistance to (1) general corrosion (uniform), (2) pitting and crevice corrosion, (3) stress corrosion cracking, (4) corrosion fatigue, and (5) galvanic attack. There is a wealth of corrosion test data on the candidate materials (except AL-6X) from test specimens and production hardware exposed to surface waters and deep ocean water for periods of from 1 to 15 years.

The corrosion test data on the candidate materials' resistance to ammonia is extensive for aluminum, copper-nickel, and titanium. Although there are limited data on the corrosion resistance of AL-6X in ammonia, there are adequate data on steel and stainless steel alloys that, in general, have lower corrosion resistance than AL-6X. These data provide assurance that AL-6X will have excellent resistance to ammonia.

Aluminum Alloys. Alloys 3003, 3004, and 5052 have been shown to have a general (uniform) corrosion rate of less than 0.001-in. per year (1 mpy). Pitting corrosion test data show varied results with the 0 temper having that best resistance. Welded tubing is the lowest cost material and would normally be in the 0 temper condition and therefore should be the design choice. The maximum pit depth after 1 to 2 years has been found to be 0.005 to 0.012 in. in surface and deep ocean waters. Operating experience on alclad tubing used in seawater-cooled condensers at a California coastal oil refinery has shown only one perforation of 0.071-in. wall tube in 14 years. Table 4-7 summarizes corrosion test data from aluminum specimens and operating heat exchangers.

The deep seawater data are provided for information purposes only. The results are not considered directly applicable to the condensers because the specimens were maintained in static seawater and because the hydrostatic pressure at the test depth minimized the volumetric concentration of gases. With seawater velocities of 6 to 7 fps and at the actual heat exchanger depths, the tube surfaces will be subjected to near surface oxygen concentrations, particularly if the cold water is taken from a sump which is open to the atmosphere.

Experimental data are required to determine the effect of deep seawater on material corrosion at flow velocities and depths representative of the OTEC condenser operation. Data from the OTEC-1 facility would be applicable. However, it would be desirable to obtain near-term data for application to the module and test article design.

Crevice corrosion of the aluminum alloys does occur with barnacle-type biofouling but is no more severe than pitting attack.

The candidate aluminum alloys are immune to stress corrosion cracking in seawater; and although the seawater corrosion fatigue strength of aluminum is quite low, their fatigue strength has been well documented at various stress reversal differences with and without cathodic protection.

Aluminum alloys are susceptible to galvanic attack in seawater when coupled to all metals and their alloys except magnesium and zinc. An example of this problem would be the use of steel for the inlet piping and waterbox components and copper for the inlet screens. The solutions to this problem are (1) design the seawater system of all aluminum components, (2) electrically insulate the dissimilar materials, or (3) coat the cathodic material (steel) and provide cathodic protection. Long life epoxy coal tar

Table 4-7

ALUMINUM-SEAWATER CORROSION DATA

Material	Period (yr)	Surface Seawater		Deep Seawater	
		Min.	Max.	Min.	Max.
AL5052 – 0	1	0.0	0.014 in. (pitting)	0.0	0.012 in. (pitting)
	2	0.0	0.003 in. (pitting)	0.0	0.000
	3	–	–		
	5	0.0	0.008 in. (pitting)	0.0	
AL5052 – H34	1	0.0	0.029 in. (pitting)	0.0	0.012 in. (pitting)
	2	0.0	0.020 in. (pitting)	0.0	0.065 in. (pitting)
	5	0.0	0.006 in. (pitting)		
	10	0.0	0.012 in. (pitting)		
AL3003 Alclad	2		0.013 in. (pitting)		
	14		0.091 in. (pitting)		
3003 Alclad – H12	1				0.014 in. (pitting)
	2				0.000 in. (pitting)
	3				0.020 in. (pitting)
3003 Alclad – H14	2				0.000 in. (pitting)
3003 Alclad – 0	1				0.000 in. (pitting)
	2				0.000 in. (pitting)
	3				0.000 in. (pitting)
Data from surface and deep (2,300- to 6,300-ft) seawater corrosion tests performed in Florida, the Bahamas, Southern California, Nova Scotia, and South Carolina.					

coatings and 20-year life sacrificial anodes (cathodic protection) offer the best solution to this problem when the design requires dissimilar metals. Materials compatibility for the heat exchanger material candidates, including effects of the platform materials, is presented in Table 4-8.

Table 4-8

HEAT EXCHANGER/PLATFORM MATERIALS COMPATIBILITY

Platform Materials	Heat Exchanger Materials			
	Aluminum Alloys	Stainless AL-6X	Titanium C.P.	Copper-Nickel 90-10
CW Pipe				
Steel	NC	C	C	C
Aluminum	C	C	C	C
Concrete	C	C	C	C
GRP/Plastic/Rubber	C	C	C	C
Ducting				
Steel	NC	NC	NC	NC
Aluminum	C	NC	NC	NC
GRP/Plastic	C	C	C	C
Concrete	C	C	C	C
SST	C	C	C	C
Screens				
Cu-Ni (90-10)	NC	C	C	C
Steel-Galvanized	C	C	C	C
Steel-Coated W/A.F.	C	C	C	C
SST	C	C	C	C
CW Pipe Transition				
Steel	NC	C	C	C
Aluminum	C	C	C	C
Bronze	NC	C	C	C
Graphite	C(?)	C	C	C
Pumps				
Steel	NC	C	C	C
SST	C	C	C	C
Bronze	NC	C	C	C
Ni Cast Iron	NC	C	C	C

NC = Not compatible if heat exchanger is not electrically isolated.
Cathodic protection may make compatible.

The corrosion resistance of the aluminum alloys to ammonia is excellent as has been demonstrated by refrigeration industry experience. Recent laboratory test data on combinations of ammonia and seawater show little or no corrosion except at low ammonia percentages such as 1 to 5 percent. Subsequent field corrosion tests using natural seawater under OTEC heat exchanger flow conditions and 70 ppm ammonia have resulted in only incipient pitting (less than 0.001-in.). During these field corrosion tests it was found that the small amount of ammonia (70 ppm) caused deposition of a calcareous precipitate onto the aluminum tubing wall. The precipitate developed a bond to the aluminum but it would appear that frequent cleaning by the MAN brush system (i.e., daily) would remove this deposit. Since an ammonia leak detection system will be used to locate a leaking tube, it is probable that an ammonia leak would be of short duration.

Copper-Nickel. Copper-nickel 90-10 alloys have been successfully used as a seawater heat exchanger material for several decades on naval vessels. In more recent usage in coastal power plants, it has experienced failures in 0.035-in. wall tubing in less than 5 years. These failures have, in most cases, been attributed to inlet erosion/corrosion due to poor waterbox design or downstream fouling on the tube wall. This problem is due to the low shear strength of its oxide film, which means the alloy is seawater velocity limited. Although the velocities planned for OTEC are within the alloy's limit, these velocities are close enough that this problem must be considered in the waterbox and duct design of an OTEC plant.

Table 4-9 summarizes the seawater corrosion data for materials other than aluminum A galvanic corrosion problem could be caused by copper-nickel heat exchangers should steel or aluminum be used in the components of the seawater system. As mentioned in the previous paragraph on aluminum, this problem can be solved with the use of cathodic protection, but cathodic protection would cause the copper nickel tubing to biofoul at the inlet because its corrosion would be stopped.

The corrosion of copper-nickel 90-10 by ammonia is dependent on contamination of the ammonia by water and oxygen, with oxygen being the controlling factor. If the ammonia system can be kept uncontaminated, the copper-nickel will suffer little or no attack.

Table 4-9

SEAWATER CORROSION DATA

Material	Surface Seawater	Deep Seawater
Cu-Ni (90-10)	<1 mpy (unif)	<1 mpy (unif)[a]
AL-6X	0.000	No Data
Ti (CP)	0.000	0.000

(a) Corrosion rate data of <1 mpy based on relatively static seawater tests.

In summary, copper-nickel 90-10 has a low corrosion rate (less than 0.001-in. per year) in seawater and uncontaminated ammonia, is immune to stress corrosion cracking in the same environment, and has documented corrosion fatigue strength that can be used in design to prevent failure by fatigue. The problem of erosion/corrosion must be addressed in the seawater system design.

Stainless Steel Alloy AL-6X. Alloy AL-6X has been used in coastal power plant condenser tubing for more than 5 years without a corrosion failure. There is no corrosion data on the effect of ammonia, but experience with steel and stainless steel alloys in ammonia and water-contaminated ammonia provides assurance that AL-6X will be immune to corrosion time in these environments.

Commercially Pure Titanium. Commercially pure titanium is immune to all types of corrosion by seawater and ammonia and their combination, based on more than 15 years of marine and chemical industry experience.

A summary of 10-year seawater corrosion performance of the candidate materials is given in Table 4-10. Table 4-11 summarizes the ammonia corrosion data.

Table 4-10

TEN-YEAR CORROSION PERFORMANCE IN SEAWATER

Candidate Materials	Corrosion				
	Uniform	Pitting	Crevice	Stress Cracking	Seawater Erosion Resistance
AL5052 Alclad 3003 or 4	Nil	0.010 to 0.060 in.	0.015 to 0.070 in.	N.S.[a]	Good
AL-6X[b]	Nil	N.S.	N.S.	N.S.	Excellent
Ti	Nil	N.S.	N.S.	N.S.	Excellent
Cu-Ni (90-10)	0.000 to 0.040 in.[c]	20 to 30 mpy 0.02 to 0.03 in.	N.S.	N.S.	Fair

(a) N.S. — Not susceptible.
(b) Based on similar stainless steels.
(c) Cu-Ni tubing life very dependent on waterbox design and operating velocity.

Table 4-11

AMMONIA CORROSION DATA

Materials	Ammonia	Seawater in Ammonia (1 to 5% Water)	Ammonia in Seawater (1 to 5% Ammonia)
Aluminum	< 5 mpy[a]	000	1 mpy[a]
Cu-Ni (90-10)	< 1 mpy	< 1 mpy	1 to 5 mpy
AL-6X	0.000[b]	0.000[b]	000
Ti (C.P.)	000	000	000

(a) Corrosion rate based on 10- and 30-day tests.
(b) Corrosion data based on 6 years of coastal power plant experience.

4.4.4 Biofouling Resistance

The only candidate material possessing biofouling resistance is the copper-nickel 9-10 alloy which will resist fouling by hard shell organisms such as barnacles but will not resist attachment of bacteria in the form of slime layers.

Based on 1976 heat transfer/biofouling tests in Hawaii and the Virgin Islands, it has been stated that a slime layer no thicker than 0.002-in. can be tolerated on the heat transfer surfaces. This implies that all candidate materials will require biofouling countermeasures to control slime layer development. The use of chlorination and mechanical scrubbing at regular intervals would appear to provide high reliability of achieving relatively foul-free heat transfer surfaces. Chlorination at the rate of 0.2 to 1.6 mg/liter for 20 minutes three times per day will not have adverse corrosion effects on any of the candidate materials because of low dosage and short exposure time.

4.4.5 Wear Resistance

Aluminum alloys have poor wear or abrasion resistance to such materials as sand or marine shell particles. Abrasion by sand in an OTEC plant is very unlikely, but marine shell particles are possible if these growths are permitted upstream of the heat exchangers. For this reason the chlorination system should be located at the seawater inlet to prevent hard shell growth from occurring anywhere upstream of the heat exchangers.

4.4.6 Ammonia Compatability

In ammonia vapor and liquid ammonia systems, carbon steel and cast iron are generally suitable for anhydrous ammonia service. However, some uncertainties exist with air-contaminated anhydrous ammonia and seawater-contaminated ammonia.

Carbon steel is subject to stress corrosion cracking in air-contaminated anhydrous ammonia. It is not likely that the main cycle piping and components will be subjected to air contamination except during initial checkout, shutdown for maintenance, and restart. Air and nitrogen should be purged from the system by the ammonia through the noncondensible vents. However, this stress corrosion cracking can be prevented by post-weld heat treatment and thermally stress relieving (at 1,100° F minimum) parts containing high residual fabrication stresses. Another method used in ammonia systems is to add 0.2 percent water to the ammonia. However, for the OTEC system, the addition of seawater to the system through leakages is considered a liability for thermodynamic reasons and is minimized by a low leakage heat exchanger design. Hence post weld thermal stress relieving is recommended for carbon steel piping.

Seawater concentrations in the ammonia system can be expected to vary from very low levels in the vapor and condensate pipe and the condenser to as high as 5 percent in the evaporator. The corrosion and deposition effects caused by this seawater content are not clearly understood at this time.

Aluminum alloys are also suitable for ammonia service, common application being in refrigeration systems and storage tanks. Galvanic attack between steel and aluminum in anhydrous ammonia would not be expected since the conductivity of anhydrous ammonia is less than 1×10^{-8} mhos/cm (values as low as 1×10^{-11} mhos/cm have been reported). This is less than one-hundredth that of demineralized water. In laboratory tests, it has been shown that if anhydrous ammonia is contaminated with air, pitting attack of aluminum coupled to steel is promoted. These same tests indicate that water additions may reduce the galvanic attack. However, as discussed above, water additions are detrimental to the operation of the ammonia power cycle and, in fact, seawater contamination may promote galvanic attack.

In summary, aluminum and steel are suitable for anhydrous ammonia service provided the ammonia is not contaminated with air or seawater. Air contamination is expected to be minimal with the system purging all noncondensibles during initial checkout. Seawater inleakage will be handled by the ammonia cleanup system. Concentrations of seawater should be extremely small throughout the system except in the evaporator and distribution pumps, piping, and valves. The baseline conceptual design approach is therefore to use carbon steel piping for the ammonia cycle together with the aluminum heat exchangers. Aluminum piping is still being considered as an alternative. Fiberglass pipe was considered and rejected because commercially available resins cannot withstand ammonia.

In order to withstand galvanic corrosion on the seawater side, a suitable cathodic protection system and electrical isolation system design will be incorporated into the preliminary design using coated carbon steel pipe. The aluminum piping is the fallback option. The results from any test programs concerning the effects of seawater and/or air contamination will also be factored into the preliminary design materials selection effort.

Copper-nickel is slightly superior to aluminum but would require the same protection. AL-6X and titanium have excellent abrasion resistance and do not require protection.

4.4.7 Economic Evaluation of Heat Exchanger Tubing Materials

Comparison of the cost effectiveness of aluminum, stainless steel, and titanium as tubing materials requires an intertemporal cost trade study. This is done in this section in terms of equivalent annual cost, using the methodology presented in "The Cost of Energy From Utility-Owned Solar Electric Systems" (ERDA/JPL-1012-76/3). Of course, the cost effectiveness of any material depends on its expected lifetime, and this is the most important independent variable in this study. Copper-nickel heat exchangers were not included in the cost trade study as adequate data were not generated in the heat exchanger fabrication study. Other assumptions used in the analysis are:

• Power module configuration	Submerged
• Escalation rate	6 percent
• Cost of money	10 percent
• Other taxes	2 percent
• Insurance	1.5 percent
• Heat exchanger replacement cost	$1,000,000
• Effective power module downtime	30 days

- Heat exchanger material scrap value
  - aluminum 10¢/lb
  - steel 2¢/lb

While some of these assumptions appear to be superficial, it will be shown later that the results of the study are quite insensitive to changes in them. The study uses the submerged power module configuration as the cost relationships among heat exchangers using the three alternative tubing materials are essentially the same for all configurations. Cost of money and escalation rates were supplied by DOE. A figure of 2 percent, typical of utility experience, has been assumed for property and other taxes. A 1.5-percent charge has been used for insurance charges based on a range of 1 to 2 percent of insured investment used in previous studies. The $1,000,000 cost of scrapping the old heat exchanger and replacing it represents LMSC's estimate of labor and dock fees for a 30-day period at a graving dock. It is planned that replacement of the heat exchangers would be scheduled at the same time as routine major maintenance activities and would require additional downtime for refit at the shipyard. The 30-day downtime assumed represents the additional downtime. Finally, the scrap value of aluminum and steel has been assumed to be 10 percent of current prices for plate metal.

The curves shown in Fig. 4-16 have been derived on the basis of the assumptions discussed above. They show the equivalent annual cost of aluminum and stainless steel-tubed heat exchangers as a function of their engineering lifetime. The equivalent annual cost of titanium-tubed units with expected lifetime of 30 years is 110 $/kW(e)-year. Aluminum-tubed heat exchangers will be more cost effective than titanium-tubed units if their lifetime exceeds 8 years, and the breakeven life for stainless-steel-tubed heat exchangers is 20 years. The above breakeven lives are referenced to titanium-tubed heat exchangers with a 30-year lifetime. Should the lifetime of stainless-steel-tubed heat exchangers be also 30 years, the breakeven life required of aluminum-tubed units increases to 10 years.

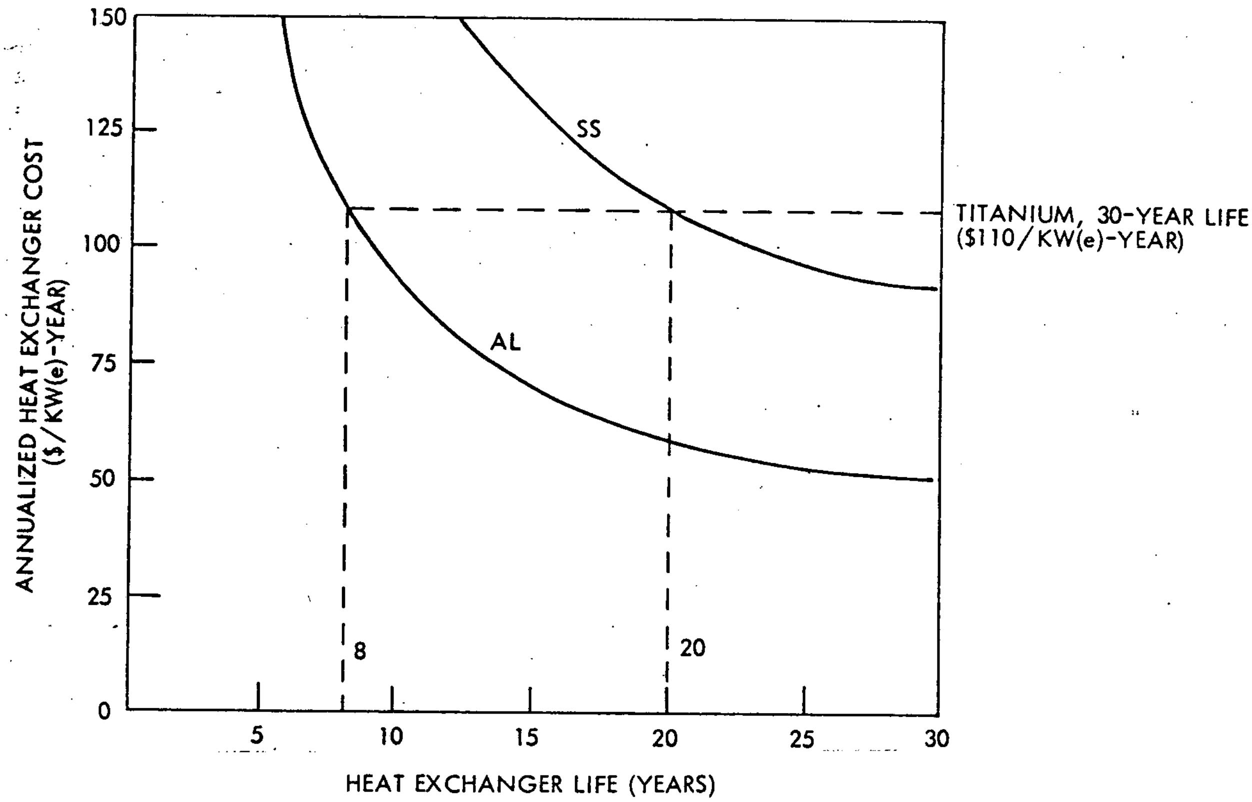


Fig. 4-16 Heat Exchanger Annualized Cost

A brief sensitivity analysis was conducted to determine the impact on the breakeven lives discussed above in terms of changes in the assumptions. To this effect, assumptions concerning heat exchanger replacement cost, material scrap value, module downtime, and cost of money and inflation rates were changed separately, and the new breakeven lives were calculated for aluminum- and stainless-steel-tubed heat exchangers vis-a-vis titanium-tubed units with a 30-year life. The results of the sensitivity analysis are summarized in Table 4-12. Clearly, breakeven lives are remarkable insensitive to changes in the variables considered; in no case do they change more than 15 percent, while changes in independent variable values range as high as 200 percent. While this insensitivity may at first sight seem surprising, it is simply the consequence of heat exchanger costs dominating, since in themselves they are much greater than, for example, replacement or downtime costs.

Table 4-12

BREAKEVEN LIFE SENSITIVITY

Paremeter	Initial Value	New Value	SS/Ti Breakeven Life (years)	Al/Ti Breakeven Life (years)
Heat Exchanger Replacement Cost	$1M	$2M	21.2	8.8
Heat Exchanger Material Scrap Value				
– Steel	2¢ /lb	4¢ /lb	19.7	–
– Aluminum	10¢ /lb	20¢ /lb	–	7.7
Module Downtime	30 days	90 days	20.5	8.5
Cost of Money	10%	8%	21.2	8.7
Escalation Rate	6%	5%		

4.4.8 Material Consideration Summary

An OTEC heat exchanger with Aluminum 5052 alloy tubing 0.075-in. wall thickness with or without alclad can provide more than 10 years life with a high reliability (less than 1 percent tube failure due to corrosion). The aluminum heat exchangers with a 10-year life are the most cost effective heat exchangers for the 30-year life power system. In addition, the aluminum heat exchangers have the least weight and smallest volume for a given size of the candidate materials evaluated.

In addition to the drawn aluminum tubing used in the heat exchanger comparison study, rolled and welded 5052 aluminum tubing is considered a candidate method for reducing costs of the heat exchangers incorporating heat transfer enhancement surfaces. While the ASTM standard for rolled and welded 5052 aluminum tubing is not part of the ASME pressure vessel code, the code allows use of various materials for unfired pressure vessels when margins and safety factors are incorporated into the design. The necessary analyses, manufacturing development, and tests will be accomplished to ensure that formed, rolled, welded, and treated 5052 aluminum tubes will provide the performance and life required.

Ammonia piping and valves made from carbon steel will be used throughout the system for cost and ammonia compatibility reasons. Carbon steel to aluminum joints will be designed to be in a controlled environment wherever possible with isolation and sacrificial anodes used as required.

4.4.9 References

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4.5 BIOFOULING CONTROL

The fouling mechanism of greatest concern is from the warm water passing through the evaporator. No comparable large-scale experience is known to exist, and although several scaled tests have been initiated there is a lack of specific data on which to base a design. A conservative approach therefore was taken to ensure power module performance. The application of biofouling control measures should consider the entire seawater system. As an example, if no biofouling control measures are applied to the platform, the water ducts in the platform could act as a breeding area for the micro-organisms (which form the slime layer) and the macro-organisms (visible organisms such as barnacles, worms, and mussels) that would continuously be released into the seawater flow going to the heat exchangers. The use of antifoulant coatings on the platform and water ducts must also consider the heat exchanger materials. However, the PSD project is concerned only with the power module, and the selected solution is based primarily on the power module configuration and operating condition with secondary consideration given to the platform.

4.5.1 Fouling Condition

The nature and extent of biofouling in the evaporator and condenser is based on the accepted assumptions that the fouling layer is essentially water which has been immobilized on the heat transfer surface by the micro-organism structure or slime layer. A second fouling which can occur under specific conditions is that of corrosion product fouling. Such effects, not fully understood today, will be avoided by minimizing leaks, following operational procedures defined to minimize fouling, and the use of coatings in upstream water ducting.

The limited test results indicate a 0.002-in.-thick slime layer corresponds to a fouling heat transfer resistance factor of about 0.0005 hr-ft^2-°F/Btu. The rate of increase of heat transfer resistance (slime layer build up) shown in Fig. 4-17 is a function of water velocity and tube surface condition. From the data points, slopes have been drawn, and for the countermeasure design condition it is assumed that the thermal resistance will

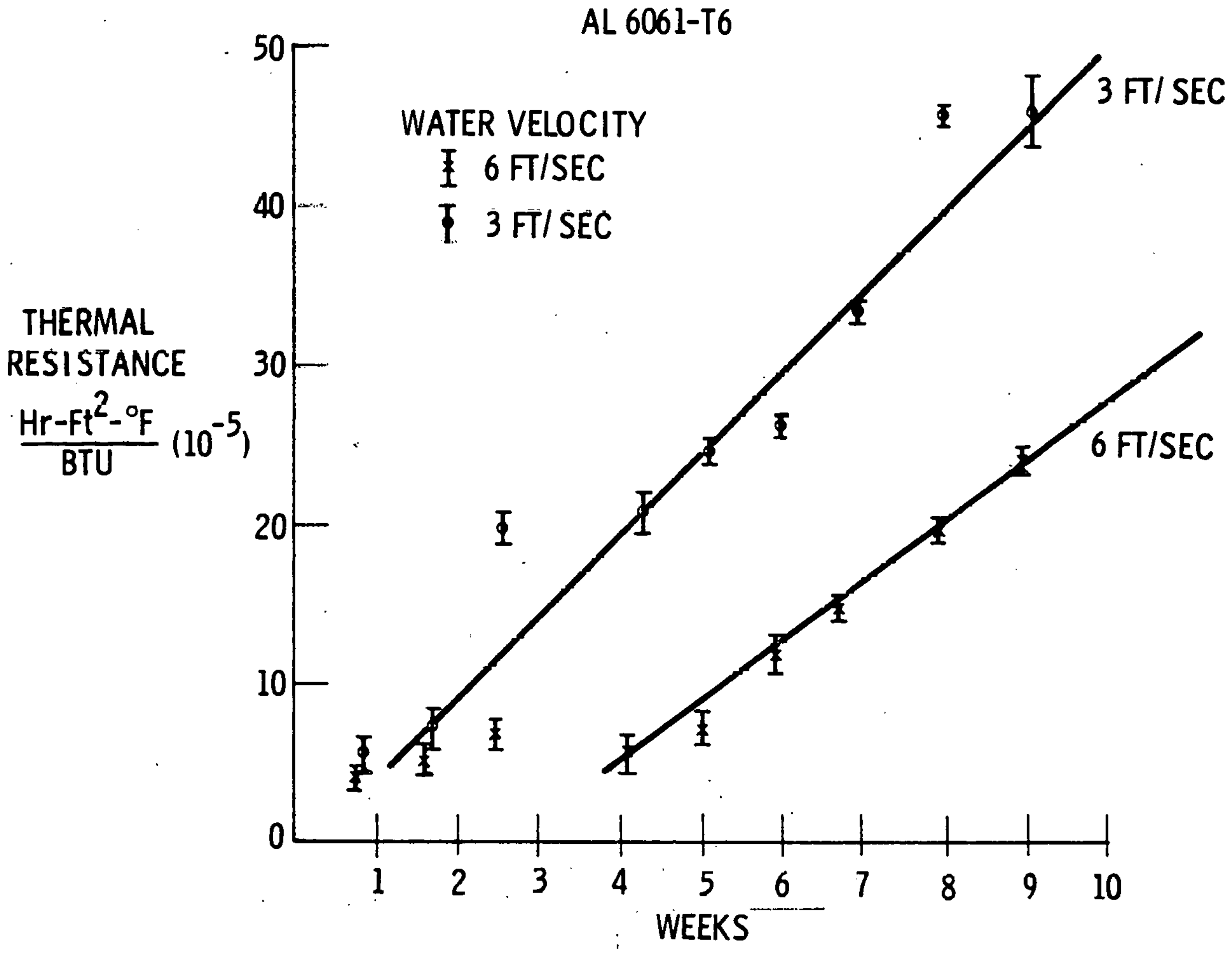


Fig. 4-17 Biofouling Resistance Buildup

drop no lower than 0.0001 hr-ft^2-°F/Btu and will increase at the rate of 0.00004 hr-ft^2-°F/Btu per week. The effect of fouling on module performance (power module net power output) is shown in Fig. 4-18. From this figure it is found that an increase in the fouling resistance factor of one ten thousandth (from 0.0002 to 0.0003 as an example) reduces the net power by 1 MW(e). At a cost of power of 4¢/kWh, this increase represents a cost of $960/day or approximately $350,000/year. The importance of controlling the biofouling layers is demonstrated by the relative thermal performance capability of the candidate heat exchanger materials. As an example, with an aluminum heat exchanger an increase in the fouling resistance from 0.0001 to 0.0003 for the baseline case (sea-water film heat transfer coefficient of 1000 and an ammonia heat transfer coefficient of 3000) causes an 11 percent reduction in the overall heat transfer coefficient.

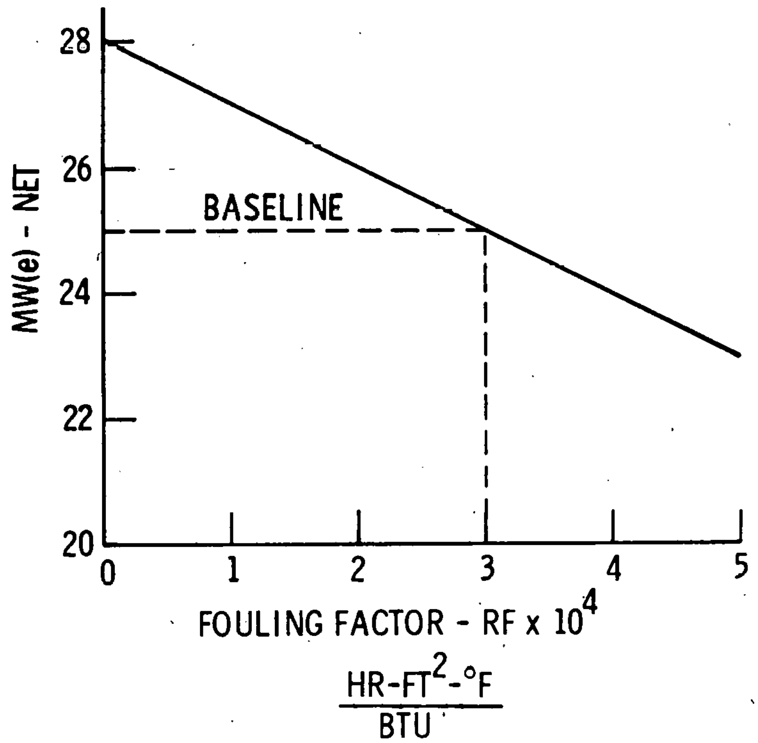


Fig. 4-18 Effect of Fouling Factor on Module Performance

A second type of fouling may also arise from corrosion products. It is possible for corrosion products formed at an upstream point in the fluid handling system to be deposited on the heat transfer surface. To minimize this form of fouling, proper selection of materials for the seawater systems is mandatory. Where feasible, materials and material coatings will be selected and integrated with the platform studies to ensure material compatibility.

4.5.2 Fouling Countermeasures

A variety of countermeasures have been proposed to control fouling of OTEC heat exchangers. The countermeasures and their application are discussed in the following paragraphs.

Chlorination. This is the best known countermeasure with considerable background in other marine-related systems. The biofouling problems encountered in condenser tubes of seawater-cooled power plants are frequently severe. Of the seawater-cooled power plants surveyed in California, Texas, and Florida, chlorination was by far the most universal treatment method employed to combat biofouling. The chlorination treatments administered varied in frequency and intensity according to the severity of the problem, the biological demand, and the power company's ability to optimize chlorine treatment.

Typically, chlorination treatments consist of injecting chlorine into the water supply until the total residual chlorine concentration at the downstream end of condenser tubes is 0.2 to 0.5 ppm for 20 minutes, or until free chlorine can be detected at the same point. A 20-minute exposure to 0.2 ppm chloramine or an instantaneous exposure to free chlorine is considered adequate to kill slime. An example of chlorine treatment (used at San Onofre, California, Unit 1) is 0.2 ppm total residual chlorine for 20 minutes, three times a day. With this treatment, a condensing temperature of 70° F is normally maintainable. The use of chlorination may pose an environmental problem. An alternative is a closed-loop chlorination soak cycle.

Based on available data, three 20-minute periods per day using between 1.0 to 2.5 ppm of chlorination with on-board generation of sodium hypochlorite will cost between $140,000 to $1,200,000 for equipment and between $18,000 and $152,000 per year for operation and maintenance based on 3 kWh/lb chlorine and 4¢/kWh power cost (12¢/lb chlorine and 3¢/lb chlorine for maintenance).

Ozonation. Ozonation is less effective than chlorination in reducing slime film. Capital equipment expense and operating costs are much greater than for chlorination. Power consumption required to generate ozone for moderate treatment is five to seven times that of chlorino.

Copper or Copper Alloy Tubes. Copper is a known passive antifouling material; but because it is incompatible with anhydrous ammonia, a bimetallic tubing was costed. The tubing cost was approximately 75- to 150-percent greater than an equivalent diameter single metal smooth tube.

Static Inserts. Inserts in the form of helical wire springs to create a helical flow path can provide net benefits in the form of heat transfer enhancement. Use of copper-bearing materials for such inserts can provide some antifouling advantages. These inserts would be installed in each tube on the basis of a periodic (3 to 5 year) replacement cycle, and would be used only with compatible heat exchanger materials. A quick evaluation of the effectiveness in either controlling or removing biofouling is questionable.

Dynamic Inserts. This class of device is related to the static insert but has the potential for simultaneously accomplishing two additional functions. The configuration consists of a twisted flat ribbon slightly narrower than the inside diameter of the tube. The ribbon is allowed to rotate freely in reaction to the water flow by means of a retention thrust bearing at its upstream end (at the tube entrance). The ribbon, even if neutrally buoyant, would wear on the tube wall. The ribbon disturbs the wall boundary layer and any incipient slime layer once every half revolution. With a 0.002-in.-thick layer giving a heat transfer resistance factor of 0.0005 hr-ft^2-°F/Btu, the dynamic insert effectiveness is questioned.

Velocity. High water velocity results in fully developed turbulent flow, which is characterized by a nearly uniform velocity profile across the tube section. The latter creates a high liquid shear stress at the wall, particularly for smooth surfaces. As surfaces become roughened (from corrosion, sedimentary deposits, or crystalline deposits), wall shear stress decreases, and microscopic zones of low velocity become focal points for biological attachment and development of a slime layer. The mechanism by which water velocity has an antifouling effect is the prevention of the organism from stabilizing its position for a sufficient time to secrete an attachment bonding agent. It appears that, to be thoroughly effective, liquid velocity as an antifoulant requires the use of a corrosion-resistant tube surface. The use of a periodic high velocity water jet in removing the slime layer incorporates two poor features. It requires compatibility of heat exchanger materials for the erosion/corrosion aspects and the mechanization of such a system with its attendent high power consumption.

Thermal Shock. Hot-water thermal-shock antifouling techniques are widely used for condensers in coastal steam plants. In these plants the method involves recirculating part of the cooling water until a temperature of about 120°F is reached and held for a about one-half hour. The warmest available heat source for OTEC is 85°F and to apply higher temperatures for even a few minutes in an OTEC plant may require an excessive amount of power. Cold thermal shock is an effective biofouling countermeasure. This method of cooling the evaporator uses the working fluid (e.g., ammonia) as the coolant. Seawater flow through the evaporator is stopped and the ammonia vaporization process continued until all the ammonia remains in a liquid state, having equalized the temperature of the stationary evaporator intake seawater with that of the cold water condenser coolant, i.e., approx. 40° to 45°F. After a suitable time at this reduced temperature, the normal flow paths of the working fluid and intake water are restored and power generation resumed. Ammonia pump power would be taken from auxiliary power or another OTEC module. An investigation into the mechanization and time durations required for cooling the evaporator deleted this approach.

<u>MAN Shuttle Brush Tube Cleaning System.</u> This commercial antifouling system is proprietary to the American MAN Corporation. It has been widely used for antifouling of shell-and-tube refrigeration heat exchangers, and to a lesser extent for seawater-cooled steam condenser cleaning. The active element of the system is illustrated in Fig. 4-19. A small nylon or "RILSAN" bristle brush is propelled through the tube

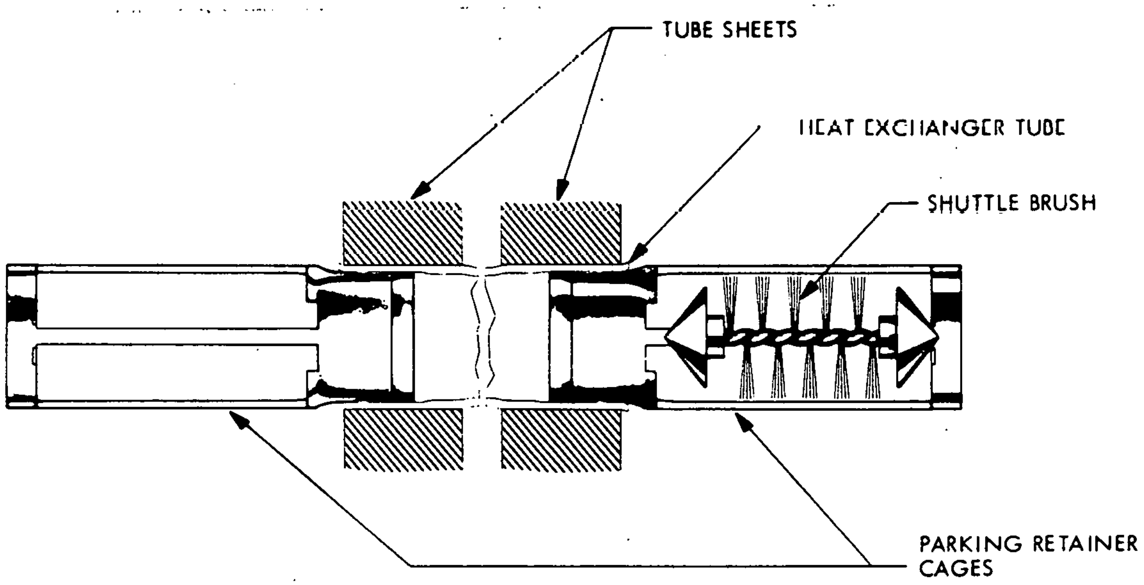


Fig. 4-19 MAN Shuttle Brush Tube Cleaning System

by the normal water flow and is captured by a pair of plastic retainer cages, one at each end of the tube. After traversing the tube, the brush is parked in the downstream position. The cleaning process requires periodic reversal of the water flow. With the single-pass heat exchanger, the reverse flow is implemented by the use of reversible rotation pump motors. The bristle stiffness can be varied to minimize the potential for tube surface erosion. Tests will be conducted to define a bristle configuration which provides the desired cleaning effectiveness with no attendant erosion problem.

The MAN configuration will use a female cage which snaps over the end of the heat exchanger tube (the heat exchanger tube will extend 3/4-in. past the tubesheet).

A pressure loss, due to the brush and cages, of 1 ft of water was estimated based on MAN data. The brush cycle has been established as four passes (based on reversing the pump rotation twice), requiring 15 minutes once a day. The brush life has a predicted 5000 pass life giving a 3 year brush/cage life. Equipment and installation is estimated at $400,000 with a $300,000 per year operating/maintenance cost.

Amertap. This system is the second major commercially available tube-cleaning system. It is based on a recirculating tube-wall contact element traveling through the tube and propelled only by the water flow itself as indicated in Fig. 4-20.

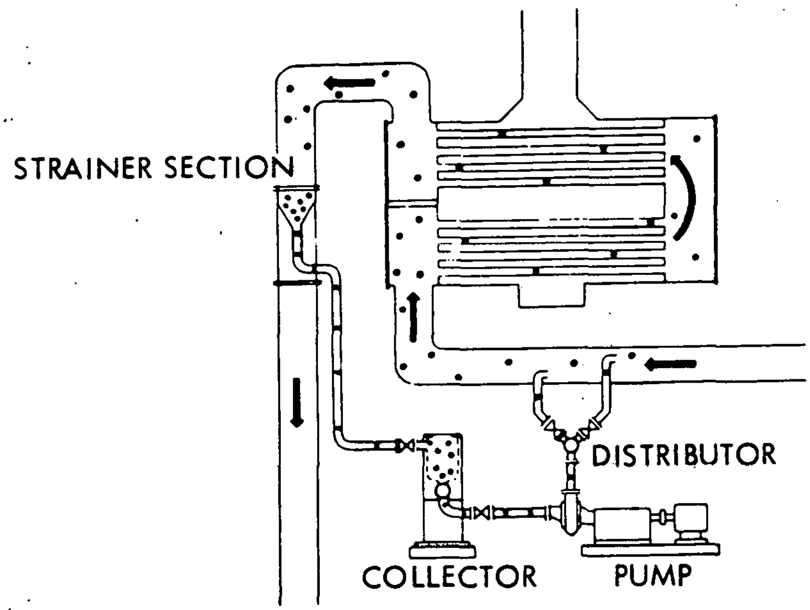


Fig. 4-20 Amertap Tube Cleaning System

Two approaches have been evaluated for the distribution and collection of the Amertap balls. The first follows the current power plant approach of random dispersion at the upstream side and screen collection on the downstream side of the heat exchanger. This approach was dropped because of the cost for the collector. The second approach follows that previously given in the June 1975 OTEC report which uses indexed arms to disperse and collect the balls. The arm mechanisms with an estimated 10-year life will cost approximately $2,000,000, use 640 balls in the circulation loop, rotate across the tubesheet once every 24 hours, and feed about three balls to each tube on each rotation. Each ball will last from 4 to 6 weeks. Replacement cost is estimated at 40 cents per ball. This system of course requires development. It is estimated that the total cost of equipment is $2,000,000 for 10 years life, plus installation and as yet undefined effect on the heat exchanger structure for the indexing and rotating equipment mounting. The yearly operational cost is estimated to be $360,000 using a 30-hp rotating arm device, a 15-hp ball recirculation and cleaning system and 1-h/day maintenance.

4.5.3 Fouling Control

Because of the lack of specific test data on biofouling rates and countermeasures, we have elected to use two systems for biofouling control – daily chlorination and MAN brush operation. The chlorination is to be injected three times per day 20 minutes each, at or very near the seawater inlets to the platform. This will help protect the platform ducts as well as the power module seawater pumps, ducts, and heat exchanger. The dosage rate and duration will be defined during preliminary design based on biofouling test results. Dosage estimates are from 0.2 mg/l to 1.6 mg/l. The MAN brush system was selected for cost and simplicity. The exact level of biofouling heat transfer resistance will be defined in preliminary design based on test results.

4.6 ROTATING EQUIPMENT

This section discusses the rotating machinery required for the operation of the power module. A compilation of the related data is presented at the end of this section following the format of Tables B and C of the DOE letter dated Jan 6, 1978.

4.6.1 Seawater Pumps

The sizing of the seawater pumps and pump motors is dependent on the seawater system components as well as the heat exchanger seawater flow rates. The seawater subsystem components which must be considered in the determination of pressure drop include:

- Cold water pipe (platform)
- Inlet plenum (platform)
- Pump installations
- Heat exchangers

In addition, inlet closure valves will be provided to permit closure of individual pumps. These valves will be included as part of the platform seawater subsystems and, if butterfly valves are used, the valve pressure drop must be included.

The seawater pump performance requirements are dependent upon the type of platform selected. The optimized installation requires a lower gross generator power due to reduction in ammonia and warm seawater pumping power requirements. The heat exchangers are slightly longer and the seawater flow rates are slightly less. However, the increased length of the total flow path results in higher pressure drops, particularly in the cold seawater system. The Design Requirements included approximate pump head requirements of 10 ft for the warm seawater subsystem and 15 ft for the cold seawater subsystem. For the power system modules, developed in this conceptual study, the flow rates and head requirements are as shown in Table 4-13.

Selection of the number of pumps for each seawater subsystem requires evaluation of the economic aspects and installation effects. Preliminary information received from KSB, Hitachi, Mitsubishi, Allis-Chalmers, Worthington, and Pleuger indicate that

Table 4-13
SEAWATER PUMP FLOW RATES AND HEAD REQUIREMENTS

Item	Configuration		
	Baseline Reference	Optimized	
		Surface	Submerged
Warm Water Pumps			
Flow (cfs)	3,590	3,450	3,600
Head (ft)	10	8	7.6
Cold Water Pumps			
Flow (cfs)	3,640	3,525	3,710
Head (ft)	15	9.5	8

there is little economy of scale in pumps sized to provide 1,000 to 4,000 cfs. The installation, therefore, has a major influence on the selection of the number of pumps. Two factors to be considered are the physical installation and the effect of pump failure on performance.

With regard to the latter, the off-design performance program was run at reduced seawater flow rates corresponding to the loss of a pump in one system and in both systems. The results are shown in Fig. 4-21. As shown, the loss of one pump in either the warm or cold subsystem has the same effect on performance. With four pumps installed, a pump loss in one subsystem results in a performance degradation of approximately 4 percent, and a pump loss in both subsystems results in a performance loss of approximately 9 percent. With three pumps installed, these performance losses are increased to 9 percent and 18 percent, respectively. The use of variable flow pumps can reduce these losses. The loss of the single pump installation obviously reduces the power output to zero.

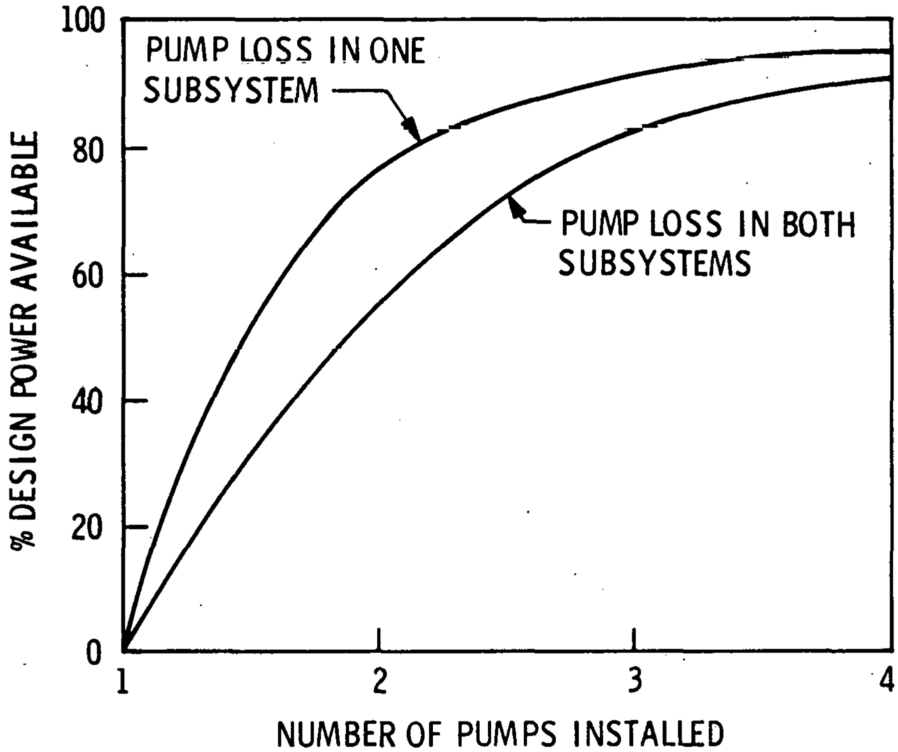


Fig. 4-21 Operational Capability With Seawater Pump Loss — Constant Flow Pumps

With regard to the physical installation, use of a single pump would require an impeller 22 to 24 ft in diameter with a related overall pump assembly length of 75 to 80 ft. A two-pump installation would reduce this length to 53 to 57 ft, but the side-by-side pump arrangement would require a long transition section to the heat exchanger face. The three- and four-pump installations are shown in Fig. 4-22 to indicate relative size effects. The maximum diameter of the pump sized for the four-pump installation permits installation of the pumps within the diameter of the heat exchanger front face. The overall diameter of the three-pump installation is greater than the heat exchanger front face diameter. In addition, a longer transition section is required to provide good flow distribution at the heat exchanger. The four-pump installation is, therefore, recommended for the 25-MW(e) power system module.

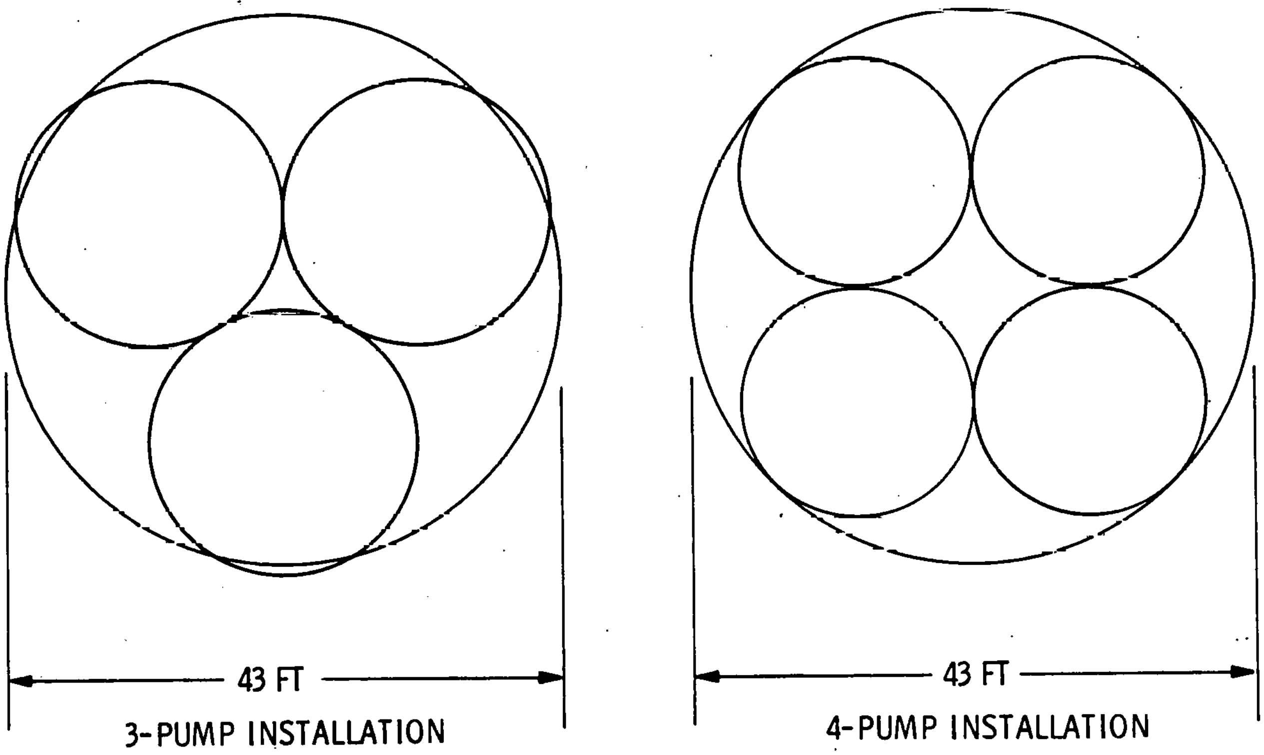


Fig. 4-22 Seawater Pump Installation for 25-MW(e) Module

The desirability of incorporating variable flow pumps is dependent upon the thermal resource variation (site sensitivity) and the cost for obtaining the flow variation (variable speed or pitch). From the data of Table 3-4, it is seen that performance gains of 1 percent or more can be realized at all sites and design paramotors. In addition, the off-design performance data of Fig. 3-8 indicate a significant improvement in self-sufficiency with optimized seawater and ammonia flow rates. Data provided by Pleuger Submersible Pumps (Appendix I-1) indicate that 10 percent speed variation can be obtained by varying voltage only. Regulation to 20 percent speed can be obtained by maintaining a constant voltage/frequency ratio using adjustable frequency ac motor speed controls. This approach has no impact on the pump costs and the cost of the speed control equipment must be evaluated versus the gain in available performance and the desirability of maintaining plant self-sufficiency to the lowest potential temperature differential. A third factor to be considered is the ability to increase the speed of the remaining pumps in the event of pump failure, thus maximizing power output.

Considering all three items, it is recommended that variable flow pumps be incorporated into the design.

Pump speed control is maintained by appropriate algorithms in the main plant power control system. Based on inputs from the warm and cold seawater temperature sensors, the control adjusts the voltage to the pump motors to predetermined values and the net power output is measured. Perturbations from these predetermined values are then performed in a specific pattern to maximize measured net power out of the module. For example, the warm water pump speed is increased by 1 percent and net power out measured. If the net power is increased, the cold water pump speed is increased and the process repeated until the maximum power is attained. If the power is decreased, the pump speed is decreased in a similar procedure to maximize power output.

Appendix I-1 presents data provided by the companies noted above. Costs, for the four-pump configuration ranged from approximately $600,000/pump to $1,840,000/pump. The lower value has been used in the cost analysis of Section 5.

4.6.2 Turbine Generator

The turbine sizes associated with the 25-MW(e) Power System Development effort do not provide an a priori selection of the type of turbine best suited for this usage. It was therefore decided to provide subcontracts to General Electric Company and Rotoflow Corporation for conceptual design studies of the axial flow and radial inflow turbines, respectively. The requirements of the study were to provide a family of turbine sizes to provide 25-MW(e) (net) or 32 to 35 MW(e) (gross) with unit sizes of 5-, 12.5-, and 25-MW(e) (net). The factors to be considered in evaluating the design include:

- Cost
- Efficiency
- Size
- Weight
- Reliability
- Off-design performance

<u>Mechanical Design Considerations.</u> The turbine subcontractors were requested to consider the following items in arriving at their recommended design configurations:

- 1800 vs. 3600 rpm (size and weight)
- Materials (ammonia compatibility)
- Seal arrangements (prevent contamination)
- Orientation (horizontal vs. vertical)
- Producibility (size limitation)
- Safety (control and shutdown)
- Reliability

The Rotoflow Corporation recommends the use of 3,600-rpm turbine-generator units. This approach requires the use of multiple wheels with the radial inflow turbine having four wheels in two cases driving the generator from both ends. The factors used by Rotoflow in arriving at this recommendation are shown in Table 4-14.

The General Electric Company recommends use of the 1,800-rpm turbine in order to maintain a reasonable bucket height/pitch diameter ratio (less than 0.25) with a dual-wheel installation.

Aluminum and steel have been selected as the materials that are compatible with the ammonia. Figures 4-23 and 4-24 show the sizes and weights of the turbines proposed

Table 4-14

FACTORS FOR SPEED SELECTION

RPM	1,800	3,600
Number of Wheels	1	4
Number of Cases	1	2
Castings Cost	$\$1 \times 10^6$	$\$0.5 \times 10^6$
Patterns Cost	\$136,000	\$67,000
Castings Weight	32,000 lb	25,000 lb
Size (typical)	232 in.	116 in.
Spare Unit	30,000 lb	12,000 lb
Lead Time	28 months	21 months

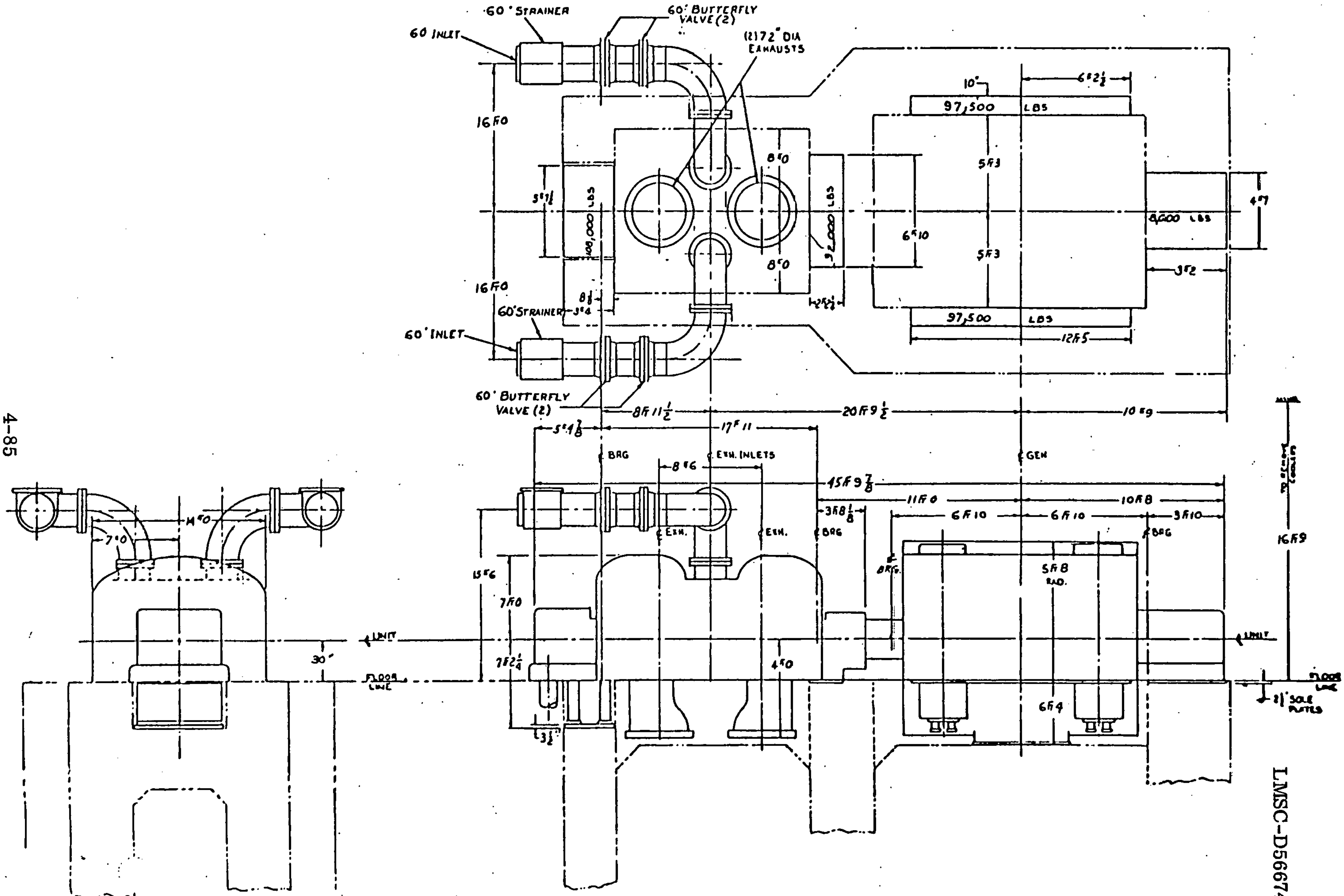


Fig. 4-23 Outline of Proposed General Electric Turbine

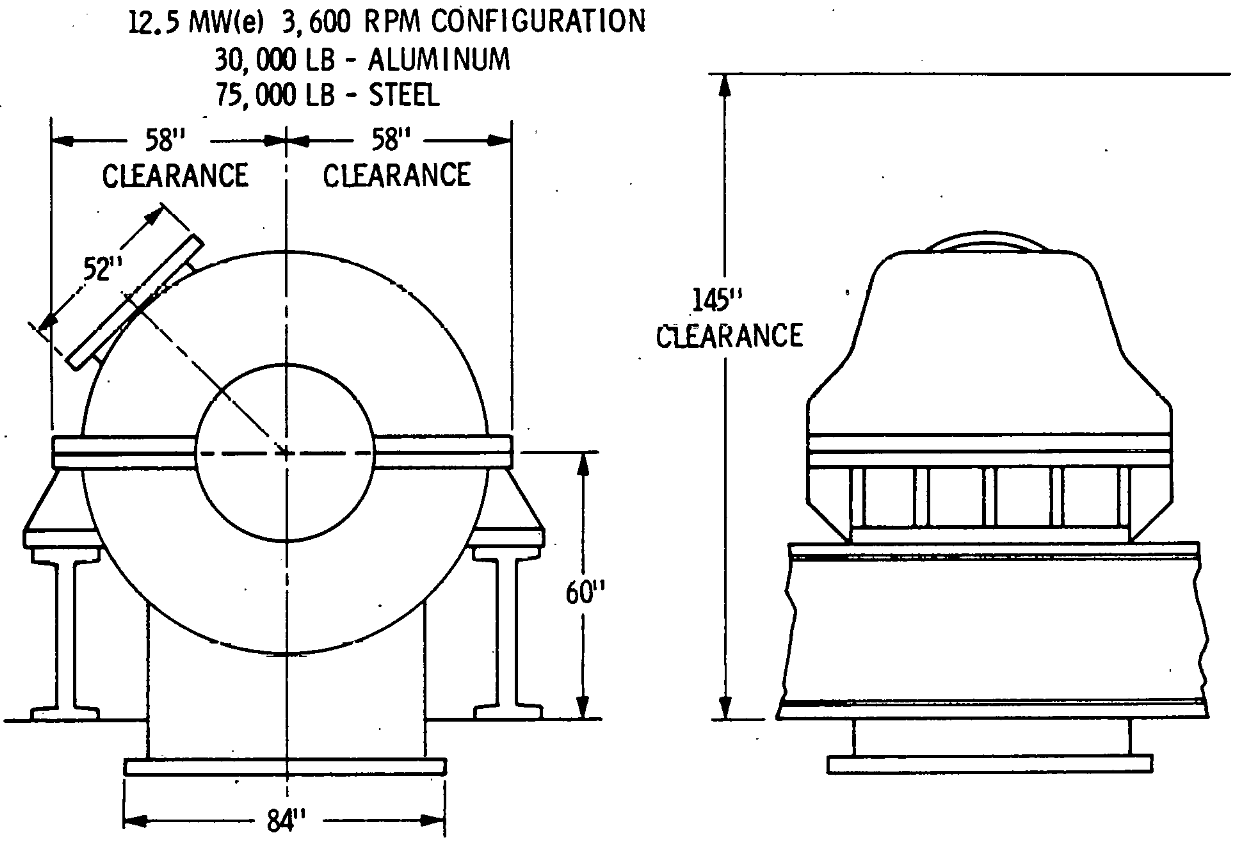


Fig. 4-24 Radial In-Flow Ammonia Turbine

by General Electric and Rotoflow, respectively. The General Electric installation is steel while the Rotoflow installation shows weights for both aluminum and steel. The materials selected for the aluminum case and wheel installation are shown in Table 4-15.

Table 4-15
MATERIALS

Cases	Cast Aluminum Alloy, C355
Wheel	K.01 Aluminum
Shaft	17-4 P.H. Stainless
Bearings	Babbited Bronze
Seals	Aluminum C355

The seal arrangement proposed by General Electric incorporates a zero leakage, floating ring concept which has been used successfully for over 35 years and has accumulated millions of hours of satisfactory service on hydrogen-cooled generators. Rotoflow proposes to use a dual labyrinth seal with the volume between the seals pressurized with nitrogen to a value slightly above the working fluid pressure. In either case, contamination of the lubricating fluid by the ammonia can be eliminated by selecting a compatible fluid such as Mobil Flowrex or Gargoyle and incorporating a means of separating any entrained ammonia.

With regard to turbine axis orientation, both subcontractors have indicated a preference for the horizontal installation. General Electric has made vertically oriented turbine-generators in the past and discontinued this approach primarily due to maintenance problems.

The size of the turbine hardware is such that the current fabricating, heat treating, handling and machinery equipment at both the General Electric Lynn River Works plant and at facilities available to Rotoflow are satisfactory for manufacturing the conceptual designs depicted.

The control and shutdown of the two designs are different in that the radial inflow turbine incorporates inlet nozzles which act as both the control and emergency shutdown systems while the axial flow turbine uses two butterfly values in series, one for speed control and one for emergency shutdown. The control logic is the same for both systems. Actual turbine speed, sensed by a permanent magnet generator on the rotor and related pick-up, is compared with a speed reference and the appropriate signal is given to the hydraulic power unit to position the control system. Excessive vibration as well as turbine overspeed will actuate the emergency shutdown system.

The multiple nozzle system for the radial inflow turbine has little inertia and can be closed very rapidly. A typical time history of a shutdown due to overspeed is shown in Fig. 4-25. An acceleration sensor opens the nozzle actuator vent at 0.025 sec after load loss. The system reaction starts to close the nozzles at 0.1 sec with complete

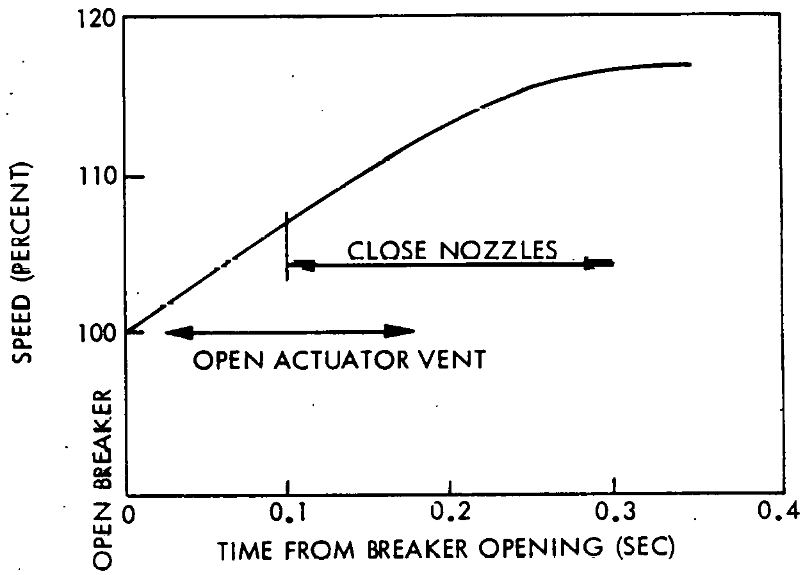


Fig. 4-25 Shutdown Capability – Radial Turbine

closure at 0.3 sec. The overspeed is limited to approximately 17 percent, which is well within the overspeed safety limit for the turbine.

Reliability of the turbine-generator can be evaluated from the data of Table 4-16. The General Electric data were obtained from 86 units reporting a total of 3,620,774 operating hours over the 7-year period. Reported forced outage hours amounted to 14,899 hours or 0.4 percent. Of these, 87 percent were due to steam path deposits (50 percent), water ingestion (21 percent) and lube oil supply (16 percent). Since the first two are not pertinent to the ammonia system, the outage factor should be reduced to approximately 0.1 percent. The Rotoflow data are for approximately 600 units and the small outage rate is due to the fact that a change-out requires from 4 to 8 hours. It is anticipated that, on the machine depicted, change-out and system restart will be accomplished in less than 12 hours.

Table 4-16

TURBINE RELIABILITY DATA

Item	General Electric	Rotoflow
Survey Period	Oct '67 – Oct '74	20 yr
Total Operating Hours	3,620,774	50×10^6
Forced Outage Rate (all causes)	0.4 percent	0.01 percent

<u>Design/Off Design Performance.</u> The ammonia state points provided to the subcontractors for the design of the turbine-generator family are as shown in Table 4-17. The isentropic head available under these conditions is 17.67 Btu/lb. Data from General Electric indicate that 8,086,000 lb/hr of ammonia are required to provide the 32.25-MW(e) gross power associated with the 25-MW(e) (net) design point. This indicates that the design point efficiency is 77 percent as compared with the 87 percent efficiency guaranteed for the radial inflow turbine. The effect of this increment in efficiency on the heat exchanger size to maintain the 25-MW(e) (net) is shown in Table 4-18. The incremental cost of the heat exchangers is estimated to be 18 to 20 percent. In addition, the vapor and condensate lines and the condensate pumps have to be increased to accommodate the 15 percent increase in ammonia flow.

Table 4-17

TURBINE DESIGN PARAMETERS

Item	Inlet	Exit
Pressure (psia)	131.1	94.26
Temperature (°F)	71	52.9
Quality	0.99	–

Table 4-18

COMPARATIVE TURBINE PERFORMANCE EFFECTS

Item	Radial	Axial	Percent Difference
Design Efficiency	0.87	0.77	13
Tube No. Required			
– Evaporator	78,014	81,915	5
– Condenser	77,924	81,820	5
Tube Lengths			
– Evaporator	42.5	50.1	18
– Condenser	45.7	54.2	18
Ammonia Flow (lb/sec)	1,960	2,250	15

Off-design performance for the radial inflow can be obtained by using the curves of Fig. 4-26 to obtain correction factors as functions of the off-design ammonia flow ratio and isentropic enthalpy ratio. The equations shown on that figure were incorporated into the off-design computer programs to obtain the results discussed in Section 3. The correction factors for the axial flow turbine are presented in Fig. 4-27. The off-design performance program is being modified to accept these correction factors.

Turbine-Generator Costs. In developing the costs for the families of turbine-generators, the subcontractors used combinations of number of wheels and wheel sizes which are most cost-effective. The results, shown in Fig. 4-28, indicate a significant difference in the axial and radial turbine-generator costs and trend. There is still considerable economy of scale for the axial configuration above 25-MW(e) (net). For the radial turbine, there is little economy of scale above 12.5-MW(e) since the same wheel size is used and the reduction in unit cost is due to a slight economy of scale for the generator. From a relative cost standpoint, the axial installation is twice as costly at the 25-MW(e) design condition.

4.6.3 Ammonia Pumps

The ammonia condensate system is configured to provide an optimum approach to the selection of pump sizes. The primary condensate system, described in Section 4.7, incorporates these pumps, each of which can provide 50 percent of the design flow rate. This approach ensures that a pump failure will not cause a reduction in performance. This system directs the flow from the condenser to the evaporator sump, which provides mixing of the cold liquid with the warm residual in the sump. With a design evaporator supply rate equal to 1.5 times the turbine design flow rate, three pumps, of the same capacity as the primary condensate pumps, can supply the total flow. A fourth pump is incorporated into the system to provide redundancy and ensure no performance degradation in the event of pump failure. As noted in Table B, the motor sizes are different for the two classes of pumps. The primary condensate pumps have to deliver flow against a head of approximately 52 psid while the maximum head requirement in the distribution system is 15 psid.

CRIGINAL USED
IN PSD-II

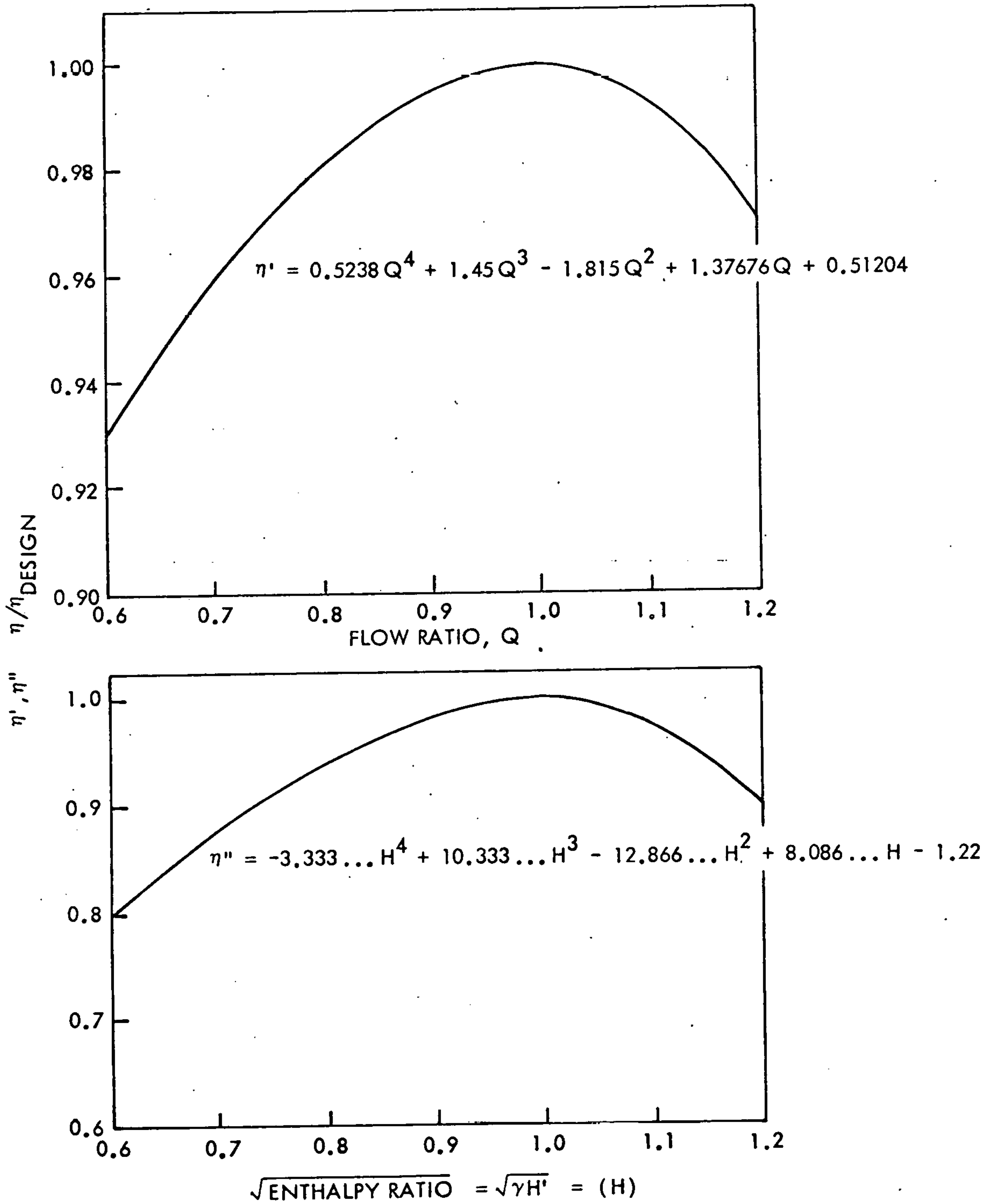
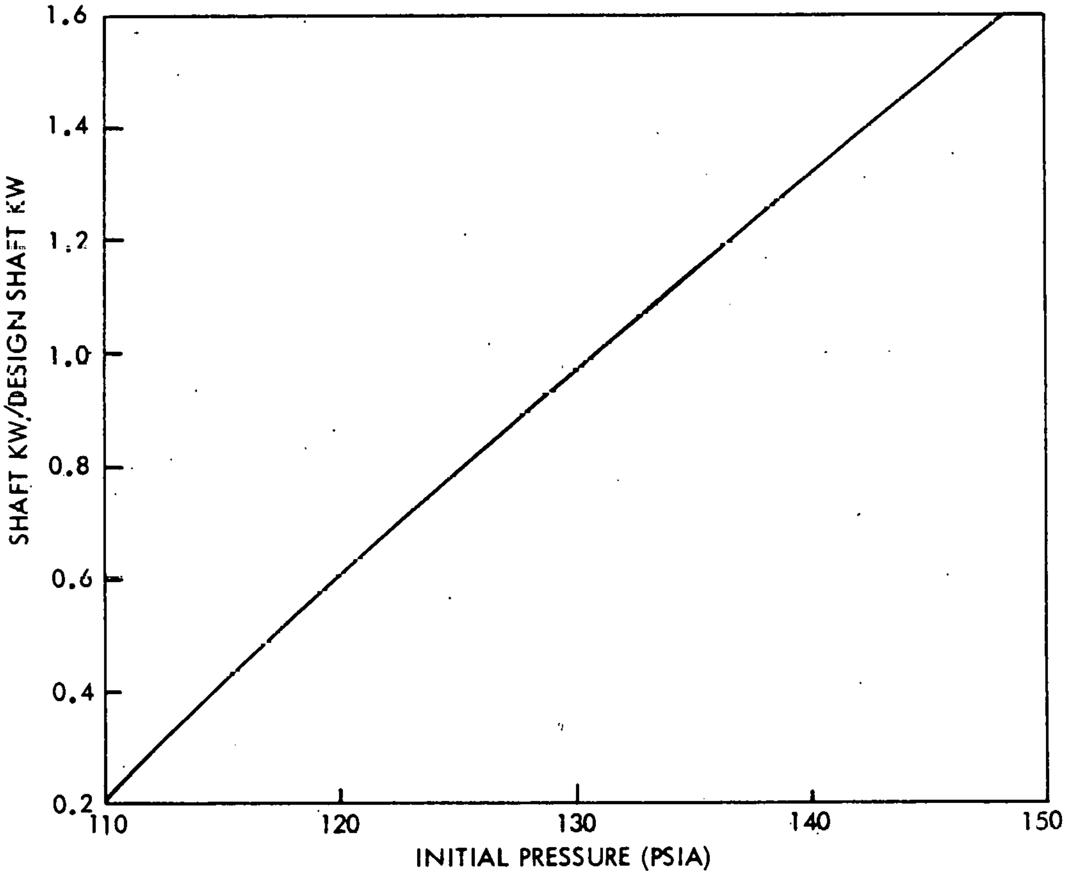


Fig. 4-26 Off-Design Performance – Radial Turbine



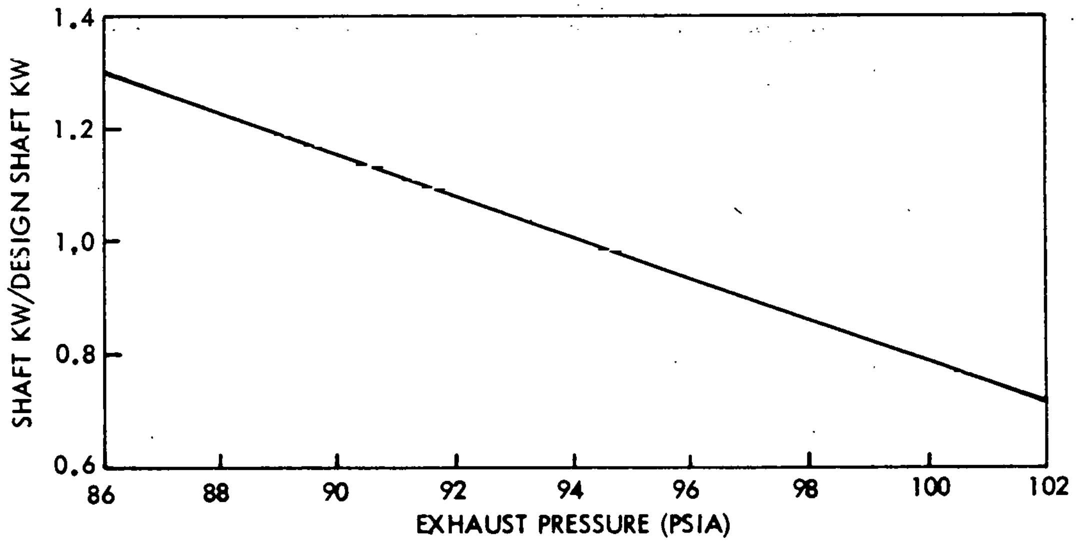


Fig. 4-27 Off Design Performance for 25-MW(e) Module – 131.1 psia, 1 Percent Moist., 94.26 psia

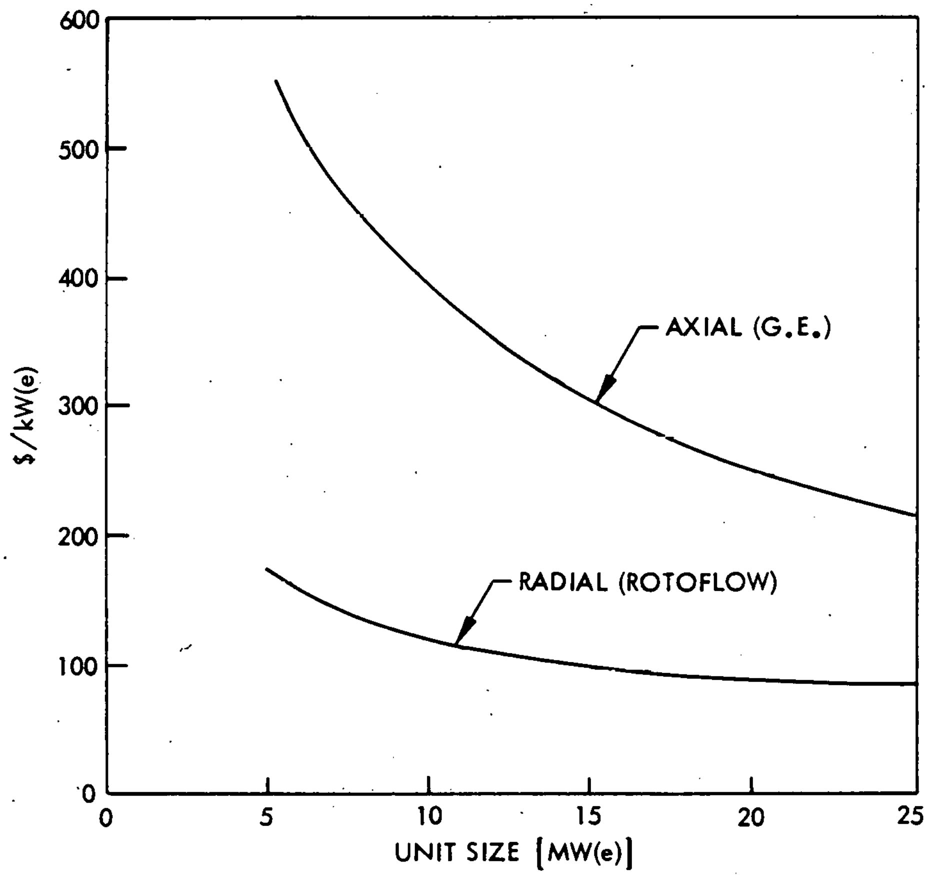


Fig. 4-28 Turbine Generator Unit Cost vs. Size

Table B

ROTATING MACHINERY (PER BASELINE MODULE OF 25-MW(e) (NET)

Item	Number	Type	RPM	Horsepower	Efficiency	Comments
Turbine and Diffuser	2	Radial Inflow	3,600	–	0.86	18.57 MW each
Generator	1	AC	3,600	–	0.985	
Warm Water Pump	4	Axial	100 ± 10%	–	0.86	
Warm Water Pump Motor	4	Submerged	870 ± 10%	1,500	0.90	
Cold Water Pump	4	Axial	100 ± 10%	–	0.86	
Cold Water Pump Motor	4	Submerged	870 ± 10%	2,250	0.90	
Ammonia Feed Pump	3	Centrifugal	–	–	0.90	
Ammonia Feed Pump Motor	3	–	1,200	600	0.90	
Ammonia Distribution Pump	4	Centrifugal	–	–	0.90	
Ammonia Distribution Pump Motor	4	–	1,800	75	0.90	

Table C

FLOW RATES AND CONDITIONS AT DESIGN POINT OPERATION

Item	Reference Baseline	Optimized	
		Surface	Submerged
TURBINES			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
INLET			
Pressure (psia)	136.9	136.9	137.5
Temperature (°F)	73.49	73.49	73.74
Velocity (fps)	150	150	150
Quality (%)	99	99	99
OUTLET			
Pressure (psia)	95.94	96	95.93
Temperature (°F)	53.84	53.87	53.83
Velocity (fps)	75	75	75
Quality (%)	96.5	96.6	96.5
PUMPS			
AMMONIA FEED			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
Pressure Rise (psia)	52.1	52.1	110
AMMONIA DISTRIBUTION			
Flow Rate (lb/hr) each	3.8×10^6	3.39×10^6	3.36×10^6
Pressure Rise (psia)	15	15	15
WARM SEAWATER			
Flow Rate (gpm) each	402,600	386,700	403,800
Pressure Rise (ft)	10	8	7.6
COLD SEAWATER			
Flow Rate (gpm) each	408,500	395,500	416,200
Pressure Rise (ft)	15	9.5	8

4.7 MODULE SUBSYSTEMS

This section describes the remainder of the module subsystems and related auxiliaries. The ammonia cycle vapor and liquid plumbing is described and the results of a trade study on velocity vs. cost are presented. Descriptions of the ammonia cycle auxiliaries, such as the noncondensible gas removal system and the fluid sampling system, are also included.

Brief descriptions of the corrosion control system, the data acquisition and control system, safety and emergency systems are also included.

4.7.1 Ammonia Cycle

The ammonia cycle systems and components discussed below include the ammonia vapor and condensate systems and the two major ammonia cycle auxiliary systems – the ammonia cleanup system and the ammonia storage and transfer system.

Ammonia Vapor and Condensate Systems. The purpose of the ammonia vapor and condensate systems is to provide the necessary piping, valving, and pumps to interconnect the major components of the power cycle, i.e., the condenser, evaporator, and turbine. The ammonia vapor system for the 100-MW(e) spar configuration OTEC plant consists of two 60-in.-diameter vapor pipes running down either side of each 25-MW(e) power module from the evaporator outlet connections (downstream from the mist extractors) to the turbine inlet main stop valves. Part of the pipe is exposed to the external seawater environment while the remainder is contained within the power module and is exposed to atmospheric conditions. The pipe is configured to avoid the formation of liquid-ammonia pockets which could be carried into the turbine and cause damage to the turbine blades. The pipe exposed to the seawater environment is designed to preclude collapse when ammonia evacuation is being performed. Pipe material is carbon steel with coating on the outside to minimize corrosion. A cathodic protection system is also provided to further minimize corrosion and to provide protection if the coating is penetrated. To provide electrical isolation, an insulating flange or pipe section is provided between the aluminum evaporator and the carbon steel pipe and at the penetration point of the pipe into the power module. Appropriately isolated pipe supports, snubbers, and expansion joints are provided to accommodate the expected static and dynamic loads. No valves are included in the vapor system since the main turbine stop valves can serve as isolation valves for the evaporator, thereby avoiding unnecessary additional pressure drop. Bypass lines and control valves are provided around the turbine for startup purposes. Refer to Fig. 4-29 for a schematic representation of the ammonia vapor system.

The ammonia condensate system consists of the pipe, pumps, and valving required to remove the ammonia condensate from the condenser sump and deliver it to the evaporator sump and from there to the piping distribution system within the main ammonia evaporator. A 36-in. header from the condenser sump is provided with 14-in. lines which supply three one-half capacity, vertical centrifugal multi-stage condensate pumps. A sketch of one condensate pump showing estimated outline dimensions is shown in Fig. 4-30. The pumps are provided with butterfly isolation valves and minimum flow recirculation valves and lines. The discharge side isolation valves automatically close on pump trip. A butterfly level control valve is provided in the 20-in. condensate pump discharge header to control level in the evaporator sump. Four one-third capacity, vertical centrifugal single-stage distribution pumps take suction from the evaporator sump and deliver liquid ammonia through 14-in. headers to distribution points in the evaporator. These pumps are also provided with butterfly isolation valves. A condenser level controller operates ammonia make-up and/or drain control valves which are connected to the central ammonia storage system. The ammonia condensate system pipe is coated carbon steel and is provided with thermal insulation within the power module to prevent excessive condensation. Pipe supports, snubbers, and expansion joints are provided as necessary to accommodate the expected static and dynamic loads.

Pipe sizes are based on flows and state points for the design presented in Section 4.1 above utilizing a design flow rate of 1,970 lb/sec. The ammonia vapor and condensate header velocities are 160 fps and 30 fps, respectively.

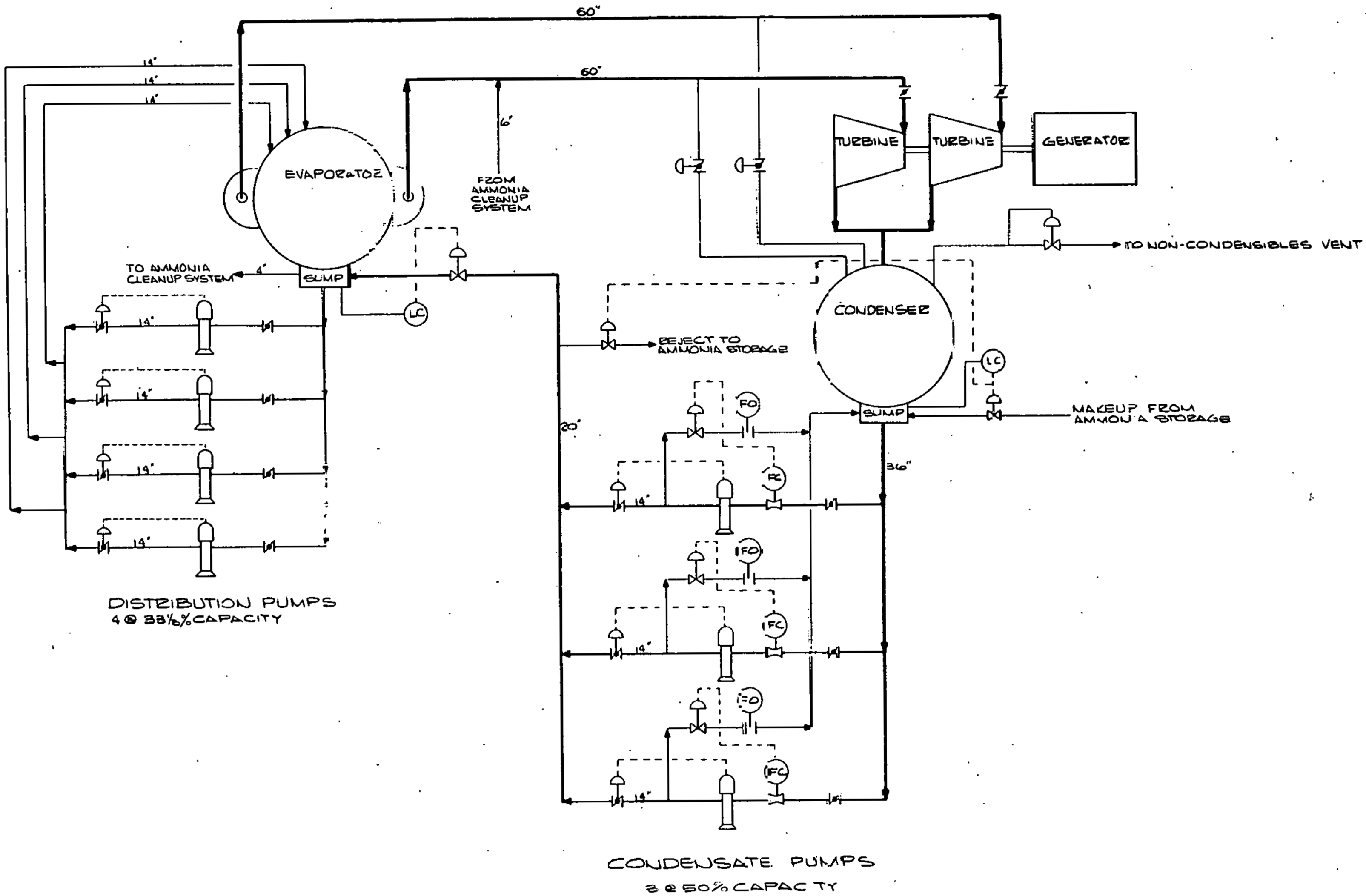


Fig. 4-29 Bechtel Schematic Flow Diagram

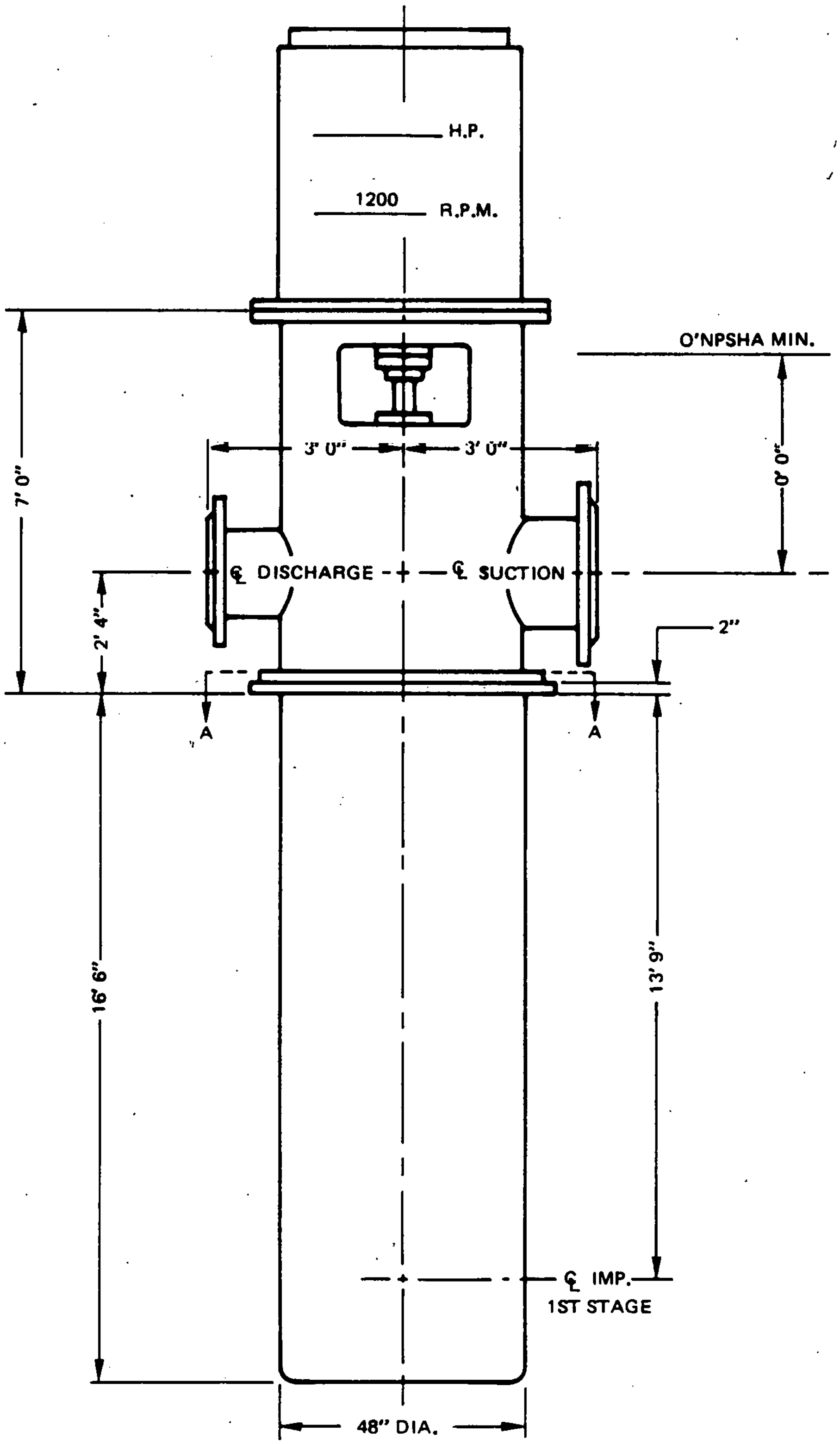


Fig. 4-30 Condensate Pump Outline Dimensions

In arriving at this system arrangement and description, a number of trade studies were performed and integrated into overall selection criteria. For the ammonia vapor and condensate systems, these studies centered mainly on establishing the ammonia vapor and condensate piping costs for several plant configurations and also, once a configuration was selected, to determine the optimum ammonia vapor and condensate piping size and flow velocity. This last trade study involved consideration of both pipe cost and pressure drop.

The configuration costing studies included three surface-ship, immersed heat exchanger configurations (four 25-MW(e), eight 12.5-MW(e)), and twenty 5-MW(e) heat exchangers) and two submerged spar configurations (four 25-MW(e) and twenty 5-MW(e) heat exchangers). The surface ship configuration is representative of the reference platform as well as the supertanker installation. Each of these configurations was in turn costed at three design thermal gradient conditions (ΔT = 32°F, 36°F, and 40°F) for a total number of 15 configuration cost combinations. These costs are shown in Fig. 4-31 for steel piping and include the vapor and condensate piping, valves, and condensate pumps but exclude the distribution pumps, piping, and valves. Internal design pressures for the pipe varied from a minimum of 0 psia for evacuation conditions to a maximum of 250 psig in accordance with OSHA requirements. External pressures varied from a minimum of 0 psig for atmospheric conditions to a maximum of 155 psig for the maximum submergence condensate pipe. For these configuration studies, piping design velocities were held at 60 fps for the vapor lines and 30 fps for the condensate lines. The results show the general trend of lower cost for higher design thermal gradient as expected, although the slope is not very steep. For the supertanker configurations, the cost is the highest for the twenty 5-MW(e) case because of the complex interconnecting piping and valving arrangements required. The eight 12.5-MW(e) case requires a simpler arrangement and actually results in the lowest cost. The even more simple four 25-MW(e) piping shows a slight upturn in cost due to the effects of large vapor pipe diameters and wall thicknesses. This effect takes place at pipe diameters greater than 90-in. for the design pressures under considerations. If a baseline vapor velocity of 160 fps had been used for the configuration costing exercise instead of 60 fps, all pipe sizes would be below 90-in. in diameter and the four 25-MW(e) cases would have been the least expensive.

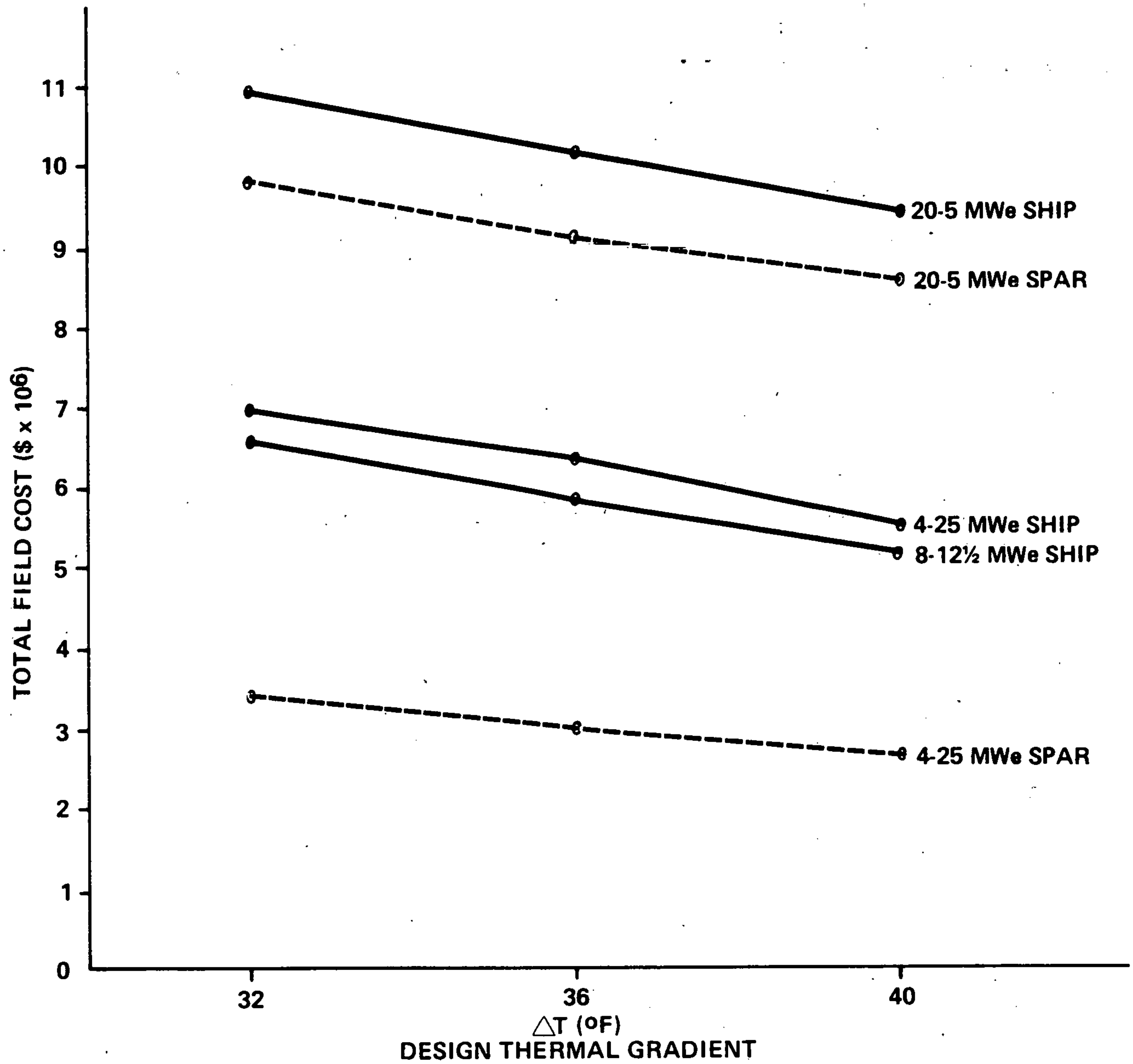


Fig. 4-31 Ammonia System Costs (Piping, Valves, and Pumps)

The four 25-MW(e) case is the less expensive of the two submerged spar cases considered. The four 25-MW(e) submerged spar case is less expensive than the four 25-MW(e) ship case because the average length of pipe between evaporator and condenser is greater for the ship arrangements, since in the supertankers, the evaporators and condensers are arranged in common blocks to facilitate seawater flow, resulting in a greater average distance between an evaporator and its associated condenser.

When it became apparent that vapor piping costs could be reduced significantly by increasing velocity and reducing diameter, cases were run for the four 25-MW(e) configuration at 60, 100, 160, and 200 fps. The results of these cases are shown in Fig. 4-32. Aluminum pipe is included for comparison and is seen to cost approximately twice as much as carbon steel pipe. Using the ammonia cycle computer optimization program, the cost significance of higher velocities including the effect of pressure drop increases in the vapor lines and module component costs, it was determined that 160 fps was the most cost-effective velocity for the spar considering the cost curves shown in Fig. 4-32. Similarly, a cost curve was produced for the four 25-MW(e) spar configuration for the condensate piping as shown in Fig. 4-33. Computer runs of pressure drop penalties show the baseline case of 30 fps to be the most cost effective. Computer output data are presented in Appendix A.

Figures 4-34 and 4-35 show similar curves for the four 25-MW(e) ship configuration vapor and condensate pipe costs, respectively. LMSC cycle computer runs showed the optimum vapor and condensate velocities in this case to be 160 and 30 fps, respectively.

Examination of these results, together with other LMSC and Foster Wheeler parametric costing studies, resulted in the selection of the four 25-MW(e) spar configuration as the most economical. Consequently, the baseline plant uses the 60-in. vapor lines and 14-in. condensate lines at 160 and 30 fps, respectively.

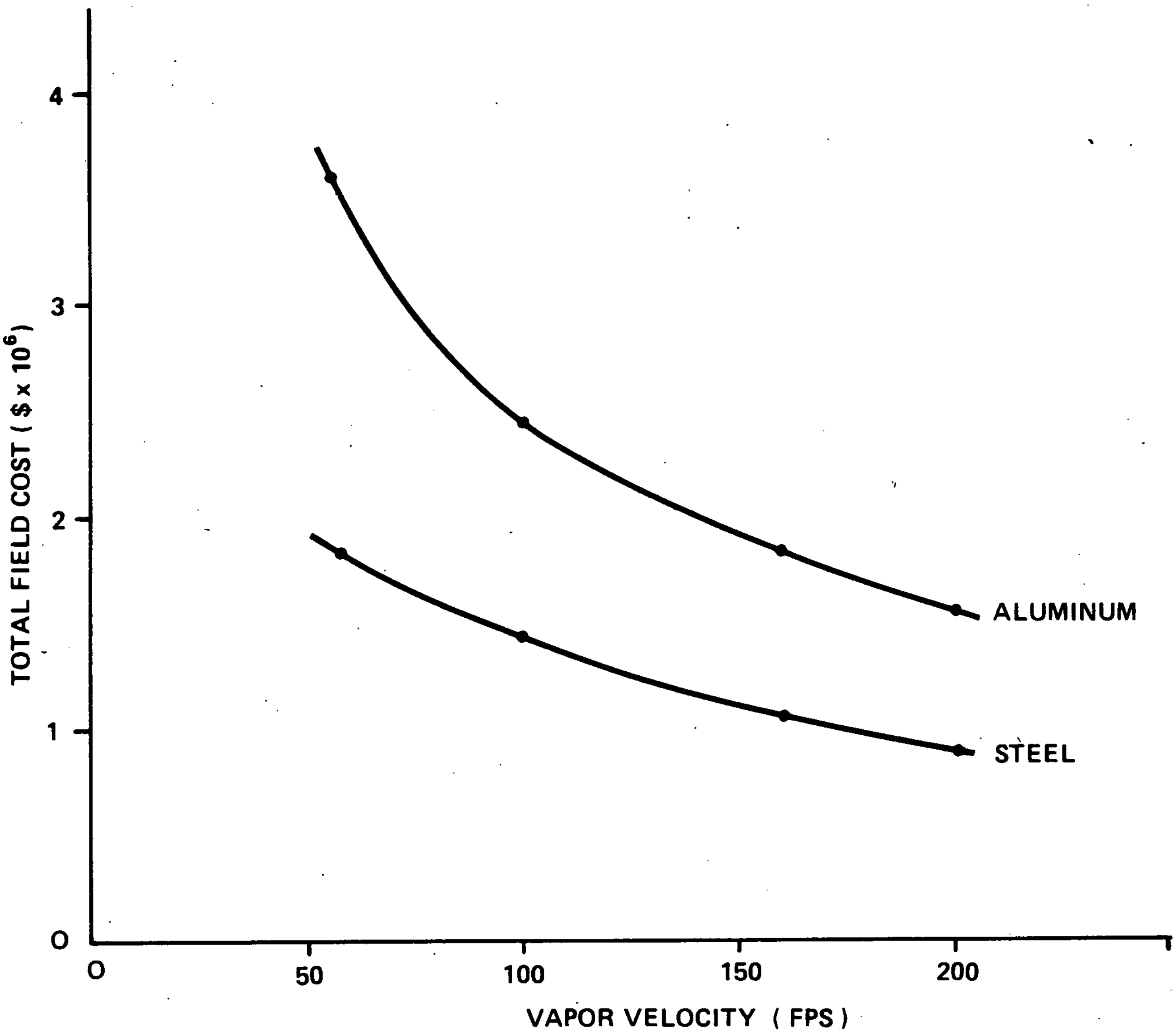


Fig. 4-32 Ammonia System Vapor Pipe Costs vs. Velocity (Spar Configuration, Four 25-MW(e), 36° F ΔT)

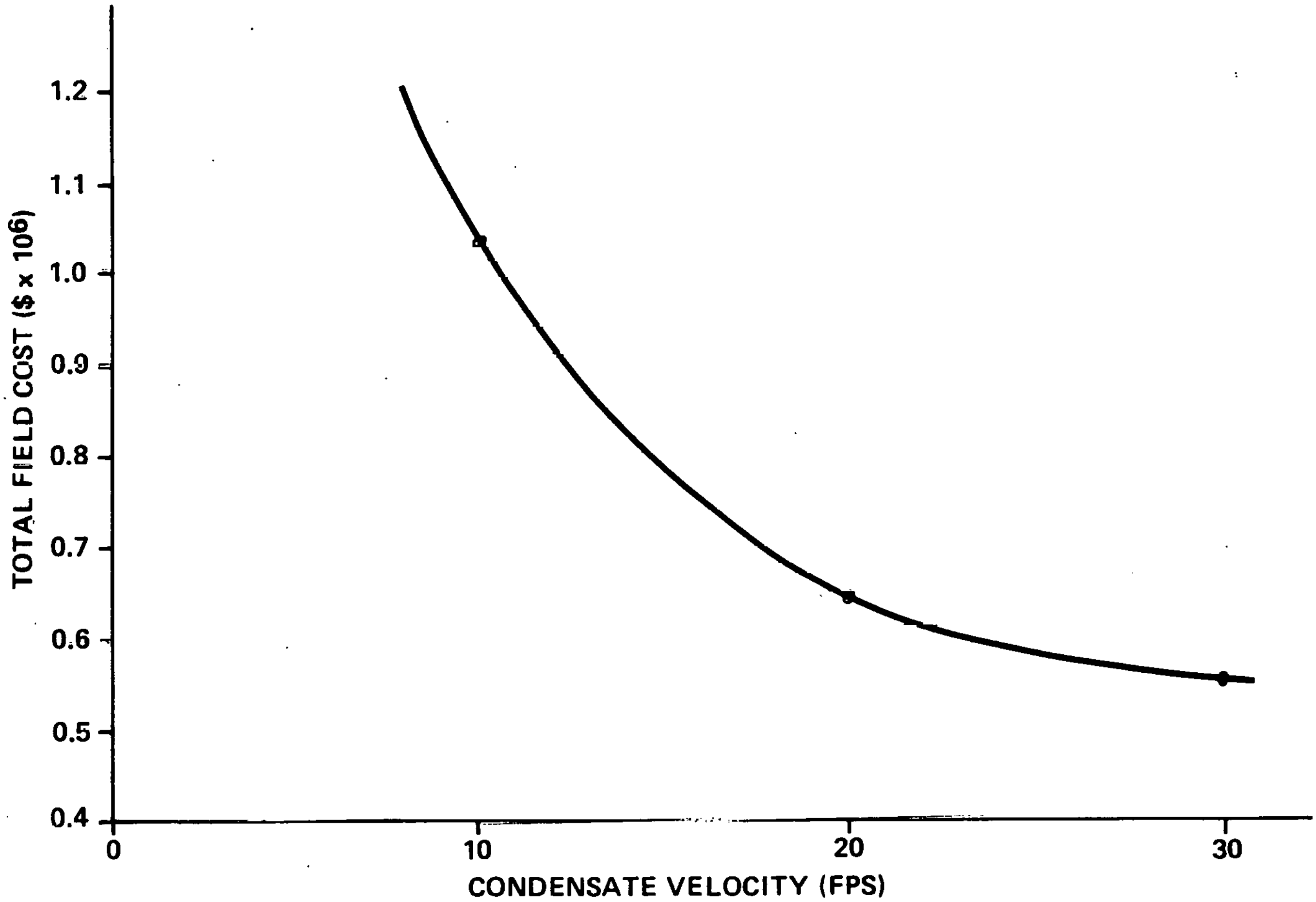


Fig. 4-33 Ammonia System Condensate Pipe Costs vs. Velocity (Including Valves) (Spar Configuration, Four 25-MW(e), 36° F ΔT

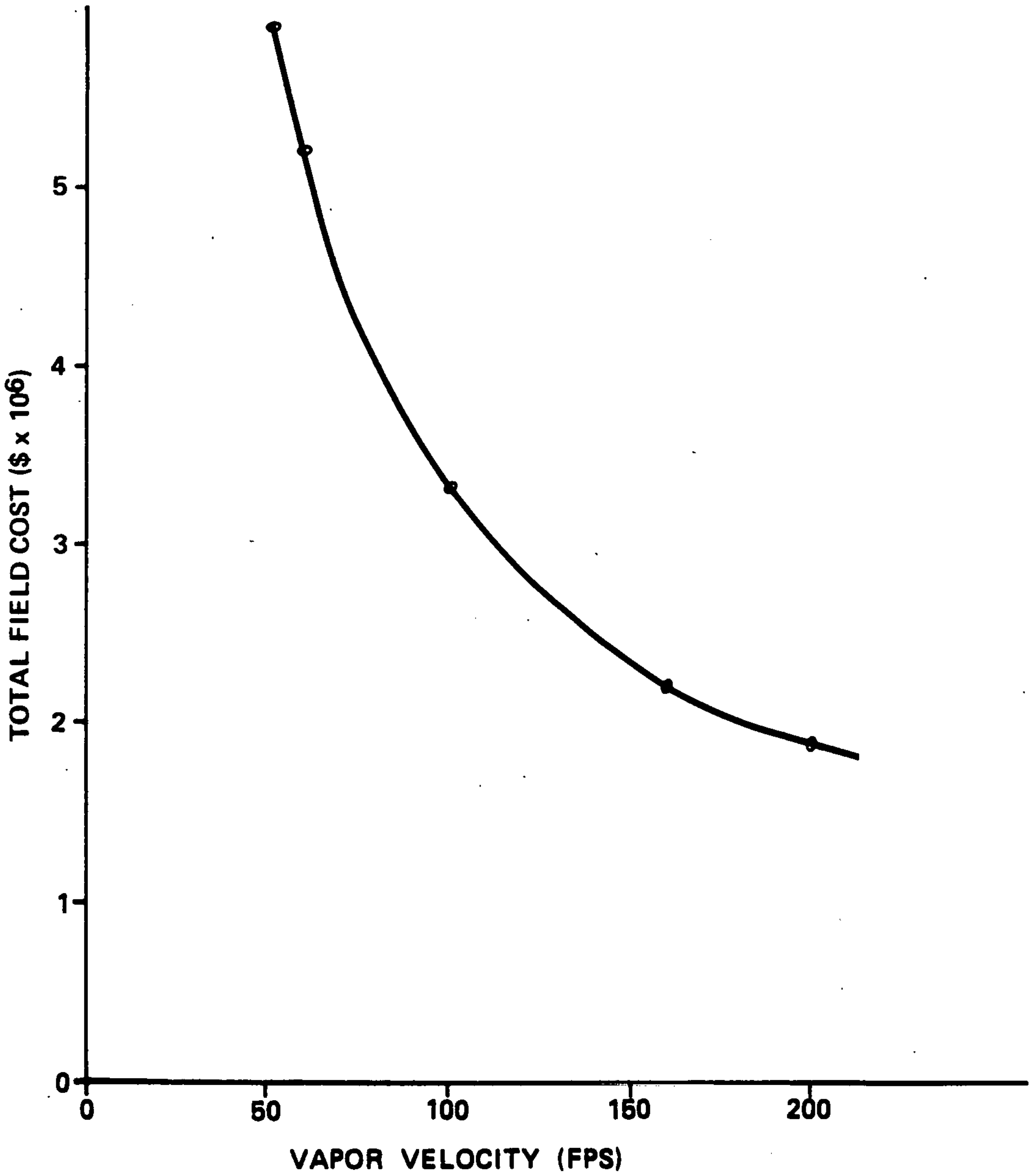


Fig. 4-34 Ammonia System Vapor Pipe Costs vs. Velocity (Ship Configuration, Four 25-MW(e), 36° F ΔT)

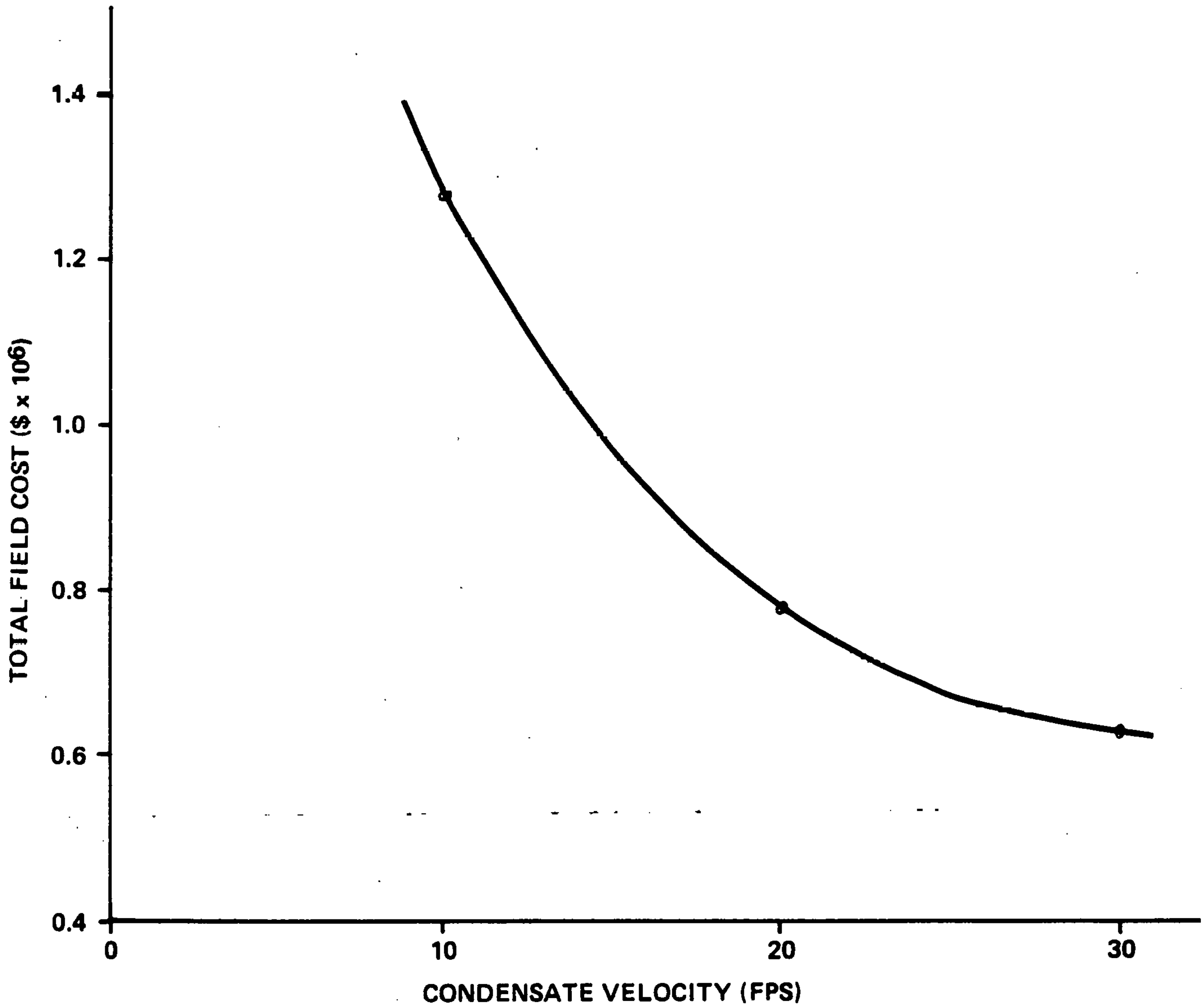


Fig. 4-35 Ammonia System Condensate Pipe Costs vs. Velocity (Including Valves) (Ship Configuration, Four 25-MW(e), 36° F ΔT)

Ammonia Subsystem Off-Design Performance. The OTEC power plant, although designed for a particular difference in temperature between the hot and cold ocean water, will be subject to ocean ΔT's that vary throughout the year. The design ΔT will fall somewhere between the summer maximum and the winter minimum, depending on how the plant design is optimized. At all times, however, the module will be operated to obtain the largest available electrical power output.

The percentage variation in ammonia flow rate during the year is approximately plus 15 percent and minus 35 percent from the design condition. The condensate pumps will operate at a relatively high pump efficiency within this range of flow rates. However, since they are constant speed pumps, the ammonia flow rate will be controlled by throttling with the control valve, which introduces pumping power losses.

Consequently, the electric motor drivers must be sized for the maximum required brake horsepower within the expected range of ammonia flow rates. During turndown of the power cycle in wintertime, it may be possible to take one of the half-capacity condensate pumps out of service to minimize throttling losses. However, subject to a more detailed pump curve analysis, the reduction in ammonia flow rate from the design value to the minimum value is probably insufficient to permit this. Four, one-third capacity, constant speed pumps may be better suited to turndown pump tripping. Variable speed pumps would eliminate throttling losses, but would be more expensive.

Based on reasonable estimates of maximum and minimum ocean ΔT's during the year, generator gross power output can be expected to increase from less than 20 MW(e) in winter to more than 40 MW(e) in summer depending upon the selected design ΔT. To allow for this increase, the power conditioning equipment (consisting of rectifiers, transformers, circuit breakers, bus, and cable) must be sized for at least the maximum expected electrical power output.

As shown in Fig. 4-36, the electrical efficiency of the rectification equipment is relatively high and varies only slightly between 25 percent and 100 percent of the design full capacity.

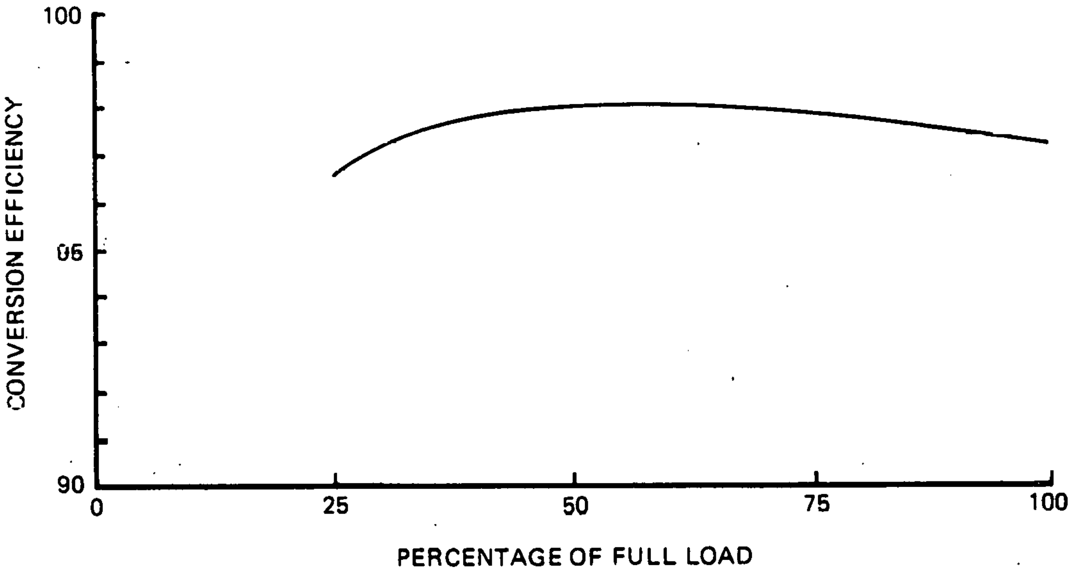


Fig. 4-36 Conversion Efficiency of Rectification Equipment vs. Percentage of Full Load

The cooling water system discussed in Section 4.8.2 is the only auxiliary system that may be affected by off-design performance of the ammonia power cycle. Many auxiliary systems, such as the ammonia storage, fill and purge system, and the fire protection system, do not operate on a continuous basis. The operation of other systems is not related to the level of power generated by the plant. For example, the flow rate of ammonia processed by the distillation column in the ammonia cleanup system is related only to the rate that seawater leaks into the ammonia system.

The cooling water system is designed to remove heat rejected by various plant systems. Some of these heat rejection loads will vary with the output level of the OTEC plant, which is estimated to be about 50-percent less in winter than in summer. Other heat rejection loads will remain constant as the output level of the plant changes.

For the constant load systems, the seawater and freshwater flow rates for cooling will remain constant at all times. For the variable heat rejection systems, it may be feasible to vary the cooling water flow rates, thereby maintaining constant temperatures in these systems as well. Further studies during the preliminary design will examine the feasibility of variable cooling water flow rates.

4.7.2 Module Auxiliary Subsystems

The module auxiliary subsystems discussed below include those subsystems that are not directly within the ammonia cycle but which are required for power system operation. These are the electrical distribution system, the noncondensible gas removal system, the overpressure vent system, the fluid sampling and leak detection system, and the seawater pump pod pressurization system.

Electrical Distribution System. A block diagram of the module electrical distribution system and the interrelation between the modules and the platform is shown in Fig. 4-37. Each module uses a stepdown transformer to provide 4,160-volt power from the 13,800-volt generated power. Each 4,160-volt bus supplies power to the warm and cold seawater pumps, the three ammonia condensate pumps, and the ammonia cleanup system. A stepdown transformer reduces the 4,160 volts to 480 volts for use by the ammonia distribution pumps, the cleanup system pump, the ammonia transfer pump, and the purging system vacuum pump. Additional stepdown transformers reduce the voltage to 208/120 for hotel loads, controls, and safety items.

During normal operation at the design condition, each warm water pump requires 760 kW, each cold water pump requires 825 kW, each ammonia condensate pump requires 635 kW, and each ammonia distribution pump requires 60 kW. In addition, the hotel loads and intermittent equipment operation average to approximately 1,050 kW. Internal transmission losses of approximately 78 kW result in a module power output to the core of approximately 23,810 kW. As the design matures during the Preliminary Design task, the module components will be resized to provide 25 MW(e) module output at the busbar.

Noncondensible Gas Removal. With the use of nitrogen as an inerting gas during transit and purging operation, and the procedures described in Section 4.8.1 for making the transition from an operational mode to a safe access condition, the noncondensible gas removal system serves a dual purpose. During the transition phases, the gas mixture will be drawn off, at an appropriate point in the condenser, and routed to the noncondensible gas removal system where the auxiliary condenser will liquefy the ammonia.

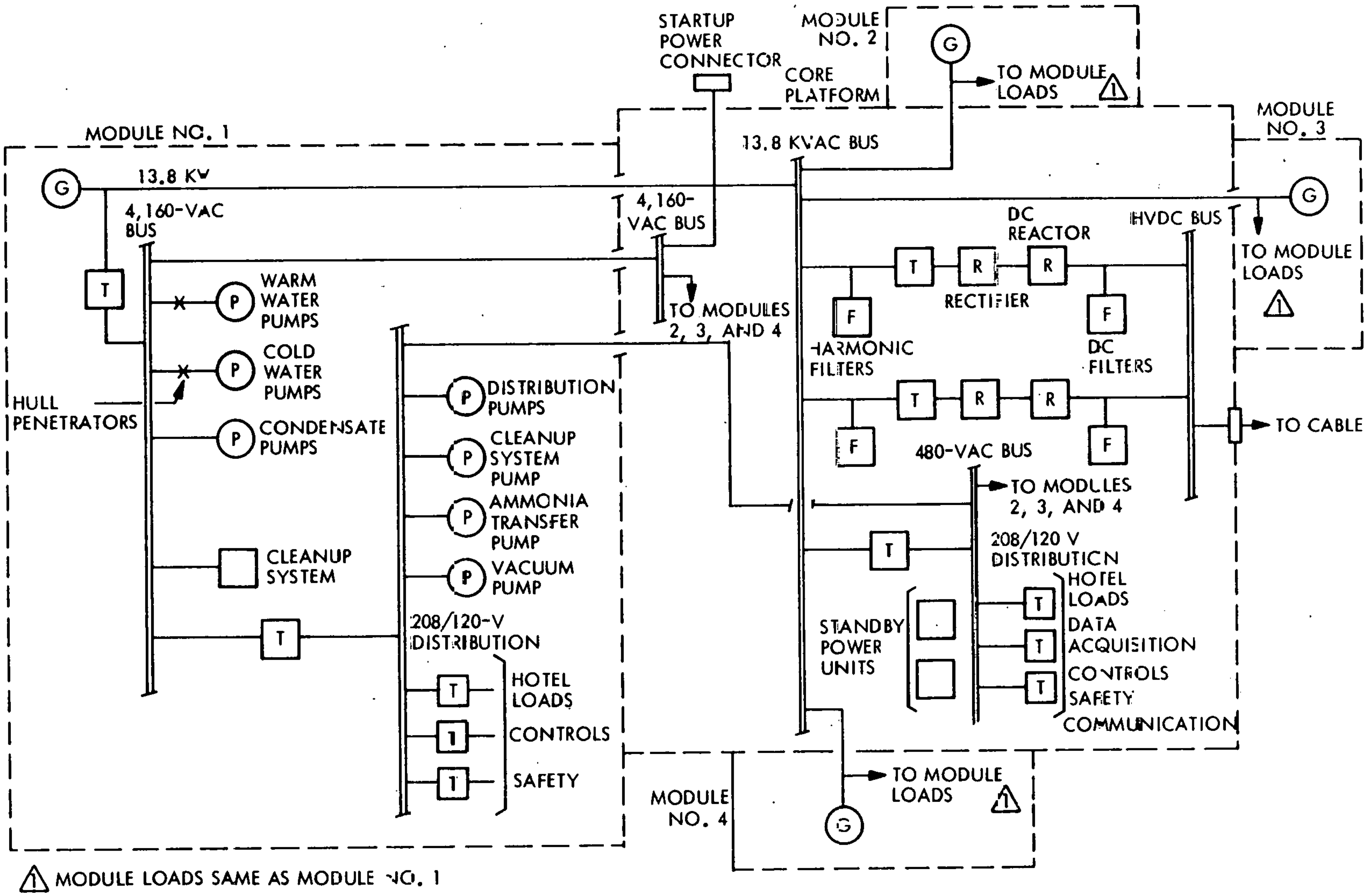


Fig. 4-37 OTEC 25-MW(e) PSD Electrical Power Block Diagram Baseline Configuration

Dependent upon the transition phase, the ammonia will be returned to storage and the nitrogen pressurized for reentry to the evaporator or vice versa.

The system for removing noncondensible gases, during normal operation, includes a perforated duct for ingesting the gases and routing them out of the condenser and a baffle to prevent impingement of liquid ammonia. This duct is presently located at the center of the condenser tube bundle where most of the ammonia vapor will have been liquefied. Data from the Heat Exchanger and Proof-of-Concept test articles will be used to determine the optimum location for this duct.

Overpressure Vent System. The overpressure vent system consists of relief valves and ducting incorporated into the evaporator and condenser. The yield design pressure for the heat exchangers is 127 psid. The relief valves will be set to crack at 100 psid and to flow full at 110 psid. The external pressure sensed by the valve is the local hydrostatic pressure. Ammonia vapor flowing through the valve will then bubble through the seawater prior to entering the atmosphere.

Fluid Sampling and Leak Detection. As discussed in Section 4.8-1, Ammonia Cleanup System, leakage of seawater into the ammonia system can cause drastic thermodynamic changes in the cycle and reduce the available power output of the ammonia turbines. The primary source of seawater inleakage is expected to be through a defective tube in the condenser. This defect cannot be readily located. The condenser has 56,500 tubes each approximately 40 ft long, submerged to an average depth of 275 ft, and any leakage occurs into the interior of the shell to which there is no access. A fluid sampling system is required to determine the severity of inleakage and the need for operating the ammonia cleanup system or for shutdown to detect and plug leaking tubes.

Samples are taken continuously from the condenser and evaporator sumps and passed through ion detectors (sodium ion). Contamination levels by seawater will be continuously recorded and, at a preset level, the ammonia cleanup system is activated and an alarm sounded in the control room. If the contamination level increases to the level which is determined to be unacceptable, the system is shut down.

Leak detection is accomplished by introducing nitrogen at pressures above the local hydrostatic head and providing visual observation of nitrogen bubbles from the leaking tubes. Tubes will be plugged and operations resumed.

Seawater Pump Pod Pressurizing System. The warm seawater pumps are installed at an average depth beneath the surface of 150 ft and the cold seawater pumps at an average depth of 275 ft. These depths correspond to external pressures on the pods of approximately 82 and 137 psia. The pods will have a rotating seal on the large propeller shaft and static seals on other hull penetrations. A rotating seal of the size required is complex and will require maintenance in proportion to the pressure load. Pressurizing of the pod interior to offset the external pressure of the sea will relieve the seal of an undesirable pressure differential. In addition, any other leakage will be minimized.

The pump pod pressuring system consists of pressure controllers, piping, valves, and flow indicators. The system receives its air from the central compressed air system. Each module is provided with its own system. Compressed air is admitted to each group of pump pods by an automatic controller, such that the interior pressure is only slightly below the seawater pressure. If maintenance is required, the pump is shut down and the pod depressurized. The pods may be depressurized or pressurized from the control room.

4.7.3 Corrosion

Corrosion in metallic seawater systems has been a maintenance problem for many years. The corrosion is caused by the electrical coupling of dissimilar metals. In the OTEC seawater systems, corrosion-resistant metals will be used. Where large areas of corrosion-resistant metals are coupled to small areas of carbon steel, very high and destructive corrosion rates can occur. This situation may occur due to the use of carbon steel for piping, pump pods, pump shafts, intake screens, etc., which may be exposed to larger areas of corrosion-resistant metals. Cathodic protection is an electrical method of preventing galvanic corrosion, and therefore is considered as a means of protecting such submerged components.

Design of specific elements of the cathodic protection system will be initiated during the preliminary design phase when the power system configuration has been finalized. If carbon steel piping, included in the design, requires an extensive cathodic protection system, it may be more economical to install pipe which is less susceptible to galvanic action. Alternatively, it may be less expensive to reroute the carbon steel vapor and condensate pipe so that it is never exposed to seawater.

Coatings will be used to reduce the amount of current required for cathodic protection. Heavy localized corrosion may occur at breaks in the coating and the impressed current adjusted to compensate for the exposed areas.

4.7.4 Data Acquisition and Control System

The Data Acquisition and Control System (DACS) concept, developed to satisfy the power system module requirements, employs distributed system components and intelligence. This concept provides a strategically located control center with communication paths to remote terminal units (RTUs). The RTUs interface with the energy conversion process via instrumentation sensors and transducers (Fig. 4-38).

By employing programmable (intelligent) RTUs, the concept provides for a reduction in the data processing load at the control center (thereby minimizing control system costs). Additionally, because independent operational capability is located within each RTU, overall system reliability is enhanced. At each RTU, the concept employs redundancy, to the level of the interface to individual monitor and control points, with automated intelligent RTU component failover. Figure 4-39 is a functional schematic of the proposed RTU.

The DACS concept employs a redundantly configured control center (see Fig. 4-40) to permit automated failover to provide for the required system reliability. The backup computer system additionally allows software enhancing developments and optimizing program executions concurrent with on-going plant energy conversion activities. Appendix K of this report discusses control system availability/reliability with application to DACS and state-of-the-art equipment.

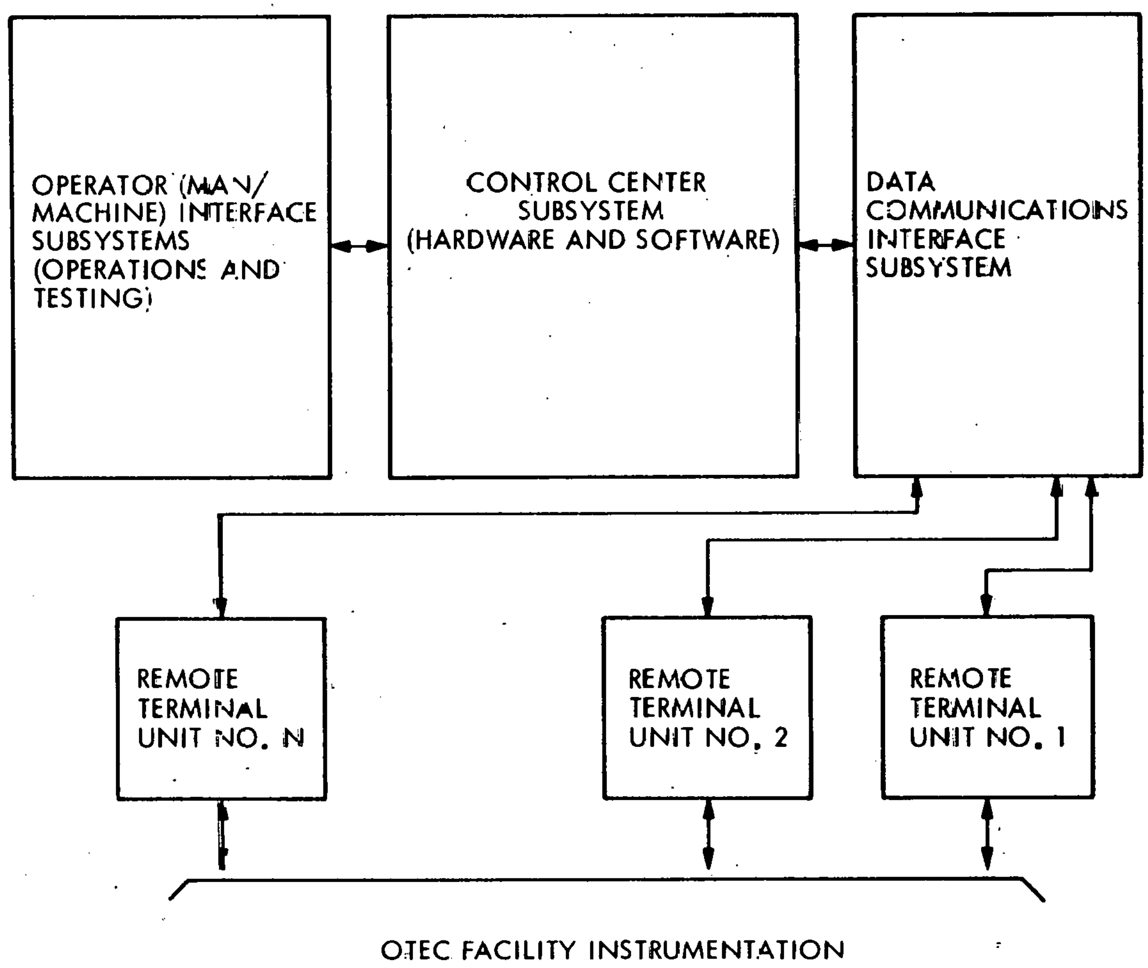


Fig. 4-38 Control System Generic Configuration

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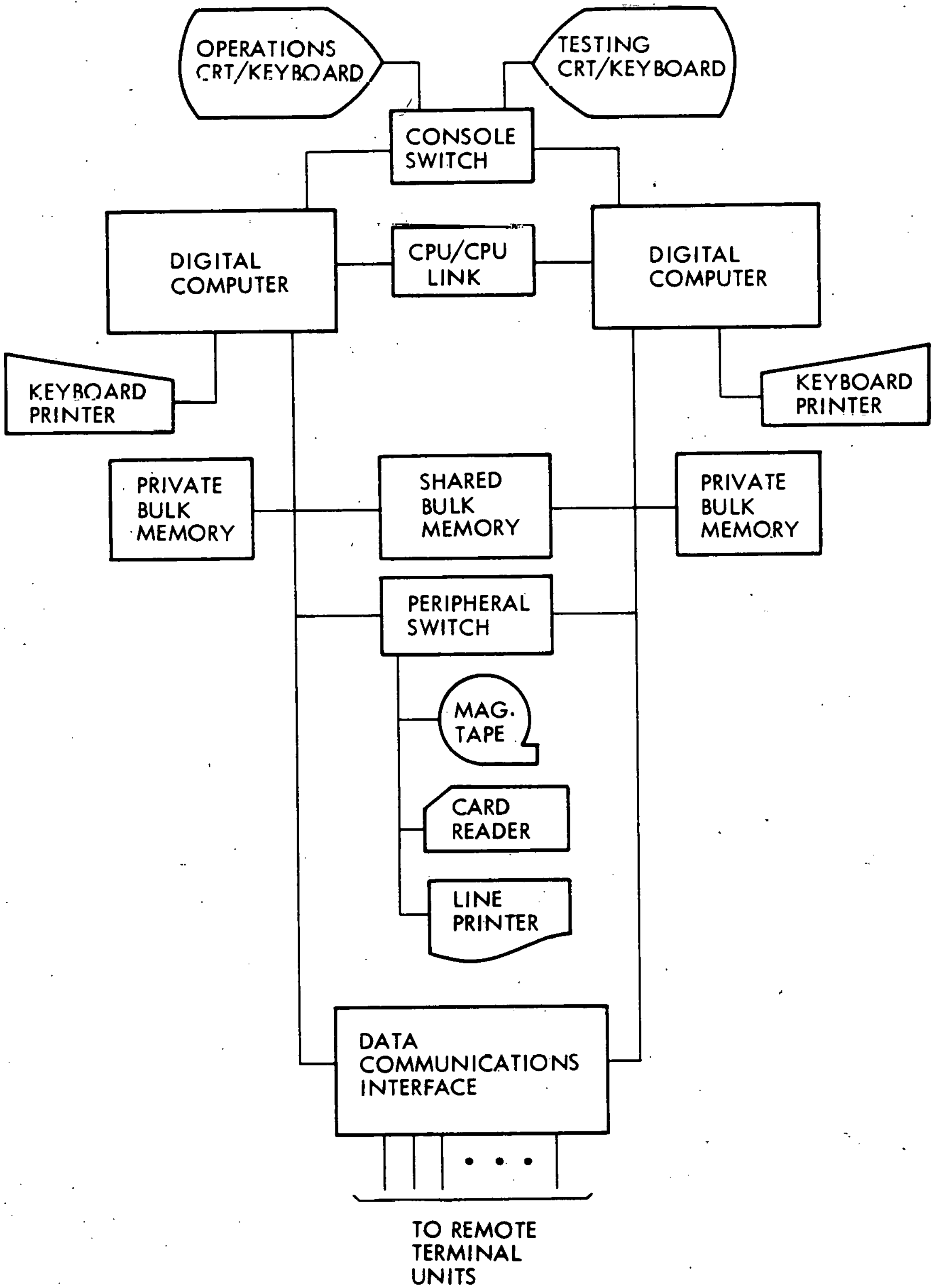


Fig. 4-40 Control Center, Computer System Configuration

The control center DACS equipment functionally will acquire operational data to provide for the maintenance of the power system module data base permitting the annunciation of operational alarms and the generation of module status logs and reports. The control center additionally will periodically execute optimizing control algorithms and provide for the automatic or manual control of the energy conversion process.

The operator or test conductor will communicate with the DACS via control center CRT display/keyboard consoles. The DACS concept employs two operator consoles:

- Operations Console – the console normally employed for power cycle performance monitoring and control
- Testing Console – the console used to evaluate power system components and operational system parameters

Figure 4-41 shows DACS control center communications devices supporting test conductor input/output and system logging.

The DACS test conductor interface will be an interactive dialog between the test conductor and the DACS. Prompt messages, as well as annunciated events, will be displayed via the CRT terminal. Figure 4-42 provides a conceptual CRT utilization showing the operator/DACS dialog zone, an alarm/advisory zone, a tabulated data zone, and a command input zone.

The DACS concept, through the use of distributed components, inherently incorporates modularity. By using independently programmable RTUs, and a redundantly configured control center, the concept offers usage flexibility, which appears desirable, particularly in the developmental stage. The DACS concept provides the capability to monitor both analog and discrete status points. The concept system may be implemented to supervise (monitor and control) in excess of 1,000 points at eight or more RTUs. The monitor and/or control rate (scan rate) will be program selectable from a fraction of a second to virtually any slower scanning rate. Individual point monitor or control points may be scanned or controlled at individually selected rates.

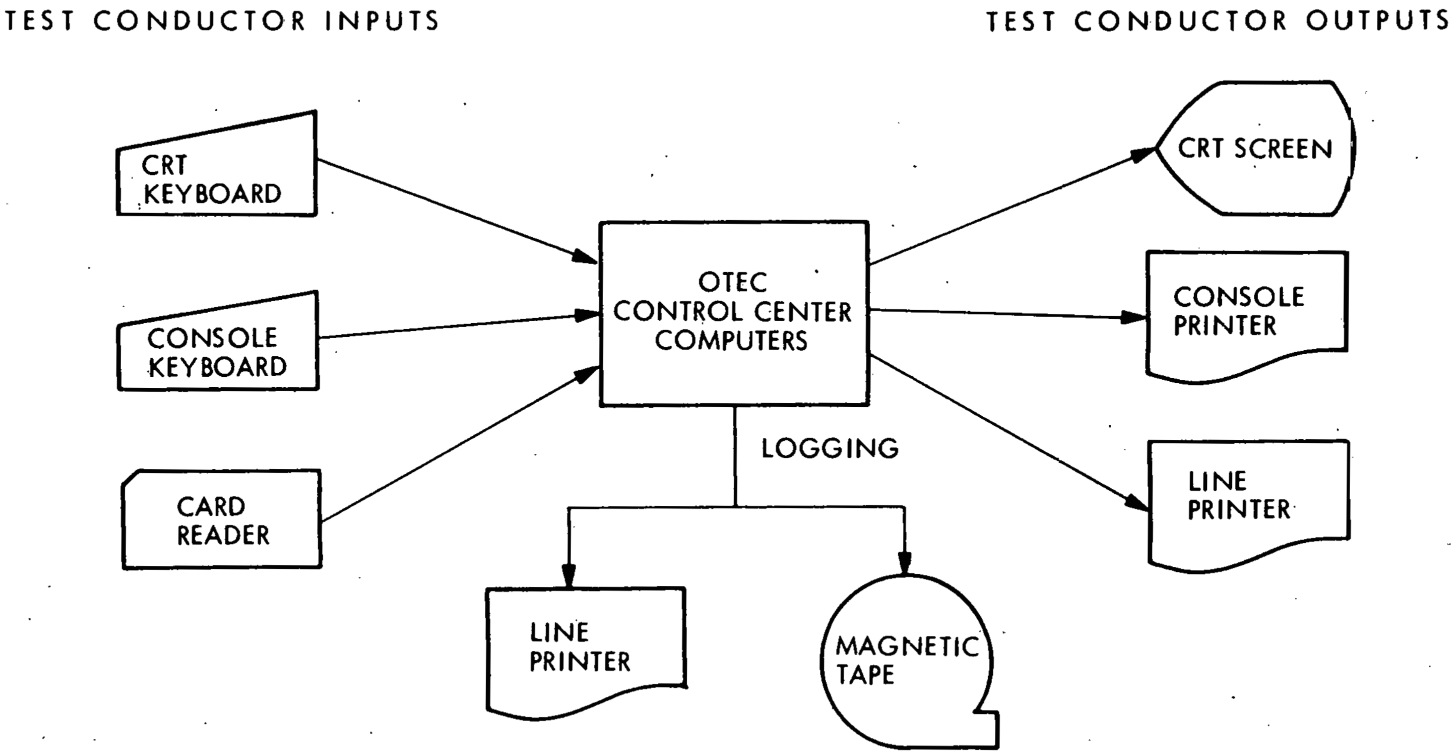


Fig. 4-41 OTEC Test Conductor Communications

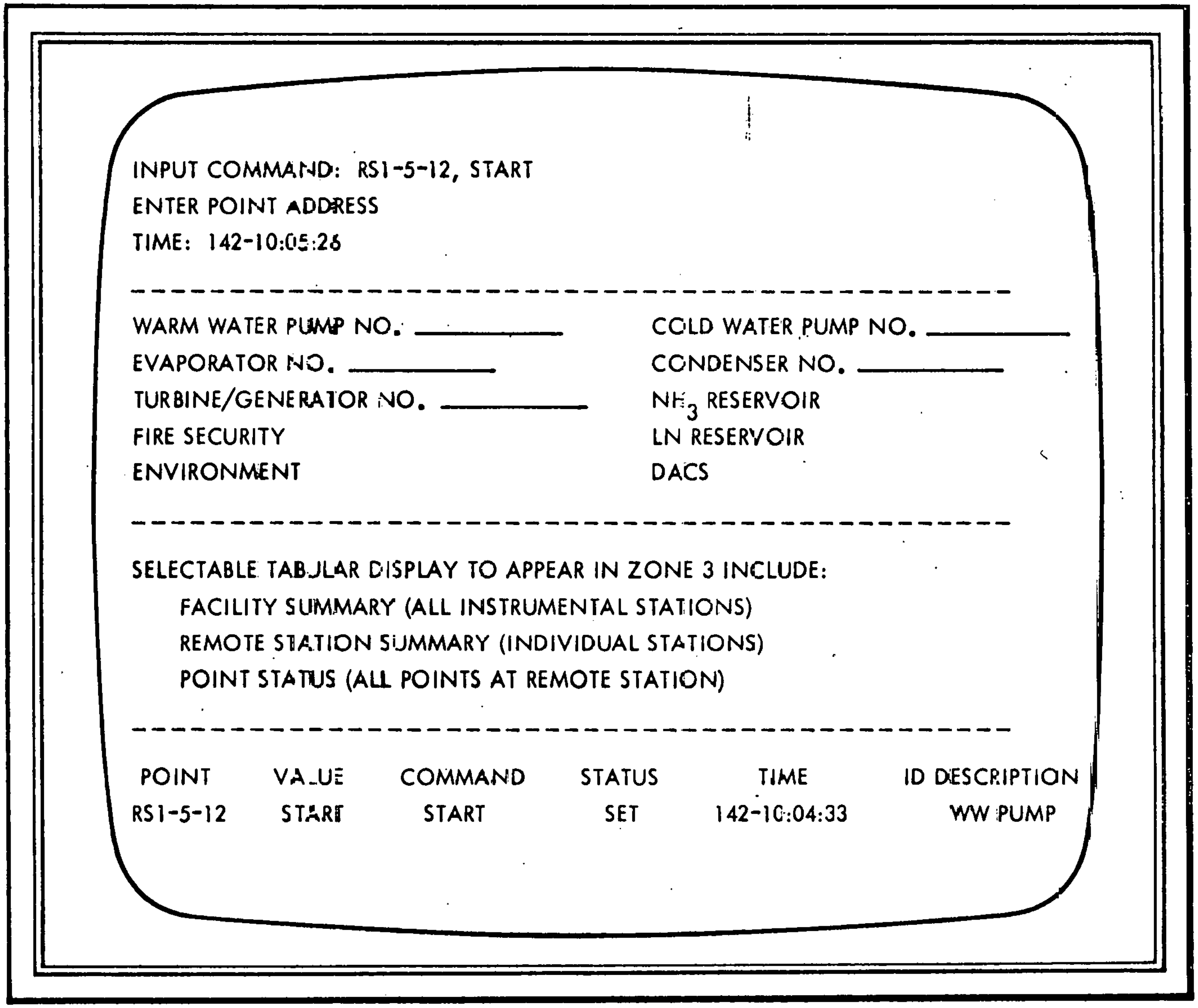


Fig. 4-42 Conceptual CRT Utilization

The DACS concept system, with redundantly configured hardware and periodically activated self-test capability, will support 24-hr/day testing and evaluation activities lasting in excess of one year.

In terms of DACS implementation or producibility, no major problems are foreseen. It is expected that RTU enclosures will be gas-tight and that all system components will be shock mounted both at the base and/or to a bulkhead as required.

The estimated cost of the 25-MW(e) power module DACS is approximately $750,000 installed. For a four-module 100-MW(e) plant, the cost is approximately $1,500,000.

4.7.5 Safety

An OTEC power system using ammonia as a power cycle fluid must be designed and operated in consideration of the particular characteristics of ammonia and their potential impact on personnel health and safety. Adherence to the appropriate codes, standards and good engineering practices during the design and construction phases will ensure a safe system. It is important to emphasize that rigorous training programs for operators are also essential for safety. In addition to being completely familiar with ammonia cycle operation and control, operators should also be thoroughly aware of the toxic, corrosive, and flammable properties of ammonia. Personnel must know the location and use of the various pieces of protective equipment and be instructed in first aid measures.

The major hazard associated with ammonia derives from its toxicity. Exposure to ammonia vapor concentrations of 700 ppm is limited to 30 minutes. The least detectable odor is 20 ppm. Additional limits are given in Table 4-19.

Experience has shown that ammonia is extremely difficult to ignite and under normal conditions is a very stable compound. The flammable limits at atmospheric pressure are 16 to 25 percent by volume of ammonia in air. Ammonia is classified by United States Department of Transportation and the United States Coast Guard as a nonflammable compressed gas for the purpose of transportation. However, despite these favorable characteristics, flash fires and explosions have occurred around ammonia handling equipment and storage tanks.

The USCG will probably have prime regulatory authority over OTEC systems since their general responsibilities include:

> "upon the high seas........"
> "Administering laws and promulgating and enforcing regulations for promoting safety of life and property, covering all matters not specifically delegated by law to some other executive department or reserved to the States........."

Table 4-19

PHYSIOLOGICAL EFFECTS OF GASEOUS AMMONIA OVER A RANGE OF CONCENTRATIONS[a]

Ammonia Vapor (ppm)	Effect on Unprotected Worker	Exposure Period
20	Least detectable odor	Permissible for 8-hour working exposure.
40	Some slight eye irritation in a few persons	
100	Noticeable eye and nasal irritation after a few minutes exposure	
400	Severe irritation of throat, nasal passage, and upper respiratory tract	Ordinarily no serious effects following infrequent exposures if less than one-half hour.
700	Severe eye irritation	
1700	Causes convulsive coughing	No exposure permissible. May be fatal after one-half hour exposure.
5000 or more	Causes respiratory spasm, strangulation, asphyxia	No exposure permissible (rapidly fatal).

[a]Data from Manufacturing Chemists' Association Chemical Safety Data Sheet, SD-8.

DOT and the USCG consider ammonia – as a bulk cargo – dangerous to life and property. Major points of current regulations are:

- No copper
- Fittings – tongue and groove with gasket
- Brazing is prohibited
- No threaded joints greater than 2 in. in diameter

- Minimum container design pressure 250 psi
- Water dispersant to be provided
- ASME VIII Div 1 pressure vessels
- Cannister masks to be provided for all personnel

A state code also may be invoked on the OTEC system if personnel embark from a port of that state and/or if their wages are subject to compensation withholding by that state. As an example, the California Code covers ammonia plants in some detail and invokes the Uniform Mechanical Code (essentially identical to ANSI B 9.1) and requires pressure vessels per ASME Code and piping per ANSI B 31.5, Refrigerant Piping. Other major call outs are:

- NEC Class I electrical installations
- No flame, spark producer, or surface above 800° F
- Seamless piping not less than Schedule 40
- Iron or steel valves and fittings
- No brazing – welding where possible
- No screwed joints greater than 2-in.
- No column gage glasses
- Hose burst pressure 5 x working pressure
- BuMines approved masks
- Machinery room (Class T) shall be:
  - Gas tight
  - Equipped with self-closing closures
  - Equipped with remote controls immediately outside
  - Provided with two exits each not less than 3 ft by 6 ft 8 in.

ANSI K61.1 and CGA G-2.1 reiterate most of the foregoing plus:

- Provision for dispersal with 100 parts water per part ammonia
- Vapor dispersion by water spray of fog
- Flex connections rated not less than 250 psig with a safety factor of 4

The foregoing standards and regulations are directed at conventional refrigeration plants ashore and at marine transport of bulk cargo ammonia. Such plants and transporation are commonplace. The originating agencies obviously have never had to consider a plant such as contemplated for OTEC. Accordingly, the USCG, ABS, and any agencies claiming jurisdiction must, of necessity, undertake what they term "special consideration" of the OTEC plant. All agencies make provision for such action in their regulations or standards. This means they will be guided by existing Federal standards as well as those of technical and classification societies and industry groups, but will consider and evaluate the design or unique features of OTEC. Therefore, except where an existing standard can be applied without deviation, a sound engineering basis must be prepared and documented for every innovative feature. Safety margins should approximate those of current standards unless a convincing case can be made for a deviation.

Ammonia need pose no undue hazard to the plant or personnel. Safety is enhanced by isolation of heat exchanger volumes from manned spaces and isolation of machinery spaces. Mechanical machinery spaces will not be regularly manned. For short duration tasks such as routine inspections and minor maintenance actions they could be entered via locks and, if necessary, in masks and suits. For tasks of longer duration, such as major inspections or major maintenance with the plant shut down, they will be purged to a safe contaminant level.

Based on a review of codes, standards, and good engineering practices relating to the design and use of ammonia facilities with emphasis on safety considerations, some preliminary design requirements and guidelines were established for the conceptual design phase. These requirements and guidelines fall within the general categories of protective measures and equipment, fire protection, electrical equipment, and piping and mechanical equipment.

Protective measures and equipment include adequate ventilation, well-marked exits, and strategically located and clearly marked safety equipment such as safety showers, bubbler eye fountains, protective suits, and breathing apparatus.

Ammonia system fire protection is most effectively achieved through the use of water spray and dilution combined with adequate ventilation. Carbon dioxide units will be used in areas with electrical equipment. Electrical equipment will generally be arranged such that contact with ammonia will not occur, even during accidental leaks or spills. This will be achieved through the use of separate compartments provided with adequate positive pressure ventilation from a source of clean air in conjunction with effective safeguards against ventilation failure. This equipment will be standard nonhazardous rated. Where equipment may encounter flammable ammonia mixtures in the event of accidental rupture, electrical equipment and wiring will be in accordance with National Electrical Code, NFPA 70 (ANSI-CI), for Class 1, Division 2, Group D locations and ANSI C110.1.

Pipe and mechanical equipment will be specified and designed in accordance with acceptable existing codes and standards applicable to ammonia service.

Other safety related areas which have been identified as requiring special attention during the preliminary design task are:

- Location and integrity of the control room
- Secondary containment of major spills including pressurization effects
- Instrumentation and safety interlocks
- Emergency power
- Requirements for the inert blanketing of equipment and tankage

Integrating the power system with a sea-going platform introduces considerations which would not be encountered in a shore installation. The ocean is an extremely variable environment and the plant and the platform must survive its vagaries. The power system design must accommodate motions and accelerations imposed by sea conditions. The platform must be capable of withstanding sea conditions; conventional marine hazards such as collision, loss of buoyancy, loss of stability, and inability to maintain position; and, in addition, must be capable of coping with the hazards associated with the ammonia cycle; namely, large volumes of ammonia and seawater.

In some platform configurations, the large seawater loops will pose potential flooding hazards. Sea-valves will require special consideration. Platform subdivision must be such that neither ammonia loops nor seawater-loop casualties will result in uncontrollable hazards to personnel or to the buoyancy or stability of the platform.

Maintenance in the shipboard environment will require special attention principally directed to ammonia precautions and the handling of large and heavy equipments. Obviously, well conceived operating procedures and thorough training of personnel for both normal and emergency situations will be required. Timely consideration of the foregoing matters and demonstrably sound engineering, planning, and training will reduce hazard potential to an acceptable level.

4.8 SUPPORTING SUBSYSTEMS

This section provides descriptions of those subsystems which are incorporated into the the platform and are mutually shared by all modules. These include the ammonia clean-up system, the ammonia fill and purge system, the electrical distribution system, the standby power system, fire protection, and the cooling water system.

4.8.1 Ammonia Subsystems

Ammonia Cleanup System. The ammonia cleanup system separates seawater contaminant (from condenser or other inleakage) from the ammonia fluid. This is necessary because contamination by seawater will depress the boiling point pressure of the fluid, causing a drop in available pressure differential across the turbine. For example, a mixture of 90-percent ammonia and 10-percent water will result in almost 10-psi less available pressure differential than would be the case for 100-percent ammonia. Since the design pressure differential across the turbine is approximately 40 psi, this pressure reduction would result in a significant power loss. In order to maintain the water contamination at a reasonable level when the condenser is leaking, it is necessary to maintain a continuous blowdown from the evaporator which must be cleaned and returned to the main system. The condenser must be considered as a possible source of inleakage, since the external water pressure is greater than the internal ammonia pressure. Double tubesheets can be incorporated in the condenser design to eliminate leaks through the tube-tubesheet joint. However, there are 56,000 40-ft tubes in the unit, and it has been stated that the tubes will be the major source of leaks. Leaking tubes will be plugged as stated in Section 5.3.

If the assumption is made that total inleakage in the condenser is 1 gpm, and that the maximum permissible concentration of water in the evaporator will be 5 percent (discussed below), then a continuous blowdown of 0.16 percent of the total ammonia flow will be required. The cleanup system is sized on this basis, resulting in a blowdown flow of 10,270 lb/hr. Water concentration increases with vertical distance downwards through the evaporator and reaches a maximum at the lowest tubes. Since condensate flow into the evaporator sump dilutes this maximum percentage, blowdown flow must be taken from an intermediate overflow catch basin, or the line flowing into the sump, to minimize cleanup system sizing.

A number of methods were considered for this separation process including molecular sieve or silica bed adsorption beds, freezeout, flash separation, and a distillation column. Adsorption beds would normally be the most desirable method. However, they are not satisfactory in this case since the high affinity between ammonia and water precludes satisfactory separation. Distillation is considered to be the most desirable of the three other methods based on experience, performance, and power requirements. A sketch of the distillation-type cleanup system is shown in Fig. 4-43. The blowdown from the evaporator is pumped into the distillation column. The pressure in the column is maintained high enough so that the resulting vapor may be ducted to the turbine vapor line without further compression. The water that separates out flows into the sea after cooling to 100° F. The column is heated by an electric heater which draws power from the main generating system.

In order to determine the optimum cleanup system size, a trade study was conducted of column power requirements and power loss in the main turbine. As the blowdown flow and distillation column size are reduced, the equilibrium maximum water concentration in the evaporator increases, causing a reduction in power output. Although the pressure reduction at the turbine inlet is relatively easy to calculate, there is also a pressure reduction in the condenser caused by the water vapor carryover through the turbine. This condenser pressure reduction tends to ameliorate the turbine power loss. Figure 4-44 shows plots of distillation column power requirements and turbine power loss against equilibrium maximum water concentration in the evaporator. The power loss curve is based on computer runs of ammonia water mixtures in the OTEC cycle by Dr. Kenneth Starling of the University of Oklahoma. The resultant composite curve shows a minimum overall power loss of 3.7 MW(e) (1.46 MW(e) for the column, 2.24 MW(e) turbine power loss) at 5-percent water concentration. This represents an approximate 1-MW(e) per module power savings over the previous baseline 2-percent water concentration design value.

It should be noted that this tradeoff assumes that other effects which may result from increased water concentration (e.g., changes in heat transfer coefficient and amount of heat transferred) are secondary.

<u>Ammonia Storage and Transfer System.</u> The OTEC plant will utilize large quantities of ammonia in the power systems. Not only will the systems require an initial fill and makeup, but there will occasionally be a need to transfer ammonia, pump down and

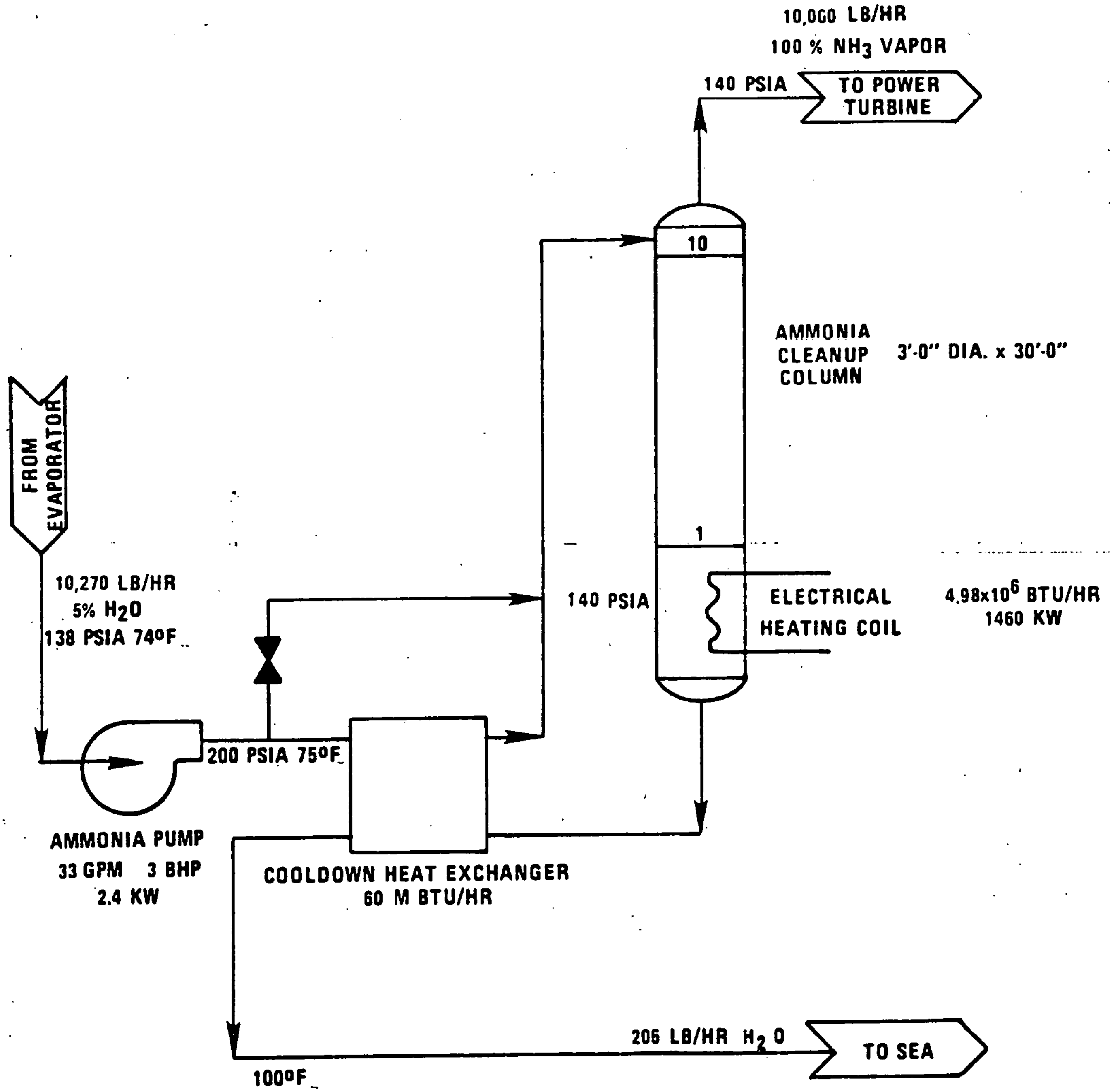


Fig. 4-43 Ammonia Cleanup System

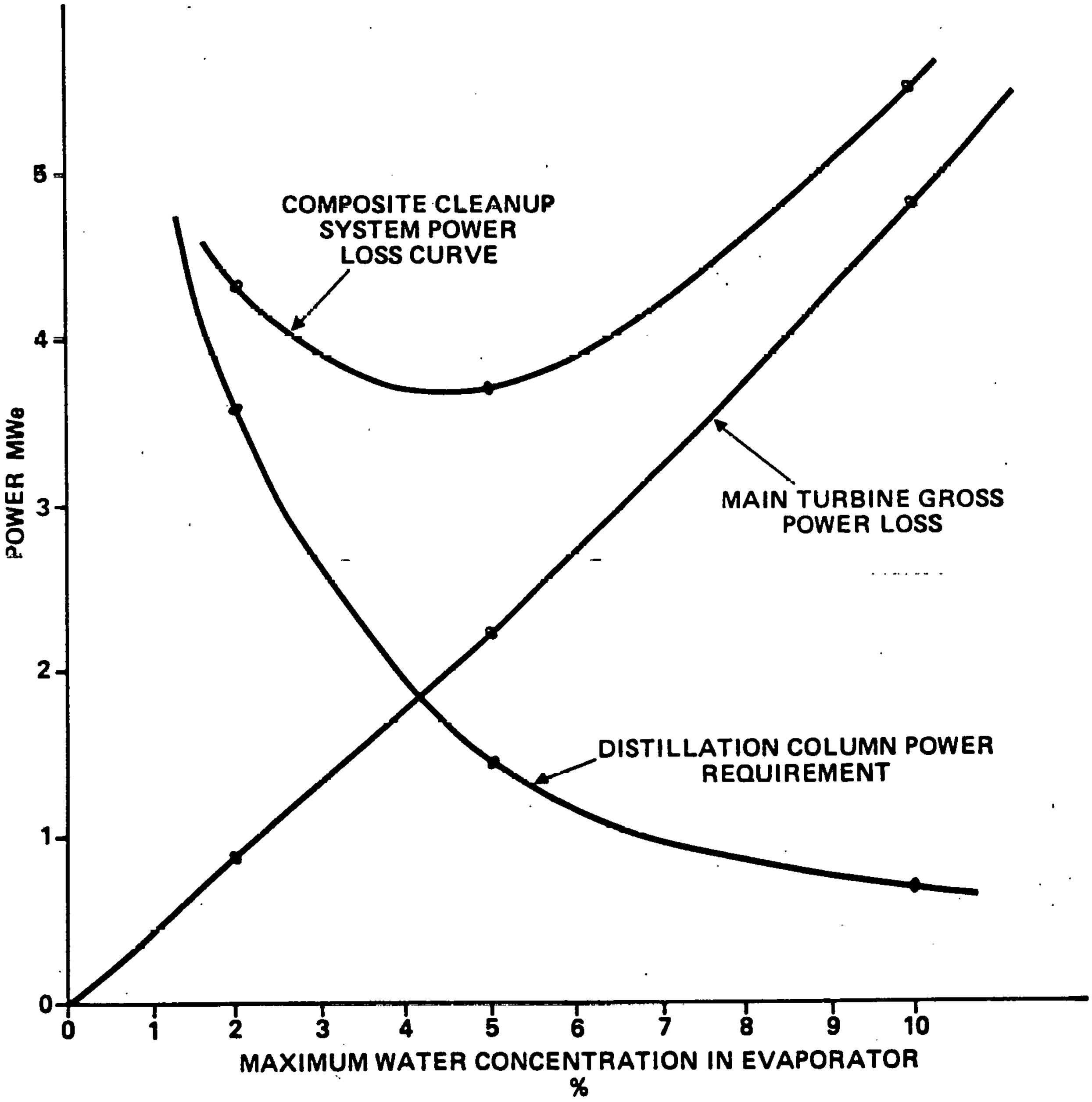


Fig. 4-44 Cleanup System Power Considerations

purge a system, and refill or dispose of fluid. For these procedures, an ammonia storage fill-and-purge system is required.

The system consists of a main storage tank, compressors, and the necessary tanks, piping, valves, and safety provisions, as shown in Fig. 4-45.

The ammonia storage tank is located on the platform or in the core structure section. The tank is designed for a pressure of 250 psig, as required by OSHA standards for nonrefrigerated storage of ammonia. The capacity of the tank is 55,000 cu ft, including OSHA ullage requirement, sufficient to hold approximately 1.8×10^6 lb of liquid ammonia. Based on 45-sec evaporator and 30-sec condenser sumps, one 25-MW(e) power module requires approximately 360,000 lb of ammonia. Therefore, the capacity of the storage tank is sufficient to contain the ammonia inventory of four modules plus one spare refill. Operating experience with test articles will probably modify this required capacity in later full-scale designs.

Each power module is provided with a transfer pump capable of pumping liquid ammonia from the power system into the tank at the tank's highest pressure. An ammonia vaporizer is used when transferring ammonia from storage into the system initially when throttling to low pressures could otherwise cause extremely low temperatures. The compressors are used to compress and liquefy gaseous ammonia when this is necessary and to evacuate either air or ammonia vapor from the system.

At certain times during operation of the OTEC plant, it will be necessary to shut down and allow workmen to enter the equipment or piping of the ammonia system for the purpose of performing required maintenance. During these times, part of the ammonia system or the entire ammonia system must be purged of ammonia and replaced with air that is safe for breathing. Air cannot be mixed with ammonia, except at very low concentrations, since ammonia in higher concentrations may form flammable or even explosive mixtures. To avoid mixing air and ammonia directly, the ammonia vapor is replaced with nitrogen before the air is allowed to fill the space. The compressor and liquefier are used to compress and liquefy gaseous ammonia, and then the liquid ammonia is returned to the storage tank.

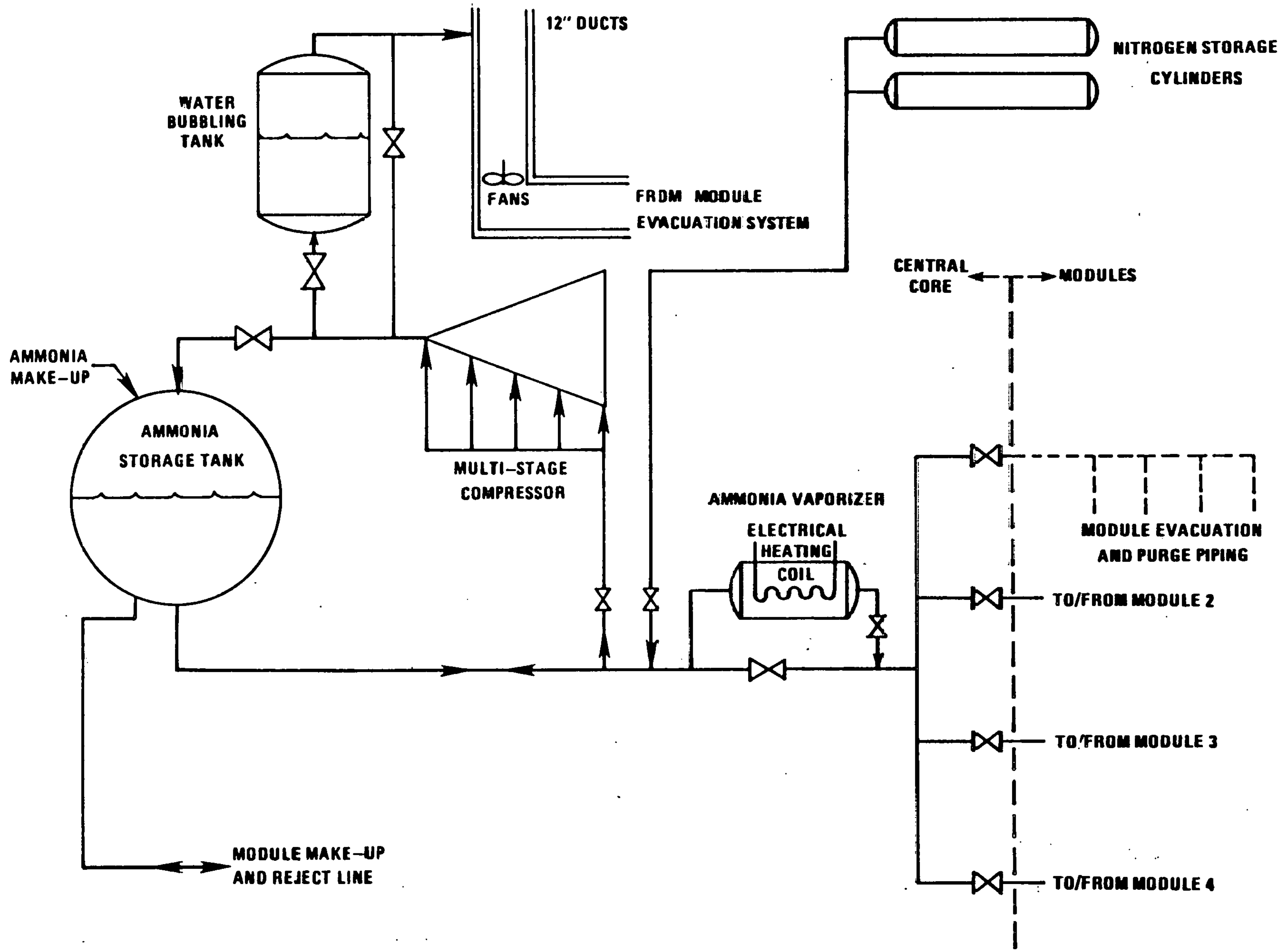


Fig. 4-45 Ammonia Storage, Fill, and Purge System

The nitrogen required for the purge operation is stored in a central bank of high-pressure (2,400 psig) cylinders. The cylinders discharge into the same piping that is used for transferring ammonia. Nitrogen is also used to pressurize the space between the evaporator and condenser double tubesheets, if utilized. The nitrogen pressure in this space is maintained slightly higher than that of the ammonia or the seawater to prevent any ammonia outflow or seawater inflow. Each module has its own pressurizing system and the module cylinders can be recharged as necessary from the central bank of high-pressure cylinders.

The following procedure is used when a small portion of the ammonia system must be isolated for maintenance of a pump or valve.

After isolating the volume that is to be purged,

1. Pump any liquid ammonia into storage.
2. Evacuate the volume to a pressure of 3 psia while compressing, liquefying, and storing the ammonia.
3. Fill the volume with air at one-atmosphere pressure.
4. Repeat Steps 2 and 3 and sample mixture for ammonia content.
5. Evacuate the volume to 3 psia again and vent the mixture of gases at the discharge of the compressors by bubbling through water.
6. Fill the volume with air at one-atmosphere pressure.
7. Circulate fresh air through the volume until measurements indicate that the air is safe for breathing.
8. After completing the required work, evacuate the volume to 3 psia.
9. Fill the volume with nitrogen at one-atmosphere pressure.
10. Repeat Steps 8 and 9 and sample mixture for air content.
11. Open valves in the piping from the ammonia storage tank and vaporize ammonia to fill the system to the required pressure.
12. Transfer liquid ammonia to the condenser sump.
13. Circulate liquid ammonia through the ammonia system, commence startup procedures and resume normal operation. The remaining nitrogen is vented from the system through an ammonia trap at the condenser noncondensibles vent.

Residual gases are ducted to the sea surface open air by two 12-in. ducts which lead from the central machinery space to the open air at the top of the platform structure. Each duct is provided with a fan at the lower end, which assists in expelling the air from the vent duct.

For the evaporator or condenser, the transition from an active to inert stage, or vice versa, is handled in a different manner. Since access to the internal portion of the heat exchangers is not required when the module is attached to the platform, this operation takes place as a part of system startup or system shutdown for transit to the refurbishing area. During transit, the system is pressurized with nitrogen at 25 psia.

The installation and startup procedure is as follows:

1. Prior to submergence, evacuate the system to 1 psia using equipment on board the support vessel to compress and store the nitrogen.
2. Fill the system with sufficient ammonia to raise the internal pressure to 60 psia.
3. Complete the submergence and installation of the module.
4. Fill the system with the remainder of ammonia required to attain stable operation and use the noncondensible gas removal system to remove the residual nitrogen.

The shutdown and detachment procedure is as follows:

1. Remove the liquid ammonia from the system.
2. Remove gaseous ammonia to reduce the system pressure to 60 psia. Use the noncondensible gas removal system secondary condenser to liquefy the ammonia and return to storage.
3. Detach the module and raise to surface.
4. Evacuate the system to 1 psia using the support vessel equipment to liquefy and store the ammonia.
5. Fill with nitrogen to 25 psia for transit.

Only one common system is used, as all four modules should not require filling or evacuation at the same time. Typical evacuation time for a portion of one module is about 12 hours. The evacuation time for the entire ammonia system of one module would not take longer than 24 hours.

In arriving at the recommended scheme for ammonia purging of the system, two other options were considered and rejected. These were (1) compression and evacuation and (2) nitrogen purge.

The compression and evacuation scheme did not use nitrogen as an inerting gas and was dropped since even after evacuation, compression, and storage of most of the ammonia from the system it is not possible to avoid flammable ammonia-air mixtures for short periods of time when filling the system with air to perform maintenance or to reduce the ammonia quantity to a safe physiological level. Similarly, a flammable condition would be temporarily experienced when filling the system with ammonia after evacuating most of the air.

The nitrogen purge scheme, on the other hand, did not use evacuation and compression but involved venting of the ammonia to atmospheric pressure through a flare stack, repressurizing the volume with nitrogen to dilute the ammonia, and then venting the ammonia-nitrogen mixture by bubbling through water. The volume was then flooded with air for maintenance and the reverse procedure followed when maintenance is complete. This scheme was dropped from consideration since, although flammable mixtures are avoided, considerable nitrogen and ammonia are lost on each turnaround purge occurrence. In addition to the expense associated with the loss of both nitrogen and ammonia, the environmental problems associated with appreciable ammonia release were considered unacceptable. The recommended scheme – which involves evacuation, compression, and nitrogen purge – is the most expensive from a capital cost standpoint, but avoids the problems associated with flammable mixtures, appreciable fluid loss, and environmental impact.

4.8.2 Auxiliary Systems

The auxiliary systems described in this subsection are those systems that are not directly ammonia related, but are considered major auxiliary systems for the OTEC power plant. These include the electrical system, standby power system, fire protection system, and cooling water system.

Electrical System. An electrical system is provided to interconnect the module generators to the platform rectifiers and to distribute power to the module and platform auxiliary electrical equipment. Direct current transmission of electrical power systems are provided from the platform to ensure the safety of the module. In the event of black startup (no power modules operating), provision is made for using startup power from a source outside the power module. The electrical system is designed to function in four distinct modes of operation: startup, normal, standby, and emergency.

Startup power is supplied at 13,800 volts into the normal power distribution system and is capable of (a) providing the estimated startup power requirement of 230 kW for the platform and a 500 kW module auxiliary load, plus an ammonia feed pump and circulation pump driver power requirement of 740 kW and (b) then sequentially starting two 1,050-horsepower cold seawater pumps and two 875-horsepower warm seawater pumps. The total level of external power required during module startup is 4400 kW. About 90 percent of the external startup power required is used at 4,160 volts, the remainder is used at 480 volts and 120 volts. Startup of the remaining seawater pumps and auxiliaries can be accomplished using module-generated power.

Normal power is generated at 13,800 volts and distributed at 4,160 volts. Each power module generator is directly connected to a 13,800-volt bus. Generating capacity available for rectification is then transmitted to rectifier summing transformers by the 13,800-volt system. Power consumed by the platform and modules is distributed at 4,160 volts and 480 volts. A single line diagram of the electrical system is shown in Bechtel Drawing 12384-E-1 (Fig. 4-46).

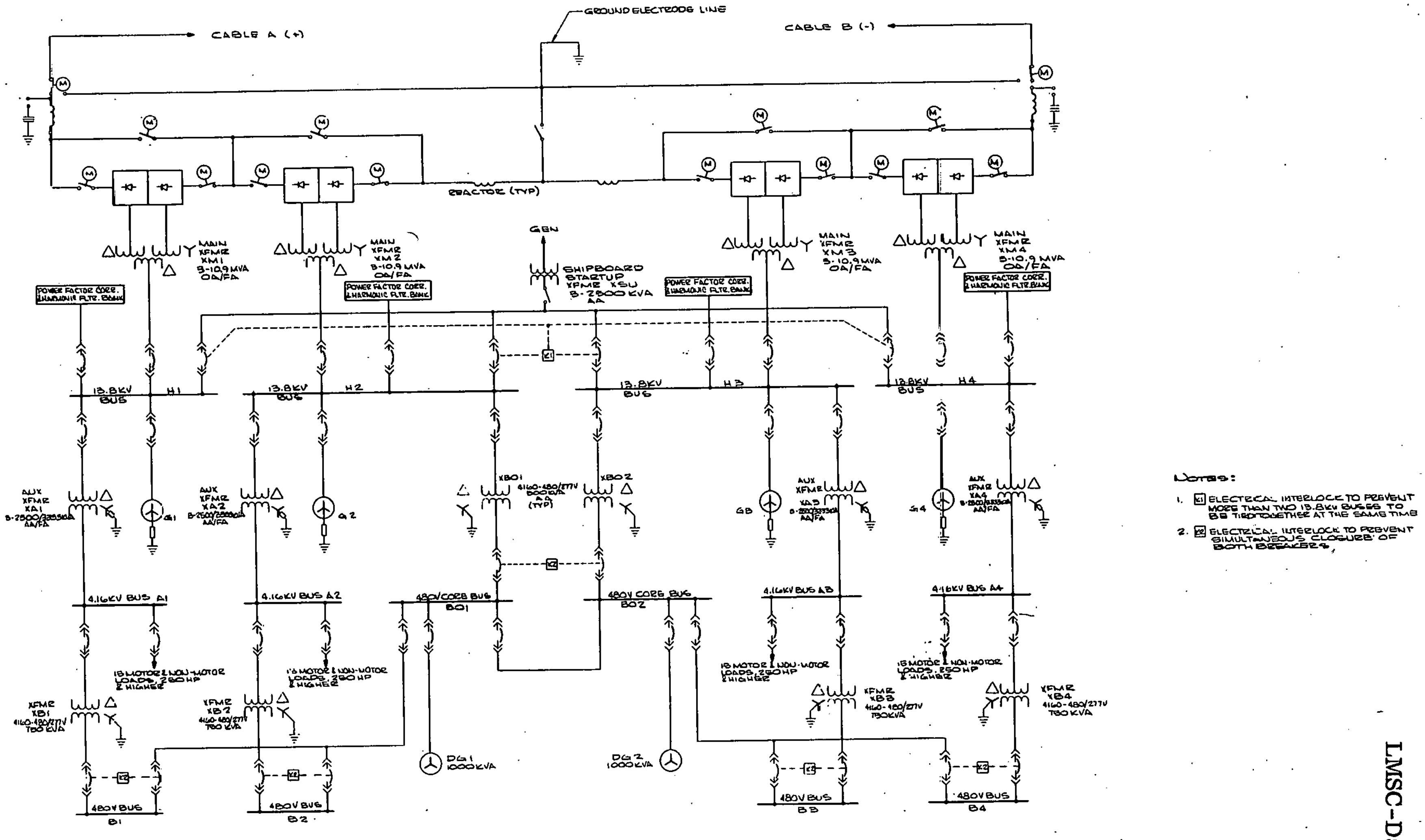


Fig. 4-46 Electrical System Diagram

Table 4-20

POWER PLANT ELECTRIC POWER OUTPUTS

(Submerged Configuration)

Item		Output (kw)
Gross Generator Output		130,934
Plant Cycle Auxiliaries	30,934	
Module Support Systems	4,212	
Core Support Systems	232	
Internal Transmission Losses	314	
Input to Step-Up Transformers		95,242
Step-Up Transformer and Rectification Losses	1,905	
Net Electric Output of Plant		93,337
		(23,334 MW per module)

During the preliminary design phase, the module will be resized so that the net power available for transmission is 100,000 kW. Under maximum thermal gradient conditions, the gross generator output is approximately 40,000 kW and the plant net power available for transmission is approximately 125,000 kW.

Standby power is generated at 4,160 volts and distributed at 480 volts into the standby power distribution system through two independent connections (one for each of the two standby generators). Total level of power required during standby conditions is approximately 930 kW. System capability is based on starting and running an additional 100-horsepower motor over and above a nominal 830-kW running load without dropping the running loads.

Electrical equipment weight is 1,314,000 lb in the platform plus 98,000 lb per module. Total cable and bus weight is 64,000 lb for a total system weight of 1,770,000 lb.

In arriving at the above baseline electrical system approach, several alternatives were considered, including operating the seawater pump motors at 13.8 kV, operating the generators at 4,160 V, modifying the bus arrangements, reducing the number of transformers, etc. However, the baseline approach represents the best combination of flexibility, backup capability, and low-cost standard equipment.

Standby Power System. The OTEC plant will be installed at a considerable distance offshore and will be physically isolated. The generated power will probably be transmitted via a dc cable to the mainland. It is therefore desirable to have standby power on board to provide auxiliary power during total shutdowns and to provide emergency services necessary to ensure the safety of the power module and its operating personnel. For reliability, standby power is provided in duplicate.

Two 1-MW(e) diesel generators will be located on the platform. The diesel-generator units were selected over gas turbines since they are considered to provide a higher level of reliability for standby service.

Fire Protection. In a large power plant, there is always the possibility of fire involving flammable liquids and gases, electrical insulation, etc. In the OTEC plant, the ammonia working fluid itself is flammable in the right concentrations. It is therefore necessary to incorporate a fire protection system into the overall plant design. As discussed in Section 4.7.5, Ammonia Plant Safety, ammonia system fire protection is most effectively achieved through the use of water spray and dilution combined with adequate ventilation. Carbon dioxide units can be used in non-ammonia related equipment areas, particularly for electrical.

For the CO_2 extinguishing system, the CO_2 is stored in a refrigerated storage tank. Remote temperature sensors or manual control triggers the CO_2 discharge which is automatically routed to nozzles in the endangered zone. Discharge duration is automatically timed to prevent exhausting the CO_2 inventory during a single discharge. Whenever the system is activated, an alarm is sounded and an annunciator in the control room shows where the sequence was initiated. The release of CO_2 will automatically order an evacuation of the space by personnel and will seal the ventilation system. Reentry into the space will be made by personnel equipped with breathing apparatus.

All other areas on the platform and in the power modules are provided with overhead spray nozzles connected by piping to the firewater storage tank. Two firewater pumps are provided. The primary pump is driven by an electric motor and the backup pump is diesel driven. Piping is also provided to the pump suction from a filtered seawater intake as a backup water supply.

Cooling Water System. The purpose of the cooling water system is to provide cooling for bearings, generators, lubricating oil, air compressors, refrigeration units, air conditioning system, and miscellaneous plant equipment. The system provides freshwater cooling to power plant machinery. A closed loop is used to isolate the machinery coolers from the external seawater pressure and to reduce corrosion in the cooling systems. Heat is received from the machinery coolers and rejected to the seawater in a freshwater/seawater heat exchanger. During the Preliminary Design Task, the use of freshwater/ammonia heat exchangers will be evaluated. The seawater and freshwater are both circulated by pumps which are installed in duplicate. A surge tank is installed in the freshwater circuit to allow for expansion and to provide a fixed suction head for the pumps. The heat exchanger is a single-pass counterflow unit with a divided waterbox at each end, so that one-half can be cleaned without a complete shutdown of the unit. A second redundant unit is provided to improve overall system reliability. The surge tank is a 1,000-gallon capacity vertical unit complete with overflow, suction, discharge and drain connections, and gauge glass and ladder. The pumps, heat exchangers, and surge tank constitute a common system to all power modules and are located in the central core machinery space.

Total heat rejection for the 100-MW(e) plant is estimated to be 13.0×10^6 Btu/hr, requiring seawater and freshwater flow rates of 700 gpm and 850 gpm, respectively.

4.9 MODULE ASSEMBLY

The power system module assembly study delineates various methods that could be applied to produce the completed end article at a major shipyard. It was determined that a shipyard is best suited for the structural fabrication, component assembly and installation, and final outfitting requirements inherent with each power system configuration that was considered, particularly in view of the logistics problems and the implications of potentially conflicting labor jurisdictions. Waterfront location, shops essential to support the production requirements, and every principle craft with clearly defined labor jurisdiction agreements are all found in major U.S. shipyards. And it was further determined that a major shipyard would not face any requirements for compliance with OSHA or other government regulatory agencies in the production of this article other than those associated with ship construction.

4.9.1 Basis and Conditions

Configurations Reviewed. The 100-MW(e) power systems which were reviewed from the standpoint of construction, installation, facilities, and costs are as follows:

- 300-ft × 750-ft Surface Platform 25-MW(e) size Monolithic Heat Exchangers
- 300-ft × 750-ft Surface Platform 12.5-MW(e) size Heat Exchangers
- 175-ft × 1,120-ft Supertankers consisting of:
  - Twenty 5-MW(e) size heat exchanger sets
  - Eight 12.5-MW(e) size heat exchanger sets
  - Four 25-MW(e) size heat exchanger sets

Power modules for spar platform were as follows:

- Ten 5-MW(e) power modules with 2 sets of heat exchangers each
- Four 12.5-MW(e) (Dual) power modules with 2 sets of heat exchangers each
- Four 25-MW(e) power modules with 1 set of heat exchangers

The 5-MW(e) test article in a T2 tanker was also reviewed.

Basis of Study. This study evaluated the above configurations for assembly and cost comparisons, establishment of construction approaches, the identification of potential problems inherent with the installation of equipments, and to make recommendations of methods and facilities to support the installation and construction. This study did not include costs and problems associated with outfitting. While the installation of small equipment, auxiliaries, accommodations, and outfitting constitutes much of the final construction costs, it was difficult in the conceptual design stage to establish with any degree of certainty their costs and potential problems. Eventually these systems will affect producibility and maintenance; therefore, they will be considered in the preliminary design when more adequate definition exists.

4.9.2 Facility Requirements for Installation

Handling Requirements. Studies of the various power system configurations indicated that the first consideration for their installation and assembly must be the handling of heavy lifts. The heat exchangers comprise the maximum weight and size lifts of all components in all systems under study. Approximate weights of the various heat exchangers are indicated in Table 4-21.

Table 4-21
HEAT EXCHANGER WEIGHTS

Heat Exchanger Configuration	Approximate Weight (tons)					
	Condenser			Evaporator		
	Al. Alloy	Titanium	Stainless	Al. Alloy	Titanium	Stainless
25-MW(e) (Surface)	1100	1830	2200	1050	1750	2100
25-MW(e) (Submerged)	1000	1550	1830	900	1400	1650
12.5-MW(e) (Surface)	700	1160	1390	660	1100	1320
12.5-MW(e) (Submerged)	640	990	1170	570	890	1050
5-MW(e) (Surface)	310	530	640	290	500	590
5-MW(e) (Submerged)	280	450	530	260	430	510

Surface Platforms and Supertankers. Various alternatives for installing the heavy components into surface platforms and supertankers are as follows:

- Graving dock with overhead gantries, since whirley-type cranes fall far short of the heavy lift requirements
- Gantry loading facility where components could be lifted from seagoing barges and positioned in hull of ship or platform
- Stiff-leg or shear-leg at a pier where loads could be lifted from seagoing barges and positioned in hull of ship or platform
- Barge cranes with a spreader beam between the cranes lifting from each side of seagoing barge then moving out the seagoing barge while bringing in the ship or surface platform

Pier-Side Installation. Foundations for the components determined by their weight would be installed at pier side or by the foregoing heavy lift means. Piping, valves, and other light equipment would be installed pier side. The major concern, however, is the extreme reach required to install equipment pier side in a surface platform with a 300-ft beam and a supertanker with a 176-ft beam.

Power Modules Installation. The power modules for a spar platform have approximately 10 percent less heat exchanger weight and present far less problem in the reach of the lift during assembly. If one were to consider the power modules assembled pier side on a 100-ft beam barge the maximum reach from the pier for lifts would be 50 ft as compared to 88 ft for a supertanker and 150 ft for the 300-ft beam surface platform. Furthermore, in the case of the surface platform and supertanker, clearance over the side of the hull is required, necessitating elevation of cranes to clear the hull. In the case of a gantry over a graving dock, the crane must clear the superstructure of the ship.

4.9.3 Existing Facilities

United States Shipyards. An investigation of U.S. shipyards disclosed that the maximum handling capacity was that of General Dynamics Corporation, Quincy Shipbuilding Division. A 1,200-ton Goliath Crane – the largest in the western hemisphere – has been installed for transferring spherical LNG tanks from the barge on which they are delivered to LNG ships. Their largest dock (936 ft by 143 ft) is not large enough for the supertanker under study.

The second largest handling capacity is located at Newport News Shipbuilding and Dry Dock Company. A 900-metric-ton (992-ton) Gantry Crane services their number 12 building dock. This dock is 1,600 ft long and 240 ft breadth at entrance, which is large enough for the supertanker under consideration; however, the handling capability falls about 10 percent short of the lightest 25-MW(e) size heat exchangers. (Refer to Table 4-21.)

Bethlehem Steel Company at Sparrows Point, Maryland, was the only other U.S. facility capable of dry docking the supertanker, but their handling is limited to 200 tons.

U.S. major dry docks and handling is given in Table 4-22.

Table 4-22

MAJOR SHIPBUILDING AND REPAIR DOCKS – UNITED STATES, 300,000 DWT TONS OR OVER

Length (ft)	Breadth (ft)	Draft (ft)	Weight (DWT Tons)	Service	Owner	Location
1,600	250	33	500,000	One 990-ton crane	Newport News Shipbuilding and Dry Dock Company	Newport News, Va.
1,200	200	28	400,000	Two 100-ton cranes	Bethlehem Steel Company	Sparrows Pt., Md.
1,200	120[(a)]	42	500,000	Not Known	U. S. Navy	Boston, Mass.
1,152	157[(a)]	53	500,000	Five 50-ton cranes	U. S. Navy	Bremerton, Wash.

(a) Fall below minimum for 176-ft beam of supertanker.

Other U. S. shipyards with single lift handling in excess of 200-tons are:

- Sun Shipbuilding and Dry Dock Company, one 800-ton barge crane and three 250-ton gantry cranes
- Avondale Shipyard, Inc., one 600-ton barge crane
- Alabama Dry Dock and Shipbuilding Company, one 275-ton Goliath Bridge Crane
- Marathon LeTourneau Company, Gulf Marine Division, one 250-ton gantry crane

Figure 4-47 is a layout of the Newport News Shipbuilding and Dry Dock Company's dock and crane facility. It was determined that this facility is the most favorable for the supertanker installation. Note that the 900-metric-ton gantry crane spans the final assembly area, a barge unloading area, and the large graving dock.

Commercial Barge Cranes. Due to the limited crane capacities of shipyards, an investigation was made into commercial barge cranes available for subcontract. Although barge cranes capable of handling about 3,000 tons have been built, they are for special application on northern oil exploration and production platforms and unless some unique agreement could be reached with the owners they would not be available for normal commercial use. Barge cranes that were available showed an absence of ample capacity for the 25-MW(e) size heat exchangers. Various barge cranes, capacities, owners, and location are indicated in Table 4-23.

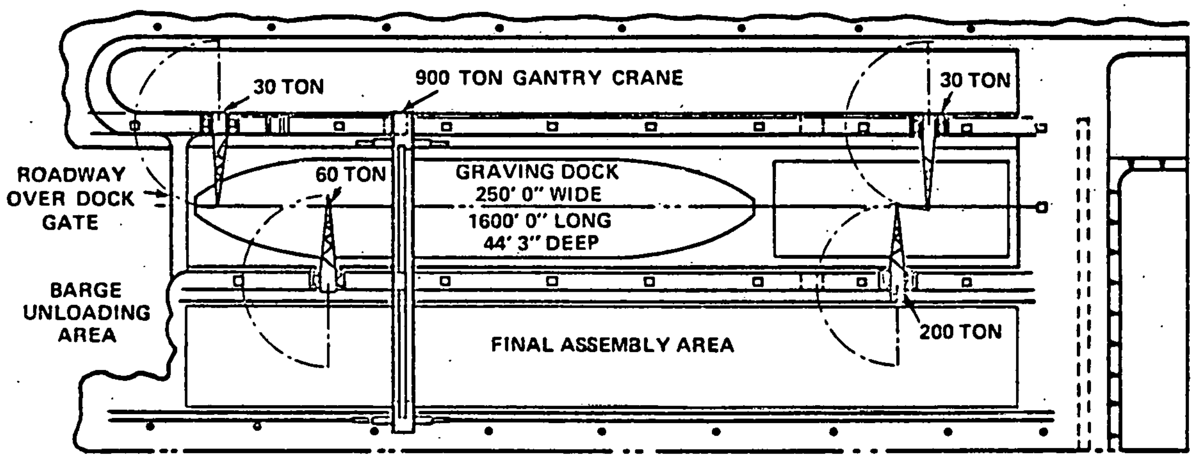


Fig. 4-47 Dock and Crane Facility

Table 4-23

AVAILABLE BARGE CRANES

Location	Owner	Cap. Over Stern (tons)	Distance From Stern to Hook (ft)	Cap. Revolving (tons)
Seattle	Manson Construction	400	55	300
S.F. Bay Area	Smith-Rice	410	65	325
Los Angeles	Crowley Maritime	385	60	275
Los Angeles	U. S. Navy	385	54	250
Houston	Brown and Root	550	72	390
Houston	Brown and Root	350	67	230

Heavy Lift Specialists. Further search for heavy lift capabilities lead to consideration of heavy lift specialists, such as the firms that handle nuclear reactors and oil platform modules for the Alaska North Slope. Rigging International with headquarters in Oakland, California, was contacted. They currently have a 1,550-ton twin luffing derrick at Pinto Island, Mobile, Alabama. This facility was designed and constructed to install LNG tanks in large LNG ships under a contract with Kaiser Aluminum and Chemical Sales Corporation.

Another facility, owned by Rigging International, is a 660-ton stiff-leg at Longview, Washington. This facility can lift 600 tons 90 ft from pier side.

Shipyard Facilities in Japan. For information purposes a tabulation of building docks and repair docks in Japan along with crane capacities is presented in Tables 4-24 and 4-25. It appears that the only graving dock in the world capable of handling the 300-ft beam of the surface platform is in Japan. Although handling facilities in Japanese shipyards in general are much better than in the United States, they still fall short of the heavy lift requirements. The best handling in Japan's docks would indicate 800 tons and several at 600 tons.

Table 4-24

MAJOR SHIPBUILDING DOCKS – JAPAN, 300,000 DWT TONS OR OVER

Length (ft)	Breadth (ft)	Draft (ft)	Weight (DWT Tons)	Service	Owner	Location
3,248	328	38	Not Ltd.	Two 300-ton cranes	Mitsubishi Heavy Industrial, Ltd.	Koyagi
2,657	301	46	500,000	Two 400-ton cranes One 100-ton crane	Ishikawajima - Harima Heavy Industrial Company, Ltd.	Chita
1,667	262	27	500,000	Two 200-ton cranes Two 50-ton cranes	Ishikawajima - Harima Heavy Industrial Company, Ltd.	Kure
1,230	183	38	500,000	Two 300-ton cranes	Mitsubishi Heavy Industrial, Ltd.	Nagasaki
1,131	213	20	300,000	Two 150-ton cranes	Ishikawajima - Harima Heavy Industrial Company, Ltd.	Kure
1,837	262	41	500,000	Two 300-ton cranes Three 30-ton cranes	Sumitomo Shipbuilding and Machinery Company, Ltd.	Oppama
1,640	246	39	600,000	Two 300-ton cranes Two 100-ton cranes	Nippon Kokan	Tsu
1,362	246	30	500,000	Two 300-ton cranes Three 150-ton cranes	Kawasaki Heavy Industrial, Ltd.	Sakaide
1,312	236	28	500,000	Two 300-ton cranes One 40-ton crane	Mitsui Shipbuilding and Engineering Company, Ltd.	Chiba
1,312	180	25	300,000	Three 200-ton cranes	Hitachi Shipbuilding and Engineering Company, Ltd.	Osaka
1,181	197	25	300,000	Two 250-ton cranes	Hakodate Dock Company, Ltd.	Hakodate

Table 4-25

MAJOR SHIP REPAIR DOCKS – JAPAN, 300,000 DWT TONS OR OVER

Length (ft)	Breadth (ft)	Draft (ft)	Weight (DWT Tons)	Service	Owner	Location
1,312	328	40	500,000	Two 300-ton cranes	Mitsubishi Heavy Industrial, Ltd.	Nagasaki
1,493	203	25	400,000	One 120-ton crane	Hitachi Shipbuilding and Engineering Company, Ltd.	Osaka
1,450	223	34	500,000	Two 200-ton cranes	Kawasaki Heavy Industrial, Ltd.	Sakaide
1,233	203	27	300,000	Two 150-ton cranes	Kawasaki Heavy Industrial, Ltd.	Sakaide
1,230	246	46	600,000	Two 250-ton cranes	Nippon Kokan	Tsu

4.9.4 Proposed Facility Concepts

Graving Dock and Gantry Crane Facilities. Figures 4-48 and 4-49 depict a graving dock facility with a gantry crane over the dock for the supertanker under study. Wing walls extend the gantry coverage beyond the miter gates allowing the gantry to remove loads from a barge and transfer to the ship. Capacities are approximate; actual capacities required under this study are those shown in Table 4-21 for the various heat exchangers. This concept is the most practical for the lowest installation costs of the supertanker configurations, and would also provide the lowest installation costs for the 300-ft beam surface platform; however, it would be too expensive and impractical to build such a facility due to the enormous span of the gantry crane (approximately 425- to 450-ft span).

Stiff-Leg or Luffing Derrick. Due to the extreme beam (300 ft) of the surface platform, the only concept determined feasible for the handling of the heavy lifts is a stiff-leg or luffing derrick crane. Due to the back stays, luffing and lifting is the only capable movement of the load. Heavy components would be lifted from a seagoing barge

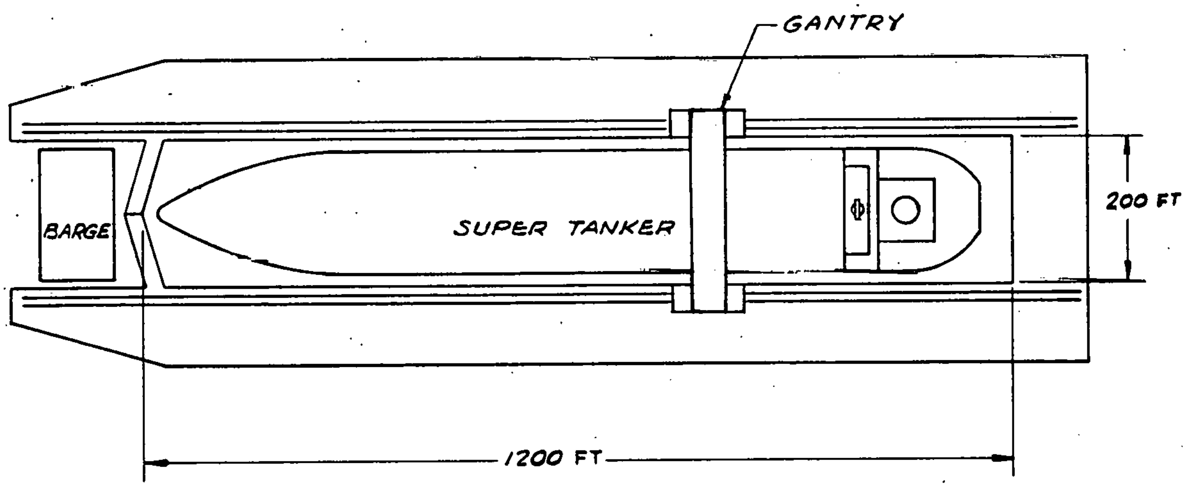


Fig. 4-48 Graving Dock Facility (Plan)

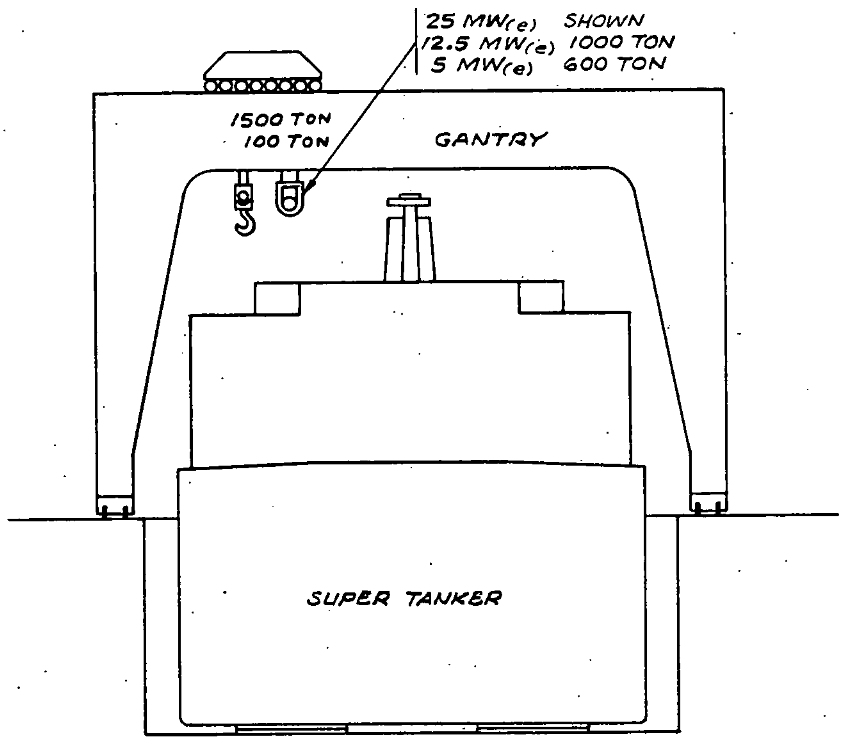


Fig. 4-49 Graving Dock Facility (Elevation)

positioned at pier side, then the barge moved out and the platform shifted into exact longitudinal position. The load can now be lowered into the hull and positioned transversely. Light lifts during outfitting would be handled by whirley-type cranes, although few facilities could handle lifts to the center of the platform (150 ft from pier side). The stiff-leg is also considered a means for the supertanker installation. Illustration of the stiff-leg is shown in Fig. 4-50.

Graving Dock or Building Ways for Power Modules. Figure 4-51 depicts a plan of either a graving dock or building ways for construction of a 25-MW(e) power module. The graving dock would be more effective for installation than the building ways. The graving dock also has the capability to inload a power module for repair or refurbishment. Another concept having the inload capability is a flat or near flat building way with track and wheel launch onto a sloping ramp. This, however, would be somewhat

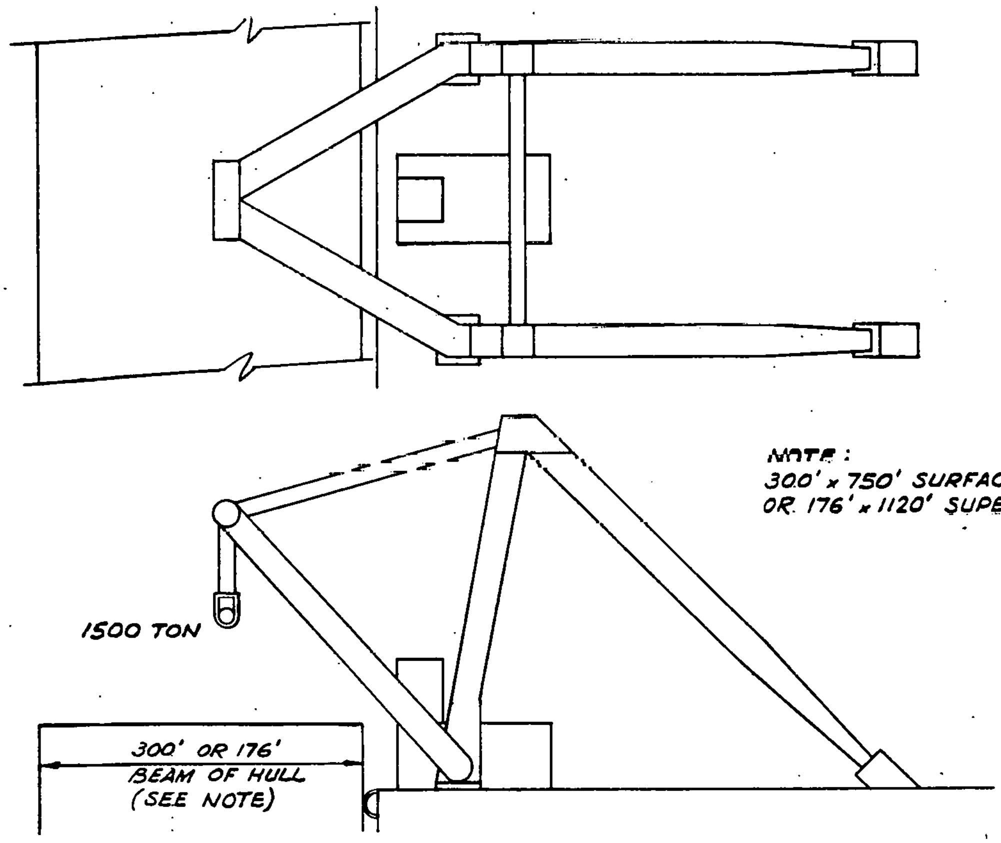
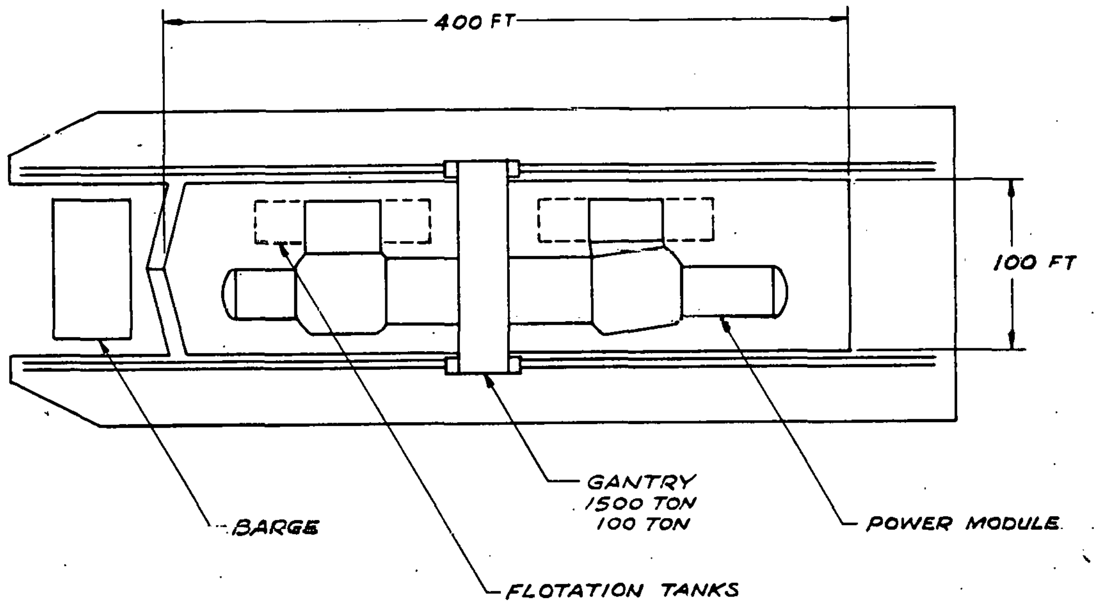


Fig. 4-50 100-MW(e) Surface Platform Heavy Lift Facility



ALTERNATE CONCEPT : SHIPWAY IN PLACE OF GRAVING DOCK

Fig. 4-51 Graving Dock Facility (Power Module)

more cumbersome and costly in receiving barge loads of heavy components, requiring an extremely long gantry extension seaside to unload the barge.

Conceptual Facility for Power Modules. Perhaps the most effective facility for the power modules would be similar to the concept illustrated in Fig. 4-52. This concept has the advantage of working undercover with the exception of the heavy lift transfers by the stiff-leg. Production could be made more efficient by adding slips and shop barges. Two slips could be built side by side using a common center column line for overhead support of bridge cranes. Water in the slip could be relatively shallow and the barge allowed to bottom out during low tide. Movements could be scheduled during high tide. The barge would be made floodable for the launching or receiving of a power module. The stiff-leg should be built to handle the largest lift proposed. Fabrications from shops or slabs would be moved to the loading pad at the rear of the slip where the shop bridge crane could move outside, pick up the load, and move it into the shop assembly area. The shop crane capacity should be designed capable of handling all lifts received by railroad cars or trucks, as well as yard capacity.

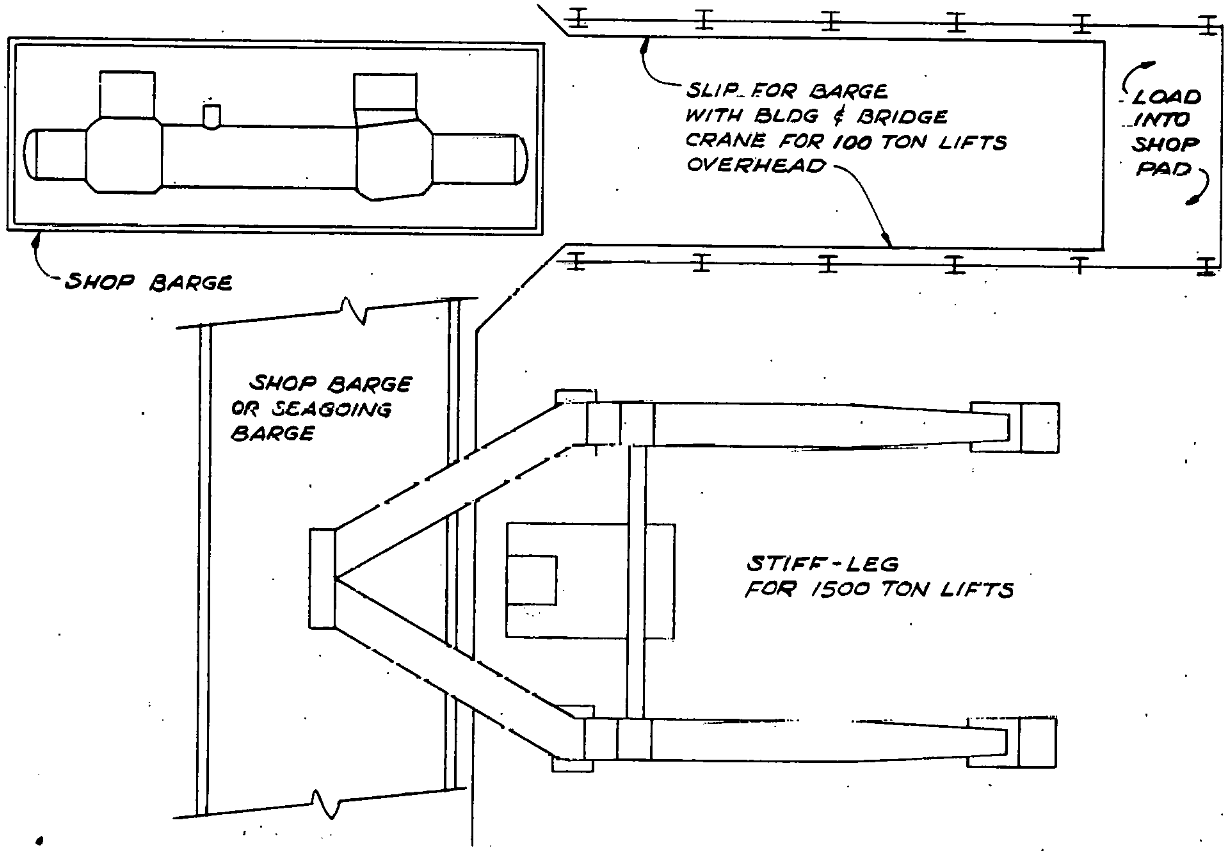


Fig. 4-52 25-MW(e) Assembly and Refurbish Facility

Movements by Other Than Cranes. Methods for the movement of heavy components by other than cranes that could be considered are:

- Wheel dollies
- Rolling track dollies
- Wheel carriages on tracks
- Pneumatic lift pallets
- Hydro lift pallets

All these methods are widely used in the industry, but are best suited to rigid structures. In the case of heat exchangers for OTEC, care must be taken to avoid the transfer of concentrated loads into the vessel shell. Though the load to be lifted would be increased, the mounting foundations could be assembled to the heat exchangers and used to guard against load concentration. Transfers of loads without cranes is normally accomplished by hydraulic jacking. Methods for the actual movements must be determined and developed for the specific item being moved and the facilities available at a definite location.

4.9.5 Costs for Facility Handling Concepts

Acquisition costs of crane handling facilities are shown in Fig. 4-53. This graph is based on a stiff-leg capable of reaching the load capacity at 120 ft from a pier, and on a gantry crane that would span a 220-ft-wide dock and lifting a capacity load on center. Cost figures to develop these curves were received from Rigging International for the stiff-leg, and Paceco for the gantry over dock.

4.9.6 Installation Cost Comparisons

100-MW(e) installation costs estimates are given in Table 4-26. In order to have a better cost comparison of the various configurations, installation of equipment in the surface platforms and supertankers was limited to those components that are evident in the power modules, including piping and valves.

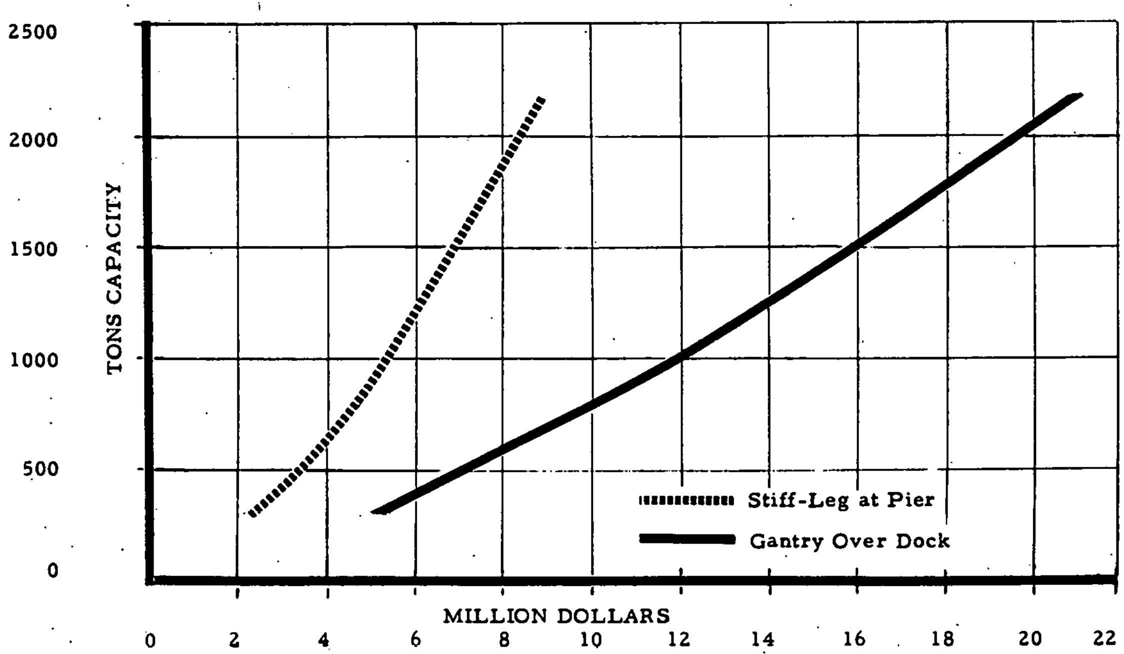


Fig. 4-53 Acquisition Cost of Crane Handling Facilities

Table 4-26

100-MW(e) SYSTEM INSTALLATION COSTS

	Power Modules			Supertankers			Surface Platforms	
MW(e) System Size	5	12.5	25	5	12.5	25	Monolithic	Multiple
Condensers	270	200	160	250	220	170	190	230
Evaporators	270	200	160	250	220	170	190	230
Generators	100	80	60	150	120	80	100	120
Turbines	80	60	40	110	90	60	80	100
Seawater Pumps	600	400	250	470	390	320	340	320
Condensate Pumps, Distribution Pumps, Piping and Valves	3,450	2,530	2,000	6,650	5,700	4,650	5,300	6,400
Foundations (Install)	4,350	3,000	2,500	8,350	7,200	5,850	7,000	8,450
Foundations (Make)	6,100	4,200	3,500	11,950	10,250	8,480	9,800	11,800
TOTAL	15,220	10,670	8,670	28,180	24,190	19,780	23,000	27,650

NOTE: Costs in Thousands of Dollars

The basis of the costs indicated in Table 4.9-6 are

- Mature industry
- A 100-MW(e) system
- Labor, burdens, including profits at $20 per hour
- Aluminum alloy tubed heat exchangers
- Costs for labor only, except foundations (make) (material included)
- Engineering and detail working drawings furnished by others
- No structure labor or materials included
- Best judgment factors

4.9.7 5-MW(e) Test Article in T2 Tanker

Construction Plan. The sequences and construction approach for this installation would be similar to construction plans outlined in Section 4.9.9.

Installation and Handling. The major problem confronting the installation is the handling of the condenser at approximately 310 tons and the evaporator at 290 tons. Due to the fact that this is a test article, it is assumed that it would be a one-time installation; consequently, the approach is to accomplish the handling without modifications of facilities.

One method would be to prepare the ship with foundations ready to accept the heat exchangers and then move to a facility where lifting capabilities exist, such as the 660-ton shear-leg at Longview, Washington. There the transfers from a seagoing barge to the ship could be made.

Another method would be to use two floating cranes with a spreader beam between them and the transfer lifts could be accomplished at a shipyard finger pier. Lifts of other equipment that may be beyond pier-side handling limits could be loaded by a single crane barge.

Much of the final fit-out could be handled at pier side, which is a minimum cost approach for a shipyard.

Dry docking the complete ship with system installed would require approximately a 9,000-ton dry dock, which is available at most U. S. ports.

4.9.8 Installation Schedules

12.5-MW(e) Units for 100-MW(e) Supertanker. This configuration was chosen to be representative of surface concepts. A conceptual schedule indicating 3-3/4-years completion is illustrated in Fig. 4-54.

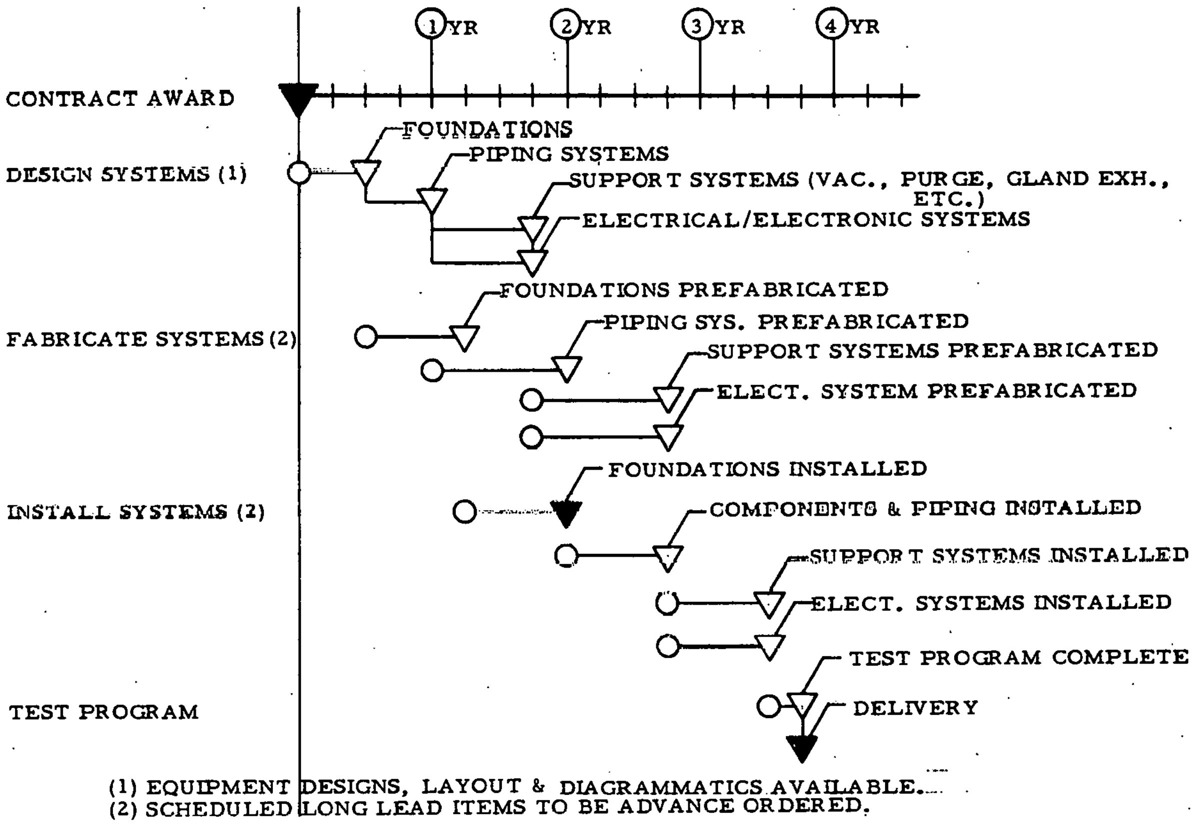


Fig. 4-54 Schedule for 12.5-MW(e) Units for 100-MW(e) Super Tanker

25-MW(e) Power Modules for 100-MW(e) Spar Platform. A conceptual schedule indicating 3 years to complete four power modules is illustrated in Fig. 4-55.

Basis of Schedule. Schedules are based on one-time installations at one facility. Schedules also require support in advance ordering of long lead items.

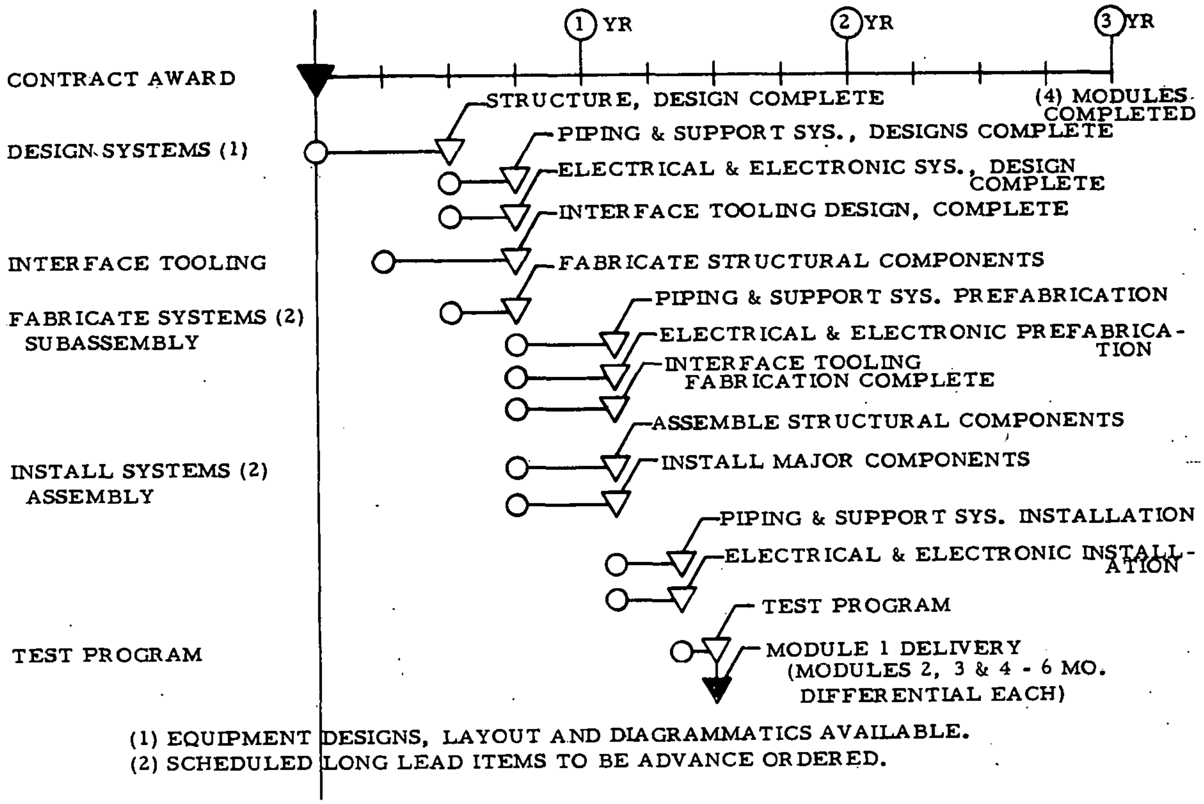


Fig. 4-55 Schedule for 25-MW(e) Modules for Spar Platform

4.9.9 Construction Plans

The construction plans that are considered to be most favorable with the system concepts are outlined below with areas of major impact mentioned in each.

Supertanker or Surface Platform. Of major consideration in the construction of a supertanker or surface platform power system is the structural analysis, from which it will be determined the size and number of access areas for loading of major components. A loading plan will be derived to prevent excessive concentrated or eccentric loads in the structure as assembly progresses.

Effective design will yield a minimum of internal structure that must be provided to support major and minor components, and allow a maximum of modular-type assemblies and prefabrications that will expedite assembly.

Construction planning must integrate the structural analysis, design, loading plan, and schedule of prefabrications to permit the effective construction of these vessels as displayed in the following assembly sequence.

Supertanker or Surface Platform Assembly Sequence.

a. Structurally analyze and design equipment supports and added structure
b. Plan construction and preplan manufacturing.
c. Fabricate modular foundations, structure, and supports.
d. Sequentially install access openings as required for direct load of major components. Due to structure removal for the direct loading of major components, specific areas of the ship may require major system and component installation completion to permit replacement of structure required to carry loads generated by the placement of other major components in other areas.
e. Sequence install additional structure and foundations and secure.
f. Prefabricate piping systems.
g. Install piping components or equipment that will be restricted due to ship's structure or major components and secure.

h. Install and secure major components in accordance with loading plan.
i. Perform nondestructive tests as required.
j. Install electrical and support systems to completion.
k. Test systems (dock trials).

Power Module Assemblies. Interface connections that will connect to the central spar must be controlled on all power modules to permit interchangability. To accomplish this, interface tooling must be designed and utilized to control placement of all the spar connections.

Construction planning to control distortion and tolerances begins with design. Emphasis must be placed on welding joint design and production welding sequence. Good joint design has the effect of minimizing weld deposits and distortion and expedites production. Welding sequence tends to have the adverse effect, but is essential in minimizing distortion.

In the case of the power module structure, modular construction planning appears the best approach. Structure, along with component foundations, constructed to maximum size or weight (within the limits of lifting capabilities) would be fabricated. Interior piping and equipment would also be incorporated in the assemblies, with make-up piping installed after assemblies are fitted and welded together, as indicated in the following assembly sequence.

Power Module Assembly Sequence. The sequence is as follows:

a. Plan design and construction.
b. Fabricate support structure.
c. Fabricate bottom-half subassemblies.
d. Laydown, fit, and weldout bottom-half subassemblies until bottom-half structure is complete.
e. Fit and assemble make-up piping, valves, etc., in bottom assembly.
f. Set major components with interface tooling and secure.
g. Install top-half modules in sequence that best suits distortion control and provides maximum access to interior.

h. Fit and weldout top-half assemblies to structure closure.
i. Assemble make-up piping and valve assemblies to structure closure.
j. Nondestructive test as required.
k. Install exterior piping and fittings.
l. Test systems as required.
m. Remove interface tooling and finish.

SECTION 5

Section 5
COST ANALYSIS

This section on Cost Analysis discusses a variety of issues bearing directly on the cost of energy from OTEC power plants. Section 5.1 presents capital cost estimates of the various components included in the baseline and alternative selected power system configurations. Since heat exchangers are the major cost component in the overall power system estimate, a detailed heat exchanger cost breakdown is presented in Section 5.2. Recurrent operating and maintenance cost estimates are not discussed in this report, as relevant data are not yet complete. System availability and reliability considerations are addressed in Section 5.3. Finally, some areas of potential power cost reductions are discussed in Section 5.4.

5.1 POWER SYSTEM COMPONENT COSTS

This section presents a summary of OTEC power system costs with all costs expressed in terms of 1977 dollars. A combined allowance of 30 percent of material and labor cost is included in the estimates, comprising separate allowances for contingency, engineering and home office charges, contractor and integrator fees, and owner costs (Table 5-1). While the cost of installing OTEC equipment on board various platforms has been estimated, the cost of the required foundations is not included here because it is felt that this is more properly regarded as a component of platform, rather than power system, costs; consequently, it is included in the platform cost estimates presented in the following subsection.

Table H presents a listing of power system component costs for prototype and production 25-MW(e) power systems. No tooling or facility costs have been included in the production unit cost estimates because these costs assume the existence of an OTEC industry. No labor learning is assumed to have taken place and consequently the production unit cost estimates presented should be interpreted as being applicable specifically to the first production units.

Table 5-1

STRUCTURE OF PRODUCTION UNIT COST ESTIMATES

Material and Labor (Including Overhead)		1.00
Contingency		0.10
Engineering	0.06	
Home Office	0.04	
Contractor Fees	0.04	
TOTAL SERVICES		0.14
Owner Costs and Integrator Fees		0.04

Overall Multiplier to be Applied to Material and Labor = 1.10 × 1.14 × 1.04 = 1.30

The costliest components of the power system are the two heat exchangers, whose cost structure is discussed in Section 5.2; however, it might be noted that the baseline configuration heat exchangers are approximately 30-percent costlier than those of the submerged configuration. The seawater system is the next costliest item; nearly 85 percent of its cost is accounted for by the seawater pumps. The cost of this system is the same for both configurations because the lower baseline flow requirements are compensated by its greater head loss. The cost of seawater pumps is somewhat higher than anticipated by LMSC on the basis of previous studies. However, on the basis of several quotes from large pump manufacturers, the cost given in the table appears to be correct. Heat exchangers and seawater systems jointly account for roughly two thirds of total power system costs; the next largest cost items are turbogenerators and the various auxiliary systems, accounting for another 25 percent of total cost. The cost of turbogenerators is roughly 10 percent higher for the baseline, as parasitic power consumption is almost double that of the selected configuration. The cost of auxiliary systems is approximately the same for both configurations. It should be noted that the costs assigned in Table H to "Other Auxiliary Systems" have been estimated provisionally on the basis of earlier LMSC studies. A more accurate estimate will be produced during the preliminary design task. The last item listed in the table, equipment installation, is more than double for the baseline configuration than for the one recommended. This is a direct consequence of the two different layout philosophies involved:

Table H

POWER SYSTEM COST SUMMARY

Item	Power System Costs ($/kW)			
	Prototype Unit		First Production Unit	
	Reference Baseline	Submerged	Reference Baseline	Submerged
Evaporator	375	272	327	225
Condenser	428	310	371	256
Demister	7	7	7	7
Turbine	83	74	62	55
Generator	57	51	50	45
Seawater System	260	260	200	200
Ammonia Feed Pumps	7	7	6	6
Ammonia Circulation Pumps	10	10	9	9
Ammonia Piping	20	14	17	12
Control Systems	17	17	15	15
Cleaning Systems	23	22	20	19
Other Auxiliary Systems	114	114	100	100
Chlorination	34	35	30	31
Equipment Installation	92	40	81	35
FIRST COST TOTAL	1,527	1,233	1,295	1,015
Heat Exchangers Replacement (10-Year HE Life)	823	570	823	570
Heat Exchangers Replacement (15-Year HE Life)	405	280	405	280
Life Cost Total (10-Year HE Life)	2,350	1,803	2,118	1,585
Life Cost Total (15-Year HE Lift)	1,932	1,513	1,700	1,295

the baseline configuration fits power system componentry into a predetermined hull shape, while in the submerged configuration the hull is designed around the OTEC power system components. Total first costs for the first production power system of the recommended configuration are slightly more than $1,000/kW, while those of the baseline configuration approach $1,300/kW. A prototype system is expected to be about 20-percent costlier; approximately half of this difference is accounted for by prototype allowances higher than those indicated in Table 5-1 for contingency, services, and fees.

The selected heat exchanger material is aluminum, and the costs given in Table H reflect this material choice. The operational life of the aluminum heat exchangers envisaged is likely to fall short of the required 30-year life of the power system, and consequently they probably will have to be replaced at some point. LMSC's studies indicate that heat exchanger operational life will range from 10 to 15 years. The present costs of replacing heat exchangers on 10- and 15-year cycles are also shown in Table H, as are the resulting total capital costs over the life of the power system The present costs shown for heat exchanger replacement assume no cost reduction beyond the first production unit and are thus likely to be somewhat conservative. Present worthing of future disbursements is based on a cost of money of 10 percent, and a 6-percent cost escalation rate; both provided by DOE.

The two power system configurations discussed in this report impose considerably different requirements on the platform portion of the total OTEC plant. Since platform design in sufficient detail for cost estimating purposes is beyond the scope of the Power System Development study, it is only possible to discuss the approach to be taken in evaluating total OTEC plant costs.

As indicated in Section 3.5, the effects of module component size and weight on the platform size and associated costs are important criteria in determining the optimum plant size and related design parameters. The cost per unit volume or per unit weight must be developed by the OTEC Platform Configuration and Integration Study contractors in order to accomplish this optimization. In addition, the modules discussed in this report incorporate the seawater pumps and related structure; their costs are included in the Cost Summary of Table H. Care should be taken to ensure that these costs are not duplicated in considering the optimum combination of platform and power system module.

5.2 HEAT EXCHANGER COST DATA

Table I gives a breakdown of the baseline and selected heat exchangers' costs including evaporator, condenser, and demister. Material costs represent roughly two-thirds of the first production unit's total cost, and a somewhat smaller proportion of the prototype unit's. Tubing material by itself accounts for roughly one-half of total cost in all cases. The cost of inserting and securing the tubes is the second largest cost item, and the largest among labor costs. Other cost components of some importance are associated with the waterboxes, tubesheets, tube supports, and shell erection and final assembly. Material costs are assumed to be similar for prototype and production units, while labor costs have been estimated separately for both cases. In comparing labor costs for the prototype and production units, it is of interest to note that the cost of shell erection has the greatest reduction upon entering production. This reflects the use of assembly and welding fixtures designed specifically for repeated use and ease of manufacturing rather than the single-purpose, least-cost tooling envisaged for prototype units. As already mentioned, heat exchangers of the submerged configuration are considerably less expensive than those designed for baseline conditions. The source of approximately half of the cost differential is to be found in the lower material requirements of the submerged configuration, which is designed for significantly lower differential pressure regimes than those found in surface operation. This is also a major cause of the differences in labor costs between the two configurations because the thicker shell and tubesheets of the baseline configuration require greater labor inputs to erect, perforate, and assemble. In addition, the baseline configuration heat exchanger is inherently the larger of the two because it must provide sufficient power to supply the greater parasitic power requirements of the seawater pumps while still yielding a net output of 25 MW(e).

Table I

HEAT EXCHANGER COST SUMMARY
(EVAPORATOR AND CONDENSER)

Item	Prototype Unit Costs ($/kW)				First Production Unit Costs ($/kW)			
	Reference Baseline		Submerged		Reference Baseline		Submerged	
	Material	Labor	Material	Labor	Material	Labor	Material	Labor
Tubes	346[a]	101	248[b]	89	346[a]	71	248[b]	63
Tubesheets	19	42	11	26	19	25	11	16
Tube Support Plates	18	60	14	51	18	37	14	31
Shell[c]	57	123	31	75	57	94	31	36
Waterboxes	8	22	8	22	8	16	8	16
Ammonia Distribution[d]	7	2	7	2	7	2	7	2
Nozzles	1	1	1	1	1	1	1	1
Miscellaneous	1	2	1	2	1	2	1	2
TOTAL	457	353	321	268	457	248	321	167

(a) Tubes at $0.84/ft basic + $0.44/ft enhancement
(b) Tubes at $0.70/ft basic + $0.44/ft enhancement
(c) Includes final structural assembly
(d) Includes demisters

5.3 AVAILABILITY AND RELIABILITY ANALYSIS

The definition of the power system module reliability and maintainability characteristics provides the estimates for module availability. Reliability and maintainability are discussed separately and summarized in terms of power system module availability.

5.3.1 Reliability

The reliability of the power system is the probability that the power system will perform its intended function for the defined period of time under specified conditions. The basic factors associated with reliability therefore are (1) the operating cycle conditions, (2) the output capability, (3) operating time, and (4) environmental and support functions.

Base loading is applied to the operating cycle – the power system operates continuously at the maximum possible output based on available warm/cold-water ΔT. If the operating cycle is changed to on-off operation or if pressures are caused to fluctuate drastically within short time spans, then the reliability will be lower because the conditions will violate the base loading concept.

The output capability of the power system is a direct function of the available ΔT and therefore dependent on seasonal, site, and platform conditions. Power system energy output below a stated nominal value is a reliability problem only when the factors beyond the control of the power system are within stated limits for nominal power system operation and the power system does not perform to its stated limits.

The longer a system operates, the more opportunity it has for failure. Thus its reliability decreases with time. The design life goal requirement of 6,000 hours per year for a 25-MW(e) power system module can be equated to 17.12 MW(e) for 8,760 hours per year (365 days at 24 hours per day) on a MW(e)/year basis or a 28.54-MW(e) module at 5,256 operating hours per year (60 percent duty cycle of 8,760 hours). See Fig. 5-1 for operating time and module size to obtain 150,000 MW-hr per year (25-MW(e) output at 6,000 hours per year).

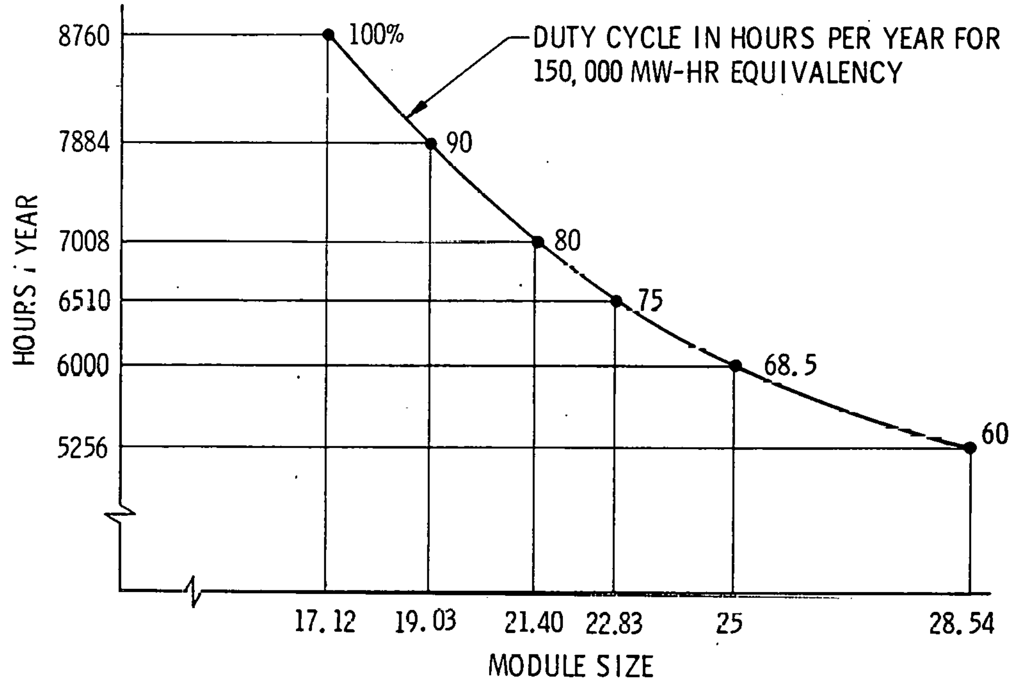


Fig. 5-1 Equivalency Duty Cycle

For the power module to provide its output at a stated level, the conditions which allow the power system to function properly must be maintained. In this context platform functions such as water supply at given rates, temperatures, and pressures must be realized as well as limits on the platform motions such as pitch, roll, and heave. In addition, the support functions such as preventive or scheduled maintenance schemes are to be followed.

The requirement of plant availability, "Operability (i.e., actual operational time ÷ required operational time) shall be 90 percent at a 60 percent confidence level" is interpreted to be based on the 6,000 hours per year operation allowing periodic replacement of components and maintenance during the remaining 2,760 hours per year. To demonstrate the 90-percent operability with a 60-percent confidence statistically would require a total of nine tests (nine sets of power system equipment), each operating for the 6,000 hr without a failure. Failure in these tests could be interpreted to be power output below prescribed limits.

The 30-year design life is interpreted to mean that the power system module shall survive a wearout service life of 30 years economically, with reasonable periodic replacement of components. At a point in time when a reasonable maintenance schedule can no longer keep a component operating, the item has reached its useful life (wearout has occurred). Reliability does not account for wearout nor does it account for the early failures covered by defects in the manufacturing processes or poor quality (these failures are noted as "infant mortality").

Reliability then is the random failure of a component during its useful life period and can be caused by factors such as corrosion, biofouling, fatigue, and operating instabilities. The function of reliability is the failure rate – the instantaneous rate at which devices/components fail. The failure rate, λ, (in terms of failure per year or 1,000,000 hours) is related to the probability of successful operation for t hours by

$$R = e^{-\lambda t}$$

and to the Mean-Time-Between-Failures (MTBF) by

$$m = 1/\lambda$$

The MTBF is not wearout life. Further, once a device/component fails, a finite time period is required to perform the repair (Mean-Time-To-Repair – MTTR).

Redundancy has been incorporated into the power system to ensure providing a high-reliability power module. Four critical areas have redundant equipment:

- Evaporator/condenser – tubes
- Seawater system – pump
- Ammonia cycle – condensate and distribution pumps
- Turbine/generator – turbine

As stated in the section on heat exchangers, the evaporator and condenser design incorporates approximately 7 percent excess tubes to provide margin for uncertainties or tolerances in tube wall thickness and material properties. The percent excess tubes will be specifically defined during preliminary design when the tolerances for parts of the heat exchanger are established.

The selected seawater pump configuration has eight identical pumps, four for each heat exchanger. One strong rationale for this selection was the redundancy provided. On seven-pump operation, a power module net power output drops to approximately 95 percent, and with six pumps operating (three operating on each heat exchanger) the module output is approximately 90 percent of normal.

In the ammonia cycle, three 50-percent capacity pumps are used for the condensate pumping (two will provide 100-percent flow); and in the ammonia distribution system at the evaporator, four 33-percent capacity pumps are provided (three will provide 100 percent ammonia flow to the evaporator). This redundancy of pumps ensures essentially uninterrupted flow capability in the ammonia liquid system. Because of the 50-percent reflux rate in the evaporator, all seven of the pumps have the same flow capacity and therefore will be the same, thus minimizing spare parts and maintenance procedures.

As the cost of the selected radial inflow turbine showed little saving in going from 12.5-MW(e) to 25-MW(e) size, two 12.5-MW(e) turbines were selected for the power system module. This turbine redundancy virtually ensures power output from the module at all times even though it be half rated power.

The power system critical component MTBF and MTTR estimates are in Table 5-2. These values have been assembled based on vendor data; subcontractor supplied data; Electric Power Research Institute Reports; Edison Electric Institute Reports; and Mechanical Design and Systems Handbook, McGraw-Hill, M.A. Rathbart, Editor-in-Chief.

Table 5-2

POWER MODULE CRITICAL COMPONENT FAILURE RATES

Item	MTBF (yr)	Failure Rate (failure/yr)	MTTR (days)
Evaporator			
– Tube	10	0.100	3
– Shell	20	0.050	10
– Tubesheet, Baffles, Support	15	0.067	3
Condenser			
– Tube	10	0.100	3
– Shell	20	0.050	10
– Tubesheet, Baffles, Support	15	0.067	3
Turbine	6	0.167	3
Generator	10	0.100	4
Ammonia Vapor			
– Piping	40	0.025	4
– Valves/Controls	10	0.100	3
Ammonia Liquid			
– Piping	40	0.025	4
– Valves/Controls	10	0.100	3
Seawater			
– Warm	8	0.125	2
– Cold	6	0.067	3
Subsystems			
– Ammonia	10	0.100	3
– Electrical	15	0.067	2
– Corrosion	10	0.100	5
– Controls	20	0.050	1
– Biofouling	6	0.167	1
Structure	50	0.020	10
		1.645	

The best estimate of critical power system module equipment MTBF is 222 days. For the failure, approximately 3.5 days will be required for repair, during which the power module will not be operating. The heat exchanger tube failure rate is based on 95 percent of the tubes surviving to the tenth year (5 percent of the tubes will be plugged) and the preventive maintenance procedures at level 4 (minor repair/replace), accomplished every 6 months will account for the downtime for plugging. The biofouling failure rate is based on cleaning system failure (loss of brush or brush failure), which will be repaired at the same time as the tube failure.

5.3.2 Maintenance

The two maintenance schemes of importance are:

- Preventive or scheduled maintenance carried out to keep the equipment in a satisfactory operational condition by providing systematic cleaning, adjustment, replacement, and refurbishment of components before they are expected to fail.
- Repair maintenance carried out on a nonscheduled basis to restore a component to a satisfactory condition by providing immediate correction of a failure after it has occurred.

The emphasis here is on the first category as the second category has been discussed in the reliability section.

Seven levels of maintenance are planned and the estimated impact on performance is as follows:

- LEVEL 1 – Fill/change weekly/1 day duration/no power reduction
- LEVEL 2 – Adjust/clean monthly/2 days duration/90% power for 12 hours
- LEVEL 3 – Adjust/change 3 months/3 days duration/ 0 power 24 hours
75% power 12 hours
90% power 12 hours
- LEVEL 4 – Minor repair/replace 6 months/5 days duration/ 0 power 3 days
75% power 1 day

- LEVEL 5 – Repair/replace, refit 12 months/10 days duration/ 0 power 5 days
  75% power 2 days
- LEVEL 6 – Overhaul/refurbish 36 months/50 days on-board/0 power 50 days
  60 days off-board/0 power 60 days
- LEVEL 7 – Heat exchanger change out 10 years/90 days/0 power 90 days

The activities for maintenance are arranged such that when Level 2 is accomplished Level 1 is also accomplished and when Level 3 is accomplished Levels 2 and 1 are also accomplished. This carries through such that Level 6 accomplishes all activities through Level 1.

Two variations to the levels of maintenance were briefly investigated. The first concerned an automated power system module (unattended) with Level 1 maintenance deleted and Level 2 accomplishing all necessary maintenance activities. The second is the method of accomplishment of Maintenance Level 6 at the platform or at a shipyard and includes the use of a spare detachable power system module.

The automated operation approach impacts the power output of the power system and the operational costs (manpower) and controls (transmission of signals signifying a parameter out of a prescribed limit). At the present time the impact to the power system output is not known because the response time for a crew to arrive at the plant and perform corrective action is unknown. The operational cost savings were not estimated because a plant operational crew was not established.

The accomplishment of all preventive maintenance activities on-board the platform requires each module to be out of service approximately 740 days in a 30-year period (24.7 days per year) with 30-year life heat exchangers. For on-board platform change-out of heat exchangers (or retubing) every 10 years and adding 30 days each time for the activity, the preventive maintenance is approximately 800 days (26.7 days per year).

For the case where a module is detached and returned to a shipyard for overhaul and refurbishment with a 30-year heat exchanger, the preventive maintenance time is 830 days (using 10 days transit time every 3 years) or 27.7 days per year average. For a 10-year heat exchanger, the total time is 890 days or 29.7 days per year. Finally, with a spare detachable module and an 8-day changeout period (from shutdown of the on-plant module to full operating capability of its replacement), the preventive maintenance time per power system module is reduced to 362 days (12.1 days per year average).

The variation in preventive maintenance time for the first four modes is small – 25 days/year on-board activities with a 30-year heat exchanger, 27 days/year for on-board activities with a 10-year heat exchanger, 28 days/year for a detachable module with 30-year life heat exchangers, and 30 days/year for detachable module with 10-year life heat exchangers. The concept of a spare detachable module, however, reduces the preventive maintenance time significantly – to less than half of the others (12.1 days/year).

If yearly power output is the dominant criterion for power system configuration selection, a detachable power module is the choice. For the four variations discussed, the detachable module presents a savings, at $0.04/kWh, over the 30-year span of $9.1 million for on-board maintenance with 30-year heat exchangers; $10.5 million for on-board maintenance with 10-year heat exchangers; $11.2 million for detachable power module with 30-year life heat exchangers; and $12.7 million for detachable power module with 10-year life heat exchangers. These costs do not include manpower or support equipment to accomplish the maintenance modes on-board, at shipyard, or for transit between.

5.3.3 Availability

The power system module availability exceeds the required 6,000 hr/yr operation design life goal. With critical component redundancy, the estimated equipment failure rate and the use of a spare detachable module concept, the maximum down time is about 18 days per year. On an 8,760 hr/year basis, this module provides approximately 8,325 hr/year operation or 95 percent of the time. The availability estimates, including maintenance and failure rate, are:

Overhaul Location	Heat Exchanger Life (yr)	Downtime (hr/yr)	Time/Year Operating (%)
On-Board	30	736	91.6
On-Board	10	784	91.1
Shipyard	30	808	90.8
Shipyard	10	856	90.2
Shipyard/Spare	–	434	95.0

5.4 POTENTIAL FOR COST REDUCTION

As is the case with any developing technology, the greatest reduction in OTEC costs are likely to result from advances in related technologies which are, by their very nature, extremely difficult to forecast. Consequently, the discussion below focuses on selected potential cost reduction areas where cost improvements can be expected to materialize through further development along the lines currently comtemplated by the DOE Program Plan, but does not attempt to evaluate the impact on OTEC costs of progress in related fields.

Perhaps the most obvious source of cost reductions is provided by the economies of scale available in the manufacture of various OTEC components, particularly heat exchangers, turbines, and seawater pumps. While 25-MW(e) power systems already capture very significant economies of scale, they do not by any means exhaust them. Preliminary studies indicate that doubling heat exchanger size would result in a reduction of approximately 10 percent in their cost per kilowatt. Economies of scale of similar magnitude appear to be available in turbine manufacture. The situation regarding seawater pump costs is somewhat less clear, but various quotes received, in the course of this study, tend to support the data presented in Deep Water Pipe, Pump, and Mooring Study Ocean Thermal Energy Conversion Program (00-2642-3), which suggests the presence of significant economies of scale in large pump manufacture.

Within heat exchanger manufacture, several improvements on current manufacturing techniques suggest themselves as likely sources of cost reductions. Some of these developmental techniques are discussed in Section 4. Two other manufacturing techniques deserving mention in this context are high-speed machining (instead of drilling) of tubesheet and baffle holes, and explosive welding of tubes to tubesheets. Research in both areas is being carried out at LMSC under separate Independent Development programs. It might be noted that tubing material and labor costs represent the single major cost components in their respective categories. Tubing material being the largest single cost element in heat exchanger costs, it is perhaps reasonable to expect further cost reductions to arise from development of more cost-effective forms of heat transfer enhancement. As currently envisaged, the process of inserting tubes and attaching them to the tubesheets is essentially a manual operation involving up to ten operators at a time; this is an obvious area for methods improvement and manufacturing automation. Before leaving the subject of heat exchanger cost reduction potential, it should be stressed that this study has been limited to tube-and-shell heat exchanger concepts. It is possible that other types of heat exchangers, such as the panel type, will result ultimately in lower costs for OTEC applications.

Seawater pump costs are to a great extent driven by the cost of aluminum-bronze impellers. This suggests that substituting a less expensive material (such as mild steel) for aluminum-bronze might result in significant savings. However, the impeller itself would then require protection from direct contact with the seawater; a neoprene or epoxy coating might be applied to provide such protection. While this approach to impeller design is apparently beyond the current state of the art, its promise for reducing the cost of OTEC power systems renders its development desirable.

Large-scale biofouling control systems are likely to experience improvement as a result of OTEC technology development. As already noted, it is quite difficult to forecast the direction such advances will take. However, in this particular case it may be surmised that decreases in cost and increases in cleaning efficiency will result from the development of special-purpose systems, designed from the outset for the scale and environmental conditions characteristic of OTEC applications. By contrast, currently envisaged biofouling control systems are direct extrapolations of systems designed for quite different conditions and their estimated cost is almost certain to overstate cost levels achievable by specialized designs.

It should be stressed again that the production unit costs, given in Table H, are applicable only to the first production unit and consequently do not take into account any cost reductions arising from labor learning. While the various components of an OTEC power system are likely to experience different labor learning rates, previous LMSC studies suggest that an experience curve slope of 0.93 is probably appropriate for the OTEC plant as a whole. As a result of the labor efficiency improvements implied by such a curve, the cost of the 25th OTEC power system would be expected to be roughly 10 percent lower than the first production unit cost given in Table H.

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SECTION 6

Section 6
TEST ARTICLES

The test articles for Power System Development are the proof of concept (5-MW(e) nominal) power system scaled to the design of the 25-MW(e) module and the 1-MW(e) size heat exchangers (evaporator and condenser subsystems) which are representative of the heat exchangers of the 25-MW(e) power system design. These test articles are intended to provide a scaled approach to the development of the 25-MW(e) power system modules for the 100-MW(e) OTEC Demonstration Plant.

This section presents the guidelines followed in deriving the test article configurations and sizes and their recommended configurations and sizes.

6.1 GUIDELINES

The major guidelines used for development of test article configurations and sizes are to provide, within reasonable cost, hardware representative of the 25-MW(e) power system which would yield data sufficiently accurate to evaluate the validity and relative merits of 25-MW(e) power system concepts. Further, the data are to provide confidence in (or upgrade) the analytical methods used for predicting performance and cost of the power system configuration.

In the 1-MW(e) heat exchangers, the major concern is understanding the performance of both the evaporator and the condenser. Performance includes heat transfer rates as well as effects of vapor velocities, fluid distribution, and the seawater environment. The primary purpose of the 1-MW(e) evaporator and condenser test series is to determine the tube bundle heat transfer characteristics of selected tubes and tube arrangements under actual OTEC operating conditions. Mechanical effects of tube-to-tubesheet joint, support structure, tube vibration, etc., are not the major objectives in the 1-MW(e) test article case because these parameters can be resolved in design by known methods or independent tests.

In the 5-MW(e) (nominal) scaled power system, the performance of selected components and the verification of component interactions as a system, the verification of predicted physical phenomena, and the demonstration of the feasibility of selected manufacturing processes are paramount.

The 5-MW(e) scaled proof of concept power system will lead to the first technical demonstration of the OTEC concept that is representative of a power system module. It is essential that the selection of component sizes and ratings for this nominal 5-MW(e) power plant be sufficient to demonstrate manufacturing feasibility of the 25-MW(e) power plant. The size of the primary components of the power plant (i.e., heat exchangers, seawater pumps, and turbines) will be adequate to demonstrate technical as well as manufacturing maturity of the industrial sector with regard to fabricating components for the power system. The technical and engineering data generated by successful operation of the 5-MW(e) power plant at sea will be available as the basis for design and construction of the 25-MW(e) commercial demonstration plant.

6.2 SCALING

Verification of thermal design, fluid dynamics, structural integrity, and producibility depends on proper application of scaling laws, derived in general through dimensional analysis and experimental correlations. The justification for the use of models or subscale components to substantiate a design is basically an economic one because it is generally less costly to build and test models of large power system components than to rely entirely on design calculations. In order to obtain complete similarity between the subscale component and the prototype, it is necessary that geometric, kinematic, and dynamic similarity be maintained. Geometric similarity is obtained by maintaining the ratios between corresponding lengths in the subscale component as in the prototype. Kinematic similarity requires that the same ratios of fluid velocity and acceleration exist between the subscale and prototype component. This is achieved by maintaining dynamic similarity, which requires that all independent force ratios be the same, which in turn is accomplished for viscous forces by maintaining the same Reynolds number in the model as the prototype. The ratio of inertial to gravity forces is given by the Froude number, while the ratio of inertial to surface forces is given by the Weber number. In most cases it is not necessary to maintain all of the force ratios the same and, generally, all but one or two can be omitted.

Heat Exchangers. Scaling of the heat exchangers will present the greatest difficulty because maintaining kinematic similarity will be difficult for the tube bundle diameters that are much less than the full-scale diameter. Assuming that the predominant forces in vapor-liquid entrainment are viscous, dynamic similarity will be primarily dependent on maintaining the same Reynolds number in the subscale test as that for the prototype. The relationship for Reynolds number as a function of tube bundle diameter can be shown to be given by:

$$Re = \frac{\bar{U} \, \Delta T_{LM} D_B}{\lambda \, \mu \, P \, (P - 1)}$$

where $\bar{U}$ is the average overall heat transfer coefficient, ΔT_{LM} is the log mean temperature difference, λ is the latent heat of vaporization, μ is the viscosity, P is the pitch-diameter ratio, and D_B is the tube bundle diameter. In order to maintain the same Reynolds number in the subscale heat exchanger tests with a smaller tube bundle diameter, it generally will be necessary to increase the log mean temperature difference or decrease the pitch diameter ratio. Scaling of the liquid film will not be as critical because the shell-side heat transfer coefficient is relatively independent of flow rate for the evaporator as well as the condenser.

The variation of vapor velocity to heat exchanger power level size and tube diameter is shown in Fig. 6-1 for an unenhanced tube installation. With a 1.5-in.-diameter tube at 25-MW(e) size, a 6-fps vapor velocity exists at the outer row of tubes. For the same velocity in a 5-MW(e) size evaporator, the tube diameter would be about 0.67 in., and at 1-MW(e), 0.30 in. in diameter. For a 2-in.-diameter tube, the velocity is 4.3 fps, and the corresponding equal velocity tubes for 5-MW(e) and 1-MW(e) heat exchanger power level sizes are 0.91 and 0.40 in., respectively. A cross plot of Fig. 6-1 is shown in Fig. 6-2 where the vapor velocity is a function of the bundle radius (or power level size) for various tube diameters.

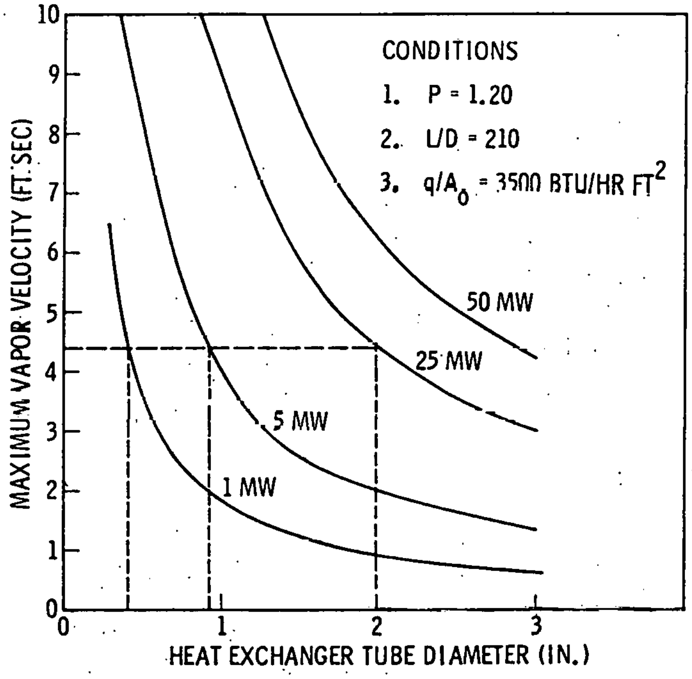

Fig. 6-1 Maximum Evaporator Vapor Velocity

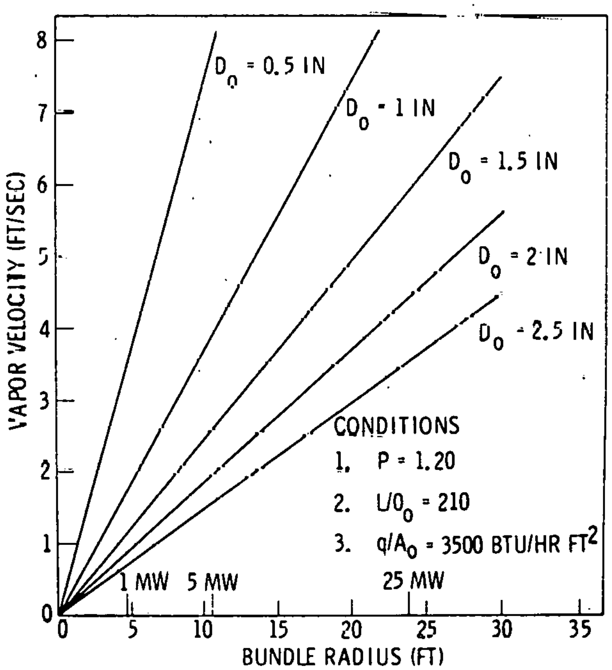

Fig. 6-2 Vapor Velocity as a Function of Bundle Radius

The effect of vapor velocity on ammonia droplet displacement is shown in Fig. 6-3. Three different droplet diameters for the vapor velocities up to 10 fps are shown. The vertical spacing between the lower surface of the upper tube and the upper surface of the lower tube for a 1.5-in.-diameter tube is 0.30 in. for a tube-pitch-to-tube-diameter ratio (P/D) of 1.2 and 0.375 in. for P/D = 1.25. For a 2-in. diameter, the values are 0.40 and 0.50 in., respectively. The effect on overall heat exchangers performance of droplet displacement is not precisely known and must be verified through tests.

The heat exchanger variables classified as the (1) SAME for both scale model and prototype, (2) PROPORTIONAL TO POWER LEVEL, or (3) as INDEPENDENT variables are:

1. SAME
   - Ammonia velocity at outer row of tubes
   - Water velocity through tubes
   - Ammonia density at outer row of tubes
   - Water density through tubes
   - Tube-pitch-to-tube-outer-diameter ratio
   - Log mean temperature difference
   - Overall heat transfer coefficient
   - Tube wall thickness
2. PROPORTIONAL TO POWER LEVEL
   - Heat load
   - Ammonia flow rate
   - Water flow rate
3. INDEPENDENT
   - Tube outer diameter
   - Tube bundle diameter
   - Active tube length

The numerical relationship of key parameters is summarized in Table 6-1 for the recommended evaporator sizes.

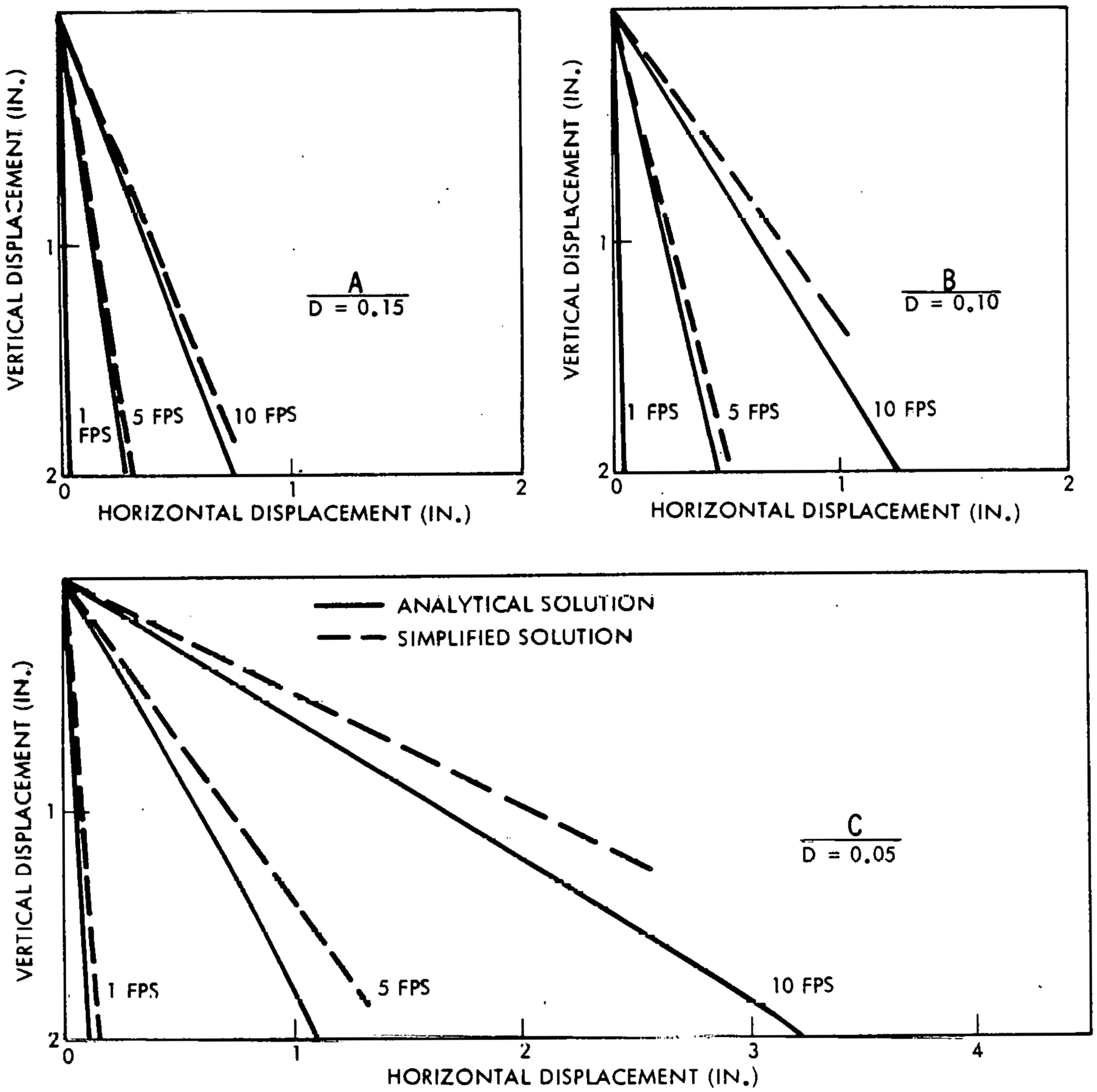

Fig. 6-3 Ammonia Droplet Trajectory

Table 6-1

SCALE FACTORS FOR TEST ARTICLE CONFIGURATIONS
(EVAPORATOR)

Parameter	Scale Factor	Parameter Value		
Power, M [MW(e) net]	–	25	5	1
Tube Wall, t (in.)	–	0.088	0.088	0.088
Tube Diameter, d (in.)	$d_x = \sqrt{M_x/M_{25}}\,(d-2t)_{25} + 2\,t_x$	1.500	0.768	0.441
Tube Pitch, P (in.)	$P_x = (d_x/d_{25})\,P_{25}$	1.875	0.960	0.551
Tube Bundle Diam., D (ft)	$D_x = (d_x/d_{25})\,D_{25}$	41.00	21.00	12.05
Active Tube Length, L (ft)	$L_x = (M_x\,d_{25}/M_{25}\,d_x)\,L_{25}$	38.00	14.84	5.17
Number of Tubes, N	$N_x = N_{25}$	50,250	50,250	50,250

Rotating Machinery. Scaling laws and relationships for the various pumps and turbines are well developed and no difficulty is expected for these subscale power system components. The basic relationships arising from similarity considerations are specific speed and specific diameter, with Reynolds number having a second order influence. In addition, scale effects must be considered when the relative roughness of the model and prototype inner surfaces differ by an appreciable amount. For subscale testing at the same head and similarity parameters of specific speed and diameter as the prototype, the following relationships between rotational speed and power, and between diameter and power, are obtained:

$$N_M/N_P = \sqrt{P_p/P_M}, \quad D_M/D_p = \sqrt{P_M/P_p}$$

These relationships are shown as a function of the net electrical power in Fig. 6-4. The relationship for Reynolds number, given by $\upsilon_M = \upsilon_P (D_M/D_p)$, which would

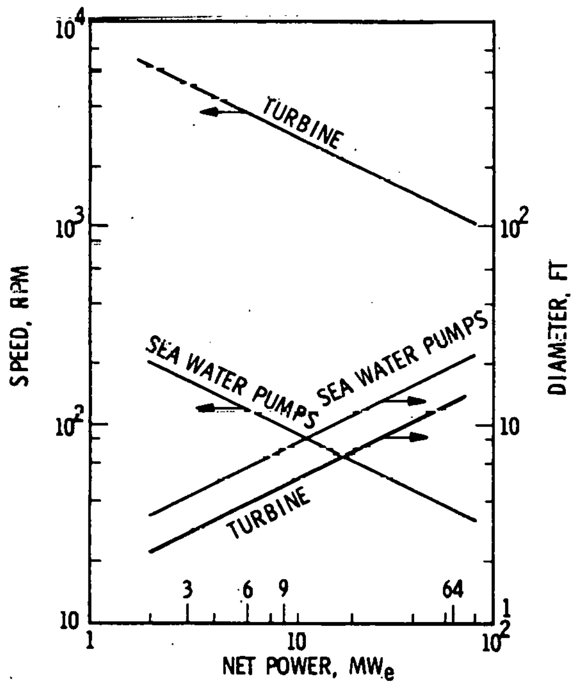

Fig. 6-4 Turbine and Seawater Pump Diameter and Speed as a Function of Power

require either a change in fluid kinematic viscosity or full-scale tests to scale the viscous forces. Subscale test results for the 5-MW(e) power system should yield excellent performance data for design of the 25-MW(e) turbine and pumps as the scale factor does not go below approximately 0.2 (see Fig. 6-5). This value is somewhat arbitrary and represents a general minimum in use within the industry. There are additional requirements imposed on the condensate and distribution pumps in order to preclude cavitation.

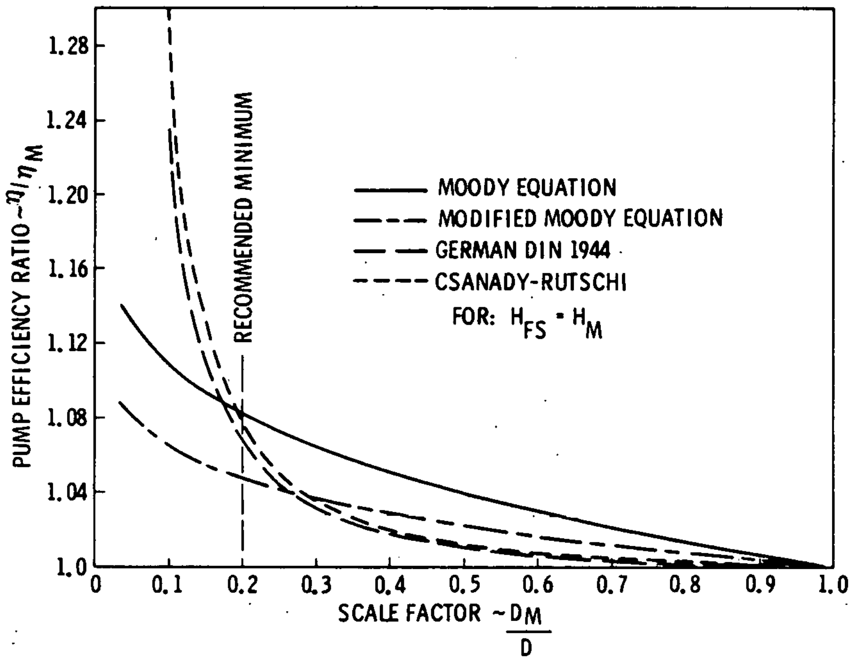

Fig. 6-5 Full-Scale Pump Efficiency From Model Pump Data

6.3 1-MW(e) HEAT EXCHANGERS

The data of Table 6-1 indicate that the number of tubes in scaled heat exchangers should be the same as the full-scale article and the tube diameter reduced to maintain identical bundle periphery ammonia vapor velocity. Since the effect of vapor velocity on fluid flow and tube vibration can be evaluated with an array of tubes under laboratory conditions, it is considered more meaningful to evaluate production techniques and long-term effects on material and performance in the OTEC environment. In addition, the thermodynamic performance can be compared with the predicted performance in order to provide confidence in the analytical methods or to indicate areas of deficiencies in the analyses.

The objectives of the tests to be conducted on the 1-MW(e) nominal sized heat exchangers therefore would be to provide sufficient data to verify the predicted hydraulic and thermodynamic performance, to verify the structural integrity and fabrication methods, and to determine long-term effects of fouling and corrosion. The results of these tests should provide the confidence to proceed with the scaled proof-of-concept power system. It is therefore desirable to provide the largest scale model compatible with resources available.

The thermodynamic resources are as specified in the OTEC-1 RFP. Previous studies have indicated that the condenser requirements are more critical. Considering that the nominal flow rate through the cold water system is 68,000 gpm, the condenser configuration can be established as follows: The condenser seawater velocity in the tubes optimizes at approximately 6.1 fps. For an anticipated operating life of 4 years, a tube wall thickness of 0.058 in. is adequate for an aluminum tube. The condenser could incorporate approximately 7,900 tubes, 0.875 in. OD, or approximately 5,825 tubes, 1 in. OD. The tube bundle diameter is 12 ft for the 0.875 in. tube and 11.8 ft for the 1-in. tube.

The tube configuration should allow determination of the effects of external stimuli, such as vessel motion or vibration induced by rotating machinery. Additionally, the heat exchanger tube length does not have to be restricted by a requirement to provide a specified thermal flux. The tube length should therefore be such that two support baffles can be incorporated at the appropriate position to provide simulation of the tube sections which are rigidly supported at the tubesheet and one tube section which is supported only at the baffles. The distance between baffles and between the tubesheet should be such that the dynamic response characteristics of the tubes are equivalent to those of the full-scale tube. An estimate of the length can be obtained by maintaining the ratio of unsupported length to tube diameter the same as for the full-scale unit. From the data of Table E, this ratio varies from 50.3 to 52.7 for the units shown. Based on the latter value, the heat exchangers would be 13.2 ft long for the 1-in. tube and 11.5 ft long for the 0.875-in. tube.

An alternative approach, which doubles the simulation scale of the test article, is to use one-half of the heat exchanger for the installation of the selected number of tubes and to install a bulkhead on the vertical centerline. The unused portion of the heat exchanger can be separated by a pressure balanced thin sheet bulkhead to prevent crossflow. With this approach, the diameter of the condenser tube bundle will be 17 ft for the 0.875-in. tube and 16.7 ft for the 1-in. tube.

A similar approach can be taken for sizing the evaporator considering the warm water flow rate of 60,000 gpm. The optimization program data indicate that the evaporator seawater velocity optimizes at a value of approximately 7.5 fps. The evaporator would thus incorporate approximately 5,750 tubes, 0.875 in. OD, or 4,240 tubes, 1 in. OD. The tube lengths are the same as for the condenser, and the tube bundle diameters are 10.25 ft for the 0.875-in. tubes and 10 ft for the 1-in. tubes. For the configuration that uses one-half of the heat exchanger for tube installation, the equivalent bundle diameter is 14.5 ft for the 0.875-in. tube and 14.2 ft for the 1-in. tube.

Cost trade studies will be accomplished during the preliminary design task. The recommended heat exchanger test articles will be based on the results of these studies.

6.4 5-MW(e) SCALED POWER SYSTEM

The scaled proof of concept power system is a performance-scaled power system. All componentry and subsystems would be configured for performance size in order to ensure component compatibility and operation as a system. Selected subsystems would not have redundant equipment in order to provide cost savings as the scaled power system is not a life test system. The seawater pump, ammonia liquid (condensate and distribution) pumps, and turbine would not have component redundancy as in the 25-MW(e) module. All operational features of the power system as well as the manufacturing feasibility and functional aspects (startup, shutdown, off-design operation control) would be demonstrated.

The recommended size of the scaled proof of concept power system is 10- to 12-MW(e) for the following reasons:

- Reasonable size to attract attention of potential users
- Reduce unknown-unknowns of scaling as the most cost effective size is in the range of 25- to 50-MW(e)
- Reduce $/kW capital investment by DOE
- Provide basis for validating commercial cost projections
- Provide reasonably sized component data and system data

An option to the 10- to 12-MW(e) size power system is a 5-MW(e) size which is considered as a lower limit in size in deriving valid test data. The size would provide sufficient data to allow the verification of the analytical techniques used in predicting operation and performance. This size would dilute the ability to reduce the unknown-unknowns.

Appendix A
SYSTEM ANALYSIS

This appendix contains additional details of the analytical data used in the development of the design performance data and the trade studies presented in Sections 3 and 4 of the main body of the report. The data are presented in two sections, as follows:

A-1 Design Performance Data
A-2 Trade Study Data

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Appendix A-1
DESIGN PERFORMANCE DATA

This section presents the data from the LMSC developed proprietary optimization program which defines an optimum configuration at specific design conditions. The on-design conditions were 32°, 36°, 40° and 44° F ΔT, with the systems sized to provide 25-MW(e) net. Each set of data (Figs. A-1 through A-4) consists of four sheets which present the cycle state points, the heat exchanger physical and thermodynamic characteristics, rotating machinery requirements, and a description of the cold water pipe assumed for the determination of cold water pump head requirements. The various stations are shown schematically on the ammonia cycle diagram in Fig. A-5.

The heat exchanger configurations – shown in terms of number of tubes, tube length, tube diameter, and spacing ratio – were used by Foster-Wheeler to develop cost data used in the trade studies discussed in Section A-2.

```
OPTIMIZE ON AREA.  THE OPTIMIZATION VARIABLES ARE:
     T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
     MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
     SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
     SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
RUNS FOR BILL CUTLER - SEPT  1977 -
THIS RUN IS FOR THE  25-MW POWER SYSTEM DEVELOPMENT WORK
SERIES AT FORTY , 36, 32 -DEGREE DELTA TEMPERATURES    - ALSO 44 DEGREES
NO DEMISTER - FOULING FACTORS ARE 0.0003   TUBE PITCH IS 1.2
EXPONENTT  USED IN KAYS EQUATION IS 0.17
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.
```

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0	68.04	124.410	.99000	623.66	.4232
01	68.04	124.410		[118.29] LIQUID AT AVERAGE DEMISTER PRESSURE	
1	68.04	124.410	.99000	623.66	.4232
2	68.19	124.624	.98897	623.66	.4239
3	51.90	92.476	.96350	609.62	.3244
4	51.93	92.542	.96348	609.62	.3246
45	51.61	91.980		[99.71] LIQUID AT AVERAGE CONDENSER PRESSURE	
5	51.29	91.421	.00000	99.35	38.9380
6	51.29	96.484	.00000	99.35	38.9380
7	51.34	206.326	.00000	99.93	38.9354
8	51.34	134.947	.00000	99.93	38.9354
89	68.16	124.678		[118.43] LIQUID AT AVERAGE BOILER PRESSURE	
9	68.04	124.410	.99000	623.66	.4232

```
GENERATOR EFF. =  .985
DEMISTER BACKFLOW MASS RATE =     -.00 LB/SEC
RANKINE CYCLE EFF. =    .025691
CARNOT CYCLE EFF. =    .030533
EFFICIENCY RATIO =    .841428
```

***** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP (PSI)
1-2	10.957	55.350	.140	110.000	-90.0	.009	.042	-.265	-.214
3-4	12.515	55.350	.107	30.000	-30.0	.002	.000	-.068	-.066
5-6	1.552	30.000	3.782	20.000	-20.0	.345	.000	-5.408	-5.063
7-8	1.552	30.000	3.782	310.000	240.0	5.352	1.135	64.892	71.379
9-0	10.957	55.350	.140	2.000	.0	.000	.000	.000	.000

Fig. A-1 Optimization Program Output for ΔT = 32° F

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR	
NUMBER OF TUBES	95466.	98609.	
TUBE PITCH	1.20	1.20	
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500	
HEAT EXCHANGER TUBE THICKNESS (IN)	.0580	.0580	
TUBE LENGTH (FT)	50.295	45.974	
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0	
SEA WATER TEMP. IN (F)	43.875	75.875	
SEA WATER TEMP. OUT (F)	47.497	72.357	
TUBE SEA WATER VELOCITY (FPS)	5.203	5.381	
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	279.424	273.099	
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000	
SEA WATER FLOW RATE (LB/SEC)	332818.14	352830.79	
SEA WATER FLOW RATE (CFS)	5189.24	5522.27	
SEA WATER REYNOLDS NO. THRU TUBES	41104.	55060.	
SEA WATER DENSITY (LBM/FT**3)	64.136	63.892	
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D.	407.	439.	BTU/HR-SQ FT
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS	850.	1006.	USING KAYS EQ.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000	
WORKING FLUID HEAT TRANSFER COEFFICIENT	1000.	1000.	
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300	
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000060	.000060	
LOG MEAN TEMP. DIFFERENCE (F)	5.737	5.774	
NTU'S	.6313	.6093	
EFFECTIVENESS	.4681	.4562	
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000	
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.921	.537	
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000	
TUBE CORE DIAMETER (FT)	48.67	49.46	
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.5362	.5362	
TOTAL HEAT FLOW ACROSS (BTU/SEC)	1127092.	1156812.	
TOTAL HEAT FLOW ACROSS (MW)	1189.3664	1220.7281	
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	1812611.	1711390.	TOTAL =3524001.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	72.5	68.5	TOTAL = 141.0

Fig. A-1 (Cont.)

***** 25.000 (MW) NET POWER OUTPUT ***** WORKING FLUID AMMONIA ***** RUN IDENTIFICATION 28 SEP 77 08:34:13

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	1297.	1381.
FLOW RATE PER PUMP (GPM)	582273.	619642.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	338.7	271.1
PUMP HEAD (FT)	5.281	4.274

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

	CONDENSER	EVAPORATOR
PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75].	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75].	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.5029
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0237
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.198
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

	CONDENSER	EVAPORATOR
PUMP DIAMETER (FT)	13.00	13.00
PUMP SPEED (RPM)	70.3188	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	3.079	2.642

Fig. A-1 (Cont.)

***** COLD WATER PIPE VARIABLES	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	-545.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	1.665	3.746
C.W. PIPE REYNOLDS NO.	7183734.	10775600.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	1	
STATIC HEAD OF COLD WATER (PSF)	53.0	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	59.3	

***** TURBINE VARIABLES

GROSS TURBINE POWER OUTPUT OF 1 TURBINES	32.713	MW
NO. OF TURBINES	1	
FLOW RATE AT TURBINE EXHAUST / TURBINE	6808.77	CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	2208.81	LB/SEC
NET TURBINE EFFICIENCY	.879	
TURBINE WORK	14.035	BTU/LB
ISEN. TURBINE WORK	15.973	BTU/LB
AIRCO AXIAL FLOW		
SPECIFIC SPEED	91.570	RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710	DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420	FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.382	DIMENSIONLESS
DRY EFFICIENCY	.902	
FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION		
TURBINE SPEED	1306.	RPM
TURBINE DIAMETER	11.10	FT
REYNOLDS NUMBER	.380+09	
ROTOR TIP SPEED	759.08	FT/SEC
FLOW COEFFICIENT	.2288	DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.6941	DIMENSIONLESS
VELOCITY RATIO	.849	DIMENSIONLESS
SPOUTING VELOCITY	894.37	FT/SEC

***** CONDENSATE PUMP VARIABLES

NO. OF CONDENSATE PUMPS	2	
PUMP EFF.	.900	
PUMP MOTOR EFF.	.900	
PUMP WORK	.580	BTU/LB
ISEN. PUMP WORK	.522	BTU/LB
MASS FLOW RATE / PUMP	1104.41	LB/SEC
VOLUME FLOW RATE / PUMP	28.3632	CFS
PUMP HEAD	109.842	PSI

Fig. A-1 (Cont.)

OPTIMIZE ON AREA. THE OPTIMIZATION VARIABLES ARE;
T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
RUNS FOR BILL CUTLER - SEPT 1977 -
THIS RUN IS FOR THE 25-MW POWER SYSTEM DEVELOPMENT WORK
SERIES AT FORTY , 36, 32 -DEGREE DELTA TEMPERATURES - ALSO 44 DEGREES
NO DEMISTER - FOULING FACTORS ARE 0.0003 TUBE PITCH IS 1.2
EXPONENTT USED IN KAYS EQUATION IS 0.17
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0	70.93	130.909	.99000	624.17	.4443
01	70.93	130.909		[121.58]	LIQUID AT AVERAGE DEMISTER PRESSURE
1	70.93	130.909	.99000	624.17	.4443
2	71.03	131.133	.98997	624.17	.4451
3	52.90	94.255	.96722	608.67	.3312
4	52.94	94.323	.96721	608.67	.3315
45	52.62	93.750		[100.84]	LIQUID AT AVERAGE CONDENSER PRESSURE
5	52.29	93.179	.00000	100.48	38.8886
6	52.29	98.212	.00000	100.48	38.8886
7	52.35	213.060	.00000	101.08	38.8859
8	52.35	141.402	.00000	101.08	38.8859
89	71.04	131.156		[121.70]	LIQUID AT AVERAGE BOILER PRESSURE
9	70.93	130.909	.99000	624.17	.4443

GENERATOR EFF. = .985
DEMISTER BACKFLOW MASS RATE = -.00 LB/SEC
RANKINE CYCLE EFF. = .028483
CARNOT CYCLE EFF. = .033911
EFFICIENCY RATIO = .839935

***** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP PSI)
1-2	10.099	55.350	.147	110.000	-90.0	.010	.044	-.278	-.224
3-4	11.697	55.350	.110	30.000	-30.0	.002	.000	-.069	-.067
5-6	1.466	30.000	3.777	20.000	-20.0	.369	.000	-5.401	-5.032
7-8	1.466	30.000	3.777	310.000	240.0	5.715	1.133	64.810	71.658
9-0	10.099	55.350	.147	2.000	.0	.000	.000	.000	.000

Fig A-2 Optimization Program Output for $\Delta T = 36°F$

***** 25,000 (MW) NET POWER OUTPUT ***** WORKING FLUID AMMONIA ***** RUN IDENTIFICATION 26 SEP 77 12:48:13

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF TUBES	77924.	78014.
TUBE PITCH	1.20	1.20
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500
HEAT EXCHANGER TUBE THICKNESS (IN)	.0580	.0580
TUBE LENGTH (FT)	45.003	42.034
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0
SEA WATER TEMP. IN (F)	43.875	79.875
SEA WATER TEMP. OUT (F)	47.511	76.231
TUBE SEA WATER VELOCITY (FPS)	5.641	5.831
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	297.930	298.789
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000
SEA WATER FLOW RATE (LB/SEC)	294511.15	303447.77
SEA WATER FLOW RATE (CFS)	4591.96	4752.70
SEA WATER REYNOLDS NO. THRU TUBES	44561.	59894.
SEA WATER DENSITY (LBM/FT**3)	64.136	63.847
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D. (BTU/HR-SQ FT)	420.	455.
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS (USING KAYS EQ.)	909.	1095.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000
WORKING FLUID HEAT TRANSFER COEFFICIENT	1000.	1000.
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000060	.000060
LOG MEAN TEMP. DIFFERENCE (F)	6.761	6.857
NTU'S	.5378	.5314
EFFECTIVENESS	.4159	.4122
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.938	.493
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000
TUBE CORE DIAMETER (FT)	43.97	43.99
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.5362	.5362
TOTAL HEAT FLOW ACROSS (BTU/SEC)	1001139.	1030491.
TOTAL HEAT FLOW ACROSS (MW)	1056.4542	1087.4276
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	1323884.	1237969. TOTAL =2561849.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	53.0	49.5 TOTAL = 102.5

Fig. A-2 (Cont.)

***** 25.000 (MW) NET POWER OUTPUT ***** WORKING FLUID AMMONIA ***** RUN IDENTIFICATION 26 SEP 77 12:48:3

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	1148.	1188.
FLOW RATE PER PUMP (GPM)	515254.	533290.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	355.8	298.8
PUMP HEAD (FT)	5.548	4.680

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75]	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75]	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.5153
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0243
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.227
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

PUMP DIAMETER (FT)	12.08	12.08
PUMP SPEED (RPM)	77.5654	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	2.862	2.488

Fig. A-2 (Cont.)

***** COLD WATER PIPE VARIABLES

	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	-545.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	1.473	3.314
C.W. PIPE REYNOLDS NO.	6356894.	9535341.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	1	
STATIC HEAD OF COLD WATER (PSF)	53.0	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	57.9	

***** TURBINE VARIABLES

GROSS TURBINE POWER OUTPUT OF 1 TURBINES	32.236	MW
NO. OF TURBINES	1	
FLOW RATE AT TURBINE EXHAUST / TURBINE	5947.45	CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	1970.00	LB/SEC
WET TURBINE EFFICIENCY	.877	
TURBINE WORK	15.506	BTU/LB
ISEN. TURBINE WORK	17.674	BTU/LB

AIRCO AXIAL FLOW

SPECIFIC SPEED	91.570	RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710	DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420	FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.382	DIMENSIONLESS
DRY EFFICIENCY	.902	

FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

TURBINE SPEED	1508.	RPM
TURBINE DIAMETER	10.11	FT
REYNOLDS NUMBER	.371+09	
ROTOR TIP SPEED	798.48	FT/SEC
FLOW COEFFICIENT	.2288	DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.6941	DIMENSIONLESS
VELOCITY RATIO	.849	DIMENSIONLESS
SPOUTING VELOCITY	940.79	FT/SEC

***** CONDENSATE PUMP VARIABLES

NO. OF CONDENSATE PUMPS	2	
PUMP EFF.	.900	
PUMP MOTOR EFF.	.900	
PUMP WORK	.607	BTU/LB
ISEN. PUMP WORK	.546	BTU/LB
MASS FLOW RATE / PUMP	985.00	LB/SEC
VOLUME FLOW RATE / PUMP	25.3288	CFS
PUMP HEAD	114.849	PSI

Fig. A-2 (Cont.)

OPTIMIZE ON AREA. THE OPTIMIZATION VARIABLES ARE:
T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
THIS RUN IS FOR THE 25-MW POWER SYSTEM DEVELOPMENT WORK
ALUMINUM TUBES
NO DEMISTER - FOULING FACTORS ARE 0.0003 - TUBE PITCH IS 1.2
SURFACE DESIGN - 58 MIL TUBES
NEW EXPONENT USED IN KAYS EQUATION(.17)
THIS GROUP HAS FORTY-DEGREE DELTA TEMPERATURE:
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0 evap out	73.99	138.084	.99000	624.70	.4675
01	73.99	138.084		[125.08] LIQUID AT AVERAGE DEMISTER PRESSURE	
1 demister	73.99	138.084	.99000	624.70	.4675
2 turb in	74.09	133.318	.98997	624.70	.4683
3 turb out	53.97	95.180	.96487	607.67	.3386
4 cond in	54.01	95.248	.96482	607.67	.3389
45	53.71	95.703		[102.06] LIQUID AT AVERAGE CONDENSER PRESSURE	
5 cond out	53.41	95.161	.00000	101.72	38.8336
6 pump in	53.41	102.161	.00000	101.72	38.8336
7 pump out	53.46	223.493	.00000	102.36	38.8307
8 evap in	53.46	149.552	.00000	102.36	38.8307
89	74.09	133.318		[125.19] LIQUID AT AVERAGE BOILER PRESSURE	
9 evap out	73.99	138.084	.99000	624.70	.4675

GENERATOR EFF. = .985
DEMISTER BACKFLOW MASS RATE = -.00 LB/SEC
RANKINE CYCLE EFF. = .031385
CARNOT CYCLE EFF. = .037446
EFFICIENCY RATIO = .838144

***** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP (PSI)
1-2	9.310	55.350	.155	110.000	-90.0	.011	.046	-.292	-.235
3-4	10.939	55.350	.112	30.000	-30.0	.002	.000	-.071	-.069
5-6	1.388	30.000	3.772	20.000	-20.0	.393	.000	-5.394	-5.000
7-8	1.388	30.000	3.772	310.000	240.0	6.091	1.131	64.718	71.941
9-0	9.310	55.350	.155	2.000	.0	.000	.000	.000	.000

Fig. A-3 Optimization Program Output for $\Delta T = 40°F$

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR	
NUMBER OF TUBES	61982.	67356.	
TUBE PITCH	1.20	1.20	
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500	
HEAT EXCHANGER TUBE THICKNESS (IN)	.0580	.0580	
TUBE LENGTH (FT)	42.504	37.167	
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0	
SEA WATER TEMP. IN (F)	43.875	83.875	
SEA WATER TEMP. OUT (F)	47.654	80.272	
TUBE SEA WATER VELOCITY (FPS)	6.074	6.103	
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	328.736	294.376	
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000	
SEA WATER FLOW RATE (LB/SEC)	252239.76	273990.30	
SEA WATER FLOW RATE (CFS)	3932.87	4294.35	
SEA WATER REYNOLDS NO. THRU TUBES	47982.	62681.	
SEA WATER DENSITY (LBM/FT**3)	64.136	63.802	
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D.	431.	465.	BTU/HR-SQ FT
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS	967.	1154.	USING KAYS EQ.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000	
WORKING FLUID HEAT TRANSFER COEFFICIENT	1000.	1000.	
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300	
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000060	.000060	
LOG MEAN TEMP. DIFFERENCE (F)	7.790	7.849	
NTU'S	.4851	.4591	
EFFECTIVENESS	.3844	.3681	
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000	
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.878	.468	
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000	
TUBE CORE DIAMETER (FT)	39.21	40.88	
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.5362	.5362	
TOTAL HEAT FLOW ACROSS (BTU/SEC)	891280.	920160.	
TOTAL HEAT FLOW ACROSS (MW)	940.5253	971.0005	
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	994566.	945085.	TOTAL =1939651.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	39.8	37.8	TOTAL = 77.6

Fig. A-3 (Cont.)

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	983.	1074.
FLOW RATE PER PUMP (GPM)	441299.	481860.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	385.3	294.4
PUMP HEAD (FT)	6.007	4.614

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

	CONDENSER	EVAPORATOR
PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75].	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75].	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.4901
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0231
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.167
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

	CONDENSER	EVAPORATOR
PUMP DIAMETER (FT)	10.96	10.96
PUMP SPEED (RPM)	88.9643	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	2.654	2.214

Fig. A-3 (Cont.)

***** COLD WATER PIPE VARIABLES	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	-545.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	1.262	2.839
C.W. PIPE REYNOLDS NO.	5444484.	8166727.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	1	
STATIC HEAD OF COLD WATER (PSF)	52.9	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	56.5	

***** TURBINE VARIABLES	
GROSS TURBINE POWER OUTPUT OF 1 TURBINES	31.659 MW
NO. OF TURBINES	1
FLOW RATE AT TURBINE EXHAUST / TURBINE	5202.13 CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	1761.61 LB/SEC
NET TURBINE EFFICIENCY	.875
TURBINE WORK	17.031 BTU/LB
ISEN. TURBINE WORK	19.441 BTU/LB
AIRCO AXIAL FLOW	
SPECIFIC SPEED	91.570 RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710 DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420 FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.392 DIMENSIONLESS
DRY EFFICIENCY	.902

FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION	
TURBINE SPEED	1732. RPM
TURBINE DIAMETER	9.23 FT
REYNOLDS NUMBER	.353+09
ROTOR TIP SPEED	837.45 FT/SEC
FLOW COEFFICIENT	.2288 DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.5941 DIMENSIONLESS
VELOCITY RATIO	.849 DIMENSIONLESS
SPOUTING VELOCITY	985.72 FT/SEC

***** CONDENSATE PUMP VARIABLES	
NO. OF CONDENSATE PUMPS	2
PUMP EFF.	.900
PUMP MOTOR EFF.	.900
PUMP WORK	.637 BTU/LB
ISEN. PUMP WORK	.573 BTU/LB
MASS FLOW RATE / PUMP	880.80 LB/SEC
VOLUME FLOW RATE / PUMP	22.6315 CFS
PUMP HEAD	120.332 PSI

Fig. A-3 (Cont.)

OPTIMIZE ON AREA. THE OPTIMIZATION VARIABLES ARE:
T2 - WORKING FLUID TEMPERATURE ENTERING THE TURBINE
MDOT - WORKING FLUID MASS FLOW RATE THROUGH THE TURBINE
SWVELB - SEA WATER VELOCITY THROUGH THE BOILER TUBES
SWVELC - SEA WATER VELOCITY THROUGH THE CONDENSER TUBES
RUNS FOR BILL CUTLER - SEPT 1977 -
THIS RUN IS FOR THE 25-MW POWER SYSTEM DEVELOPMENT WORK
SERIES AT FORTY , 36, 32 -DEGREE DELTA TEMPERATURES - ALSO 44 DEGREES
NO DEMISTER - FOULING FACTORS ARE 0.0003 TUBE PITCH IS 1.2
EXPONENTT USED IN KAYS EQUATION IS 0.17
TEMPERATURE PROFILE SOUTH OF HAWAII, N. LAT. 10 DEG.

STATION	TEMPERATURE (DEGREES F)	PRESSURE (PSIA)	QUALITY	ENTHALPY (BTU/LB)	DENSITY (#/CU-FT)
0	76.82	144.990	.99000	625.17	.4898
01	76.82	144.990		[128.33] LIQUID AT AVERAGE DEMISTER PRESSURE	
1	76.82	144.990	.99000	625.17	.4898
2	76.92	145.235	.98327	625.17	.4906
3	55.00	98.055	.96268	606.79	.3458
4	55.04	98.125	.96266	606.79	.3460
45	54.73	97.569		[103.22] LIQUID AT AVERAGE CONDENSER PRESSURE	
5	54.43	97.015	.00000	102.88	38.7828
6	54.43	101.990	.00000	102.88	38.7828
7	54.49	227.560	.00000	103.54	38.7798
8	54.49	155.421	.00000	103.54	38.7798
89	76.91	145.206		[128.43] LIQUID AT AVERAGE BOILER PRESSURE	
9	76.82	144.990	.99000	625.17	.4898

GENERATOR EFF. = .985
DEMISTER BACKFLOW MASS RATE = -.00 LB/SEC
RANKINE CYCLE EFF. = .033965
CARNOT CYCLE EFF. = .040619
EFFICIENCY RATIO = .836262

***** WORKING FLUID PLUMBING

STATION	PIPE DIAM (FT)	PIPE VEL (FT/SEC)	DYN PRES (PSI)	PIPE LENGTH (FT)	HEAD (FT)	FRIC DP (PSI)	TURN DP (PSI)	GRAV DP (PSI)	TOT DP (PSI)
1-2	8.733	55.350	.162	110.000	-90.0	.013	.049	-.306	-.245
3-4	10.394	55.350	.114	30.000	-30.0	.002	.000	-.072	-.070
5-6	1.333	30.000	3.767	20.000	-20.0	.411	.000	-5.386	-4.975
7-8	1.333	30.000	3.767	310.000	240.0	6.376	1.130	64.633	72.139
9-0	8.733	55.350	.162	2.000	.0	.000	.000	.000	.000

Fig. A-4 Optimization Program Output for ΔT = 44° F

***** 25.000 (MW) NET POWER OUTPUT ***** WORKING FLUID AMMONIA ***** RUN IDENTIFICATION 26 SEP 77 07:44:12

***** HEAT EXCHANGER VARIABLES	CONDENSER	EVAPORATOR	
NUMBER OF TUBES	53701.	55593.	
TUBE PITCH	1.20	1.20	
HEAT EXCHANGER TUBE O.D. (IN)	1.500	1.500	
HEAT EXCHANGER TUBE THICKNESS (IN)	.0580	.0580	
TUBE LENGTH (FT)	38.670	35.181	
TUBE MATERIAL CONDUCTIVITY (BTU/HR-FT-F)	77.0	77.0	
SEA WATER TEMP. IN (F)	43.875	87.875	
SEA WATER TEMP. OUT (F)	47.619	84.147	
TUBE SEA WATER VELOCITY (FPS)	6.497	6.582	
HEAT EXCHANGER SEA WATER PRESSURE DROP (PSF)	346.849	326.856	
SEA WATER PRESSURE DROP ENHANCEMENT FACTOR	1.000	1.000	
SEA WATER FLOW RATE (LB/SEC)	233767.68	243826.44	
SEA WATER FLOW RATE (CFS)	3644.86	3823.02	
SEA WATER REYNOLDS NO. THRU TUBES	51325.	67608.	
SEA WATER DENSITY (LBM/FT**3)	64.136	63.779	
AVG. HEAT TRANSFER COEFFICIENT BASED ON I.D.	442.	480.	BTU/HR-SQ FT
SEA WATER SIDE HEAT TRANSFER COEFFICIENTS	1022.	1246.	USING KAYS EQ.
SEA WATER HEAT TRANSFER ENHANCEMENT FACTOR	1.000	1.000	
WORKING FLUID HEAT TRANSFER COEFFICIENT	1000.	1000.	
FOULING RESISTANCE (HR-SQ FT-F/BTU)	.000300	.000300	
TUBE WALL RESISTANCE (HR-SQ FT-F/BTU)	.000060	.000060	
LOG MEAN TEMP. DIFFERENCE (F)	8.854	8.974	
NTU'S	.4229	.4154	
EFFECTIVENESS	.3448	.3399	
ASSUMED SPRAYBAR PRESSURE DROP(PSI)		10.000	
ASSUMED INTERNAL VAPOR PRESSURE DROPS(PSI)	.897	.431	
ASSUMED DEMISTER PRESSURE DROP(PSI)		.000	
TUBE CORE DIAMETER (FT)	36.50	37.14	
RATIO OF TUBE FLOW AREA TO SHELL FLOW AREA	.5362	.5362	
TOTAL HEAT FLOW ACROSS (BTU/SEC)	818317.	847088.	
TOTAL HEAT FLOW ACROSS (MW)	863.5304	893.8910	
TUBE SURFACE AREA (SQ FT) BASED ON MEAN DI.	783953.	738345.	TOTAL =1522298.
TUBE SURFACE AREA (SQ FT/NET-KW) BASED ON MEAN DI.	31.4	29.5	TOTAL = 60.9

Fig. A-4 (Cont.)

***** SEA WATER PUMP VARIABLES	CONDENSER	EVAPORATOR
NUMBER OF PUMPS PER HEAT EXCHANGER	4	4
FLOW RATE PER PUMP (CFS)	911.	956.
FLOW RATE PER PUMP (GPM)	408982.	428972.
SUM OF PRESSURE LOSSES ACROSS S.W. FLOW PATH (PSF)	402.9	326.9
PUMP HEAD (FT)	6.282	5.125

CONSENSUS DESIGN OF THREE MANUFACTURERS
THE CONDENSER PUMP SPECIFIC SPEED AND SPECIFIC DIAMETER ARE FIXED FROM 40MW BASELINE DATA.
THIS YIELDS A DIAMETER AND SPEED FOR THE CONDENSER PUMPS.
THE BOILER PUMP DIAMETER IS SET EQUAL TO THE CONDENSER PUMP DIAMETER WHICH YIELDS A SPECIFIC DIAMETER FOR THE BOILER PUMP.
THE BOILER PUMP SPEED AND SPECIFIC SPEED ARE UNKNOWN.

	CONDENSER	EVAPORATOR
PUMP SPECIFIC SPEED [RPM*SQRT(CFS)/FT**.75],	727.0	
PUMP SPECIFIC SPEED [RPM*SQRT(GPM)/FT**.75],	15401.9	
SPECIFIC SPEED (DIMENSIONLESS USING RAD/SEC)	5.64	
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(CFS)]	.5470	.5076
PUMP SPECIFIC DIAMETER [FT*FT**.25/SQRT(GPM)]	.0258	.0240
SPECIFIC DIAMETER (DIMENSIONLESS)	1.303	1.209
PUMP EFFICIENCY	.860	.860
PUMP MOTOR EFFICIENCY	.900	.900

FOLLOWING SEA WATER PUMP VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

	CONDENSER	EVAPORATOR
PUMP DIAMETER (FT)	10.43	10.43
PUMP SPEED (RPM)	95.5675	
PUMP ELECTRICAL POWER REQUIREMENT (MW)	2.573	2.189

Fig. A-4 (Cont.)

***** COLD WATER PIPE VARIABLES

	TOP	BOTTOM
C.W. PIPE I.D. (FT)	63.0	42.0
C.W. PIPE DEPTH (FT)	-545.0	-2045.0
C.W. PIPE SEA WATER VELOCITY (FPS)	1.169	2.631
C.W. PIPE REYNOLDS NO.	5045773.	7568659.
NO. OF C.W. PIPE SECTIONS	10	
NO. OF POWER MODULES	1	
STATIC HEAD OF COLD WATER (PSF)	53.0	
SUM OF C.W. PIPE PRESSURE LOSSES (PSF)	56.1	

***** TURBINE VARIABLES

GROSS TURBINE POWER OUTPUT OF 1 TURBINES	31.501	MW
NO. OF TURBINES	1	
FLOW RATE AT TURBINE EXHAUST / TURBINE	4696.13	CFS
TOTAL MASS FLOW RATE THROUGH THE TURBINE(S)	1623.92	LB/SEC
WET TURBINE EFFICIENCY	.875	
TURBINE WORK	18.383	BTU/LB
ISEN. TURBINE WORK	21.014	BTU/LB
AIRCO AXIAL FLOW		
SPECIFIC SPEED	91.570	RPM*SQRT(CFS)/FT**.75
SPECIFIC SPEED	.710	DIMENSIONLESS USING RAD/SEC
SPECIFIC DIAMETER	1.420	FT*FT**.25/SQRT(CFS)
SPECIFIC DIAMETER	3.382	DIMENSIONLESS
DRY EFFICIENCY	.902	

FOLLOWING TURBINE VARIABLES ARE ONLY USED FOR OUTPUT INFORMATION

TURBINE SPEED	1932.	RPM
TURBINE DIAMETER	8.60	FT
REYNOLDS NUMBER	.358+09	
ROTOR TIP SPEED	870.68	FT/SEC
FLOW COEFFICIENT	.2288	DIMENSIONLESS USING RPM
HEAD COEFFICIENT	.6941	DIMENSIONLESS
VELOCITY RATIO	.849	DIMENSIONLESS
SPOUTING VELOCITY	1025.85	FT/SEC

***** CONDENSATE PUMP VARIABLES

NO. OF CONDENSATE PUMPS	2	
PUMP EFF.	.900	
PUMP MOTOR EFF.	.900	
PUMP WORK	.666	BTU/LB
ISEN. PUMP WORK	.599	BTU/LB
MASS FLOW RATE / PUMP	811.96	LB/SEC
VOLUME FLOW RATE / PUMP	20.9362	CFS
PUMP HEAD	125.570	PSI

Fig. A-4 (Cont.)

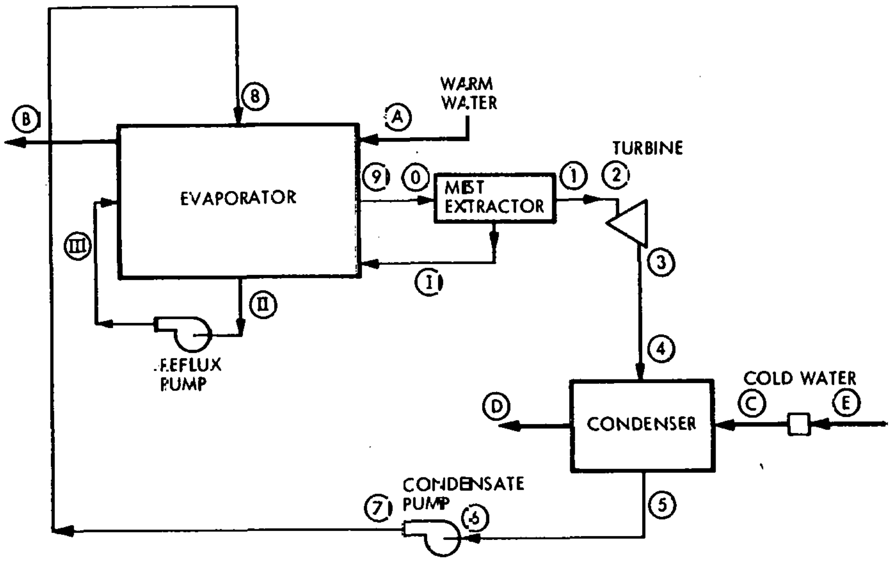
NUMBER ____________
DATE ____________
NET POWER ____________

CONDITION

ΔT (°F) ____________
TEMP PROFILE ____________
CWP LENGTH (FT)/DIA (FT) ____________
CWP INLET DEPTH (FT) ____________
CONFIGURATION ____________
EFFICIENCIES ____________
TURBINE ____________
GENERATOR ____________
SEA WATER PUMP ____________
AMMONIA PUMPS ____________

POWER (MWe)

TURBINE GROSS (SHAFT) ____________
SEA-WATER PUMP ____________
CONDENSATE PUMP ____________
REFLUX PUMP ____________
AUXILIARIES ____________
PLATFORM ____________
MISC. ____________
NET ____________



HEAT EXCHANGERS

	EVAP	COND
TUBE		
MATERIAL		
O.D. (IN.)		
THICKNESS (IN.)		
LENGTH (FT)		
SPACING		
NUMBER		
ORIENTATION		
HEAT EXCHANGER		
FOULING RES		
AREA (FT^2)		
AV. HEAT TRANS		
BUNDLE L/D		

STATION	9	0	1	2	3	4	5	6	7	8	A	B	C	D	E	I	II	III
PRESSURE (PSIA)																		
TEMPERATURE (°F)																		
ENTHALPY BTU/LB																		
DENSITY (LB/FT^3)																		
QUALITY (%)																		
FLOW (CFS)																		
FLOW (LB/SEC)																		
FLOW VEL. (FT/SEC)																		
ΔQ (BTU/SEC)																		
ΔP (PSF)																		
MIST EXTRACTOR LIQ FLOW (LB/SEC)																		

Fig. A-5 Ammonia Cycle Schematic Diagram

Appendix A-2
TRADE STUDY DATA

Section 4.7 of the main report presents data on the cost of ammonia system vapor and condensate plumbing for the submerged (spar) configuration and surface (ship) configuration. Optimization runs were made with vapor velocities of 55, 100, 160, and 200 fps with a constant condensate velocity of 30 fps for the configuration designed for 36° F ΔT to coincide with the piping cost data that were developed for flow rates at that design condition.

Cost data for the heat exchangers and ammonia plumbing were developed on an incremental basis from a baseline configuration that has plumbing sized for a vapor velocity of 55 fps. All heat exchangers incorporated the same number of tubes, and the variations in ammonia system pressure drop resulted in variations in overall heat exchanger tube lengths. The cost increments were based on data provided by Foster-Wheeler, which indicated that the slope of the incremental cost curve versus tube length has a value of approximately 2/3. The equation used to develop the incremental heat exchanger cost data is

$$\Delta \text{ cost} = \frac{2}{3}\left(\frac{L}{L_{BASE}} - 1\right) \times \text{base cost}$$

Optimization runs were also made to determine the effect of condensate velocity at a fixed vapor velocity of 160 fps.

The data used in this comparison are shown in Tables A-1 through A-3 and the results are shown in Figs. A-5 and A-6 for the submerged configuration for variations in vapor and condensate velocity, respectively, and Fig. A-7 for variations in vapor velocity for the shipboard configuration.

Table A-1

EFFECT OF VAPOR VELOCITY SUBMERGED CONFIGURATION,
CONDENSATE VELOCITY = 30 FPS

Item	Vapor Velocity (fps)			
	55	100	160	200
Condenser Tube Length (ft)	46.0	45.22	45.78	46.41
Evaporator Tube Length (ft)	42.03	42.27	42.82	43.29
Condenser Cost ($/kW)	175.00	175.56	177.02	178.63
Evaporator Cost ($/kW)	200.00	200.76	202.48	203.97
Pipe Cost ($/kW)	23.98	19.81	16.08	14.59
TOTAL ($/kW)	398.98	396.13	395.58	397.19

Table A-2

EFFECT OF CONDENSATE VELOCITY SUBMERGED CONFIGURATION
VAPOR VELOCITY = 160 FPS

Item	Condensate Velocity (fps)		
	10	20	30
Condenser Tube Length (ft)	45.80	45.69	45.78
Evaporator Tube Length (ft)	42.43	42.65	42.82
Condenser Cost ($/kW)	177.06	176.79	177.02
Evaporator Cost ($/kW)	201.24	201.96	202.48
Pipe Cost ($/kW)	20.78	16.98	16.08
TOTAL ($/kW)	399.08	395.73	395.58

Table A-3

EFFECT OF VAPOR VELOCITY SURFACE CONFIGURATION
CONDENSATE VELOCITY = 30 FPS

Item	Vapor Velocity (fps)			
	55	100	160	200
Condenser Tube Length (ft)	45.44	45.67	46.72	47.43
Evaporator Tube Length (ft)	42.48	42.86	43.37	44.24
Condenser Cost ($/kW)	176.14	176.73	179.46	181.28
Evaporator Cost ($/kW)	201.41	202.63	204.24	207.01
Pipe Cost ($/kW)	58.70	39.50	28.70	25.00
TOTAL ($/kW)	436.25	418.86	412.4	413.29

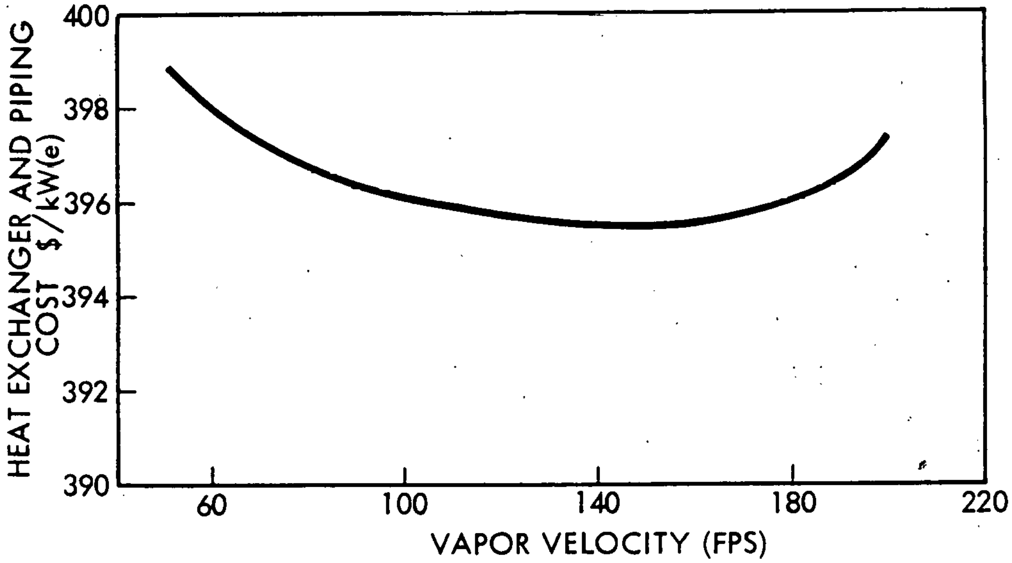


Fig. A-5 Effect of Vapor Velocity, Submerged Configuration, Condensate Velocity = 30 fps

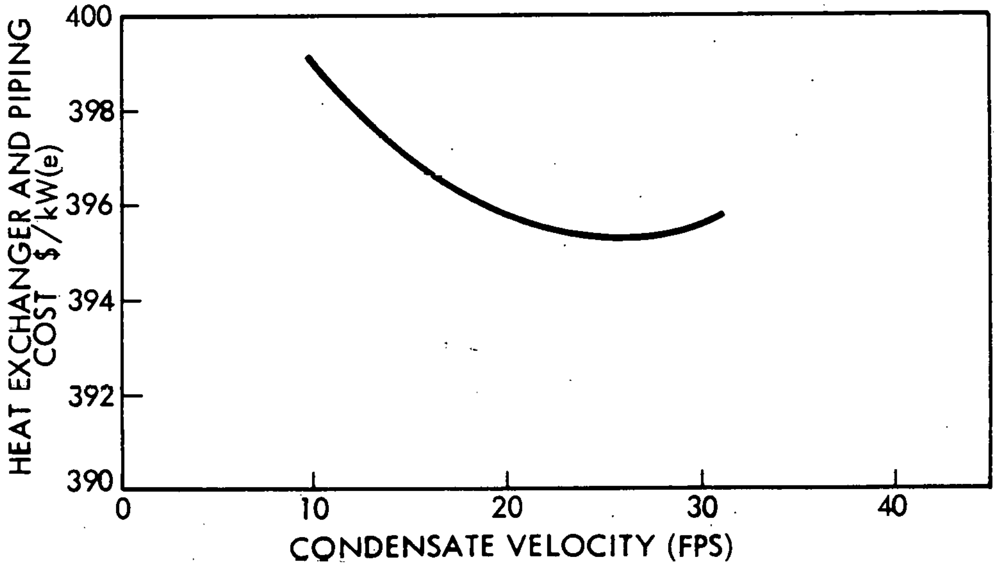


Fig. A-6 Effect of Condensate Velocity, Submerged Configuration, Vapor Velocity = 160 fps

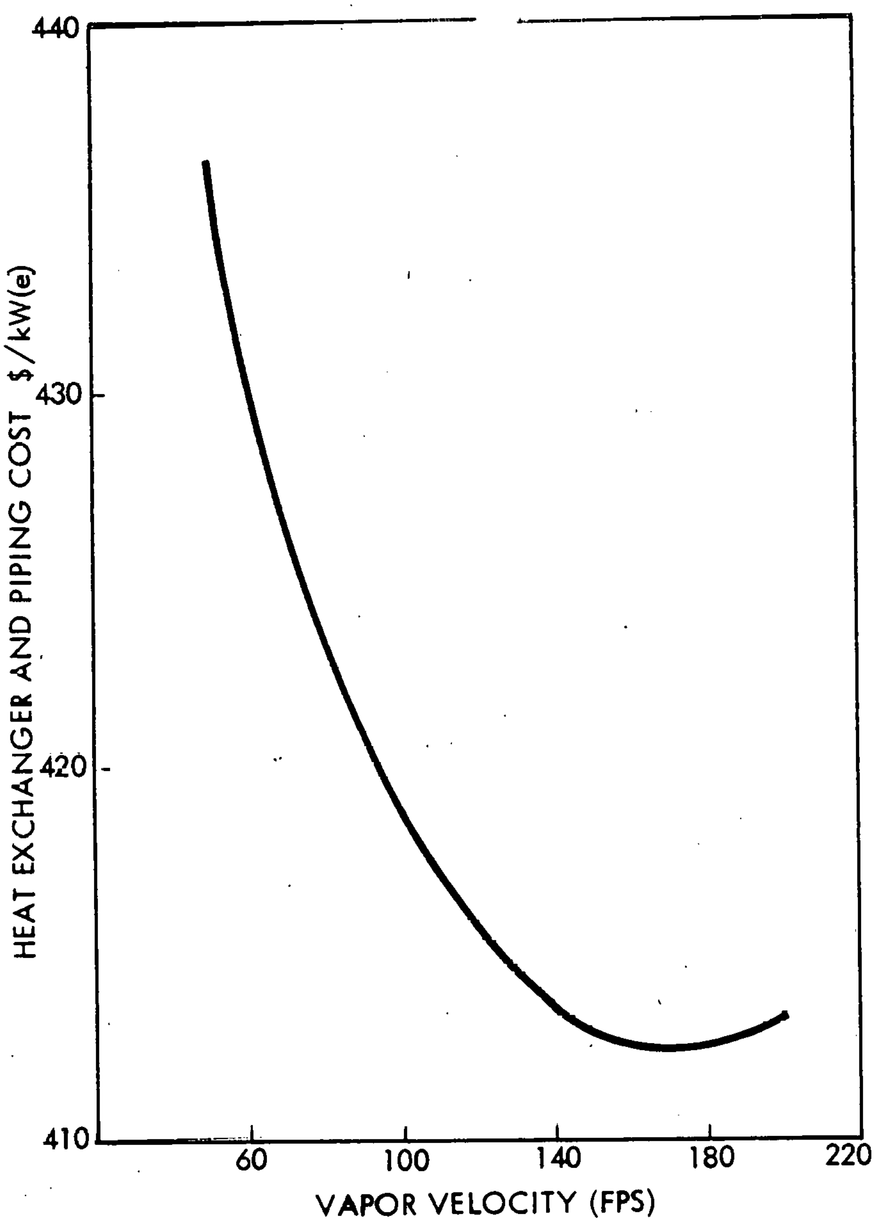


Fig. A-7 Effect of Vapor Velocity, Surface Configuration, Condensate Velocity = 30 fps

Appendix B
GENERAL ARRANGEMENTS

The power system module configurations derived in the Power System Development project and the original concepts are contained in this appendix.

Part A contains the two derived configurations – the surface platform (reference baseline) and the detachable external module.

Part B provides the 25-MW(e) and the 5-MW(e) power system schematics.

Part C contains the hardware breakdown structure for the power system module. This listing is applicable to both the surface and the submerged modules.

Part D contains the nine arrangements used in many of the trade studies. These arrangements are for three platforms each containing three sizes and multiples of heat exchangers to provide 100 MW(e). The arrangement sketches are for five 5-MW(e), two 12.5-MW(e), and one 25-MW(e) heat exchanger sizes and the three platforms are barge/ship (surface platform), supertanker (immersed heat exchangers), and the spar.

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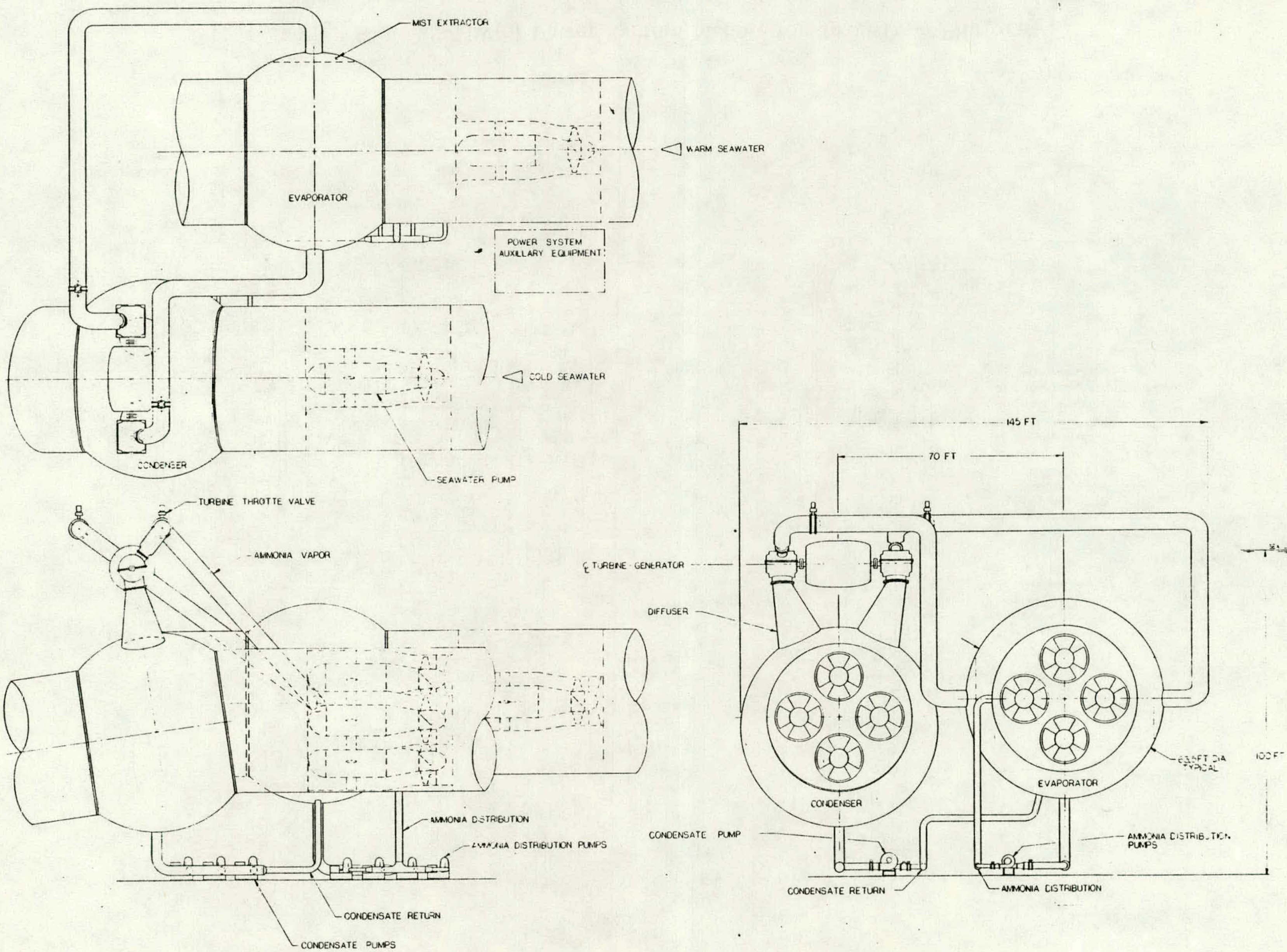

Fig. B-1 25-MW(e) Power System Module for 100-MW(e) Surface Platform

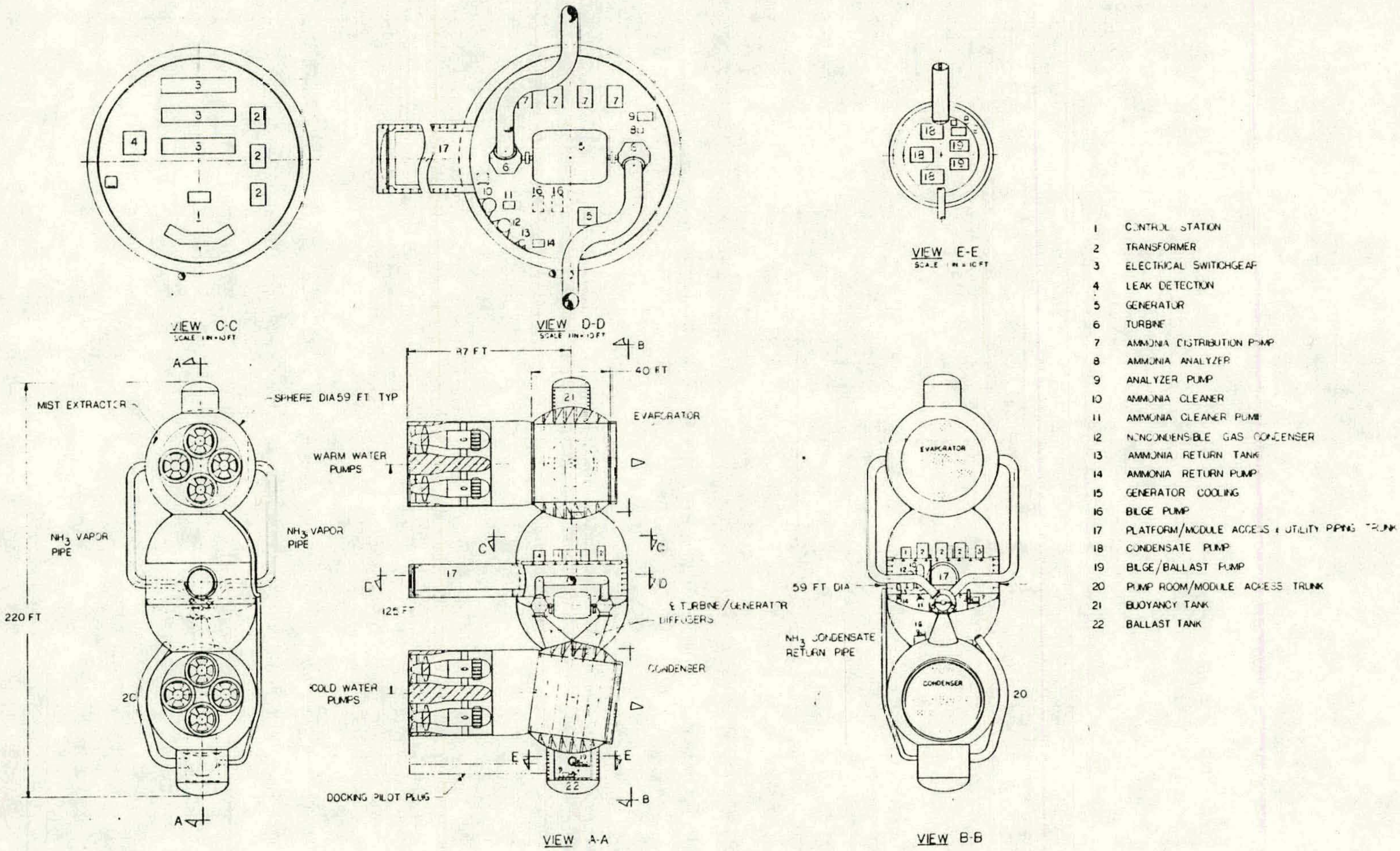

Fig. B-2 25-MW(e) Power System Module for Spar-Type Platform

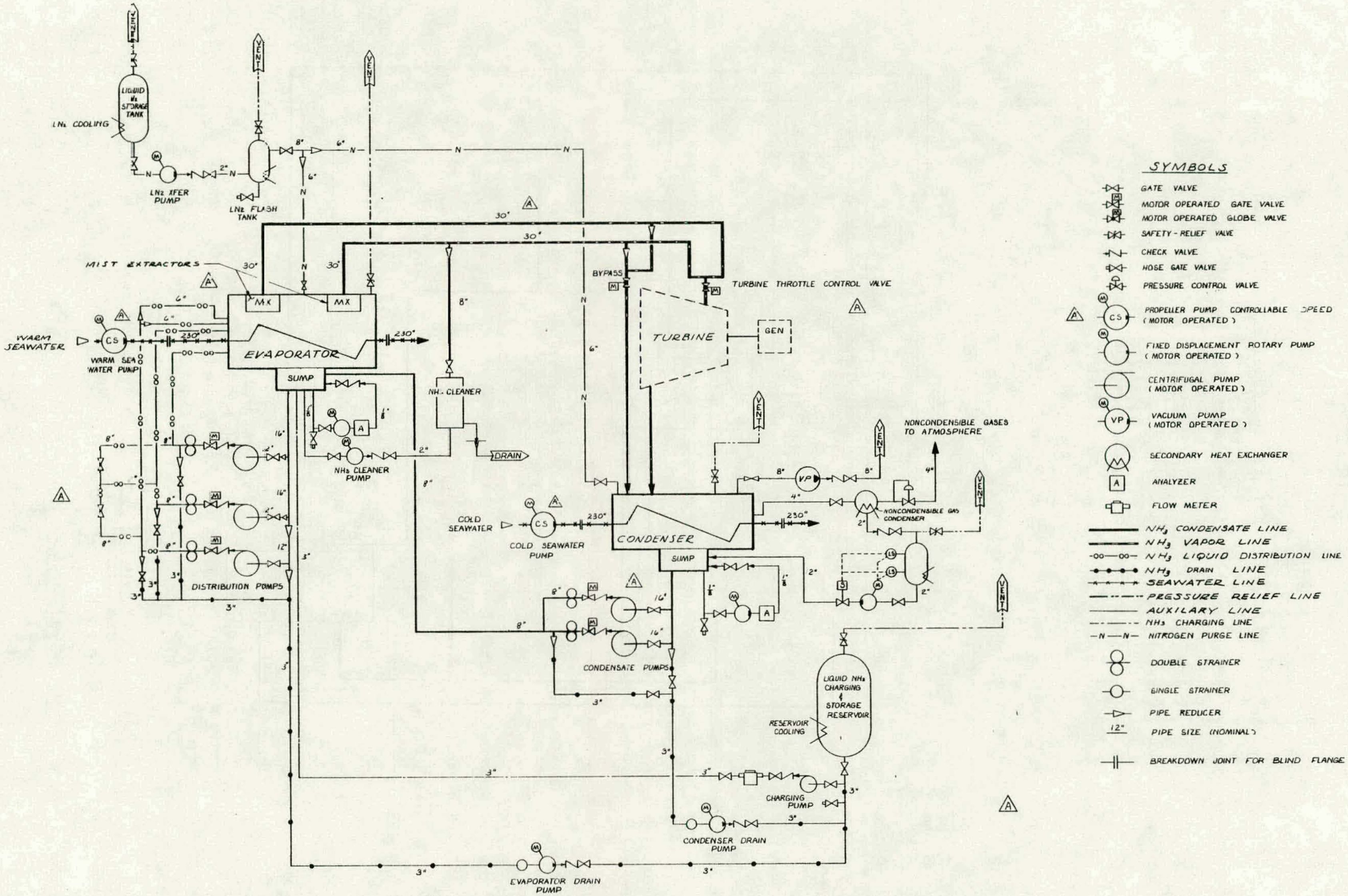

Fig. B-3 5-MW(e) System Schematic

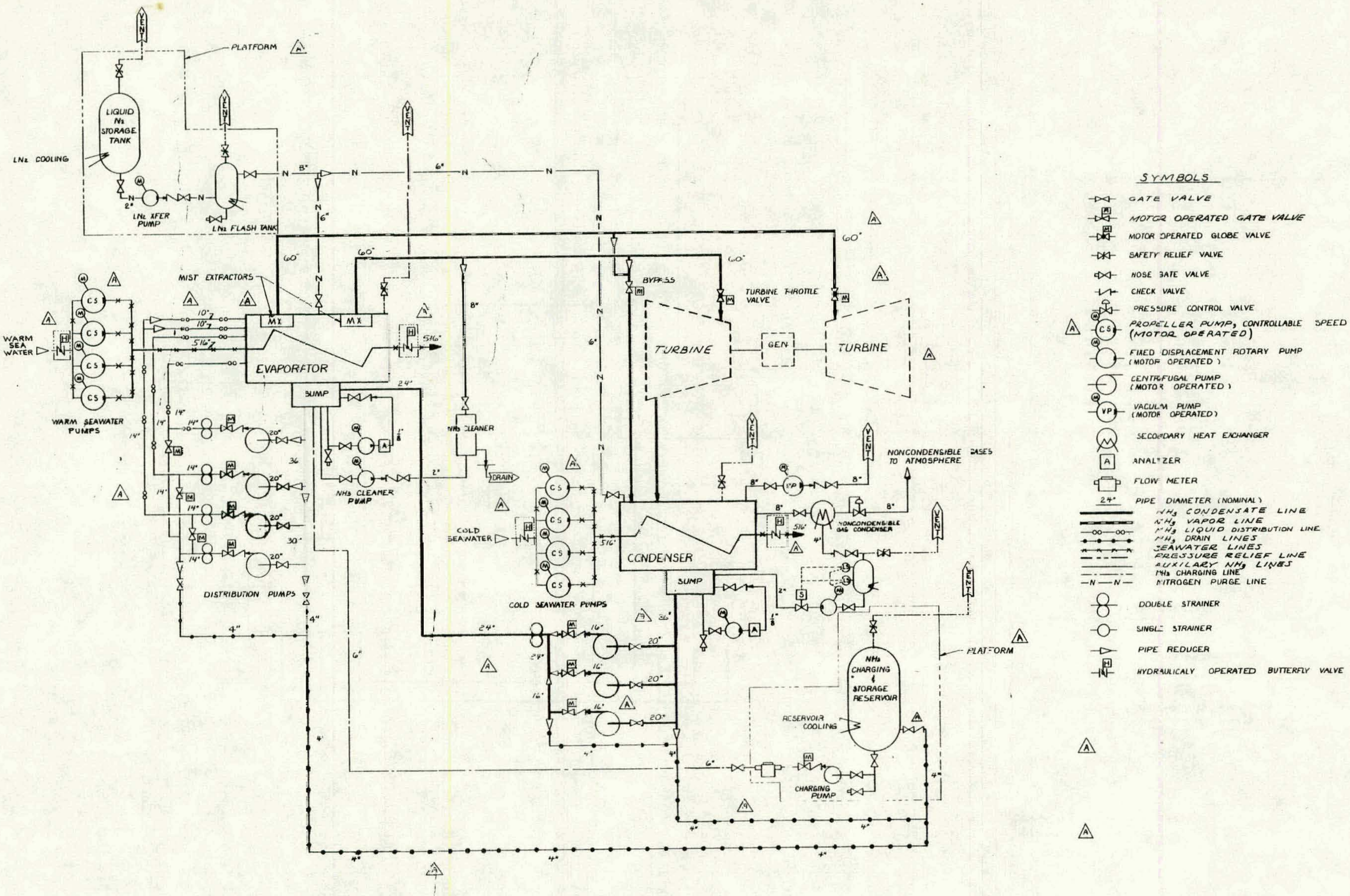

Fig. B-4   25-MW(e) System Schematic

HARDWARE BREAKDOWN STRUCTURE

1. POWER CYCLE

1.1 Evaporator

1.1.1 Pressure Shell
1.1.2 Transition Ring Assembly
1.1.3 Tubesheets
1.1.4 Tubes
1.1.5 Tube Support Sheet
1.1.6 Tube Support Sheet Structure
1.1.7 Vapor Port Rings (Adapter/Cone)
1.1.8 Mounting Stiffeners
1.1.9 Liquid Distribution Assembly

1.2 Condenser

1.2.1 Pressure Shell
1.2.2 Transition Ring Assembly
1.2.3 Tubesheets
1.2.4 Tubes
1.2.5 Tube Support Sheet
1.2.6 Tube Support Sheet Structure
1.2.7 Diffuser Port Rings (Adapter/Cone)
1.2.8 Mounting Stiffeners
1.2.9 Impingement Plate
1.2.10 Noncondensible Gas Removal Assembly
1.2.11 Sump Assembly

1.3 Turbine/Generator

1.3.1 Turbine/Generator
1.3.2 Diffuser
1.3.3 Ammonia Flow Control
1.3.4 Speed Control

1.4 Ammonia Vapor Assembly

1.4.1 Mist Extractor
1.4.2 Piping

1.5 Ammonia Liquid Assembly

1.5.1 Condensate Pumps
1.5.2 Condensate Pump Isolation Valves
1.5.3 Condensate Control Valve
1.5.4 Distribution Pumps
1.5.5 Distribution Pump Isolation Valves
1.5.6 Distribution Pump Control Valves
1.5.7 Fill/Drain Valve(s)
1.5.8 Liquid Piping

2. SEAWATER SYBSYSTEMS

2.1 Warm Water

2.1.1 Seawater Pump Assemblies
2.1.2 Pump Housings and Foundations
2.1.3 Inlet Assemblies
2.1.4 Outlet Assemblies

2.2 Cold Water

2.1.1 Seawater Pump Assemblies
2.1.2 Pump Housings and Foundations
2.1.3 Inlet Assemblies
2.1.4 Outlet Assemblies

3. AUXILIARY SYSTEMS

3.1 Power Cycle

3.1.1 Ammonia Cleanup (Dryer)
3.1.2 Ammonia Analysis
3.1.3 Ammonia Leak Detection
3.1.4 Ammonia Storage, Fill, and Purge
3.1.5 Overpressure Vent Assembly
3.1.6 Noncondensible Gas Removal

3.2 Operation Support

3.2.1 Auxiliary Power Generation (Standby Power)
3.2.2 Electrical Power Distribution
3.2.3 Turbine/Generator Lubrication
3.2.4 Generator/Transformer Cooling

3.3 Ancillary Subsystems

3.3.1 Nitrogen Purge/Pressurization System
3.3.2 Vacuum System
3.3.3 Compressed Air
3.3.4 Hydraulic Power
3.3.5 Ballasting and Attitude Control
3.3.6 Fresh Water

3.4 Fouling Countermeasures

3.5 Corrosion Control

4. OPERATIONAL CONTROL SYSTEM

4.1 Cycle Control Station

4.2 Control Data Processor and Processing Equipment

4.3 Cycle Operator's Console

4.3.1 Indicators
4.3.2 Displays
4.3.3 Controllers
4.3.4 Switches
4.3.5 Enclosure
4.3.6 Data Display/Recorders

4.4 Cycle Operator Communications

4.5 Control Data Instrumentation

4.6 Cabling and Connectors

4.7 J Boxes, Penetrators, Conduit

4.8 Equipment Environmental Control

4.8.1 Cooling
4.8.2 Humidity Control

5. DATA ACQUISITION AND CONTROL SYSTEM

5.1 Test Operators Console

5.1.1 Input Controls (Generalized and Dedicated)
5.1.2 Output Displays (Generalized and Dedicated)

5.2 Computer

5.3 Test Data, Recorder

5.4 Signal Processing Equipment

5.5 Test Operator's Communications

5.6 Diagnostic Data Instrumentation

5.7 Equipment Environmental Control

6. SAFETY AND EMERGENCY SYSTEMS

6.1 Sensors and Alarms (fire, ammonia, flood, toxic conditions, damage)

6.2 Fire Suppression

6.3 Equipment Deluge

6.4 Personnel Washdown

6.5 Emergency Breathing

6.6 Damage Control

6.7 Emergency Power

6.8 Emergency Lighting and Communications

6.9 Protective Clothing

6.10 Safety Line, Harnesses, Hoists

6.11 First Aid (Kits, Stretchers, Baskets)

7. PERSONNEL ACCOMMODATIONS AND OUTFITTING/FURNISHINGS

7.1 Lighting

7.2 Communications

7.3 Walkways, Ladders, Rails, Handholds

7.4 Shield and Guards

7.5 Doors and Hatches

7.6 Materiel and Consumables Storage

7.6.1 General
7.6.2 Hazardous

7.7 Food, Water, and Waste Disposal

7.8 Control Room Furnishings

7.8.1 Work Station
7.8.2 Technical Data Storage

7.9 Equipment Handling

8. STRUCTURE

8.1 Equipment Interfaces/Foundations

8.2 Interconnecting Structure (External Module)

8.3 Mechanical Attachments

9. SPECIAL SUPPORT EQUIPMENT

9.1 Mechanical/Handling

9.2 Special Test Equipment

9.2.1 Checkout Test – Operational
9.2.2 Checkout Test – Test Data
9.2.3 Checkout Cables
9.2.4 Auxiliary Power and Lights

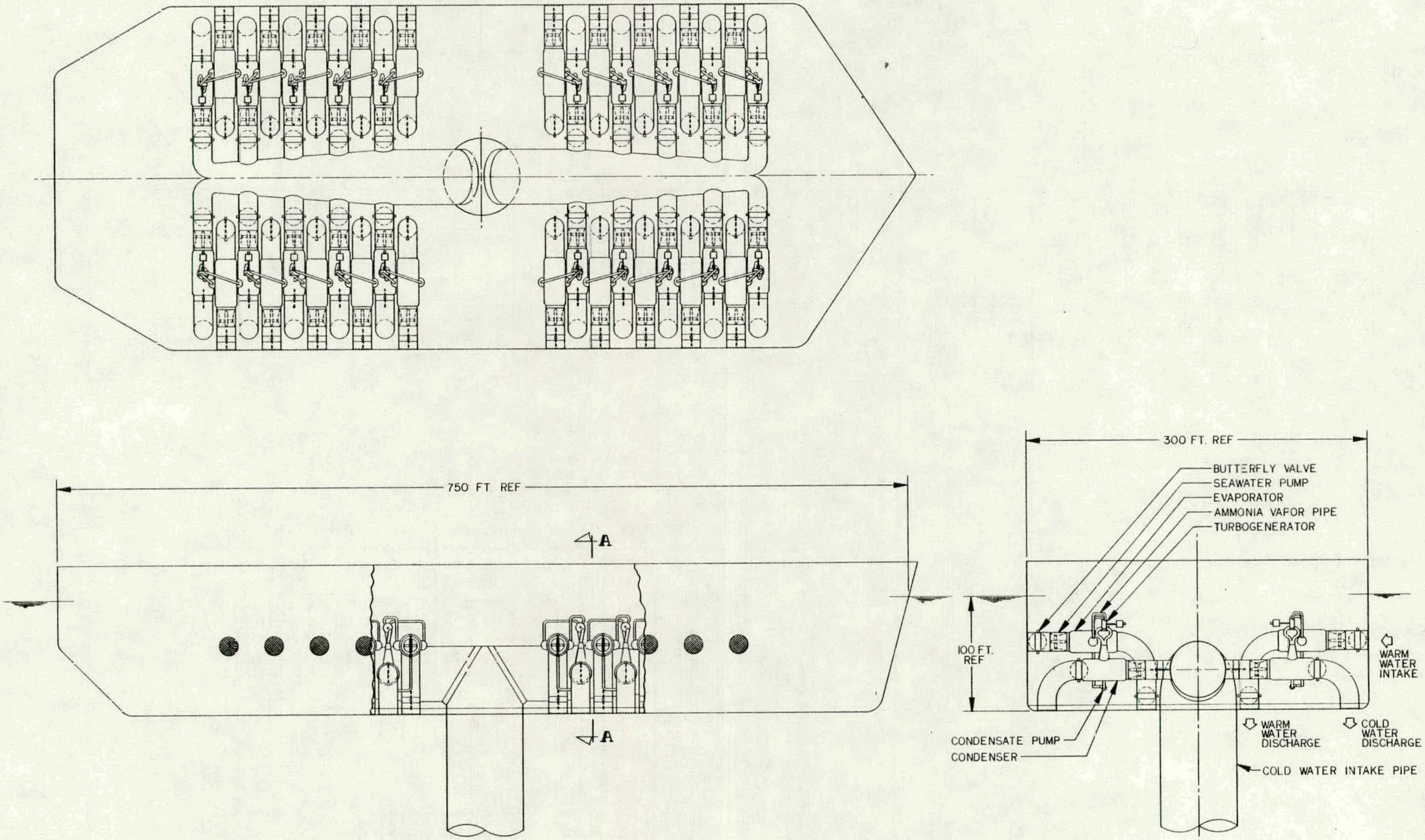

Fig. B-5 Surface Platform 5-MW(e) Heat Exchanger Concept

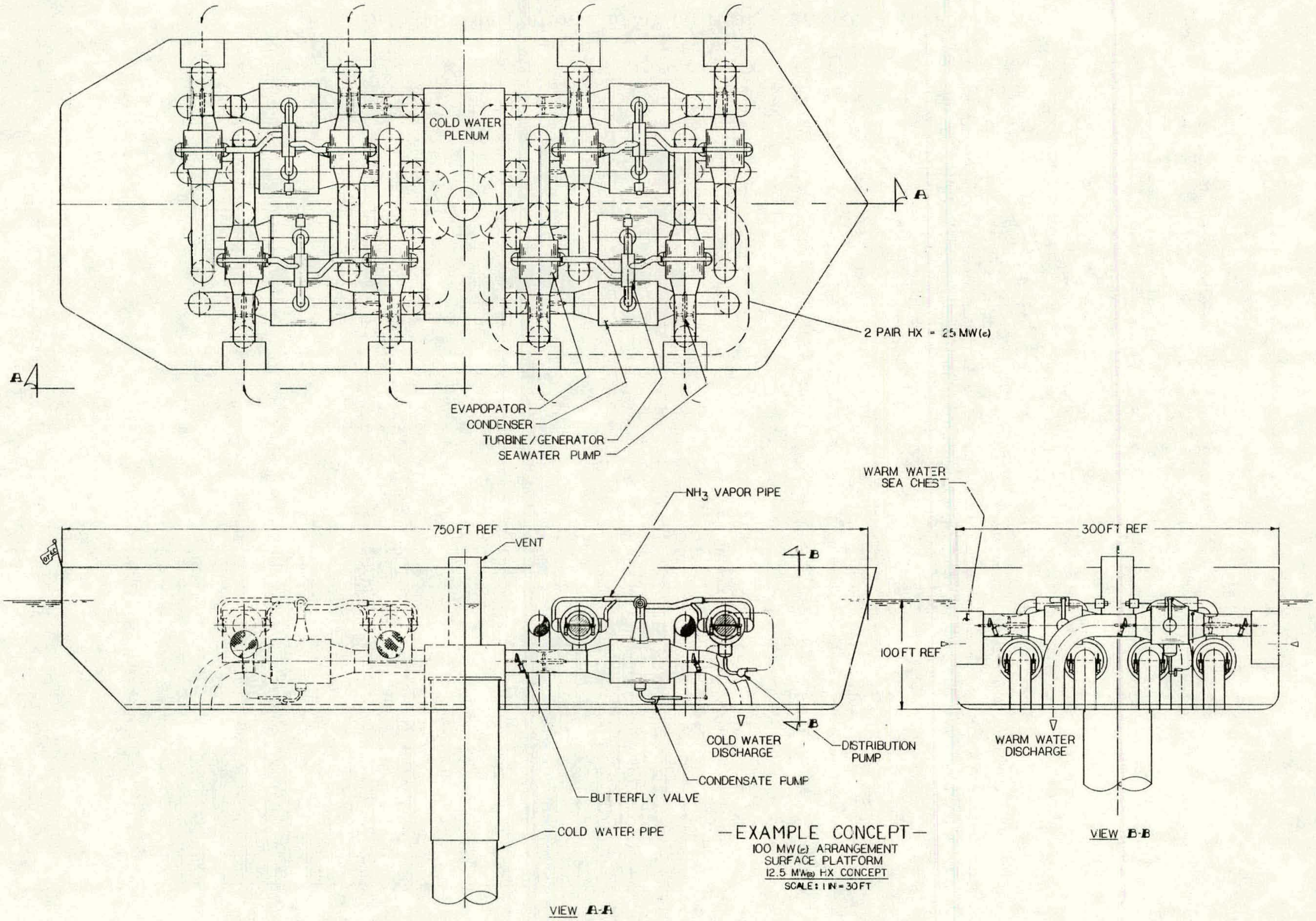

Fig. B-6 Surface Platform 12.5-MW(e) Heat Exchanger Concept

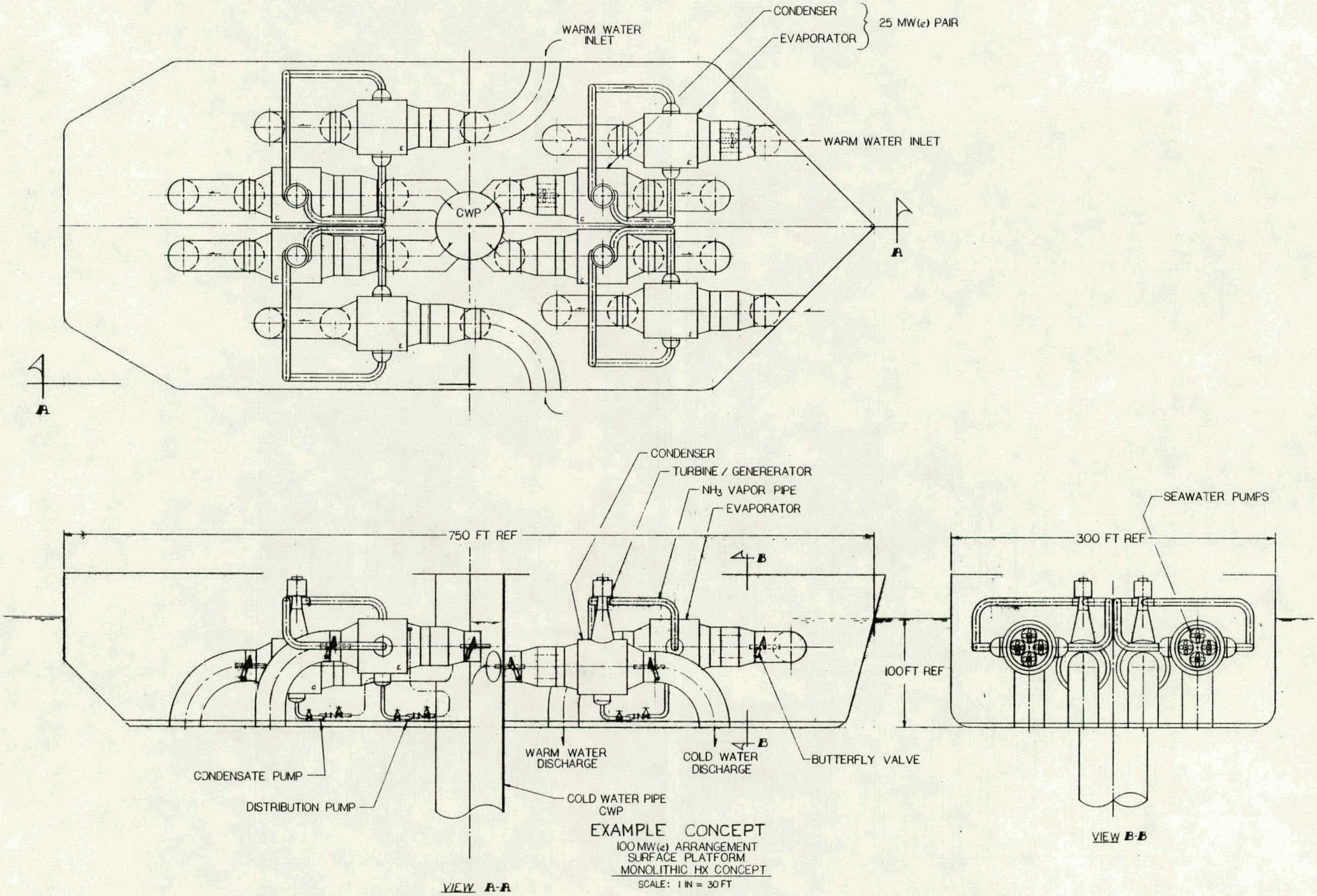

Fig. B-7 Surface Platform Monolithic Heat Exchanger Concept

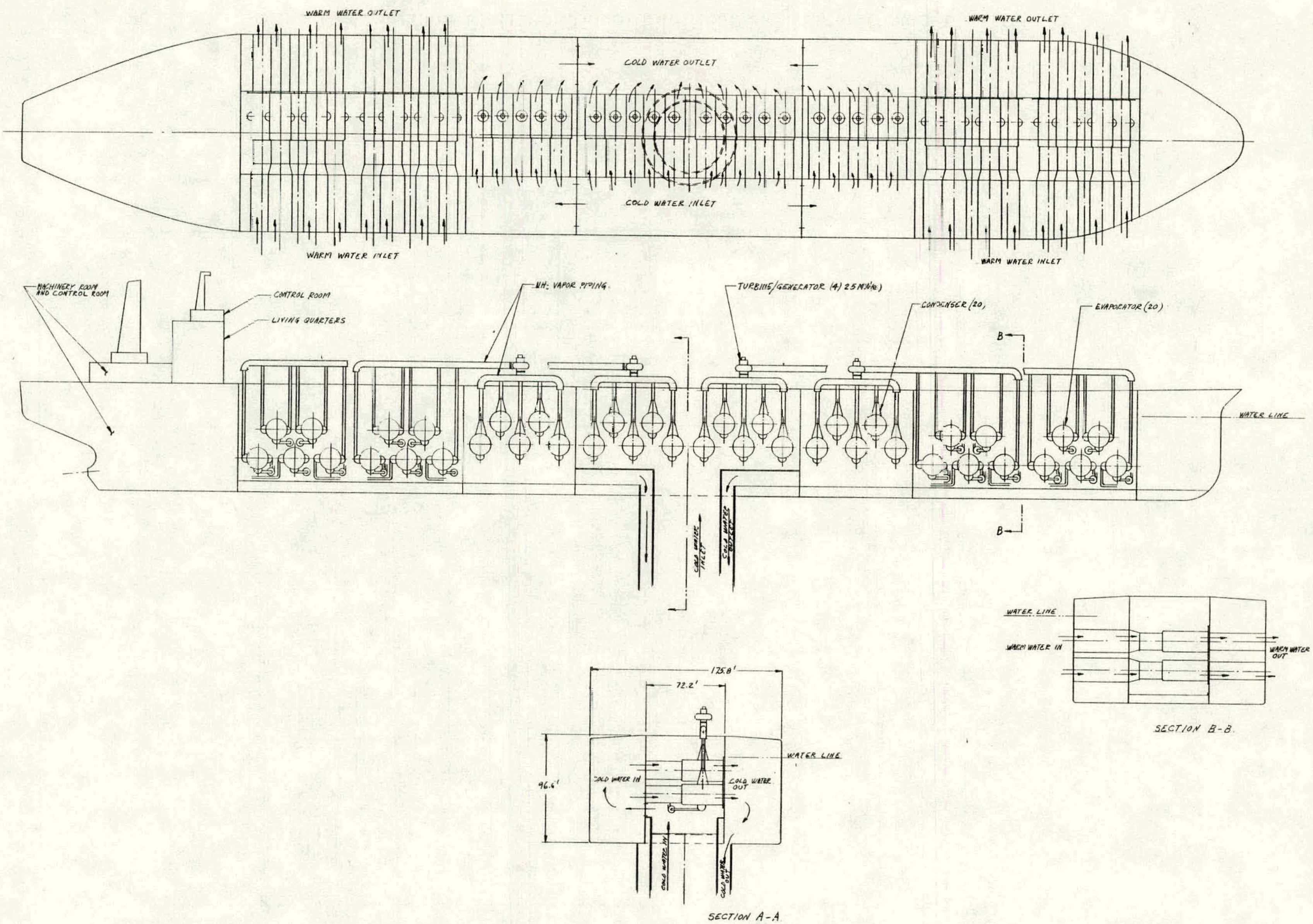

Fig. B-8 Supertanker Twenty 5-MW(e) Heat Exchanger Concept

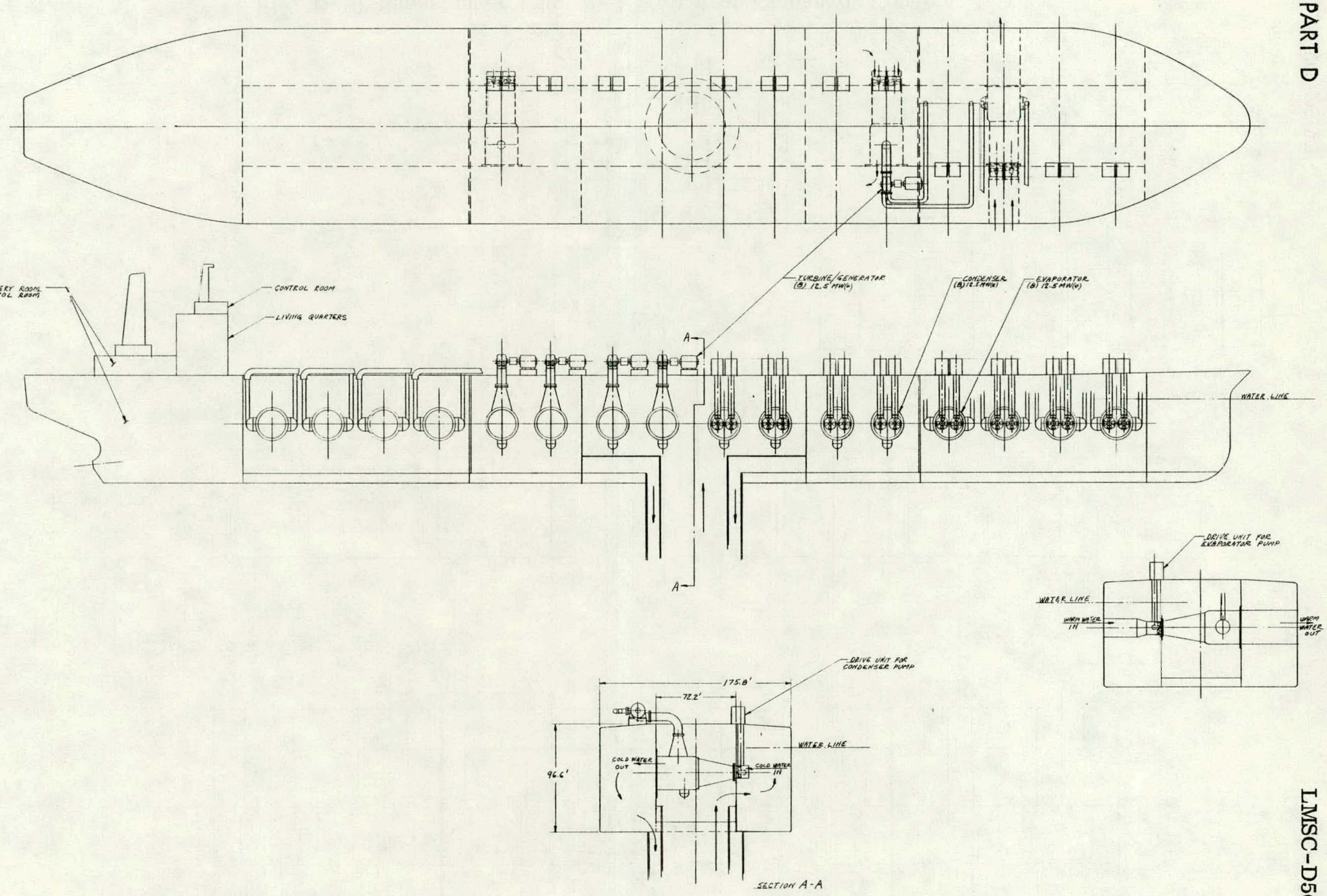

Fig. B-9 Supertanker Eight 12.5-MW(e) Heat Exchanger Concept

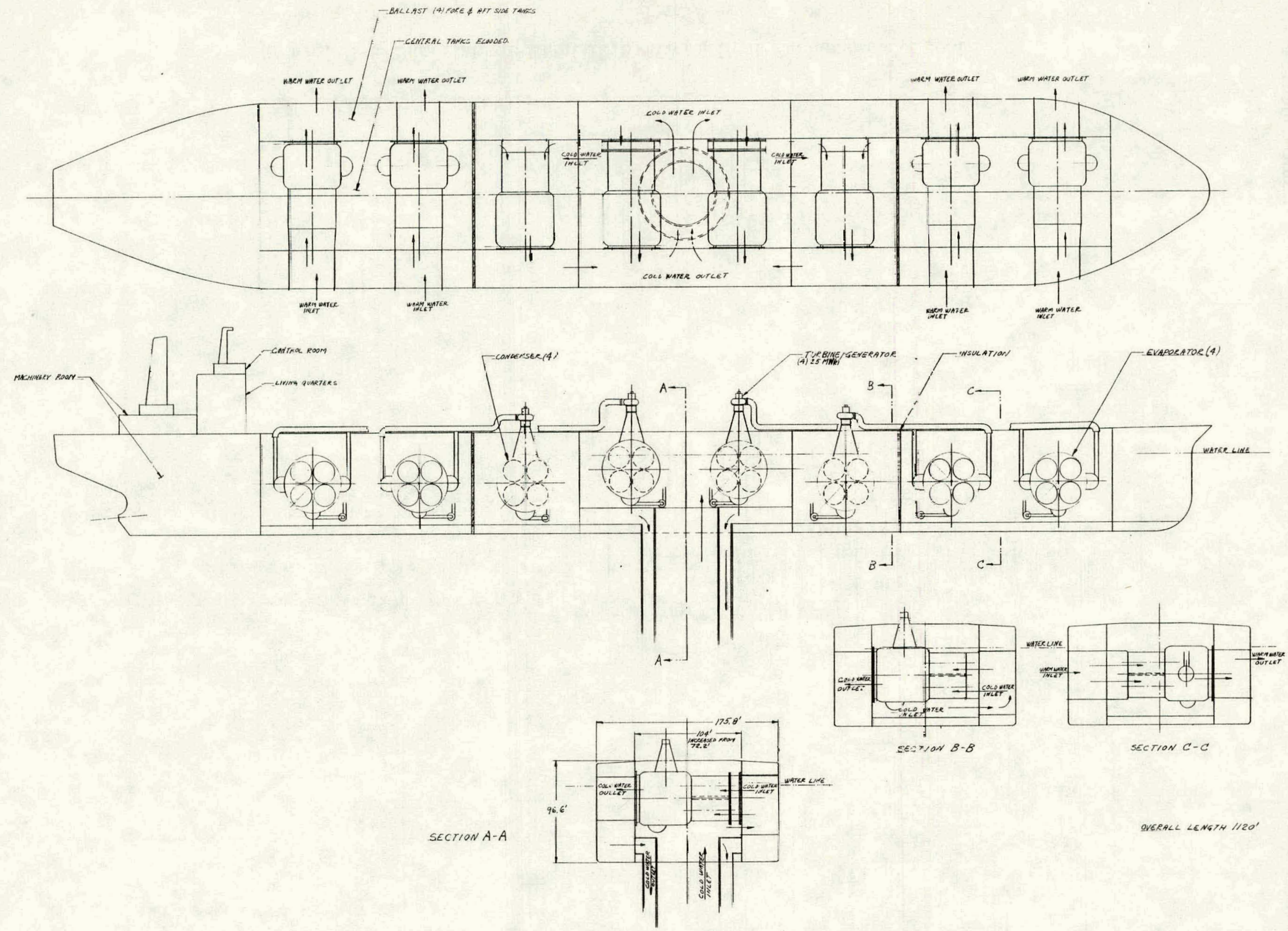

Fig. B-10 Supertanker Four 25-MW(e) Heat Exchanger Concept

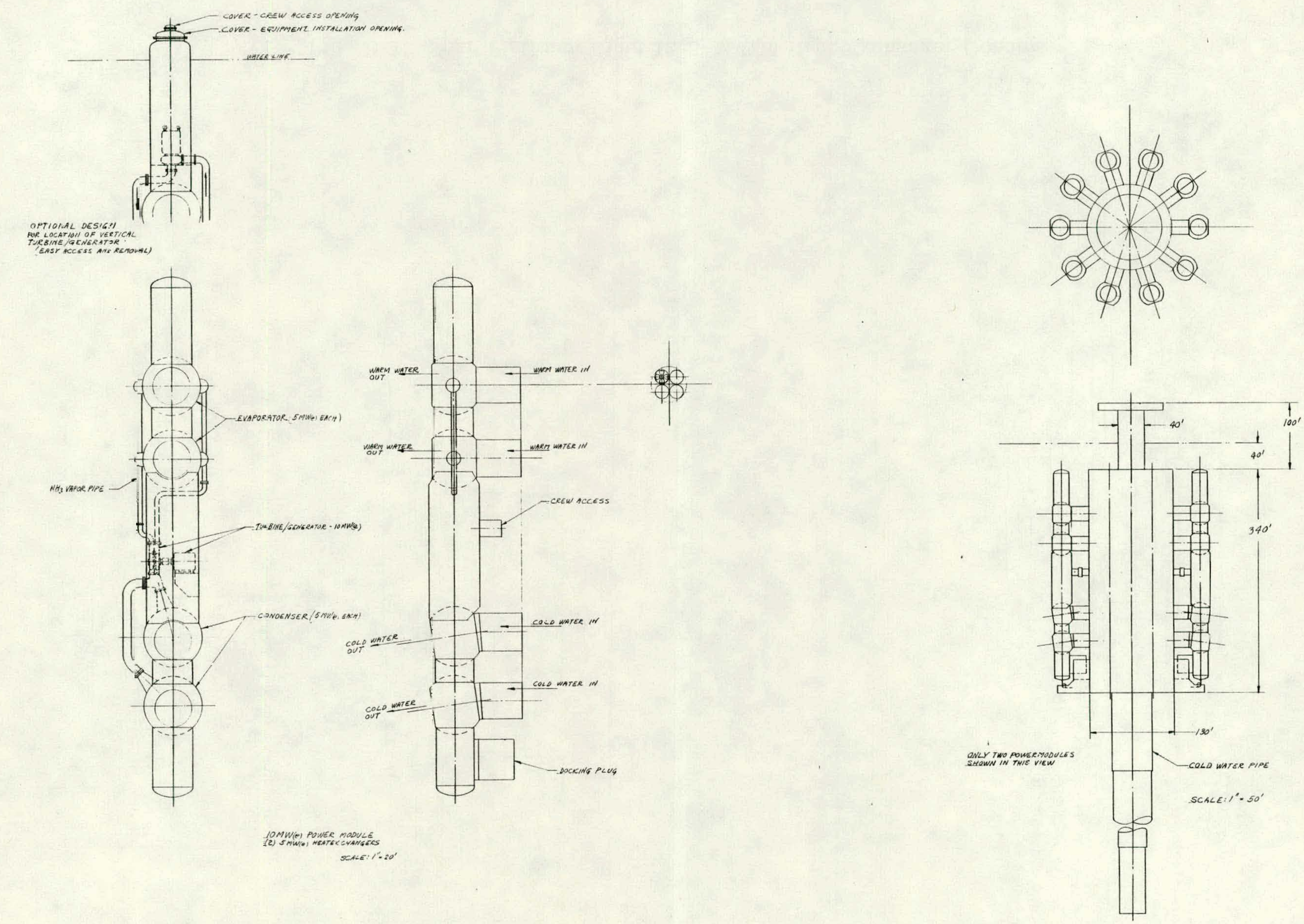

Fig. B-11 Spar Platform Twenty 5-MW(e) Heat Exchanger Concept

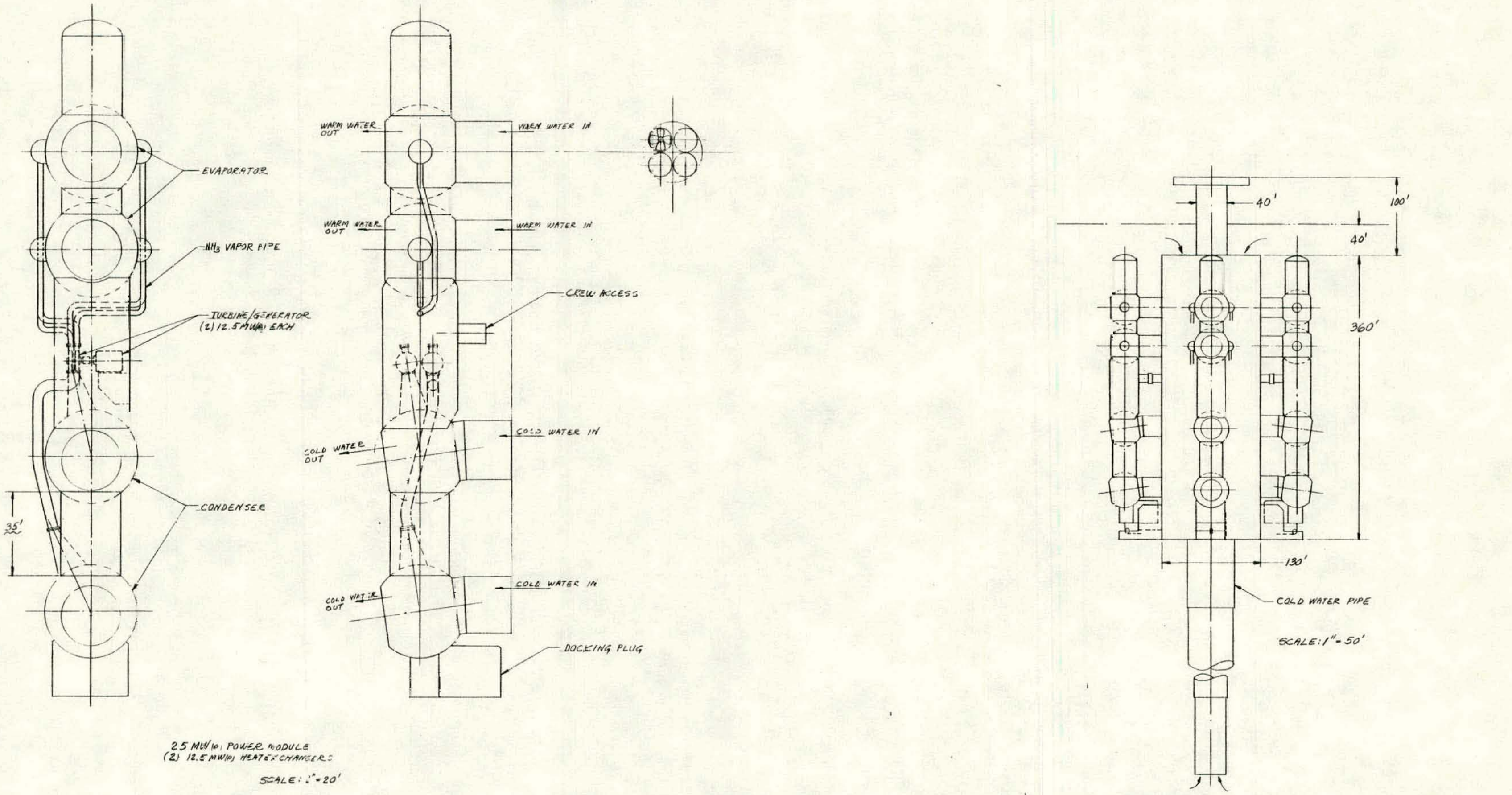

Fig. B-12 Spar Platform Eight 12.5-MW(e) Heat Exchanger Concept

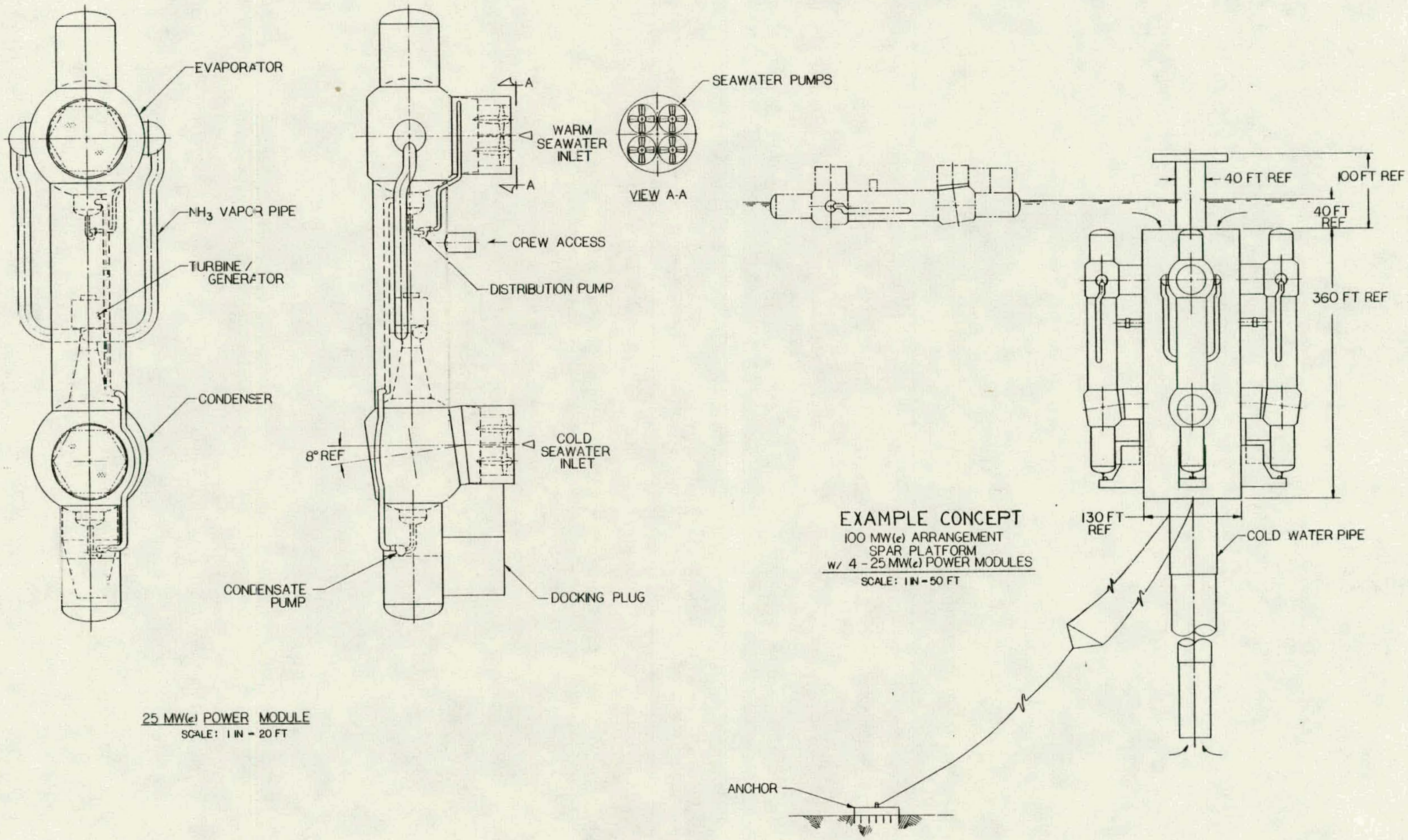

Fig. B-13 Spar Platform Four 25-MW(e) Heat Exchanger Concept

Appendix C
SYSTEM EQUIPMENT

This appendix presents a listing of the major components required for operation of the power system including auxiliary system components which are shared with or located on the platform. As indicated, some pertinent data will be determined during the Preliminary Design task.

Table C-1

PLATFORM AND POWER SYSTEM MAJOR EQUIPMENT

Component	Description	Number	Dimensions	Weight (lb)	Comments
Evaporator	Aluminum Sperical Shell w/1.500 in. OD Aluminum Tubing	1	63.5 ft OD Effective Tube Length 44 ft	1,656,000	Surface/Baseline Platform 0.102 in. Tube Wall Thickness
			59 ft OD Effective Tube Length 38 ft	1,297,500 (Dry) (2,998,000)	Submerged Configuration 0.088 in. Tube Wall Thickness (Positive Buoyancy Fully Submerged)
Condenser	Aluminum Sperical Shell w/1.500 in. OD Aluminum Tubing	1	63.5 ft OD Effective Tube Length 44 ft	1,656,000	Surface/Baseline Platform 0.102 in. Tube Wall Thickness
			59 ft OD Effective Tube Length 38 ft	1,297,500 (Dry)	Submerged Configuration 0.088-in. Tube Wall Thickness (Positive Buoyancy Fully Submerged)
Turbine	Rotoflow Radial Inflow Turbine	2	7.5 ft × 11.25 ft × 9.7 ft	75,000 (Each) 30,000 (Each)	Steel Aluminum
	or				
	General Electric Axial Flow Turbine	1	27 ft × 14 ft × 14 ft	200,000	Steel
Generator	Rotoflow, 13,900 Volts 1800 Amps, 60 Hz	1	20.8 ft × 11.1 ft × 11.1 ft	179,000	37-MW(e) Generator, Self-Contained Cooling and Lubrication System
	or				
	General Electric 13,900 Volts, 1800 Amps, 60 Hz	1	19 ft × 12 ft × 12 ft	203,000	37-MW(e) Generator, Self-Contained Cooling and Lubrication System
Condensate Pumps	Two-Stage Centrifugal	3	23.5 ft Length × 6 ft Width Plus Motor	23,000 (Each)	Baseline Platform 3,800,000 lb/hr at 52.1 psi, 500-hp Motor Submerged Platform 3,360,000 lb/hr at 110 psi, 825-hp Motor
Distribution Pumps	Two-Stage Centrifugal	4	23.5 ft Length × 6 ft Width Plus Motor	20,000 (Each)	Baseline Platform 3,800,000 lb/hr at 15 psi, 100-hp Motor Submerged Platform 3,360,000 lb/hr at 15 psi. 100 hp Motor

Table C-1 (Cont.)

Component	Description	Number	Dimensions	Weight (lb)	Comments
Seawater Pumps	Single-Stage, Axial Flow	8	11 ft Diam. Impeller Inlet/Outlet 16 ft Diam. Overall Length, 45 ft	260,000 (Each)	875-hp Motor Each Submerged ac Motor w/ Speed Control by Voltage Regulation Baseline Platform Cold Water Pumps 1,050-hp Motors (4)
Ammonia Cleanup System	Vertical Pressure Vessel w/Pump	1	Column 3 ft Diam. 30 ft Height	11,500	Platform Mounted, 1.46-MW(e) Power Required, 100-MW(e) System
Ammonia Storage System	Storage Tank w/Pump and Compressor	1	Storage Tank 55,000 cu ft Miscellaneous Equip.	2,000,000 35,000	Platform Mounted 100-MW(e) System
Overpressure Vent	Safety Relief Valve	2	6-in. Pipe Size		
Noncondensible Gas Removal System	Secondary Condenser and Compressor With Vent	1	TBD		
Standby Power System	Diesel-Generators, 1-MW(e) Power Total	2		51.000 (Each)	Platform Mounted 100-MW(e) System
Electrical Power Distribution System	See Appendix F-1				
Equipment Cooling	Seawater-Fresh Water Cooling System			20,000	Platform Mounted
Nitrogen Purge	40 Nitrogen Bottles Plus Piping and Pressure Reducing Valve			8,000	Platform Mounted
Fouling Countermeasures					

Appendix D
AMMONIA SYSTEM MATERIAL CONSIDERATIONS

This appendix contains the results of a study made by Bechtel National, Inc., on the compatibility of materials with the ammonia system. A survey of land-based and shipboard installations was undertaken and the results are included.

CONTENTS

Introduction

Bechtel was requested to provide a brief review of the materials being proposed by Lockheed Missiles and Space Company (LMSC) for the Ocean Thermal Energy Conversion (OTEC) unit.

Conclusions

In ammonia vapor and liquid ammonia systems, carbon steel and cast iron are generally suitable for anhydrous ammonia service. However, some uncertainties exist with air contaminated anhydrous ammonia and seawater contaminated ammonia. In order to prevent stress corrosion cracking of the carbon steel components, we recommend that the welds and parts containing high residual stresses be heat treated to relieve those stresses. Aluminum alloys are also suitable for anhydrous ammonia service. It is recommended that test programs be conducted to determine the effect of seawater inleakage on material selection, including consideration of aluminum and steel separately and aluminum-steel couples. Insufficient piping service experience is presently available to allow us to provide an estimate of whether aluminum alloys will provide the required reliability in the seawater systems. Very little corrosion rate data directly applicable to the OTEC system are available. A test program which uses the proposed alloys and service conditions in the proposed locations is recommended. Alclad 5052 alloy should be considered. In order to minimize intergranular attack, the 5083 alloy components should be specified with an H116 or H117 temper.

Materials Proposed by LMSC (25 MWe system)

Condenser:

Shell	5083-H321 aluminum alloy
Tubes	5052 H32 aluminum alloy (2½ in. dia. x 0.058 in. wall)
Tubesheet	5083-H321 aluminum alloy
Tube supports	aluminum

Evaporator:

Shell	5083-H321 aluminum alloy
Tubes	5052-H32 aluminum alloy 2½ in. x 0.058 in. wall
Tubesheet	5083-H321 alloy

Service Conditions (25 MWe system)

Condenser:

Design life	30 years
Design temperature	40-100 F

	Shell side	Tube side
Fluid circulated	ammonia	seawater
Temperature, in	52.81 F	43.38 F
Temperature, out	52.45 F	45.91 F
Operating pressure	79.39 psig	
Velocity		5 ft/sec

Evaporator:

Design life	30 years
Design temperature	40-100 F

	Shell side	Tube side
Fluid circulated	ammonia	seawater
Temperature, in	52.50 F	79.75 F
Temperature, out	70.18 F	76.84 F
Operating pressure	124.7 psig	
Velocity		4.5 ft/sec

Discussion

1. Ammonia vapor and liquid ammonia systems:

In ammonia vapor and liquid ammonia systems, carbon steel and cast iron are generally suitable for anhydrous ammonia service. However, some uncertainties exist with air contaminated anhydrous ammonia and seawater contaminated ammonia.

Carbon steel is subject to stress corrosion cracking in air-contaminated anhydrous ammonia. It is not likely that the main cycle piping and components will be subjected to air contamination except during initial startup, shutdown for maintenance and restart. Air and nitrogen should be purged from the system by the ammonia through the condenser non-condensable vents. However, this stress corrosion cracking can be prevented by post weld heat treatment and thermally stress relieving (at 1100F minimum) parts containing high residual fabrication stresses. Another method used in ammonia systems is to add 0.2% water to the ammonia. However, for the OTEC system the addition of seawater to the system through leakages is considered a liability for thermodynamic reasons and is minimized by a low leakage condenser design. Hence post weld heat treatment and thermal stress relieving is recommended for carbon steel piping.

Seawater concentrations in the ammonia system can be expected to vary from very low levels in the vapor and condensate pipe and the condenser to as high as 5% in the evaporator. The corrosion and deposition effects caused by this seawater content are not clearly understood at this time. It is recommended that test programs be conducted to determine the effect of seawater inleakage on material selection.

Aluminum alloys are also suitable for ammonia service, common application being in refrigeration systems and storage tanks (ref. 1). Galvanic attack between steel and aluminum in anhydrous ammonia would not be expected since the conductivity of anhydrous ammonia is less than 1×10^{-6} mhos/cm (values as low as 1×10^{-11} mhos/cm have been reported (ref. 21)) which is less than one-hundreth that of demineralized water. In laboratory tests, it has been shown that if anhydrous ammonia is contaminated with air, pitting attack of aluminum coupled to steel is promoted (ref. 25). These same tests indicate that water additions may reduce the galvanic attack. However, as discussed above, water additions are detrimental to the operation of the OTEC cycle and, in fact, seawater contamination may promote galvanic attack. Any seawater inleakage test program should therefore consider corrosion effects on aluminum-steel couples.

In summary, aluminum and steel are suitable for anhydrous ammonia service provided the ammonia is not contaminated with air or seawater. Air contamination is expected to be minimal with the system purging all non-condensibles after startup. Seawater inleakage will be handled by the ammonia cleanup system. Concentrations of seawater should be extremely small throughout the system except in the evaporator and distribution pumps, piping and valves. The baseline conceptual design approach is therefore to use carbon steel piping together with the aluminum heat exchangers. Aluminum piping is still being considered as a more expensive alternate. Fiberglass pipe was considered and rejected because commercially available resins cannot withstand ammonia.

In order to withstand galvanic corrosion on the seawater side, a suitable cathodic protection system and electrical isolation system design will be incorporated into the preliminary design using coated carbon steel pipe. If this proves to be too unwieldy or expensive, the aluminum piping option will be exercised. The results from any test programs concerning the effects of seawater and/or air contamination will also be factored into the preliminary design materials selection effort.

2. Seawater systems:

Aluminum alloys have been proposed by LMSC for the evaporator and condenser in the seawater system - 5052-H32 for the tubes and 5083-H321 for the shell and tubesheet. Piping and pump materials apparently have not yet been proposed.

Since the design life of the system is thirty years, retubing during the lifetime may make the OTEC system uneconomical (ref. 2). Furthermore, seawater contamination of the ammonia adversely affects its fluid characteristics and ultimately results in a significant power loss. Inleakage also necessitates the operation of the ammonia cleanup system which may consume a significant percentage of the available OTEC power. In short, very high reliability is required in the heat exchangers.

It is possible that aluminum alloys, and particularly aluminum alloy tubes that have not been aluminum clad (alclad) will not provide sufficient reliability. Recent reports have indicated that aluminum alloys may be suitable in desalination plants (refs. 3-7). However, the seawater used in desalination plants is adjusted to a nonscaling condition (pH adjustment) and the oxygen generally is controlled between 10 and 50 ppb. This water treatment can have a profound effect on the corrosion resistance of aluminum. A slightly acidic pH of 6.0 to 6.5 appears to be the optimum for aluminum when all factors of pitting and general corrosion are considered (ref. 6). Reference 6 also states that the presence of 100 ppb or more dissolved oxygen in <u>concentrated</u> seawater brine increases the pitting and general corrosion susceptibility of aluminum. Conversely, deep water immersion tests indicate a lower rate of penetration should be expected as the oxygen concentration is increased, e.g. from 0.5 to 5 ppm (ref. 10 and Figure 1). Other reports indicate that pitting is not of concern at pH 7 and above (ref. 20). It should be noted that aluminum alloys are not being used in commercial desalination plants.

Much of the aluminum in seawater corrosion data is based on simple immersion tests. Since fouling does occur on the samples during these tests and since there is little or no velocity component, these test results are not directly relevant to the OTEC system environment. Table I summarizes corrosion rate data for 5052, 5083 and similar aluminum alloys. Uniform corrosion rates were generally less than 0.5 mpy and decreased with time. Maximum pit depths ranged up to 55 mils after 10 years for alloy 5083, 29 mils after 1 year for alloy 5052 and 131 mils after 10 years for

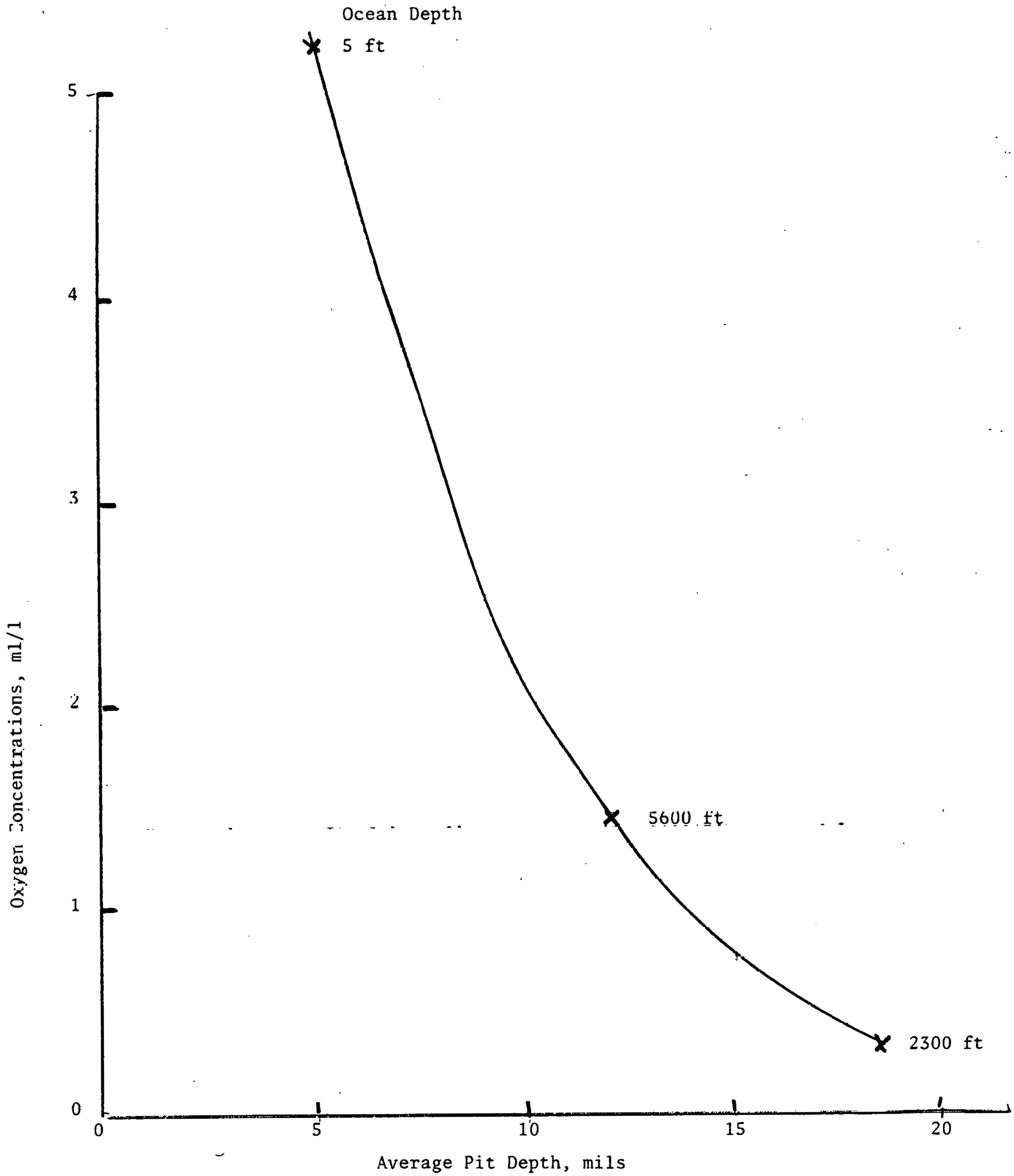

Figure 1. Relationship between Average Pit Depths of Aluminum-Magnesium Alloys (5000 series) and Oxygen Concentration of Seawater (ref. 13).

Ta I

Corrosion Rates of Aluminum Alloys
Immersed in Seawater

Alloy	Time of Exposure	Location	Pit Depth, mils Max.	Pit Depth, mils Avg.	Uniform Corrosion, mils	Max. Total Corrosion, mils	Reference
5083-0	1 year	Wrightsville Beach, NC	20.9	8.2	0.17	21.1	8
5083-0	2 years	Wrightsville Beach, NC	18.9	5.4	0.24	19.1	8
5083-0	5 years	Wrightsville Beach, NC	7.0	1.0	0.33	7.3	8
5083-0	10 years	Wrightsville Beach, NC	24.0	1.0	0.59	24.6	8
5456-0	10 years	Wrightsville Beach, NC	131.0	39.9	0.35	131.4	8
5083	1 year	Harbor Island, NC	16.0		0.4	16.0	11
5083	2 years	Harbor Island, NC	13.0		0.2	13.0	11
5083	5 years	Harbor Island, NC	6.0		0.4	6.0	11
5083	10 years	Harbor Island, NC	10.0		0.6	11.0	11
5083	1 year	Halifax, N.S.	4.0		0.2	4.0	11
5083	2 years	Halifax, N.S.	23.0		0.2	23.0	11
5083	5 years	Halifax, N.S.	16.0		0.4	16.0	11
5083	10 years	Halifax, N.S.	22.0		0.6	23.0	11
5083	1 year	Esquimalt, B.C.	29.0		1.2	30.0	11
5083	2 years	Esquimalt, B.C.	38.0		1.2	39.0	11
5083	5 years	Esquimalt, B.C.	47.0		2.5	50.0	11
5083	10 years	Esquimalt, B.C.	55.0		2.3	57.0	11
5083	10 years	Brixham, England	34.0				11
5083-H113	1.6 years	Pt. Mugu, CA	11.0	7.0			12
5083-H323	1 year	Miami, FL		0.3			9
5083-H323	2 years	Miami, FL	2.0	2.0			9
5052	5 years	Brixham, England	27.0				11
5052	10 years	Brixham, England	17.0				11
5052-H32	368 days	Key West, FL	<1.0				10

Table I (continued)

Alloy	Time of Exposure	Location	Pit Depth, mils Max.	Pit Depth, mils Avg.	Uniform Corrosion, mils	Max. Total Corrosion, mils	Reference
5052-0	.5 years	Port Hueneme, CA	Incipient	---	0.6	1.2	12, 13, 26
5052-0	1 year	Port Hueneme, CA	5.0	---	0.6	5.6	12, 13, 26
5052-H34	1 year	Harbor Island, NC	0	---	0.34	---	25
5052-H34	2 years	Harbor Island, NC	0	---	0.48	---	25
5052-H34	5 years	Harbor Island, NC	0	---	0.70	---	25
5052-H34	10 years	Harbor Island, NC	0	---	1.10	---	25
5052-H34	1 year	Halifax, N.S.	5.0	---	0.21	5.2	25
5052-H34	2 years	Halifax, N.S.	20.0	---	0.24	20.2	25
5052-H34	5 years	Halifax, N.S.	6.0	---	---	---	25
5052-H34	10 years	Halifax, N.S.	12.0	---	1.10	13.1	25
5052-H34	1 year	Esquimalt, B.C.	16.0	---	0.12	16.1	25
5052-H34	2 years	Esquimalt, B.C.	6.0	---	0	6.0	25
5052-H34	5 years	Esquimalt, B.C.	0	---	0	0	25
5052-H34	10 years	Esquimalt, B.C.	5	---	0	5.0	25
5052	1 year	Freeport, TX	3	---	0.29	3.3	25
5052	2 years	Cape Beale, B.C.	0	---	0.22	0.2	25
5052	1 year	Galiano	0	---	0.11	0.1	25
5052-H34	8.3 yrs.	Pensacola, FL	1.4	0.6	---	---	25
5052-H34	2 years	Corpus Christi, TX	14	---	1.8	15.8	25
5052-H34	1 year	Miami, FL	29	11.6	---	---	17
5052	1 year	Miami, FL	14.5	6.5	---	---	17
5052	1 year	Miami, FL	2.9	0.9	---	---	17
5052-0	2 years	Miami, FL	1.3	0.7	---	---	17
5052-0	2 years	Miami, FL	2.5	1.2	---	---	17
5052-0	3 years	Miami, FL	1.2	0.8	---	---	17
5052-0	3 years	Miami, FL	3.4	0.9	---	---	17
5052-0	5 years	Miami, FL	1.4	0.8	---	---	17
5052-0	5 years	Miami, FL	8.4	1.2	---	---	17

Table I (continued)

Alloy	Time of Exposure	Location	Pit Depth, mils Max.	Pit Depth, mils Avg.	Uniform Corrosion, mils	Max. Total Corrosion, mils	Reference
3003 alclad	2 years	Harbor Island, NC	12.0		0.9	13.0	11
3003 alclad	2 years	Halifax, N.S.	13.0		0.2	13.0	11
3003 alclad	2 years	Esquimalt, B.C.	13.0		0.1	13.0	1
Alclad	14 years	Los Angeles, CA	91 mils*				14

*Seawater cooled surface condenser at Union Oil Refinery contained 0.083 inch aluminum tubes clad with 8 mils Al-clad that were installed in 1960. To date, only one leak has been recorded and that was in 1974.

alloy 5456. All rates decreased with time. If these data are representative, it is possible the tube walls would experience a few perforations within ten to fifteen years. Corrosion rates in the OTEC system may differ from those in Table I because of velocity, oxygen concentration, tube wall condition and tube cleaning methods. Note the generally higher pitting rates for alloy 5052 immersed in deep seawater (Table II) compared to those immersed in surface seawater.

The proposed 5 to 6 ft/sec tube velocity should lead to lower pitting rates than those obtained in stagnant or low velocity seawater. The oxygen concentration can significantly affect the pit depth as shown in Figure 1. Fouling products were allowed to occur on the immersed samples and, depending on the nature of product, these may have enhanced or hindered pitting rates. (Note the tube perforations which occurred under barnacles reported in Table IV). Clean tubes must be maintained in the OTEC system. The various methods of maintaining clean tubes, Amertap, Mann system, etc., may affect the protective aluminum oxide film and thus the corrosion rates.

Due to the above uncertainties, we recommend that a corrosion test program be performed on the aluminum alloys of interest under the proposed service conditions. Ideally, the tests should be performed at the proposed OTEC system locations and should study the effect of velocity, tube cleaning methods, etc. Since the pitting rates in aluminum alloys do decrease significantly with time, a one to two year test program should provide a conservative estimate of material performance.

The consensus of several cognizant industry representatives was that alclad aluminum alloys should be used for the tubing material (refs. 15-17). A seawater cooled steam surface condenser has an excellent service record at Union Oil's Los Angeles refinery (ref. 14). Alclad aluminum tubes were installed in this heat exchanger in 1960. In 1974, one leaking tube was removed. Four other tubes were removed for inspection at that time. Isolated, shallow (0.001 inch) pits were observed in the cladding and no attack was observed on the base metal. Alloy 3003 tubes clad with alloy 7072 have also provided ten years of satisfactory service in the seawater cooled lube oil cooler on the freighter, Alcoa Clipper (ref. 20). Data in Tables III and V through also indicate that very good performance can be obtained

Ta II

Corrosion Rates of Aluminum Alloys
Immersed in Deep Seawater (ref. 25)

Alloy	Depth, ft.	Time (years)	Uniform Corrosion Rate, mpy	Max. Pit Depth (mils)	Avg. Pit Depth (mils)	Crevice Depth (mils)	Location	Reference
5052-0	2340	0.5	1.8	0	----	65 (PR)	Port Hueneme, CA	12, 13, 26
5052-H32	2340	0.5	----	Uniform	----	0		
5052-H34	2340	0.5	<0.1	12	----	0		
5052-H32	2370	0.5	0.09	26	----	-------	Port Hueneme, CA	28
5052-0	2370	1.0	0.4	0	----	20		
5052-H32	2370	1.0	----	Few	----	0		
5052-H34	2370	1.0	0.2	Incip.	----	34		
5052	2370	1.0	0.2	12	8.0	-------	Port Hueneme, CA	29
5052-H32	4200	1.0	<10	<10	----	-------	Tongue-of-the-Ocean, Bahamas	30
5052-H32	4200	1.0	<10	<10	----	-------		
5052-H32	4200	1.0	<10	<10	----	-------		
5052-H32	4200	1.0	<10	<10	----	-------		
5052-0	5300	3.0	3.1	0	----	65 (PR)	Port Hueneme, CA	12, 13, 26
5052-0	5640	0.3	3.7	0	----	65 (PR)		
5052-0	5640	0.3	<0.1	Severe	----	0		
5052-H34	5640	2.0	2.2	65 (PR)	----	65 (PR)		
5052-H34	5640	2.0	----	-------	----	0		
5052-0	6780	1.0	4.5	0	----	62 (PR)		
5052-H32	6780	1.0	----	39	----	0		

Table II (continued)

Alloy	Depth, ft.	Time (years)	Uniform Corrosion Rate, mpy	Max. Pit Depth (mils)	Avg. Pit Depth (mils)	Crevice Depth (mils)	% Clad Consumed	Location	Ref.
Alc. 3003-H12	2340	0.5	2.2	15	13.1	13	Large area	Port Hueneme, CA	12,13,26
Alc. 3003	2340	0.5	2.3	0	----	0	General	Port Hueneme, CA	12,13,26
Alc. 3003-H12	2370	1.0	2.2	14	12.9	15	20%	Port Hueneme, CA	12,13,26
Alc. 3003	2370	1.0	1.6	0	----	0	60%	Port Hueneme, CA	12,1[illegible]6
Alc. 3003-H12	5300	3.0	0.7	20	16.5	13	18%	Port Hueneme, CA	12,13,26
Alc. 3003-H12	5300	3.0	0.3	15	14.8	--	Large area	Port Hueneme, CA	12,13,26
Alc. 3003	5300	3.0	1.5	0	----	0	General	Port Hueneme, CA	12,13,26
Alc. 3003-H12	5640	0.3	0.2	18	14.6	15	Large area	Port Hueneme, CA	12,13,26
Alc. 3003-H14	5640	0.3	---	Incip.	----	0	----------	Port Hueneme, CA	12,13,26
Alc. 3003	5640	0.3	2.7	0	----	0	General	Port Hueneme, CA	12,13,26
Alc. 3003-H12	5640	2.0	0.3	13	12.8	13	18%	Port Hueneme, CA	12,13,26
Alc. 3003-H14	5640	2.0	---	0	----	0	General	Port Hueneme, CA	12,13,26
Alc. 3003	5640	2.0	1.4	0	----	0	General	Port Hueneme, CA	12,13,26
Alc. 3003-H12	6780	1.0	0.4	13	13.0	14	----------	Port Hueneme, CA	12,13,26
Alc. 3003	6780	1.0	2.5	0	----	0	General	Port Hueneme, CA	12,13,26

TAB III

S. S. ALCOA CLIPPER
SHIPBOARD HEAT EXCHANGER - 3/4" OD x .064" WALL
ALCLAD (INSIDE 7072) 3003-H14 (ref. 27)

Original Cladding Thickness(1) %	Original Cladding Thickness(1) Inch	Shell	Tubes	Years	Cladding Consumed %	Maximum Remaining Cladding Thickness Inch	Maximum Penetration of Core - Inch
16	.0104	Lube Oil	Seawater	1	-	.0100	Nondetected
16	.0104	Lube Oil	Seawater	1	5	----	Nondetected
16	.0104	Lube Oil	Seawater	2.7	10	----	Nondetected
16	.0104	Lube Oil	Seawater	4	-	.0100	Nondetected
16	.0104	Lube Oil	Seawater	4	-	.0070	Nondetected
16	.0104	Lube Oil	Seawater	5	50	.0051	Nondetected
16	.0104	Lube Oil	Seawater	7	53	.0107	Nondetected
8	.0052	Lube Oil	Seawater	8	90	.0050	.022
16	.0104	Lube Oil	Seawater	9	50	.0103	Nondetected
16	.0104	Lube Oil	Seawater	10	76	.0102	Nondetected

Note: (1) Nominal thickness: % is of tube wall thickness.

TABLE IV

OSW - TEST BED PLANT, FREEPORT, TEXAS
HEAT REJECT EXCHANGER E-21
RAW SEA WATER AT 110°F IN TUBES (ref. 27)

Tube Alloy	Months Tested	Max. Pit Depth Inches	Avg. Pit Depth Inches	Pits/In.^{2}	Remarks
3003	6	0	0	0	Excellent condition
5052	6	.049	.030	.01	Perforation and pits under barnacles
6063	6	.049	.030	.01	Perforation and pits under barnacles
5050	6	.049	.030	.01	Perforation and pits under barnacles
3003	12	0	0	0	Excellent condition
5052	12	.020	-----	-----	One deep pit
6063	12	0	0	0	Excellent condition
5050	12	.049	.020	.01	Perforation and pits under barnacles

Source: Desalination Materials Manual prepared by Dow Chemical Company for OSW, 1975-May. On page 4-108 of the data source it is reported that 4 tubes removed after the first 6 months showed no pitting. Pitting that developed on the replacement tubes and the original tubes occurred during the last 6 months of test.

TABLE V

BALTIMORE 30, MARYLAND
CONDENSER WITH 3/4" OD x .065" WALL ALCLAD (INSIDE 7072) 3003-H14 TUBES (ref. 27)

Service			Cladding Consumed	Maximum Measured Penetration
Shell	Tube	Years	%	Inches
Steam	Baltimore Harbor Water	2	16	.006
		4	11	.008
		6	29	.009
		8	66	.009

TABLE VI

FREEPORT TEXAS, CONDENSER, ALCLAD (INSIDE 7072) 3003 AND NONCLAD 3003 TUBES 3/4" OD x .065" WALL (ref. 27)

Service				
Shell	Tubes	Years	Tube Alloy	Maximum Measured Penetration Inch
Steam	Seawater	2	Alc. 3003	Cladding Not Perforated
Steam	Seawater	2	3003	.0038

TABLE VII

TEXAS CITY, TEXAS
CONDENSER - ALCLAD (INSIDE 7072) 3003 AND NONCLAD 3003 TUBES
3/4" OD x .065" WALL (ref. 27)

Service Shell	Service Tubes	Years	Tube Alloy	Remarks
Steam	Seawater	1 1/4	Alc. 3003	Scattered pits limited to cladding
			3003	Scattered small pits
		2 1/2	Alc. 3003	Broad shallow pits - cladding protecting
			3003	Scattered small pits deeper than at 1 1/4 years

TABLE VIII

ALUMINUM DESALINATION UNIT ABOARD THE ALCOA SEA PROBE
CONDENSER AND EVAPORATOR TUBED WITH 5/8" OD x .065" WALL ALCLAD (BOTH SIDES 7072)
3003-H14 TUBES (ref. 28)

Condenser

Shell 5454-H34 aluminum
Tubes 5/8" CD x .065" wall alclad (both sides 7072) 3003-H14
Water box - steel coated internally with Debecate
Service
  Raw sea water in tubes (small barnacles formed)
  Condensate in shell
  7 years - No problems due to corrosion

Evaporator

Shell 5454-H34 aluminum
Tubes 5/8" CD x .065" wall alclad (both sides 7072) 3003-H14
Tube sheet - Alclad (both sides 7072) 6061-T6, 1" thick
Water box - steel coated internally with Debecate
Service
  Engine jacket water in tubes
  Hot sea water evaporating in shell
  7 years - no corrosion problem

TA IX

LUBE OIL COOLER - PORT ARTHUR, TEXAS (ref. 27)

Tubes - Alclad (Inside 7072) 3003-H14

Coolant - Brackish Water in Tubes

Service - After 5-6 years about 50% of cladding consumed, but no penetration of the 3003 core. The cooler remained in service.

TABLE X

AMMONIA CONDENSER - LAKE CHARLES, LA (ref. 27)

Tubes - Alclad (Inside 7072) 3003-H14

Coolant - Brackish water in tubes

Service - No examination or trouble reported in 4 years. Condenser remained in service.

TAI XI

ALCLAD ALUMINUM TUBES IN BALTIC SEA WATER AT STUDSVIK, AB ATOMENERGI RESEARCH STATION (ref. 27)

Composition of Baltic Sea Water at Studsvik		Tube Alloys	Remarks
Chloride	3970 mg/l	SIS 4212, Al Si 1 Mg (6351)	10% cladding removed by
Sulfate	550 mg/l	Clad inside with Al-Zn 1 (7072)	erosion - corrosion. Pene-
Calcium	72 mg/l		tration no deeper than
Sodium	2450 mg/l	SIS 4212, Al Si 1 Mg (6351)	140 m (.0055")
Magnesium	79 mg/l	Clad inside with Al 99.99	50-60% cladding removed by
Oxygen	8-14 mg/l		erosion-corrosion. Penetra-
Hydrogen Sulfide	< 0.1 mg/l	Time 10,000 hrs. (13.7 mos.)	tion no deeper than 200 m
Ammonia	Not Detected		(.0079")
pH	7.8	Velocity 2.5 m/sec (8.2 ft/sec.)	
		Temperature 50°C (122°F)	

Concluded that at 50°C (122°F) velocity of 2.5 m/sec (8.2 ft/sec) too high for aluminum tubes tested. Further stated that heat exchangers containing aluminum tubes clad inside with Al Zn 1 (7072) have operated problem-free in "Studsvik-water" at a maximum temperature of 20°C (68°F) and water velocities of less than 2.5 m/sec (8.2 ft/sec) for more than 15 years.

Ref: "Corrosion Tests in Baltic Seawater on Heat Exchanger Tubes of Various Metallic Materials", Henrikson and Knutsson.

with alclad tubes. On the other hand, other alclad tubes at Union Oil failed in three months possibly because the cladding had not been applied properly (refs. 18, 19).

Alloy 7072 cladding was developed primarily for application in natural waters. Alcoa test work has shown that this alloy may be too active in seawater, particularly for deep sea applications (ref. 25). We concur with their recommendation that a need exists to develop an alclad aluminum alloy especially for deep sea exposures.

We also suggest that the H116 or H117 temper be considered for the 5083 tubesheets and shells to prevent intergranular corrosion in the weld areas (ref. 17). Furthermore, if the seawater is contaminated with heavy metal ions, such as copper, aluminum alloys will fail rapidly.

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Appendix E
AMMONIA CYCLE

This appendix includes supplemental details on the components incorporated into the ammonia cycle subsystems. These details are supplied in three separate sections, as follows:

E-1 Ammonia Vapor and Condensate Systems
E-2 Ammonia Cleanup System
E-3 Ammonia Storage, Fill, and Purge System

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APPENDIX E-1

AMMONIA VAPOR AND CONDENSATE SYSTEMS

PURPOSE

The purpose of the ammonia vapor and condensate systems is to provide the necessary piping, valving and pumps to interconnect the major components of the OTEC power cycle, i.e. the condenser, evaporator and turbine.

DESCRIPTION

The ammonia vapor system for the 100 MWe spar configuration OTEC plant consists of one 51" diameter vapor pipe running down each side of the 25 MWe power module from the evaporator outlet connections to the turbine inlet main stop valves. Part of the pipe is exposed to the external seawater environment and the remainder is contained within the power module and is exposed to ambient internal conditions. The pipe is configured to avoid the formation of liquid ammonia pockets which could be carried into the turbine in slugs and cause damage to the turbine blades. The pipe exposed to the seawater environment is provided with external stiffening rings to preclude collapse when ammonia evacuation is being performed. Pipe material is carbon steel coated on the outside to minimize corrosion. A cathodic protection system will also be provided to minimize corrosion. To provide electrical isolation, an insulating flange or pipe section is provided between the aluminum evaporator and the carbon steel pipe and at the penetration point of the pipe into the power module. Appropriately isolated pipe supports, snubbers and expansion joints are provided to accommodate the expected static and dynamic loads. No valves are included in the vapor system since the main turbine stop valves can serve as insolation valves for the evaporator, thereby avoiding unnecessary additional pressure drop. A bypass line and control valve are provided around the turbine for startup purposes. Refer to Bechtel Drawings 12384-M-1 and 12384-P-1 for the schematic representation and arrangement, respectively, of the ammonia vapor pipe.

The ammonia condensate system consists of the appropriate piping, pumps and valving required to remove the ammonia condensate from the condenser sump and deliver it to the evaporator sump and from there to the piping distribution system and spray nozzles within the main ammonia evaporator. A 36" suction line is provided to three, one-half capacity, vertical centrifugal condensate pumps. The pumps are provided with butterfly isolation valves and minimum flow recirculation valves. A butterfly level control valve is provided in the 20" condensate pump discharge header connected to the evaporator sump. This valve controls level in the evaporator sump. Four one-third capacity distribution pumps take suction from the evaporator sump and deliver liquid ammonia through 14" headers to distribution points in the evaporator. These pumps are provided with butterfly isolation valves. A condenser level controller operates ammonia makeup and/or reject control valves which are connected to the central ammonia storage system. The pipe is coated carbon steel and is insulated within the power module to prevent excessive condensation. Outside the power module, the pipe is electrically isolated to prevent corrosion. Pipe supports, snubbers and expansion joints are provided as necessary to accommodate the expected static and dynamic loads. Pipe sizes are based on flows and state points for LMSC Computer Run Identification 26 September 1977, 12:48:3 (ΔT = 36°F) and ammonia vapor and condensate header velocities of 160 fps and 30 fps, respectively.

TABLE 1

AMMONIA CYCLE PARAMETERS - DESIGN CONDITIONS (36°F ΔT)

	Evaporator Discharge	Turbine Entrance	Condenser Discharge
Pressure, psia	130.9	131.1	93.18
Temperature, °F	70.93	71.03	52.29
Enthalpy, Btu/lb	624.17	624.17	100.48
Density, lb/ft³	0.4443	0.4451	38.89
Quality, %	99.0	98.997	0
Flow, lb/sec	1970	1970	1970

EQUIPMENT

Pumps:

	Condensate Pumps	Distribution Pumps
Number Required:	3	4
Fluid:	- Anhydrous Ammonia (liquid) -	
Temperature, °F:	52	71
Suction Pressure, psia:	98.2	135.9
Discharge Pressure, psia:	193.1	145.9
Flow (gpm) (per pump):	11,370	11,370
Suction Static Head:	20	20
Estimated HP (per pump):	790	75
Electrical, 3 phase volts:	4,160	460
Motor Type:	- Class 1, Div. 2, Group D -	
Material:	- Carbon steel, iron with SS trim -	
Seals:	- mechanical -	

Control Valve (1 required/module):

Type:	Butterfly, control and shutoff
Operator:	Oil hydraulic with remote signal
Shutoff Δp, psia:	Maximum 260
Normal operating pressure, psia:	193

Material: Steel, SS trim

Features: Static elastomer sealing ring or inflatable elastomer (T ring) seal for minimum leakage

Piping (per module):

Vapor:

Main vapor lines - 51" diameter, 0.56" wall, coated carbon steel, 300 feet.

Condensate:

14" diameter, 0.375" wall, coated carbon steel, 250 feet

20" diameter, 0.375" wall, coated carbon steel, 300 feet

36" diameter, 0.5" wall, coated carbon steel, 55 feet

Valves:

6 - 14" Butterfly, carbon steel with SS trim, elastomer seal for minimum leakage. To be complete with oil-hydraulic or motor operators.

WEIGHTS

	Approximate Weight in Pounds	
Ammonia condensate pipe	49,000	
Ammonia vapor pipe	92,000	
Ammonia (condensate)	46,000	
Ammonia (vapor)	2,000	
Condensate pump	23,000	(each)
Distribution pump	11,000	(each)
Vapor line expansion joints	2,000	(each)
Condensate line expansion joints	1,000	(each)
Control valve	1,200	
14" Butterfly valves	800	(each)

APPENDIX E-2

AMMONIA CLEANUP SYSTEM

PURPOSE

The ammonia cleanup system separates seawater contaminant (from condenser in-leakage) from the ammonia fluid. This is necessary because contamination by seawater will affect the fluid characteristics so that available pressure drop across the turbine is reduced. For example, a mixture of 90% ammonia and 10% water will result in 4 psi less available pressure differential than would be the case for 100% ammonia. Since the design pressure difference across the turbine is approximately 40 psi, this reduction would result in a significant power loss. In order to maintain the water contamination at a reasonable level when the condenser is leaking, it is necessary to maintain a continuous blowdown from the evaporator which must be cleaned and returned to the main system.

DESCRIPTION

The condenser must be considered as a source of possible in-leakage, since its external water pressure is greater by some 60 psi than the internal ammonia pressure. Double tube sheets have been incorporated in the condenser design to eliminate leaks through the tube sheet. However, there are 56,500 40-foot tubes in the unit, and it is reasonable to suppose that there may be a few leakers.

If the assumption is made that total in-leakage in the condenser is 1 gpm, and that the maximum permissible concentration of water in the evaporator will be 5%, then a continuous blowdown of 0.16% of the total ammonia flow will be required. The cleanup system is sized on this basis, resulting in a blowdown flow of 10,270 lb/hr.

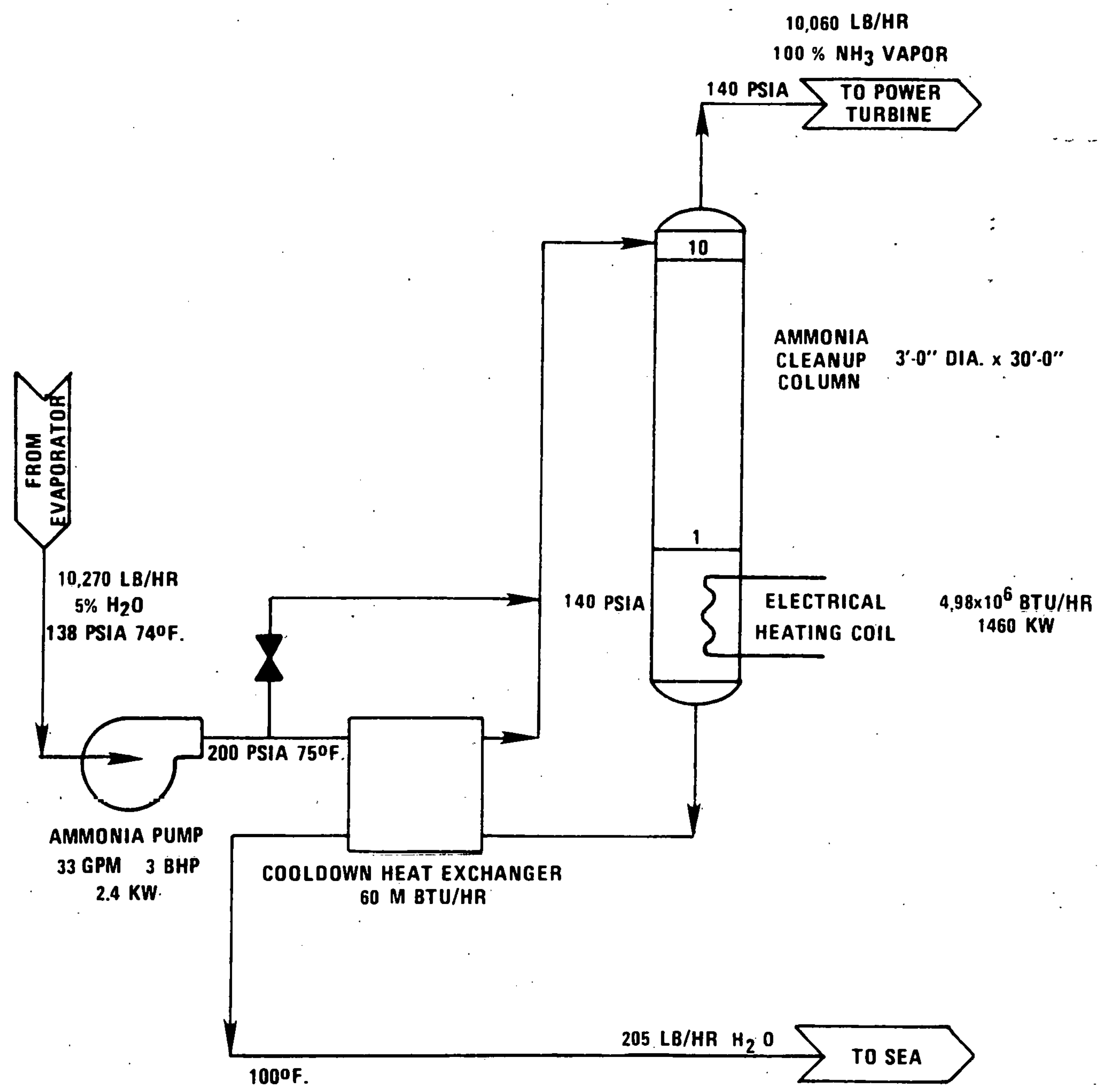
A number of methods were considered for this separation process including molecular sieve or silica gel adsorption beds, freezeout, flash separation and a distillation column. Adsorption beds would normally be the most desirable method. However, they are not satisfactory in this case since the high affinity between ammonia and water precludes satisfactory separation. Distillation is considered to be the most desirable of the three other methods based on experience, performance and power requirements. A sketch of a distillation type ammonia cleanup system is shown in SK-12384-7.

The blowdown from the evaporator sump is pumped into the distillation column. The pressure in the column is maintained high enough so that the resulting vapor may be ducted to the turbine vapor line without further compression. The water which is separated out flows into the sea after cooling to 100 F.

The column is heated by an electric heater which draws its power from the main generating system.

AMMONIA CLEANUP SYSTEM

SK – 12384 –7 REV. 1 DEC.29,1977



<u>Pump - Ammonia to Column (1 required/module)</u>:

Fluid:	anhydrous ammonia with 5% (maximum) water
Inlet pressure, psia:	138
Outlet pressure, psia:	200
Temperature, °F:	74
Specific gravity:	0.62
Flow, gpm:	33
Estimated HP:	3
Electrical:	460 volt, 3 phase, 60 cycle
Features:	Pump shall be complete with a Class 1, Division 2, Group D motor, mounted and aligned on a cast iron base. Mechanical seals are to be used.
Material:	Mfr. std. for ammonia service

<u>Distillation Column (1 required/module)</u>:

Fluid:	ammonia and seawater
Approximate size:	3' diameter by 30' vertical, 0.5" wall thickness
Design pressure, psig:	250
Temperature, °F:	370
Heater:	1460 kW, 4160 V, 60 cycle, 3 phase
Features:	Vertically mounted complete with supports, 10,270 lb/hr safety valve, drain, 4" supply and 6" discharge pipe flanges and two manholes.

Piping (per module):

Feed Pipe: 30 ft of std. wt. 4" pipe, with 4 - 90° elbows, one 200 lb gate valve.

Discharge Pipe: 30 ft of std. wt. 6" pipe, with 4 - 90° elbows, one 200 lb gate valve.

WEIGHTS

	Approximate Weight in Pounds
Pump, Ammonia to Column	200
Pipe, feed	500
Pipe, discharge	800
Distillation Column	10,000

LOCATION

The ammonia cleanup system is located in each power module machinery space.

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APPENDIX E-3

AMMONIA STORAGE, FILL AND PURGE SYSTEM

PURPOSE

The OTEC plant will utilize large quantities of ammonia in the power systems. Not only will the systems require an initial fill and makeup, but there will occasionally be a need to transfer ammonia, pump down and purge a system, refill, or dispose of fluid. For these procedures, an ammonia storage, fill, and purge system is required.

DESCRIPTION

The system consists of a main storage tank, compressors, and the necessary tanks, piping, valves and safety provisions, as shown in drawing SK-12384-9. An ammonia transfer pump is included in each power module. Operation of this system is described in Section 4.8.1 of the main body of the report.

Only one common system is used, as all four modules should not require filling or evacuation at the same time. Typical evacuation time for a portion of one module is about 12 hours. The evacuation time for the entire ammonia system of one module would not take longer than 24 hours.

EQUIPMENT

Compressors (12 stages required/plant):

Service:	Compress and condense anhydrous ammonia gas
Suction temperature, ^{o}F:	52 to 74

Suction pressure, psia:	3 to 170
Discharge pressure, psia:	170
Quantity, cfm:	1500 max
Cooling:	Seawater at 52°F
Estimated HP:	135 total for 12 stages
Drivers:	Class 1, Division 2, Group D electrical motors for first 9 stages, ammonia expanders for last three stages

Ammonia transfer pumps (1 required/module):

Service:	Liquid ammonia transfer
Fluid:	Anhydrous liquid ammonia
Temperature, °F:	from 52 to 74
Density:	39 lb/cu ft
Suction pressure, psia:	100
Discharge pressure, psia:	200
Flow, gpm:	400
Static head, ft:	200
Estimated HP:	40
Electrical:	460 V, 60 cycle, 3 phase, ac, Class 1, Division 2, Group D motor with shutdown heater
Materials:	Mfr. std. for liquid anhydrous ammonia
Features:	Mechanical seal. Pump to be assembled and aligned on rigid base with gauges and leakoff connection.

Ammonia Storage Tank (1 required/plant):

Type:	Cylindrical
Design pressure, psig	250
Storage temperature, F	70
Size, radius, ft	30
Storage volume, cu. ft.	55,000 (including 15% ullage per OSHA requirements)

Purge Fans (2 required/plant):

Service:	venting of noxious gases and nitrogen
Type:	in-duct type propeller fans
Capacity:	1000 SCFM each
Pressure:	10" H2O discharge
Supply:	ambient air at 70°F, 90% humidity
Electrical:	460 V, 60 cycle, 3 phase, explosion proof motor
Estimated HP:	5

Discharge Ducts (2 required/plant):

Light gauge, 12" diameter steel, fiberglass, or other suitable material. Maximum pressure is 10" H2O. Length, 250 ft each.

Central Nitrogen Storage Cylinders (1 set required/plant):

40 high pressure cylinders complete with rack, manifold, valving and pressure reducers. Each pressure reducer will be capable of 10 SCFM at 20 psia discharge pressure.

Module Nitrogen Storage Cylinders (4 sets required/plant):

10 high pressure cylinders, complete with rack, manifold, valving, and pressure reducers. Each pressure reducer will be capable of 10 SCFM at 150 psia discharge pressure.

PIPING AND VALVES

The piping required is that necessary to connect the central storage bank to each module. Additional piping is arranged so that either air or ammonia can be evacuated from all major equipment and piping segments. This same piping is used for filling the volume with nitrogen or air.

Ammonia transfer:	std. wt. steel, 3" size, 300 ft, 250 psig rating
Evacuation and fill piping:	std. wt. steel, 10" size, 1500 ft, 35 valves, 250 psig rating
High pressure Nitrogen piping:	3/8" stainless steel tubing, 1600 ft.
Low pressure Nitrogen piping:	Std. wt steel, 3" size, 50 ft, 10 psig rating

WEIGHTS

The estimated weight in pounds for the components are:

Ammonia storage	2.0×10^6
Nitrogen storage	8000
Ammonia transfer pump	6000 (each)
Compressors	20,000
Piping, total	4,400
Fans (each of 2)	500
Vent ducts	1000

LOCATION

The ammonia storage, fill, and purge equipment will be installed in the machinery space in the central core. The discharge ducts are installed one at each side of the central high rise section, extending from the central machinery space to the open air above the sea surface. One fan is installed at the foot of each duct.

Each module will have its own ammonia transfer pump and nitrogen pressurizing system. Nitrogen cylinders will be racked against the wall. Additional cylinders will be stored in the central structure.

Appendix F
AUXILIARY SUBSYSTEMS

This appendix includes supplemental details on several auxiliary subsystems which are not considered as integral subsystems within the power module cycle. These subsystems are required for module operation and the details are presented in three separate sections, as follows:

F-1 Electrical System
F-2 Fluid Sampling
F-3 Seawater Pump Pod Pressurizing System

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APPENDIX F-1

ELECTRICAL SYSTEM

PURPOSE

An electrical system is provided to interconnect the module generators to the core rectifiers and to distribute power to the module and core auxiliary electrical equipment. Direct current transmission of electrical power to shore is by others. Standby and emergency electrical power systems are provided to ensure the safety of the OTEC plant. In the event of a black start-up (no power modules operating), provision is made for utilizing start-up power from an outside source.

DISCRIPTION

The electrical system is designed to function in four distinct modes of operation:

- o Start - up
- o Normal
- o Standby
- o Emergency

A block diagram of the electrical system showing start-up, normal, standby and emergency power systems is shown in Figure 4.8-4 and a description of system operation is presented in Section 4.8.2 of the main report. Details of the system hardware are presented below:

CABLE AND BUS

480 volt power cable:

No. of Circuits	Wire Size	Circuit Length (ft)	Conduit size (in)	Total Circuit Feet
38	3#12	100	1	3800
18	3#4	100	1-1/4	1800
6	3#2/0	100	2	600
4	3#4/0	100	3	400
10	--	100	1	1000
14	3-500 mcm	100	4	1400

4160 volt cable:

48	3#4/0 AWG	100	3	4800
1	3-500 mcm	200	3	200

13,800 volt cable:

1800 feet, 2000 A

345,000 volt bus:

400 feet, 150 A

EQUIPMENT

Refer to table that follows this section.

WEIGHTS

Electrical equipment weight is 1,314,000 lb in the core plus 98,000 lb per module.

Cable and bus weights are as follows:

	Pounds
480 volt cable	15,000
4160 volt cable	25,000
13,800 volt cable	3,000
13,800 volt bus	15,000
345,000 volt bus	6,000
Total cable and bus:	64,000

Total Weight (Electrical equipment, cables and buses): 1,770,000 lb

CORE ELECTRICAL EQUIPMENT

	Equipment and Rating	Dimensions (ft)	Unit Weight (lbs)	Total Weight (lbs)
a)	<u>At each Core to Module Entrance (4 Entrances)</u>			
3	Summing Transformers, 10,900 KVA, 1-phase two winding, 13,800 to 345,000 Volts	12L x 10W x 8H	60,000	720,000
1	Metalclad Switchgear, 13,800 Volts, 3-phase 550 MVA class indoor type consisting of: 1 ea ACB, 2000 Amp, main generator type 1 ACB, 2000 Amp, Feeder type 2 ACB, 1200 Amp, Feeder type	17L x 7W x 8H		16,000
3	Harmonic Filter Banks	20L x 10W x 8H	5,200	74,400
3	Auxiliary Transformers, 1-phase, 2500/2875 KVA 13,800 to 2,400 Volts for 13,800-4100/2400 volt module power supply	5L x 3W x 5H	20,000	240,000
b)	<u>At Rectifier</u>			
8	Rectifier Assemblies, each 15,625 KW for 345,000 volt ac to 250,000 volt dc Rectifier	10L x 8W x 15H	2,700	21,600
8	Commutation Cubicles	6L x 4W x 8H	1,300	10,400
9	dc reactors	12L x 10W x 8H	30,000	120,000
4	pf correction breakers	5L x 3W x 8H	2,000	8,000
4	pf correction banks	15L x 10W x 8H	4,000	16,000
6	dc circuit breakers	5L x 3W x 8H	3,000	18,000
17	dc disconnect 2 transfer switches	4L x 2W x 4H	1,000	17,000

CORE ELECTRICAL EQUIPMENT (Cont'd.)

Equipment and Rating	Dimensions (ft)	Unit Weight (lbs)	Total Weight (lbs)
c) <u>At Central Core Equipment Area</u>			
2 Transformers, 3-phase, 300 KVA, 13,800 to 480 Volts, dry type	4L x 2.5W x 4H	1,500	3,000
1 Motor Control Center (MCC), 480 Volt 3-phase, 600 Amp bus, Amp momentary consisting of: 2 ea ACB, 600 Amp, 480 Volt main type, metal enclosed 4 ea size 4 combination starter 1 ea size 3 combination starter 3 ea size 2 combination starter 6 ea size 1 combination starter	10L x 3W x 7H		1,500
1 Motor Control Center (MCC) same type as above consisting of: 2 ea ACB 600 Amp, 480 volt main type, metal enclosed 6 ea size 3 combination starter 4 ea size 2 combination starter 2 ea feeder top circuit breakers	9L x 3W x 7H		1,500
2 Load Centers with 3-1600 Amp & 1-600 Amp Breakers			3,000
d) At Central Core Control Room	30L x 10W x 10H (room space)		20,000
1 Inverter, 25 KVA, 120 Volt ac, 1000 Amp - hour battery pack			

CORE ELECTRICAL EQUIPMENT (Cont'd.)

Equipment and Rating	Dimensions (ft)	Unit Weight (lbs)	Total Weight (lbs)
e) <u>On each of four levels (2, 5, 8, 11)</u>			
1 Transformer, 3-phase, 15 KVA, 480-120/208 Volts, for lighting and utilities.	3L x 2W x 3H		600
f) <u>On top level</u>			
1 Transformer, 3-phase, 5,000 KVA, ship-voltage (4160V assumed) to 13,800 Volts	12L x 8W x 9.5 H		23,000

MODULE ELECTRICAL EQUIPMENT

Equipment and Rating	Dimensions (ft)	Unit Weight (lbs)	Total Weight (lbs)
<u>In each Module (four Modules)</u>			
1 Metalclad Switchgear, 4160 Volt, 250 MVA Class, consisting of	45L x 7W x 8H	20,000	80,000
1 ea ACB, 3000 amp, Main Type 13 ea ACB, 1200 amp, Motor Feeder Type 1 ea ACB, 1200 Amp, Transformer Feeder Type			
1 Transformer, 3-phase, 250 KVA, 4160 to 480 Volt, dry type	5L x 2W x 4H	1,500	6,000
2 Motor Control Center (MCC) assemblies, 480 Volt, 22,000 short-circuit Amp, consisting of			
1 ea size 3 combination starter 2 ea size 2 combination starter 6 ea size 1 combination starter 3 ea feeder tap circuit breaker			

APPENDIX F-2
FLUID SAMPLING

PURPOSE

Fluid sampling will be required to ascertain the condition of the ammonia working fluid, especially when contamination with seawater is suspected. High contamination levels will be alarmed, resulting in operation of the leak detection system and the ammonia cleanup system.

DESCRIPTION

Samples will be taken continuously at the rate of 1 gpm from the condenser sump and the evaporator sump. They will be passed through ion detectors (sodium ion) and recorders, which will indicate contamination of the ammonia by seawater.

The ammonia will flow under pressure to the instruments, and will be pumped back into the system.

EQUIPMENT

Ion Detectors (2 required/module):

Fluid:	anhydous ammonia, with max. of 5% of seawater
Flow:	1 gpm
Pressure:	138 psia max.
Temperature:	74°F, 53°F

Features:	Sample cell must be sealed against pressure. Units to be complete with means for calibration and recorders

Pumps (2 required/module):

Fluid:	Anhydous ammonia with maximum of 5% seawater
Inlet pressure:	138 psia, and 95 psia
Discharge pressure:	150 psia
Temperature:	74°F and 53°F
Quantity:	1 gpm
Type:	chemical feed, diaphragm
Electrical:	Open drip proof motors, 110 V, 60 cycle, single phase

WEIGHTS

The weights of components in the Fluid Sampling System are negligible.

LOCATION

Fluid Sampling equipment is located near the condenser sump and evaporation sump of each module.

APPENDIX F-3

SEAWATER PUMP POD PRESSURIZING SYSTEM

PURPOSE

The warm seawater pumps are installed at an average depth beneath the surface of 150 ft, and the cold seawater pumps at an average depth of 275 ft. These depths correspond to external pressures on the pods of approximately 82 and 155 psia. The pods will have a rotating seal on the large propeller shaft, and static seals on hull penetrations. A rotating seal of the size required is complex and will require maintenance in proportion to the pressure load. Pressurizing of the pod interior to offset the external pressure of the sea will relieve the seal of an undesirable pressure differential. In addition, any other leakage will be minimized.

DESCRIPTION

The pump pod pressurizing system consists of pressure controllers, piping, valves and flow indicators. The system receives its air from the central compressed air system.

Each module is provided with its own system, to handle 4 warm and 4 cold seawater pumps. Compressed air is admitted to each group of pump pods by an automatic controller, so that the interior pressure is only slightly below the seawater pressure. The warm seawater pumps, being above the cold seawater pumps, will require less pressure than the latter. Pressure controllers are adjusted manually as required.

When the pumps are operating the pods will normally be pressurized. If maintenance is required, the pump is shut down, and the pod de-pressurized. Normal repairs or inspection may then be performed. The pods may be depressurized or pressurized from the power module control room.

EQUIPMENT

Pressure reducers (high pressure) (4 required/plant):

Service:	Regulation of air pressure to pump pods
Inlet pressure:	300 psia
Outlet pressure:	adjustable from 125 psia to 175 psia
Capacity:	100 SCFM

Pressure reducers (low pressure) (4 required/plant):

as above, except

Outlet pressure:	adjustable from 50 psia to 75 psia

PIPES AND VALVING

The same piping will be used for pressurizing as for depressurizing. An estimated 1500 ft of 1" 150 psig piping and 36 valves will be required.

WEIGHTS

	Estimated Pounds
Pressure reducers (each of 8)	10
Pipe and valving	1500

LOCATION

The pipe to each pump pod will be lead from a valved manifold in each power module.

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Appendix G
DACS AVAILABILITY ANALYSIS

G.1 INTRODUCTION

The reliability of the proposed power module Data Acquisition and Control System (DACS) may be discussed in terms of its projected availability, i.e., the probability that it will be operational at a given time. Availability is defined in terms of the statistical mean time between failures (MTBF) and the mean time to repair (MTTR) as:

$$A = \frac{MTBF}{MTBF + MTTR}$$

Individual MTBF, MTRR, and associated availability figures obtained from manufacturer experience data for the units which taken together will comprise the DACS are shown in Table G-1. The MTBF data shown has been derated by approximately 10 percent for an ocean vessel installation; the derated MTBF will be employed even though it is expected that the equipment will be operated in a protected environment. The MTTR data shown reflects the availability of a full complement of spare parts and competent systems repair technicians. Additionally the MTTR data includes time for trouble call response as well as the time necessary to return the failed unit to operational status or to make the appropriate unit replacement(s).

G.2 SYSTEM AVAILABILITY REQUIREMENT

The anticipated duration of tests to be conducted for the power system "proof-of-concept" system is 6,000 hours. It is therefore required that the DACS be capable of an overall "mean time between failure" in excess of 6,000 hours.

Table G-1

COMPONENT RELIABILITY DATA

System Component	MTBF (hr)	MTTR (hr)
Operator Console	3,200	2.1
Peripheral Switch	24,000	2.1
Computer, Memory, and Logic Systems	2,200	4.0
Private Mass Memory	14,000	4.5
Shared Mass Memory	14,000	4.5
CPU/CPU Link	40,000	2.1
Line Printer	2,000	2.1
Console Printer	2,000	2.1
Card Reader	1,000	2.1
Magnetic Tape Unit	34,000	2.5
Data Comm. I/F	6,500	2.5
Comm. Ckts. and I/F	35,000	2.1
Remote Term. Units	5,800	2.1

G.3 DACS CONFIGURATIONAL CONSIDERATIONS

A DACS system may be configured which functionally will provide the required data acquisition, recording, and control capability. The selected system configuration will provide the required capability with a predictable statistical reliability dependent, generally, upon the amount or degree of redundancy incorporated in the implemented system configurations. The employment of redundant system components on a full or partial basis usually affords the ability to expand system operations in a useful manner. For computer systems, this expansion frequently permits software development, scientific analysis program execution, and/or perhaps a management report generation activity to proceed concurrent with supervisory control and surveillance activities. This mode of utilization of the redundantly configured computer systems equipment may be termed "duplexing" of redundant system components. Where the redundant system components are receiving all system inputs and exercising all data processing capabilities necessary to provide required system outputs, the mode of utilization may be termed "dual tracking." In the "dual tracking" mode, no additional data processing operations may be achieved. The following tabulations of system configuration features shows the relative merits of the various DACS configurations considered.

1. Single System

   Advantage: Least expensive

   Disadvantage: Operations terminate with single point system failures.
   No significant background data processing capability.

2. Partial Redundancy (Duplexed CPU with switched Front-End and Data Base Storage)

   Advantage: More reliable than single string system.
   Useful processing in back-up CPU and available peripherals.

   Disadvantage: Single point failures can still terminate supervisory and control operations. Reliability limited by single point elements of system.

3. Full System Redundancy

   A. Dual (Operation Tracking) System

      Advantage: Maximum reliability – single point failures will not terminate operations.
      Immediate failover.

      Disadvantage: Costly – but only slightly more than No. 2 above.
      Backup machine not available for useful data processing.

   B. Duplexed (Non-Tracking) System

      Advantage: Maximum Reliability – single point failures will not terminate operations.
      Rapid failover (not immediate, surveillance scan required by backup system).
      Backup machine available for useful data processing tasks.

Disadvantage: Costly — but only slightly more than No. 2 above.
Failover requires scan to establish power system awareness.

Availability computation for the single versus dual system configuration (see attached computation) shows a very significant increase in the mean-time-to-failure expectation for the dually configured system. For the partially redundant system, the overall system availability is dependent upon single point failure and, as such, can be no more reliable than the component(s) where single point failures may occur.

Where life support systems are involved and/or where the loss of data may result with catastrophic failures, the dual configuration operated in the "tracking" mode appears necessary. For data acquisition and control applications, such as DACS, where operational state data is acquired at lower acquisition rates, the "duplexed" operational mode is usually employed. Upon failure by one computer system, automatic failover is implemented such that within a relatively few seconds full surveillance and control is established by the backup machine. During normal operations, the backup computer system may be employed to accomplish useful data processing work.

G.4 SYSTEM AVAILABILITY COMPUTATION

For purposes of comparison, statistical system availability has been computed for a DACS with and without critical system component redundancy. Table G-2 shows the computational results for all DACS component system parts.

If the failure of any single unit is determined to constitute a system failure, then the composite, or overall system availability is computed as the product of the individual system component availabilities.

$$A_{sys} = A_1 \cdot A_2 \cdot A_3 \cdots A_n$$

when two functionally redundant (not necessarily identical) components exist in a system such that the failure of only one of the two components does not constitute a system

Table G-2
DACS AVAILABILITY COMPUTATION

System Element	Availability	
	W/O Redundancy	With Redundancy
Operator Console	0.99934418	0.9999996
Peripheral Switch	—	0.9999125
Computer, Memory, Etc.	0.99818511	0.9999967
Private Mass Memory	0.99967867	0.9999999
CPU/CPU Link	—	0.9999475
Line Printer	0.99895110	0.9999989
Console Printer*	0.99895110	—
Card Reader*	0.9979044	—
Magnetic Tape Unit*	0.99992647	—
Data Comm. I/F	0.99961553	0.9999998
Comm. Ckts. and I/F	0.99940000	0.9999997
Remote Terminal Unit	0.99963806	0.9999999
Overall System Availability:		
Without Redundancy: 0.99450361		
With Redundancy: 0.9998544		

*Non Critical Element

failure, then the overall system availability is computed as:

$$A_{sys} = A_1 + A_2 - A_iA_2$$

With individual DACS system component availabilities, Table G-2 also shows overall system availability with and without critical system component redundancy.

G.5 DISCUSSION OF ANALYTICAL RESULTS

The results of the statistical analysis show the redundantly configured system to be enhanced, in the reliability sense, by a factor of about 36. In terms of "mean time between failure", the results imply* a decrease in failure rate from one failure per each 0.6 month period (for the nonredundant system) to one failure per each 1.8 year period. (MTTR is assumed to be 2.5 hours).

G.6 RECOMMENDATIONS AND CONCLUSIONS

Based upon the foregoing analysis, the OTEC 6,000-hr test requirement, and the opportunity for useful background computer processing**, it is recommended that the DACS be configured with critical component redundancy for employment in a duplexed (non-tracking) operational mode.

*The reader is cautioned to keep in mind that these results are based upon average failure and time to repair data - worst case results are not incorporated !

**Including software development, scientific analysis program execution, and possible management report generation.

Appendix H

HEAT EXCHANGER SUPPORTING DATA

SUMMARY

This Appendix presents detailed figures and tables to support the heat exchanger design, cost trade studies and construction technology which are discussed in Section 3.1.1 of the main volume.

The two-dimensional computer model configuration is described and the stresses and loads from the 13 runs are tabulated for the key areas.

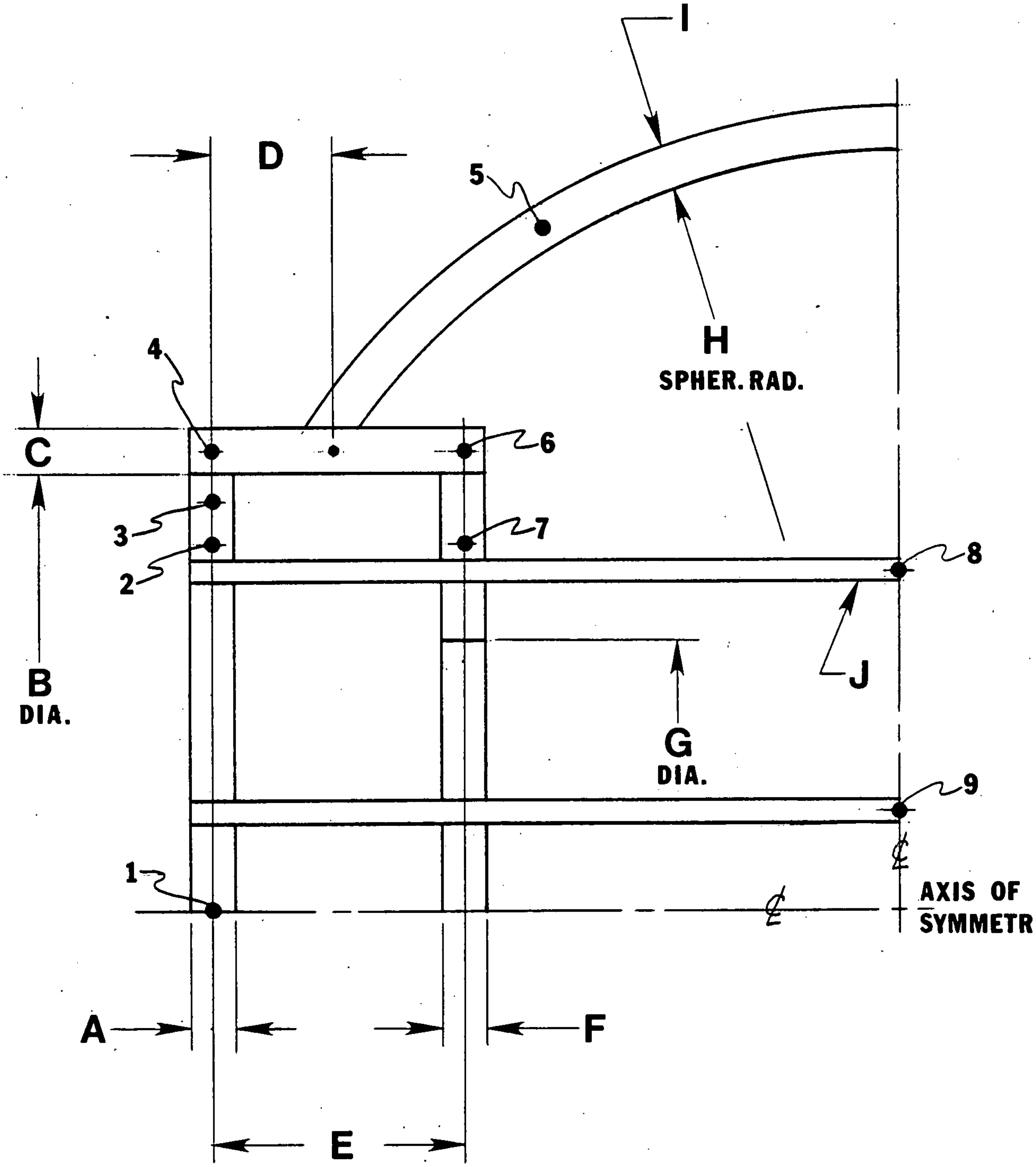
The materials and overall dimensions are tabulated for the 360 evaporators and condensers which were sized for the cost trade studies. The costs per MWt are included in the same tables for about one-third of the units which were cost estimated. A second set of tables shows the weight and cost by major categories for 120 designs. The cost per square foot for plate material and the maximum available plates sizes are tabulated for different materials and thicknesses. Tubing cost is tabulated for a variety of materials, tube diameter and tube wall thickness. Three additional cost curves are included to supplement those found in the main text.

The results of the electron beam welding test and the tube-to-tubesheet welding development program are summarized.

The cost comparisons for alternative fabrication methods are tabulated and comments offered concerning the applicability of the various methods to the heat exchanger construction.

CONTENTS

FIGURE H-1



FIXED TUBESHEET MODEL

TABLE H-1

FIXED TUBESHEET MODEL DESCRIPTION

CASE NO.	MODEL GEOMETRY												
	DIMENSIONS, INCHES (REF. FIGURE H-1)										Tubesheet Material	Ring Material	Shell Material
	A	B	C	D	E	F	G	H	I	J			
1	6.0	647	8	10.438	-	-	-	401.0	4.0	2.5"OD x .110"Wall (SB-241)	5083-0 AL (SB-209)	5083-0 AL (SB-209)	5083-0 AL (SB-209)
2	6.0	543	12	10.438	-	-	-	401.0	4.0				
3	6.0	647	8	10.438	-	-	-	401.0	5.0				
4	6.0	647	8	0	-	-	-	401.0	4.0				
5	1.0	643	8	12.0	24	1.0	0	419.31	4.0				
6	1.0	643	4	12.0	24	1.0	0	419.31	4.0				
7	1.5	647	4	12.0	24	1.0	426	419.31	4.0				
8	1.5	647	4	12.0	24	1.5	426	419.31	4.0				
9*	1.5	647	4	12.0	24	1.5	426	419.31	4.0				
10	1.5	647	4	12.0	24	1.5	426	419.31	4.0				
11	2.5	643	4	11.864	-	-	-	419.94	2.625	2.5"OD x .053"Wall AL6X	Nitronic 50 Cres	SA-516 GR.60	SA-516 GR.60
12**	1.75	653	4	12.15	-	-	-	420.0	2.5				
13	1.0	647	4	12.0	24	1.0	0	420.0	2.625				

*Added Extra Stays in Outermost Perforated Region

**4407" Radius, Spherically Dished Tubesheet

TABLE H-4

FIXED TUBESHEET ANALYSIS SUMMARY

Location (Ref. Fig. H-1)	DESCRIPTION	COMPUTED RESULTS/CODE ALLOWABLE												
		CASE NO.												
		1	2	3	4	5	6	7	8	9	10	11	12	13
1	Membrane Stress in Center of Tubesheet	6,152	5,632	5,462	5,053	7,736	10,521	9,970	10,005	10,325	11,526	10,721	17,321	11,141
		10,000	10,000	10,000	10,000	12,000	12,000	12,000	12,000	12,000	12,000	28,300	28,300	28,300
2	Bending Stress in Outermost Perforated Region	35,965	35,279	29,635	20,776	46,689	42,468	47,574	45,945	43,752	35,424	90,528	76,350	48,491
		30,000	30,000	30,000	30,000	36,000	36,000	36,000	36,000	36,000	36,000	84,900	84,900	84,900
3	Bending Stress in Outermost Unperforated Region	12,323	10,527	10,829	9,778	25,230	22,231	39,818	39,261	35,312	31,759	37,858	51,085	56,099
		30,000	30,000	30,000	30,000	36,000	36,000	36,000	36,000	36,000	36,000	84,900	84,900	84,900
4	Bending Stress at Ring Junction	15,899	15,704	9,822	12,244	66,682	58,907	56,437	55,760	50,296	46,142	48,073	66,973	84,384
		27,900	27,900	27,900	27,900	27,900	32,100	32,100	32,100	32,100	32,100	60,000	60,000	60,000
5	Bending Stress in Spherical Shell	13,700	13,381	10,920	14,206	15,892	15,814	15,677	15,626	15,547	15,561	23,110	24,992	24,544
		16,050	16,050	16,050	16,050	16,050	16,050	16,050	16,050	16,050	16,050	30,000	30,000	30,000
6	Bending Stress at Ring Junction	–	–	–	–	10,478	11,066	12,550	12,459	18,846	4,346	–	–	11,614
		–	–	–	–	27,900	32,100	32,100	32,100	32,100	32,100	–	–	60,000
7	Bending Stress in Outermost Perforated Region	–	–	–	–	29,148	24,796	18,216	21,574	28,308	5,076	–	–	33,546
		–	–	–	–	36,000	36,000	36,000	36,000	36,000	36,000	–	–	84,900
8	Maximum Load on Outermost Tubes (Lbs.) *	-3,972	-3,967	-3,176	-4,232	-3,350	-2,791	-3,936	-3,927	-4,012	-3,475	-3,782	-3,391	-3,144
		-6,922	-6,922	-6,922	-6,922	-6,922	-6,922	-6,922	-6,922	-6,922	-6,922	-20,591	-20,591	-20,591
9	Maximum Stress on Central Tubes (Lbs.)	829	829	781	865	870	830	823	789	798	838	1,589	1,641	1,648
		6,300	6,300	6,300	6,300	6,300	6,300	6,300	6,300	6,300	6,300	22,500	22,500	22,500

*Allowable Load Given is the Euler Critical Buckling Load Based on a Support Plate Spacing of 8'-6" (102")

Evaporator/Condenser Evaporator Size 5MWe= 221 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	HY80		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	2.1/2.9	1.37/2.00		3.75	2.375		.87/1.3	.50/.75		1.25	.75	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	33.50											
No. of Water Tubes	91,000											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.028	.028
Active Length, ft.	10.09	11.10	11.29	10.09	11.10	11.29	10.09	11.10	11.29	10.09	11.10	11.29
Cost/MW_t, $	11335	18585					11062					
No. of Water Tubes	40,444											
Tube OD, in.	1.5											
Tube Wall, in.	0.65	.037	.032	.065	.037	.032	.065	.034	.028	.065	.034	.028
Active Length, ft.	15.99	17.72	17.83	15.99	17.72	17.83	15.99	17.60	17.63	15.99	17.60	17.63
Cost/MW_t, $	7877	16560					7481					
No. of Water Tubes	22,750											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.072	.045	.038	.072	.045	.038
Active Length, ft.	22.03	24.97	25.12	22.03	24.97	25.12	22.07	24.74	24.84	22.07	24.74	24.84
Cost/MW_t, $	7608	18091					6846					
No. of Water Tubes	14,560											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.090	.055	.048	.090	.055	.048
Active Length, ft.	28.28	32.81	33.03	28.28	32.81	33.03	28.30	32.31	32.58	28.30	32.31	32.58
Cost/MW_t, $	8229	21198					6961					
No. of Water Tubes	10,110											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.108	.067	.057	.108	.067	.057
Active Length, ft.	34.72	41.11	41.37	34.72	41.11	41.37	34.70	40.49	40.72	34.70	40.49	40.72
Cost/MW_t, $	9259	24357					7805					

TABLE H-3

Evaporator/Condenser Evaporator Size 25 MWe=1108 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	HY80		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	3.4/4.4	2.12/3.12		6.375	4.00		1.4/1.8	.81/1.13		1.625	1.062	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	53.29											
No. of Water Tubes	244,664											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.028	.028
Active Length, ft.	14.94	16.56	16.91	14.94	16.56	16.91	14.94	16.56	16.90	14.94	16.56	16.90
Cost/MW_t, $	6828	11475	11291	6784	11470	11286	6602	11879	11694	6582	11907	11723
No. of Water Tubes	108,740											
Tube OD, in.	1.5											
Tube Wall, in.	.065	.037	.032	.065	.037	.032	.065	.034	.028	.065	.034	.028
Active Length, ft.	23.32	26.31	26.48	23.32	26.31	26.48	23.32	26.06	26.08	23.32	26.06	26.08
Cost/MW_t, $	5007	10976	8715	5014	10987	8725	4720	10748	8349	4738	10822	8423
No. of Water Tubes	61,166											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.072	.045	.038	.072	.045	.038
Active Length, ft.	31.97	37.13	37.36	31.97	37.13	37.36	31.93	36.68	36.82	31.93	36.68	36.82
Cost/MW_t, $	5084	12592	9959	5104	12600	9967	4496	12124	9913	4543	12237	10026
No. of Water Tubes	39,146											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.090	.055	.048	.090	.055	.048
Active Length, ft.	40.95	48.98	49.29	40.95	48.98	49.29	40.81	47.97	48.42	40.81	47.97	48.42
Cost/MW_t, $	5628	15263	12070	5632	15393	12312	4642	13973	11549	4702	14118	11694
No. of Water Tubes	27,185											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.108	.067	.057	.108	.067	.057
Active Length, ft.	50.11	61.56	61.93	50.11	61.56	61.93	49.94	60.33	60.67	49.94	60.33	60.67
Cost/MW_t, $	6440	18121	14550	6352	18035	14465	5351	16767	13519	5402	16893	13645

TABLE H-4

TABLE H-5

Evaporator/Condenser Evaporator Size 25MWe=1108 MWt

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	516		5083	516	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/516	SS
Shell Thickness, in.								.81/1.13			1.062	
Tubesheet Thickness, in.								1	1		1	1
Outer Tube Limit, ft.	53.29											
No. of Water Tubes	244,664											
Tube OD, in.	1.0											
Tube Wall, in.								.028	.028		.028	.028
Active Length, ft.								16.56	16.90		16.56	16.90
Cost/MW_t, $								11381	11197		11418	11234
No. of Water Tubes	108,740											
Tube OD, in.	1.5											
Tube Wall, in.								.034	.028		.034	.028
Active Length, ft.								26.06	26.08		26.06	26.08
Cost/MW_t, $								10241	7816		10284	7886
No. of Water Tubes	61,166											
Tube OD, in.	2.0											
Tube Wall, in.								.045	.038		.045	.038
Active Length, ft.								36.68	36.82		36.68	36.82
Cost/MW_t, $								11552	9331		11646	9435
No. of Water Tubes	39,146											
Tube OD, in.	2.5											
Tube Wall, in.								.055	.048		.055	.048
Active Length, ft.								47.97	48.42		47.97	48.42
Cost/MW_t, $								13326	10303		13467	11043
No. of Water Tubes	27,185											
Tube OD, in.	3.0											
Tube Wall, in.								.067	.057		.067	.057
Active Length, ft.								60.33	60.67		60.33	60.67
Cost/MW_t, $								16014	12766		16180	12932

TABLE H-6

Evaporator/Condenser: Evaporator Size: 50MWe=2216 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	HY80		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	4.1/5.3	2.62/3.75		8.25	4.875		1.7/2.1	1.00/1.38		1.875	1.25	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	67.15											
No. of Water Tubes	388,482											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.028	.028
Active Length, ft.	17.57	19.52	19.95	17.57	19.52	19.95	17.57	19.52	19.96	17.57	19.52	19.96
Cost/MW_t, $	5892	10011					5674					
No. of Water Tubes	172,659											
Tube OD, in.	1.5											
Tube Wall, in.	.065	.037	.032	.065	.037	.032	.065	.034	.028	.065	.034	.028
Active Length, ft.	27.28	30.98	31.18	27.28	30.98	31.18	27.28	30.66	30.67	27.28	30.66	30.67
Cost/MW_t, $	4444	9827					4176					
No. of Water Tubes	97,120											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.072	.045	.038	.072	.045	.038
Active Length, ft.	37.35	43.75	44.02	37.35	43.75	44.02	37.26	43.17	43.32	37.26	43.17	43.32
Cost/MW_t, $	4578	11433					4048					
No. of Water Tubes	62,157											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.090	.055	.048	.090	.055	.048
Active Length, ft.	47.80	57.78	58.15	47.80	57.78	58.15	47.57	56.49	57.05	47.57	56.49	57.05
Cost/MW_t, $	5115	13986					4215					
No. of Water Tubes	43,164											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.108	.067	.057	.108	.067	.057
Active Length, ft.	58.44	72.71	73.15	58.44	72.71	73.15	58.17	71.15	71.54	58.17	71.15	71.54
Cost/MW_t, $	5828	16686					4863					

TABLE H-7

Evaporator/Condenser Condenser Size $5MW_e$=215 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	HY80		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	2.1/2.9	1.37/2.00		3.75	2.375		1.1/1.6	.63/1.00		1.50	1.00	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	33.50											
No. of Water Tubes	95,790											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.023	.028
Active Length, ft.	11.20	12.27	12.44	11.20	12.27	12.44	11.20	12.27	12.44	11.20	12.27	12.44
Cost/MW_t, $												
No. of Water Tubes	42,573											
Tube OD, in.	1.5											
Tube Wall, in.	.065	.037	.032	.065	.037	.032	.065	.028	.028	.065	.023	.028
Active Length, ft.	17.91	19.60	19.73	17.91	19.60	19.73	17.91	19.31	19.56	17.91	19.31	19.56
Cost/MW_t, $												
No. of Water Tubes	23,947											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.065	.034	.028	.065	.034	.028
Active Length, ft.	24.72	27.57	27.75	24.72	27.57	27.75	24.85	26.86	26.91	24.85	26.86	26.91
Cost/MW_t, $												
No. of Water Tubes	15,326											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.065	.043	.035	.065	.043	.035
Active Length, ft.	31.74	36.13	36.37	31.74	36.13	36.37	31.98	34.95	34.98	31.98	34.95	34.98
Cost/MW_t, $												
No. of Water Tubes	10,643											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.065	.051	.042	.065	.051	.042
Active Length, ft.	39.02	45.14	45.42	39.02	45.14	45.42	39.25	43.36	43.42	39.25	43.36	43.42
Cost/MW_t, $												

TABLE H-8

Evaporator/Condenser Condenser **Size** 25 MWe=1076 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	516		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	3.4/4.4	2.12/3.12		6.375	4.00		1.9/2.5	1.06/1.50		2.00	1.375	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	53.29											
No. of Water Tubes	257,541											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.028	.028
Active Length, ft.	15.92	17.57	17.88	15.92	17.57	17.88	15.92	17.57	17.88	15.92	17.57	17.88
Cost/MW_t, $	7200	12277	12103				7008	12212	12041			
No. of Water Tubes	114,463											
Tube OD, in.	1.5											
Tube Wall, in.	.065	.037	.032	.065	.037	.032	.065	.028	.028	.065	.028	.028
Active Length, ft.	25.05	27.93	28.11	25.05	27.93	28.11	25.05	27.29	27.76	25.05	27.29	27.76
Cost/MW_t, $	5248	11813	9346				5021	9790	8412			
No. of Water Tubes	64,385											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.065	.034	.028	.065	.034	.028
Active Length, ft.	34.40	39.33	39.58	34.40	39.33	39.58	34.40	37.85	37.88	34.40	37.85	37.88
Cost/MW_t, $	5360	13628	10747				4480	10019	7889			
No. of Water Tubes	41,206											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.065	.043	.035	.065	.043	.035
Active Length, ft.	44.07	51.71	52.05	44.07	51.71	52.05	44.02	49.30	49.27	44.02	49.30	49.27
Cost/MW_t, $	5932	16577	13078				4298	11660	8942			
No. of Water Tubes	28,616											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.065	.051	.042	.065	.051	.042
Active Length, ft.	54.00	64.81	65.22	54.00	64.81	65.22	53.75	61.24	61.26	53.75	61.24	61.26
Cost/MW_t, $	6879	19709	15799				4464	13519	10417			

TABLE H-9

Evaporator/Condenser Condenser

Size 50 MWe= 2152 MW_t

Platform/Module	IMMERSED			IMMERSED			SUBMERGED			SUBMERGED		
Sphere/Cylinder	SPHERE			CYLINDER			SPHERE			CYLINDER		
Shell Material	5083	516		5083	516		5083	HY80		5083	HY80	
Tube Material	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS	5052	Ti	SS
Tubesheet Material	5083	Ti/516	SS	5083	Ti/516	SS	5083	Ti/HY80	SS	5083	Ti/HY80	SS
Shell Thickness, in.	4.1/5.3	2.62/3.75		8.25	4.875		2.4/3.0	1.38/1.83		2.00	1.625	
Tubesheet Thickness, in.	1	1	1	1	1	1	1	1	1	1	1	1
Outer Tube Limit, ft.	67.15											
No. of Water Tubes	408,928											
Tube OD, in.	1.0											
Tube Wall, in.	.065	.028	.028	.065	.028	.028	.065	.028	.028	.065	.028	.028
Active Length, ft.	18.41	20.38	20.78	18.41	20.38	20.78	18.41	20.38	20.78	18.41	20.38	20.78
Cost/MW_t, $												
No. of Water Tubes	181,746											
Tube OD, in.	1.5											
Tube Wall, in.	.065	.037	.032	.065	.037	.032	.065	.028	.028	.065	.028	.028
Active Length, ft.	28.82	32.36	32.57	28.82	32.36	32.57	28.82	31.52	32.11	28.82	31.52	32.11
Cost/MW_t, $												
No. of Water Tubes	102,232											
Tube OD, in.	2.0											
Tube Wall, in.	.085	.049	.042	.085	.049	.042	.065	.034	.028	.065	.034	.028
Active Length, ft.	39.51	45.60	45.88	39.51	45.60	45.88	39.44	43.67	43.70	39.49	43.67	43.70
Cost/MW_t, $												
No. of Water Tubes	65,428											
Tube OD, in.	2.5											
Tube Wall, in.	.110	.062	.053	.110	.062	.053	.065	.043	.035	.065	.043	.035
Active Length, ft.	50.59	60.04	60.42	50.59	60.04	60.42	50.35	56.92	56.85	50.35	56.92	56.85
Cost/MW_t, $												
No. of Water Tubes	45,436											
Tube OD, in.	3.0											
Tube Wall, in.	.125	.074	.063	.125	.074	.063	.065	.051	.042	.065	.051	.042
Active Length, ft.	61.92	75.34	75.81	61.92	75.34	75.81	61.38	70.76	70.75	61.38	70.75	70.75
Cost/MW_t, $												

TABLE H-10

Evaporator, 5 MWe, Aluminum, Surface, Spherical (221 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	95,765	130,152	171,756	235,671	322,910
Tubes	316,299	298,236	380,060	487,325	553,068
Balance	156,985	156,985	156,985	156,985	156,985
Total	569,049	585,373	708,801	879,981	1,032,963
Cost					
Labor @ $15.00	$1,579,245	$ 800,505	$ 530,055	$ 410,895	$ 350,730
Tubes	625,938	603,334	769,541	954,768	1,152,865
Plate	265,388	301,494	345,178	412,289	503,890
Demister	31,800	31,800	31,800	31,800	31,800
Weld Wire	2,736	3,721	4,914	6,741	9,235
Total	$2,505,107	$1,740,854	$1,681,488	$1,816,493	$2,048,520
$/MWt	11,335	7,877	7,608	8,219	9,269

Evaporator, 5 MWe, Aluminum, Submerged, Spherical (221 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	40,012	50,576	71,936	112,925	139,271
Tubes	316,299	298,236	322,438	398,979	477,606
Balance	156,985	156,985	156,985	156,985	156,985
Total	513,296	505,797	551,359	668,889	773,862
Cost					
Labor @ $15.00	$1,578,045	$ 797,505	$ 526,500	$ 406,395	$ 345,000
Tubes	625,938	603,334	710,676	811,738	1,030,066
Plate	206,847	217,939	240,367	283,406	311,069
Demister	31,800	31,800	31,800	31,800	31,800
Weld Wire	2,052	2,790	3,686	5,055	6,926
Total	$2,444,682	$1,653,368	$1,513,029	$1,538,394	$1,724,861
$/MWt	11,062	7,481	6,846	6,961	7,805

Evaporator, 25 MWe, Aluminum, Surface, Spherical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	393,196	501,402	658,222	852,836	1,138,515
Tubes	1,139,642	1,093,394	1,409,536	1,821,805	2,075,653
Balance	520,255	520,255	520,255	520,255	520,255
Total	2,053,093	2,115,051	2,588,013	3,194,896	3,734,423
Cost					
Labor @ $15.00	$4,178,910	$2,087,655	$1,361,385	$1,037,280	$ 871,320
Tubes	2,255,293	2,211,944	2,854,009	3,569,287	4,326,675
Plate	959,124	1,072,740	1,237,401	1,441,745	1,741,709
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	12,896	16,450	21,597	27,976	37,348
Total	$7,565,223	$5,547,789	$5,633,392	$6,235,288	$7,136,052
$/MWt	6,828	5,007	5,084	5,628	6,440

TABLE H-10 (continued)

Evaporator, 25 MWe, Aluminum, Surface, Cylindrical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	320,001	474,378	633,717	799,133	967,866
Tubes	1,139,642	1,093,394	1,409,536	1,821,805	2,075,653
Balance	520,255	520,255	520,255	520,255	520,255
Total	1,979,898	2,088,027	2,563,508	3,141,193	3,563,774
Cost					
Labor @ $15.00	$4,200,990	$2,115,825	$1,398,375	$1,085,190	$ 935,280
Tubes	2,255,293	2,211,944	2,854,009	3,569,287	4,325,675
Plate	882,269	1,044,365	1,211,671	1,305,357	1,562,577
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	19,344	24,675	32,396	41,964	56,022
Total	$7,516,896	$5,555,809	$5,655,451	$6,240,798	$7,038,504
$/MWt	6,784	5,014	5,104	5,632	6,352

Evaporator, 25 MWe, Aluminum, Submerged, Spherical (1108 MWt)

	1	1 1/2	2	2 1/2	3
Weight					
Shell	159,510	204,912	260,181	342,876	449,848
Tubes	1,139,642	1,093,394	1,192,689	1,485,992	1,787,750
Balance	520,255	520,255	520,255	520,255	520,255
Total	1,819,407	1,818,561	1,973,125	2,349,123	2,757,853
Cost					
Labor @ $15.00	$4,177,910	$2,085,655	$1,358,385	$1,034,280	$ 867,320
Tubes	2,255,293	2,211,944	2,628,766	3,023,305	3,855,695
Plate	713,753	761,425	819,458	906,288	1,018,608
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	9,672	12,338	16,198	20,982	28,011
Total	$7,315,628	$5,230,362	$4,981,807	$5,143,855	$5,928,634
$/MWt	6,602	4,720	4,496	4,642	5,351

Evaporator, 25 MWe, Aluminum, Submerged, Cylindrical (1108 MWt)

	1	1 1/2	2	2 1/2	3
Weight					
Shell	109,507	170,971	233,507	301,781	371,219
Tubes	1,139,642	1,093,394	1,192,689	1,485,992	1,787,750
Balance	520,255	520,255	520,255	520,255	520,255
Total	1,769,404	1,784,620	1,946,451	2,308,028	2,679,224
Cost					
Labor @ $15.00	$4,206,720	$2,138,640	$1,435,545	$1,139,250	$1,001,100
Tubes	2,255,293	2,211,944	2,628,766	3,023,305	3,855,695
Plate	661,250	725,787	791,450	863,138	936,048
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	10,872	14,536	19,395	25,378	33,605
Total	$7,293,135	$5,249,907	$5,034,156	$5,210,071	$5,985,448
$/MWt	6,582	4,738	4,543	4,702	5,402

TABLE H-10 (continued)

Evaporator, 50 MWe, Aluminum, Surface, Spherical (2216 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	736,359	922,326	1,179,463	1,542,367	1,969,250
Tubes	2,058,583	1,986,196	2,571,254	3,331,847	3,801,442
Balance	1,080,655	1,080,655	1,080,655	1,080,655	1,080,655
Total	3,875,597	3,989,177	4,831,372	5,954,869	6,851,347
Cost					
Labor @ $15.00	$ 6,732,450	$ 3,378,810	$ 2,209,455	$ 1,685,100	$ 1,406,280
Tubes	4,073,826	4,018,090	5,206,240	6,527,767	7,924,061
Plate	1,907,865	2,103,130	2,373,124	2,754,173	3,202,400
Demister	318,000	318,000	318,000	318,000	318,000
Weld Wire	24,236	30,384	38,869	50,809	64,883
Total	$13,056,377	$ 9,848,414	$10,145,688	$11,335,849	$12,915,624
$/MWt	5,892	4,444	4,578	5,115	5,028

Evaporator, 50 MWe, Aluminum, Submerged, Spherical (2216 MWt)

Weight					
Shell	299,087	377,763	488,560	625,781	796,330
Tubes	2,058,583	1,986,196	2,173,374	2,714,082	3,270,286
Balance	1,080,655	1,080,655	1,080,655	1,080,655	1,080,655
Total	3,438,325	3,444,614	3,742,589	4,420,518	5,147,271
Cost					
Labor @ $15.00	$ 6,715,450	$ 3,363,810	$ 2,185,455	$ 1,670,100	$ 1,386,280
Tubes	4,073,826	4,018,090	4,790,261	5,521,900	7,053,126
Plate	1,448,729	1,531,339	1,647,676	1,791,758	1,907,834
Demister	318,000	318,000	318,000	318,000	318,000
Weld Wire	18,177	22,788	29,152	38,107	48,662
Total	$12,574,182	$ 9,254,027	$ 8,970,544	$ 9,339,865	$10,776,902
$/MWt	5,674	4,176	4,048	4,215	4,863

TABLE H-10 (continued)

Condenser, 25 MWe, Aluminum, Surface, Spherical (1076 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	393,196	501,402	658,222	852,836	1,138,515
Tubes	1,228,825	1,192,019	1,542,893	1,997,716	2,281,491
Balance	562,585	562,585	562,585	562,585	562,585
Total	2,184,606	2,256,006	2,763,700	3,413,137	3,982,591
Cost					
Labor @ $15.00	$4,298,760	$2,102,520	$1,339,350	$ 998,310	$ 822,975
Tubes	2,431,780	2,411,463	3,124,028	3,913,933	4,755,742
Plate	1,003,570	1,117,186	1,281,847	1,486,192	1,786,155
Demister	-	-	-	-	-
Weld Wire	12,896	16,450	21,597	27,976	37,348
Total	$7,747,006	$5,647,619	$5,766,822	$6,426,411	$7,402,220
$/MWt	7,200	5,248	5,360	5,972	6,879

Condenser, 25 MWe, Aluminum, Submerged, Spherical (1076 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	223,413	296,607	389,836	511,546	696,994
Tubes	1,228,825	1,192,019	1,179,830	1,179,317	1,181,257
Balance	540,385	540,385	540,385	540,385	540,385
Total	1,992,623	2,029,011	2,110,051	2,231,248	2,418,636
Cost					
Labor @ $15.00	$4,297,760	$2,100,520	$1,336,350	$ 995,310	$ 818,975
Tubes	2,431,780	2,411,463	2,490,779	2,503,989	2,656,875
Plate	801,988	878,842	976,732	1,104,528	1,299,248
Demister	-	-	-	-	-
Weld Wire	9,672	12,338	16,198	20,982	28,011
Total	$7,541,200	$5,403,163	$4,820,059	$4,624,809	$4,803,109
$/MWt	7,008	5,021	4,480	4,298	4,464

TABLE H-11

Evaporator, 5 MWe, Titanium, Surface, Spherical (221 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	186,844	262,339	367,627	520,910	777,777
Tubesheets (2)	86,835	86,835	86,835	86,835	86,835
Tubes	245,291	309,745	409,577	526,068	640,607
Balance	342,610	342,610	342,610	342,610	342,610
Total	861,580	1,001,529	1,206,649	1,476,423	1,847,829
Cost					
Labor @ $15.00	$1,853,164	$ 940,864	$ 624,544	$ 486,600	$ 419,405
Tubes	1,711,333	2,162,492	2,796,283	3,595,849	4,369,375
Tubesheets (2)	354,026	354,026	354,026	354,026	354,026
Plate	98,478	112,520	132,104	160,615	208,392
Demister	31,800	31,800	31,800	31,800	31,800
Weld Wire	58,400	58,000	59,400	59,976	60,800
Total	$4,107,201	$3,659,702	$3,998,157	$4,684,866	$5,382,998
$/MWt	18,585	16,560	18,091	21,198	24,357

Evaporator, 25 MWe, Titanium, Surface, Spherical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	730,694	978,470	1,392,887	1,980,837	2,904,048
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	895,305	1,159,594	1,560,750	2,032,857	2,500,478
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	3,011,408	3,523,473	4,339,046	5,399,103	6,789,935
Cost					
Labor @ $15.00	$ 4,903,732	$ 2,453,697	$ 1,604,063	$ 1,228,395	$ 1,041,930
Tubes	6,246,314	8,095,732	10,656,615	13,879,801	17,054,963
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	353,690	399,777	476,858	586,217	757,935
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	176,845	178,299	180,729	184,183	189,602
Total	$12,714,260	$12,161,183	$13,951,943	$16,912,275	$20,078,108
$/MWt	11,475	10,976	12,592	15,263	18,121

Evaporator, 25 MWe, Titanium, Surface, Cylindrical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	638,780	958,443	1,321,864	1,720,357	2,142,566
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	895,305	1,159,594	1,560,750	2,032,857	2,500,478
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	2,919,494	3,503,446	4,268,023	5,138,623	6,028,453
Cost					
Labor @ $15.00	$ 4,915,425	$ 2,469,345	$ 1,626,330	$ 1,260,060	$ 1,088,355
Tubes	6,246,314	8,095,732	10,656,615	13,879,801	17,054,963
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	336,595	396,052	463,648	697,761	616,299
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	176,845	170,299	180,729	184,183	189,602
Total	$12,708,858	$12,173,107	$13,961,001	$17,055,484	$19,982,898
$/MWt	11,470	10,987	12,600	15,393	18,035

TABLE H-11 (continued)

Evaporator, 25 MWe, Titanium, Submerged, Spherical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	272,610	376,676	515,489	704,543	1,012,827
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	895,305	1,056,633	1,417,578	1,769,039	2,221,544
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	2,553,324	2,818,718	3,318,776	3,858,991	4,619,780
Cost					
Labor @ $15.00	$ 4,887,255	$ 2,432,055	$ 1,572,525	$ 1,182,480	$ 973,905
Tubes	6,246,314	7,388,333	9,693,276	12,024,018	15,152,211
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	268,487	287,843	313,662	348,827	406,167
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	174,554	175,554	176,000	176,554	177,554
Total	$12,610,289	$11,317,464	$12,799,142	$14,765,558	$17,743,516
$/MWt	11,381	10,214	11,552	13,326	16,014

Evaporator, 25 MWe, Titanium, Submerged, Cylindrical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	246,703	387,698	542,472	716,827	896,922
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	895,305	1,056,633	1,417,578	1,769,039	2,221,544
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	2,527,417	2,829,740	3,345,459	3,871,275	4,503,875
Cost					
Labor @ $15.00	$ 4,933,170	$ 2,507,655	$ 1,682,550	$ 1,336,305	$ 1,179.270
Tubes	6,246,314	7,388,333	9,693,276	12,024,018	15,152,211
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	263,668	289,893	318,681	351,111	384,609
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	174,554	175,554	176,000	176,554	177,554
Total	$12,651,385	$11,395,114	$12,904,186	$14,921,667	$17,927,323
$/MWt	11,418	10,284	11,646	13,467	16,180

Evaporator, 50 MWe, Titanium, Surface, Spherical (2216 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	1,386,371	1,813,273	2,302,011	3,603,866	5,187,156
Tubesheets (2)	391,633	391,633	391,633	391,633	391,633
Tubes	1,624,571	2,213,323	2,875,408	3,762,222	4,643,682
Balance	2,371,894	2,371,894	2,371,894	2,371,894	2,371,894
Total	5,774,469	6,790,123	8,140,946	10,129,615	12,594,365
Cost					
Labor @ $15.00	$ 7,966,126	$ 3,986,036	$ 2,605,812	$ 1,995,528	$ 1,692,615
Tubes	11,334,215	14,824,020	19,631,111	25,687,444	31,673,076
Tubesheets (2)	1,596,688	1,596,688	1,596,688	1,596,688	1,596,688
Plate	699,037	778,441	906,546	1,111,491	1,405,983
Demister	318,000	318,000	318,000	318,000	318,000
Weld Wire	270,908	273,933	278,108	284,542	290,908
Total	$22,184,974	$21,777,118	$25,336,265	$30,993,693	$36,977,270
$/MWt	10,011	9,827	11,433	13,986	16,686

TABLE H-11 (continued)

Condenser, 25 MWe, Titanium, Surface, Spherical (1076 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	730,694	978,470	1,392,887	1,980,837	2,904,048
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	963,018	1,252,556	1,686,050	2,192,067	2,691,456
Balance	1,242,558	1,242,558	1,242,558	1,242,558	1,242,558
Total	3,150,810	3,697,124	4,536,035	5,630,002	3,052,602
Cost					
Labor @ $15.00	$ 5,073,105	$ 2,500,125	$ 1,607,265	$ 1,211,745	$ 1,014,300
Tubes	6,718,729	8,744,744	11,511,070	14,966,841	18,357,567
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	367,025	413,111	490,193	599,551	771,269
Demister	-	-	-	-	-
Weld Wire	176,845	178,299	180,729	184,183	189,602
Total	$13,210,383	$12,710,958	$14,663,936	$17,836,999	$21,207,417
$/MWt	12,277	11,813	13,628	16,577	19,709

Condenser, 25 MWe, Titanium, Submerged, Spherical (1076 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	372,337	499,300	700,250	989,212	1,370,421
Tubesheets (2)	214,540	214,540	214,540	214,540	214,540
Tubes	963,018	929,060	1,129,799	1,454,609	1,758,905
Balance	1,242,558	1,242,558	1,242,558	1,242,558	1,242,558
Total	2,792,453	2,885,458	3,287,147	3,900,919	4,586,424
Cost					
Labor @ $15.00	$ 5,072,105	$ 2,498,125	$ 1,604,265	$ 1,208,745	$ 1,010,300
Tubes	6,718,729	6,661,861	7,764,664	9,870,717	11,998,064
Tubesheets (2)	874,679	874,679	874,679	874,679	874,679
Plate	300,370	323,986	361,362	415,109	486,014
Demister	-	-	-	-	-
Weld Wire	174,554	175,554	176,000	176,554	177,554
Total	$13,140,437	$10,534,205	$10,780,870	$12,545,804	$14,546,611
$/MWt	12,212	9,790	10,019	11,660	13,519

TABLE H-12

Evaporator, 25 MWe, Stainless, Surface, Spherical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	730,694	978,470	1,392,887	1,980,837	2,904.048
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,636,111	1,812,544	2,417,374	3,082,106	3,848,087
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	3,759,559	4,183,768	5,203,415	6,455,697	8,144,889
Cost					
Labor @ $15.00	$ 4,656,184	$ 2,341,321	$ 1,539,213	$ 1,185,360	$ 1,010,971
Tubes	6,881,040	6,293,794	8,394,688	10,975,304	13,720,774
Tubesheets (2)	415,369	415,369	415,369	415,369	415,369
Plate	353,690	399.777	476,858	586,217	757,935
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	45,115	46,569	48,999	52,453	57,872
Total	$12,510,447	$ 9,655,830	$11,034,127	$13,373,703	$16,121,900
$/MWt	11,291	8,715	9,959	12,070	14,550

Evaporator, 25 MWe, Stainless, Surface, Cylindrical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	638,780	958,443	1,321,864	1,720,357	2,142,566
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,636,111	1,812,544	2,417,774	3,082,106	3,848,087
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	3,667,645	4,163,741	5,132,392	6,195,217	7,383,407
Cost					
Labor @ $15.00	$ 4,667,865	$ 2,356,965	$ 1,561,470	$ 1,502,025	$ 1,057,395
Tubes	6,881,090	6,293,794	8.394,688	10,975,304	13,720,774
Tubesheets (2)	415,369	415,369	415,369	415,369	415,369
Plate	336,595	396,052	463,648	537,768	616,299
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	45,115	46,569	48,999	52,453	57,872
Total	$12,505,034	$ 9,667,749	$11,043,174	$13,641.919	$16,026,709
$/MWt	11,286	8,725	9.967	12,312	14,465

Evaporator, 25 MW, Stainless, Submerged, Spherical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	272,610	376,676	515,489	704.543	1,012,827
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,636,111	1,563,405	2,158,871	2,799,111	3,415,279
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	3,301,475	3,334,835	4.067,114	4,896,408	5,820,860
Cost					
Labor @ $15.00	$ 4,639,695	$ 2,319,675	$ 1,507,665	$ 1,139,445	$ 942,945
Tubes	6,881,090	5,435,199	7,899,562	9,972,680	12,174,812
Tubesheets (2)	415,369	415,369	415,369	415,369	415,369
Plate	268,487	287,843	313,663	348,827	406,167
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	42,405	43,010	43,818	44,700	46,638
Total	$12,406,046	$ 8,660,096	$10,339,077	$12,080,021	$14,144,931
$/MWt	11,197	7,816	9,331	10,903	12,766

TABLE H-12 (continued)

Evaporator, 25 MWe, Stainless, Submerged, Cylindrical (1108 MWt)

Tube, Dia., in.	1	1 1/2	2	2 1/2	3
Weight					
Shell	246,703	387,698	542,472	716,827	896,922
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,636,111	1,565,405	2,158,871	2,799,111	3,415,279
Balance	1,170,869	1,170,869	1,170,869	1,170,869	1,170,869
Total	3,275,568	3,345,857	4,094,097	4,908,692	5,704,955
Cost					
Labor @ $15.00	$ 4,685,610	$ 2,395,275	$ 1,617,690	$ 1,293,230	$ 1,148,310
Tubes	6,881,090	5,435,199	7,899,562	9,972,680	12,174,812
Tubesheets (2)	415,369	415,369	415,369	415,369	415,369
Plate	263,668	289,893	318,681	351,111	384,609
Demister	159,000	159,000	159,000	159,000	159,000
Weld Wire	42,405	43,010	43,818	44,700	46,638
Total	$12,447,142	$ 8,737,746	$10,454,120	$12,236,130	$14,328,738
$/MWt	11,234	7,886	9,435	11,043	12,932

Condenser, 25 MWe, Stainless, Surface, Spherical (1076 MWt)

	1	1 1/2	2	2 1/2	3
Weight					
Shell	730,694	978,470	1,392,887	1,980,837	2,904,048
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,755,311	1,957,909	2,612,341	3,324,786	4,143,287
Balance	1,242,558	1,242,558	1,242,558	1,242,558	1,242,558
Total	3,950,448	4,400,822	5,469,671	6,770,066	8,511,778
Cost					
Labor @ $15.00	$ 4,813,380	$ 2,382,330	$ 1,539,360	$ 1,166,760	$ 981,990
Tubes	7,382,413	6,798,553	9,070,234	11,837,831	14,773,340
Tubesheets (2)	415,369	415,369	415,369	415,369	415.369
Plate	367,025	413,111	490,193	599,551	771,269
Demister	-	-	-	-	-
Weld Wire	45,115	46,569	48,999	52,453	57,872
Total	$13,023,302	$10,055,932	$11,564,155	$14,071,964	$16,999,840
$/MWt	12,103	9,346	10,747	13,078	15,799

Condenser, 25 MWe, Stainless, Submerged, Spherical (1076 MWt)

	1	1 1/2	2	2 1/2	3
Weight					
Shell	372,337	449,300	700,250	989,212	1,370,421
Tubesheets (2)	221,885	221,885	221,885	221,885	221,885
Tubes	1,755,311	1,694,732	1,673,979	2,127,179	2,604,432
Balance	1,242,558	1,242,558	1,242,558	1,242,558	1,242,558
Total	3,592,091	3,658,475	3,838,672	4,580,834	5,439,296
Cost					
Labor @ $15.00	$ 4,812,380	$ 2,380,330	$ 1,536,360	$ 1,163,760	$ 977,990
Tubes	7,382,413	5,884,709	6,126,762	7,575,048	9,271,778
Tubesheets (2)	415,369	415,369	415,369	415,369	415,369
Plate	300,370	323,986	361,362	415,109	486,014
Demister	-	-	-	-	-
Weld Wire	45,115	46,569	48,999	52,453	57,872
Total	$12,955,647	$ 9,050,963	$ 8,488,852	$ 9,621,739	$11,209,023
$/MWt	12,041	8,412	7,889	8,942	10,417

TABLE H-13

PLATE SIZE AND COST

Vendor	Material	Thickness	Max.Size	Price/ft^2 $
Uddeholm	Steel 904L	3/4"	96" x 240"	92.16
		1"	96" x 240"	125.86
		2"	96" x 240"	251.23
		3"	96" x 240"	376.80
		4"	96" x 240"	510.96
		5"	-	-
		6"	-	-
G. O. Carlson	Nitronic 50	3/4"	96" x 360"	60.15
		1"	96" x 300"	79.89
		2"	96" x 180"	156.91
		3"	60" x 260"	236.35
		4"	60" x 190"	314.22
		5"	60" x 150"	390.97
		6"	60" x 120"	468.75
Lukens	Steel A516	3/4"	180" x 475"	5.70
		1"	180" x 475"	7.60
		2"	180" x 480"	15.21
		3"	180" x 480"	22.82
		4"	180" x 440"	30.10
		5"	180" x 300"	37.63
		6"	180" x 260"	45.16
Lukens	Steel HY80	3/4"	180" x 475"	17.40
		1"	180" x 475"	23.20
		2"	186" x 400"	46.19
		3"	186" x 400"	68.85
		4"	186" x 400"	97.69
		5"	186" x 300"	122.10
		6"	186" x 245"	146.53
Alcoa	Aluminum 5083	3/4"	188" x 960"	11.55
		1"	190" x 960"	15.11
		2"	180" x 750"	29.14
		3"	130" x 690"	41.85
		4"	120" x 560"	55.79
		5"	110" x 490"	69.79
		6"	100" x 450"	80.68

Note: 904L steel was not used in cost estimate.

TABLE H-14

TUBING COST

Vendor	Material	Tube OD	Tube Wall	Price/ft $
Allegheny Ludlum (welded)	Stainless AL-6X	1"	.028"	1.30
		1 1/2"	.032"	1.84
			.028"	-
		2"	.042"	3.22
			.038"	3.07
			.028"	-
		2 1/2"	.053"	-
			.048"	-
			.035"	-
			.028"	-
		3"	.063"	7.44
			.057"	6.73
			.042"	-
			.028"	-
Alcoa (drawn)	Aluminum 5052	1"	.065"	.47
		1 1/2"	.065"	.72
		2"	.085"	1.26
			.072"	1.16
			.065"	1.00
		2 1/2"	.110"	1.97
			.090"	1.67
			.065"	1.26
		3"	.125"	2.86
			.108"	2.55
			.065"	1.60
Timet (welded)	Titanium	1"	.028"	1.20
		1 1/2"	.037"	2.38
			.034"	2.19
			.028"	1.85
		2"	.049"	4.11
			.045"	3.78
			.034"	2.87
			.028"	2.46
		2 1/2"	.062"	6.50
			.055"	5.74
			.043"	4.48
			.028"	3.26

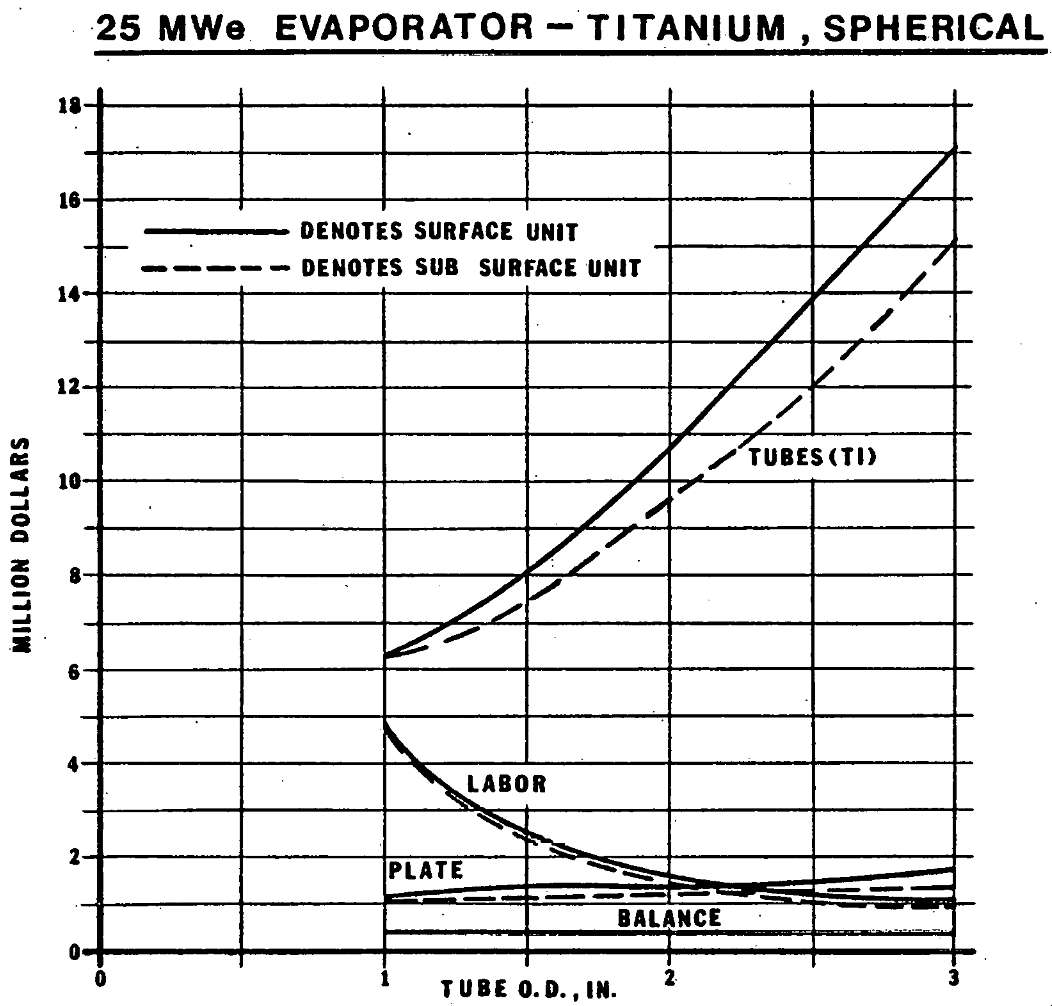
TABLE H-14 (continued)

TUBING COST (cont'd.)

Vendor	Material	Tube OD	Tube Wall	Price/Ft $
Timet (cont'd.) (welded)	Titanium	3"	.074"	-
			.067"	-
			.055"	-
			.028"	-
Wolverine (seamless)	Titanium	1"	.028"	2.74
		1 1/2"	.037"	-
			.034"	-
			.028"	-
		2"	.049"	8.27
			.045"	8.00
			.034"	-
			.028"	-
		2 1/2"	.062"	12.09
			.055"	-
			.043"	-
			.028"	-
		3"	.074"	-
			.067"	-
			.055"	-
			.028"	-

Note: Wolverine titanium was not used in cost estimate.
- indicates vendor cannot supply this size.

FIGURE H-2



25 MWe EVAPORATOR – TITANIUM, SPHERICAL
DENOTES SURFACE UNIT
DENOTES SUB SURFACE UNIT
TUBES (TI)
LABOR
PLATE
BALANCE
MILLION DOLLARS
TUBE O.D., IN.
0
2
4
6
8
10
12
14
16
18
0
1
2
3

FIGURE H-3

25 MWe EVAPORATOR – STAINLESS, SPHERICAL

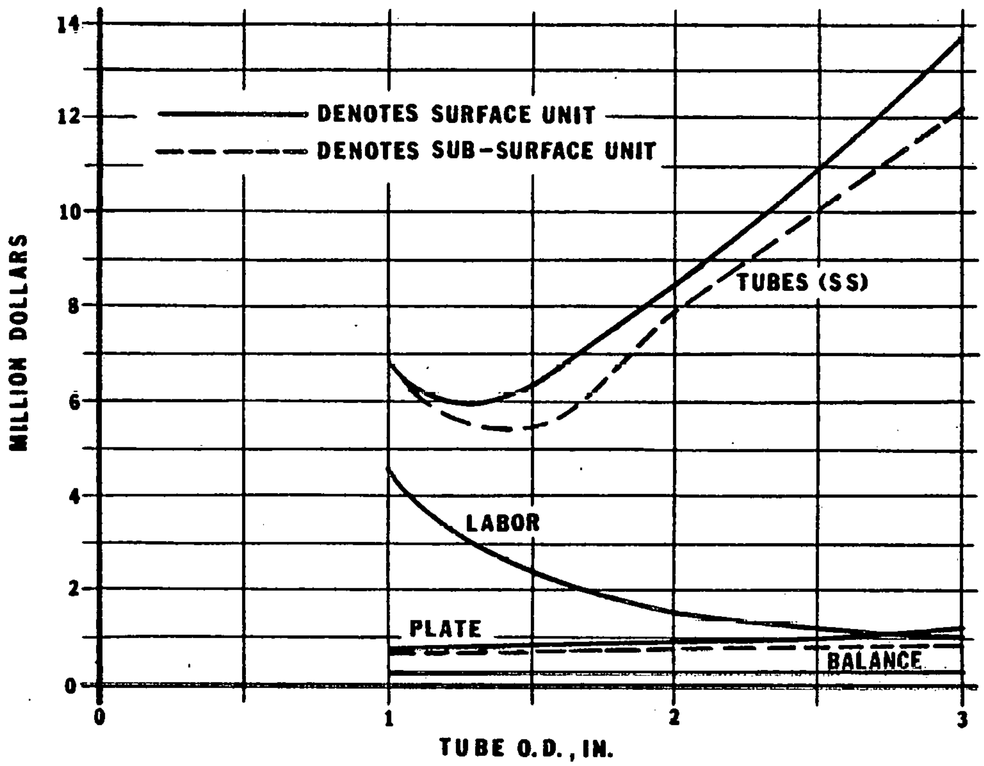
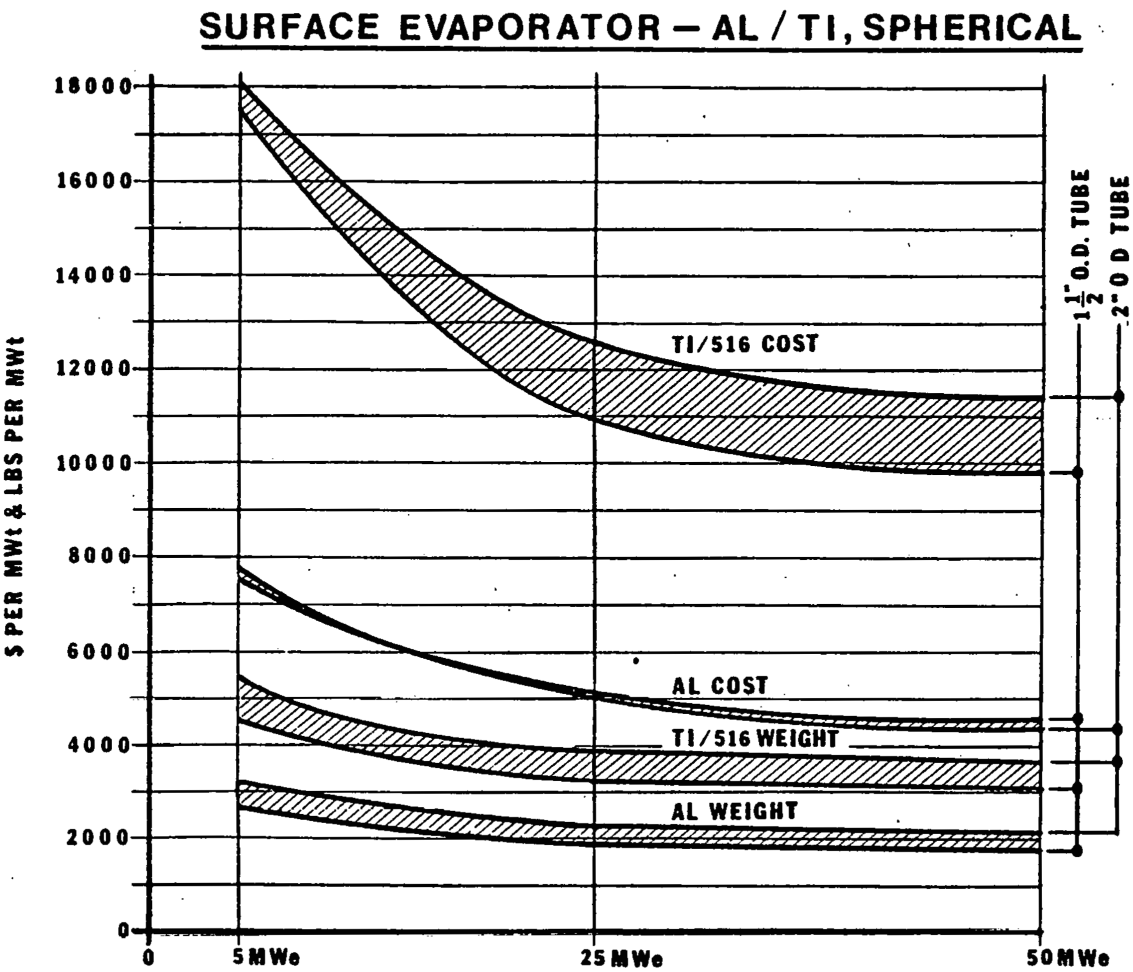
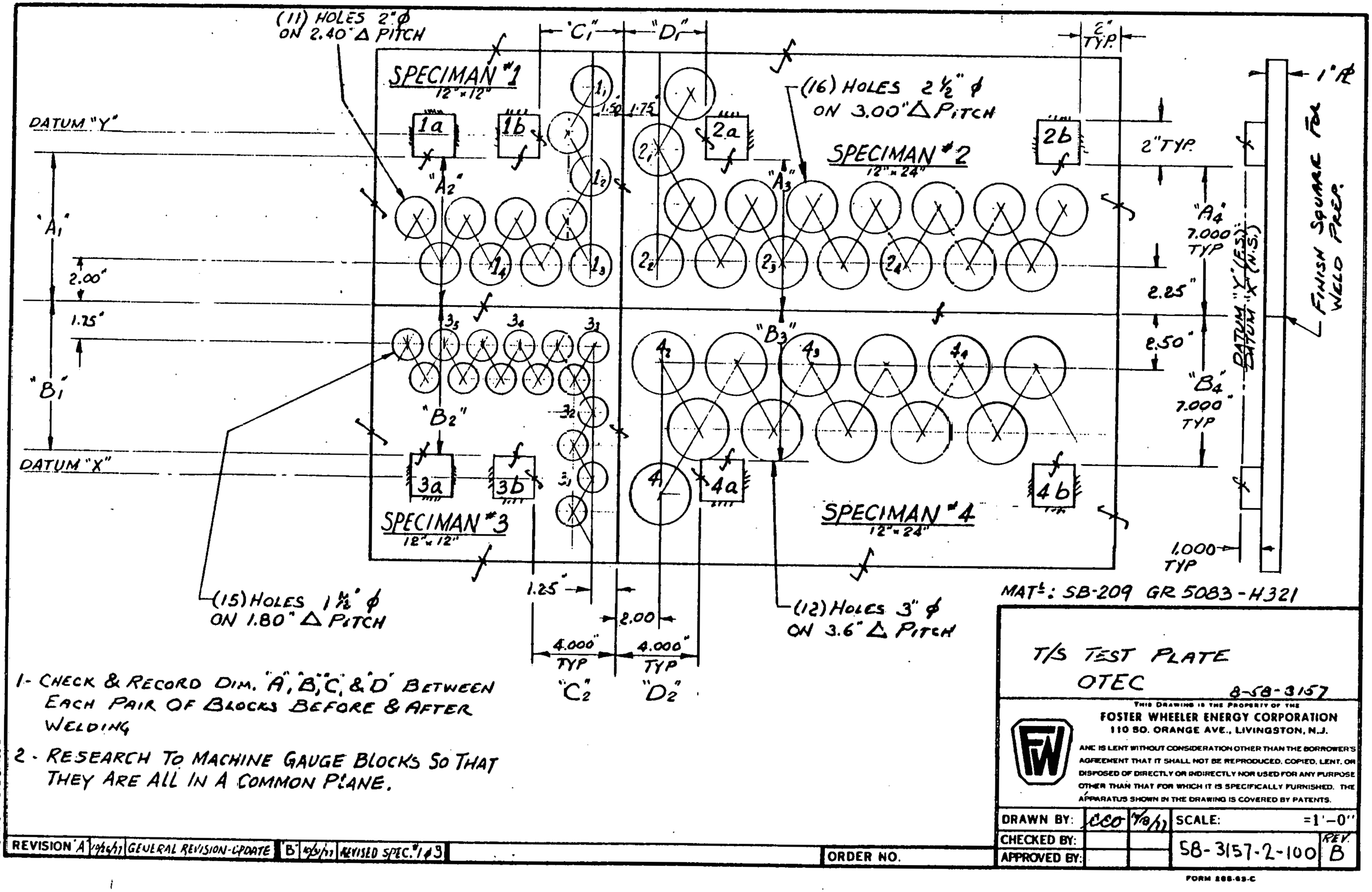


FIGURE H-4



SURFACE EVAPORATOR – AL / TI, SPHERICAL
$ PER MWt & LBS PER MWt
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0
5 MWe
25 MWe
50 MWe
TI/516 COST
AL COST
TI/516 WEIGHT
AL WEIGHT
1 1/2" O.D. TUBE
2" O D TUBE


(11) HOLES 2"ϕ ON 2.40" Δ PITCH
"C1"
"D1"
2" TYP.
SPECIMAN #1 12"×12"
(16) HOLES 2½" ϕ ON 3.00" Δ PITCH
1" R
DATUM "Y"
1a
1b
2a
2b
2" TYP.
SPECIMAN #2 12"×24"
"A2"
"A3"
"A1"
"A4" 7.000 TYP
2.00"
2.25"
1.25"
2.50"
"B3"
"B1"
"B2"
"B4" 7.000 TYP
DATUM "Y" (F.S.)
DATUM "X" (N.S.)
FINISH SQUARE FOR WELD PREP.
DATUM "X"
3a
3b
4a
4b
SPECIMAN #3 12"×12"
SPECIMAN #4 12"×24"
1.000 TYP
1.25"
2.00
(15) HOLES 1½" ϕ ON 1.80" Δ PITCH
(12) HOLES 3" ϕ ON 3.6" Δ PITCH
MAT'L: SB-209 GR 5083-H321
4.000" TYP "C2"
4.000" TYP "D2"
1- CHECK & RECORD DIM. "A", "B", "C", & "D" BETWEEN EACH PAIR OF BLOCKS BEFORE & AFTER WELDING
2 - RESEARCH TO MACHINE GAUGE BLOCKS SO THAT THEY ARE ALL IN A COMMON PLANE.
T/S TEST PLATE
OTEC
8-58-3157
THIS DRAWING IS THE PROPERTY OF THE
FOSTER WHEELER ENERGY CORPORATION
110 SO. ORANGE AVE., LIVINGSTON, N.J.
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DRAWN BY: CCO 7/8/77
SCALE: =1'-0"
CHECKED BY:
APPROVED BY:
58-3157-2-100
REV. B
REVISION A 10/26/77 GENERAL REVISION-UPDATE
B 10/31/77 REVISED SPEC. #1 & 3
ORDER NO.
FORM 268-65-C
BRUNING 44-512 27405-2

FIGURE H-5

TABLE H-15 ELECTRON BEAM WELDING TEST DATA

HOLE DIAMETER, IN.

Hole No.	Before Welding		After Welding	
	A	B	A	B
11	2.017	2.017	2.014	2.024
12	2.017	2.017	2.017	2.015
13	2.017	2.017	2.014	2.017
14	2.017	2.017	2.016	2.017
21	2.5105	2.5115	2.511	2.508
22	2.509	2.509	2.508	2.512
23	2.5105	2.512	2.509	2.512
24	2.511	2.510	2.510	2.5115
31	1.497	1.497	1.499	1.493
32	1.497	1.497	1.498	1.496
33	1.497	1.497	1.496	1.496
34	1.4975	1.4975	1.4965	1.4965
35	1.4975	1.4975	1.497	1.4975
41	2.998	2.998	3.002	2.993
42	2.998	2.998	2.999	2.996
43	3.001	3.001	3.000	2.999
44	3.0015	3.002	3.001	3.0015
Gage Blocks				
1a - 3a	13.995		13.973	
1b - 3b	13.997		13.973	
2a - 4a	14.004		13.973	
2b - 4b	14.003		13.976	
1b - 2a	7.995		7.975	
3b - 4a	8.005		7.987	

Note: A = along 3 ft long axis

B = along 2 ft long axis

TABLE H-16 TUBE-TO-TUBESHEET WELDING DATA

Aluminum Tubes, SB-210, Alloy 5052, Temper H32, 31,000 psi

Hole & Tube No.	Hole I.D.	Tube O.D.	Tube Wall	Tube I.D.	Exp'd I.D.	Pull Out P.S.I.	Comments on Rolling Length
1	1.014	0.999	.065	.869		21,477	
2	1.011	1.000	.065	.868	.914	25,667	1/2" Beyond T.S.
3	1.013	0.999	.066	.869	.896	*30,906	1/4" Beyond T.S.
4	1.016			.867		21,477	
5	1.011			.869		20,953	
6	1.012			.869	.904	23,577	1/2" Beyond T.S.
7	1.012			.869	.895	*30,906	Across Weld
8	1.010			.868		20,429	
9	1.010			.869	.903	23,577	1/2" Beyond T.S.
10	1.014		.065	.868	.905	29,334	1/4" Beyond T.S.
11	1.015		.066	.868		15,715	
12	1.012		.066	.868		20,429	
13	1.021		.065	.867	.906	28,810	1/2" Beyond T.S.
14	1.013		.066	.868	.900	*30,906	Across Weld

Titanium Tubes, SB-338, Ti Grade 2, 50,000 psi

Hole & Tube No.	Hole I.D.	Tube O.D.	Tube Wall	Tube I.D.	Exp'd I.D.	Pull Out P.S.I.	Comments on Rolling Length
1	1.013	1.000	.064	.872		60,207	
2	1.013	1.000	.065	.872	.891	65,134	1/2" Beyond T.S.
3	1.013	1.000	.064	.872	.885	58,018	1/4" Beyond T.S.
4	1.001	0.999	.065	.871		63,492	
5	1.009	1.000	.064	.872		54,734	
6	1.008		.065	.872	.886	67,323	1/2" Beyond T.S.
7	1.008		.064	.872	.887	68,965	Across Weld
8	1.009		.065	.872		62,944	
9	1.009		.065	.872	.893	65,134	1/2" Beyond T.S.
10	1.009		.064	.871	.889	66,776	1/4" Beyond T.S.
11	1.008		.065	.871		48,713	
12	1.008		.065	.872		62,944	
13	1.009		.065	.872	.890	**62,944	1/2" Beyond T.S.
14	1.042		.065	.871	.903	72,249	Across Weld

* Fracture in tube, all other fractures in weld
**Tube end plug weld fractured before tubesheet weld

TABLE H-17 COST COMPARISONS FOR ALTERNATE FABRICATION METHODS

Operation	Material	Size Limits: Thickness	Size Limits: Size L x W	Process	Production Cost $/Foot	Inspection Need: % Insp.	Inspection Need: $/Foot	Rework: % Rej.	Rework: $/Foot	Overall Cost $/Foot	Investment: Production	Investment: Development	Comments
Cutting and Edge Preparation Straight & Contour Cuts	Aluminum	3"	12' x 22'	Plasma Cut	.40	5	.02	5	.02	.44	$175K		Minor Dressing Required
			12' x 22'	Abrasive Cut	.84	5	.04	3	.02	.90	40K		For Square Edges without Machining
			12' Width	Shear 1" Maximum	.20	5	.01	3	.01	.22	60K		To 1" Thick Only
			12' Width	Machine Edge Preps.	1.33	5	.07	3	.04	1.44	100K		"J" Grooves Possible
			-	Laser Cut	.67	10	.07	10	.07	.81	-		Not Recommended
			-	Band Saw Cut	.80	5	.04	3	.02	.86	10K		Small Cuts Only
			-	Circular Saw Cut	1.0	5	.05	3	.03	1.08	5K		Small Cuts Only
Cutting and Edge Preparation Straight & Contour Cuts	Steel	2"	12' x 22'	Flame Cut	.24	5	.01	3	.01	.26	$125K		Minor Dressing Required
			12' x 22'	Plasma Cut	.20	5	.01	3	.01	.22	175K		Minor Dressing Required
			12' x 22'	Abrasive Cut	.83	5	.04	3	.02	.89	40K		Square Cut No Machining
			12' Width	Shear	.10	5	.01	3	.01	.12	60K		To 1" Thickness Only
			-	Laser Cut	.50	10	.05	10	.05	.60	-		Not Recommended
Welding Seams and Nozzles	Aluminum	3"	22' and 70' Circle	GMAW (MIG)	10.00	5	.50	10	1.00	11.50	$500K		30 Passes @ 20"/min.
				GTAW (TIG)	33.33	5	1.67	10	3.33	38.33	500K		
Welding Seams and Nozzles	Steel	2"	22' and 70' Circle	SAW (Sub Arc)	8.33	5	.41	10	.83	9.57	$500K		20 Passes @ 18"/min.
				Flux Core	15.87	5	.79	10	1.59	18.25	400K		
				Manual (Nozzles)	33.33	5	1.67	10	3.33	38.33	250K		
Welding TubeSheet and Support Plates	Aluminum	1"	27' Long	GMAW (MIG)	5.00	5	.25	10	.50	5.75	$250K		15 Passes @ 18"/min.
			54' Long	E. B. Welding	1.50	10	.15	50	.75	2.40	950K		1 Pass, Machined Edges
Welding Tubesheet and Support Plates	Steel	1"	27' Long	SAW (Sub Arc)	1.50	5	.07	10	.15	1.72	$250K		6 Passes @ 24"/min.
			54' Long	E. B. Welding	.50	10	.05	50	.25	.80	950K		1 Pass, Machined Edges
					Per Segment		Per Segment		Per Segment	Per Segment			
Forming Segments	Aluminum and Steel	3" 2"	12' x 22'	Crimp and Roll	120.00	5	6.00	3	3.60	129.60	$450K		$15/hr. for Cylinders Only
			12' x 22'	Press/Die Forming	62.50	5	3.17	3	1.90	67.57	2M		$50/hr.
			12' x 22'	Explosive Forming	75.00	5	3.75	3	2.25	81.00	500K	$500K	$15/hr. Development Required
				Hydraulic Forming	-	-	-	-	-	-	-		$15/hr. Not Recommended

TABLE H-17 COST COMPARISONS FOR ALTERNATE FABRICATION METHODS

Operation	Material	Size Limits: Thickness	Size Limits: Size L x W	Process	Production Cost $/Hole	Inspection Need: % Insp.	Inspection Need: $/Hole	Rework: Rej.	Rework: $/Hole	Overall Cost $/Hole	Investment: Production	Investment: Development	Comments
Tubesheet and Support Plate Drilling	Aluminum 1" Thick	Stack 4 Plates	7' x 27'	LAHR Gun Drill	.036	5	.002	3	.001	.04	$900K		4 Drills @ $20/hr.- 7' x 10' Part Size Limit
		Stack 4 Plates		Moline Drill and Ream	.028	5	.001	3	.001	.03	1.2M		20 Drills, $40/hr.-15' x 27' Part Size Limit
		Stack 4 Plates		Movable Drill and Ream	.40	5	.02	3	.01	.43	100K		2 Drills, $20/hr.
		1 Plate		Punch and Ream	1.01	5	.05	3	.03	1.09	600K		One Plate and One Punch, $20/hr.
		1 Plate		Electro Discharge	-	-	-	-	-	-	-		Not Practical for Production Rate Required--Too Much Power Required
		1 Plate		Movable Radial Drill (Drill and Ream Individual Holes)	2.22	10	.22	10	.22	2.66	50K		One Drill
Tubesheet and Support Plate Drilling	Steel 1" Thick	Stack 4 Plates	7' x 27'	LAHR Gun Drill	.15	5	.01	3	.005	.16	$900K		4 Drills @ $20/hr.-7' x 10' Part Size Limit
		Stack 4 Plates		Moline Drill and Ream	.05	5	.002	3	.001	.05	1.2M		20 Drills, $40/hr.-15' x 27' Part Size Limit
		Stack 4 Plates		Movable Drill and Ream	1.12	5	.06	3	.04	1.22	100K		2 Drills, $20/hr.
		1 Plate		Punch and Ream	3.36	5	.17	3	.10	3.63	600K		One Plate and One Punch, $20/hr. (Check Ligament Diameter Ratio)
		1 Plate		Electro Discharge	-	-	-	-	-	-	-		Not Practical for Production Rate Required--Too Much Power Required
				Magnetic Base Drill and Ream	5.60	10	.56	10	.56	6.72	5K		One Drill
Tube-to-Tubesheet Joint	Aluminum + Stainless + Titanium			Roller Expansion	1.02	10	.10	10	.10	1.22	$ 10K		Not Leak Tight
				Kinetic Bonding	1.02	10	.10	10	.10	1.22	30K	$ 50K	Development Required, Not Service Tested Yet
				Magne Forming	1.02	10	.10	20	.20	1.32	30K	50K	Development Required, Not Service Tested Yet (Not Used for Titanium and Steel)
				GTAW and Kinetic Expansion	1.88	10	.19	10	.19	2.26	25K	5K	Multiple Welding Setups Possible
				E. B. Welding and Kinetic Expansion	6.26	20	1.26	20	1.26	8.78	800K	15K	State of Art
				Laser Welding and Kinetic Expansion	6.26	20	1.26	20	1.26	8.78	400K	20K	State of Art
				Adhesive Bonding	2.50	15	.38	50	1.25	4.13	25K	10K	Novel Application
Tube Removal Cost	All	25 MW		Remove 1 Tube, Trepan Weld, Cut Tube, Pull Tube, Recondition Holes	37.50								Welded and Expanded Tube. Cost After Unit Staged in Shop @ $15/hr.

Appendix I
ROTATING MACHINERY

This appendix presents additional data on the power module rotating machinery. The material is presented in two subsections, as follows:

I-1 Seawater Pumps

I-2 Turbine-Generators

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Appendix I-1
SEAWATER PUMPS

This section presents a copy of the specification submitted to the various vendors to obtain data on a family of pumps for application to the Power System Development project. Copies of the replies received from KSB, Hitachi, Mitsubishi, Allis-Chalmers, Worthington, and Pleuger are included for information purposes.

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POWER SYSTEM DEVELOPMENT

POWER SYSTEM CIRCULATING WATER PUMPS SPECIFICATION

Preliminary Performance
Description

Prepared: M. I. Leitner

M. I. Leitner

APPROVED: C M Robidart

C. M. Robidart

OTEC POWER SYSTEM CIRCULATING WATER PUMPS SPECIFICATION

BACKGROUND:

The Power System Development Project has, as its basic objective, the design of a 25 MWe (nominal) power system module to be utilized in the development of a 100 MWe Plant and the design of a 5 MWe scaled cycle plant. Several candidate platforms are included in the study which result in modules which may be categorized as "surface" or "submerged" and the design values for component sizing are different. Part of the module trade studies will include the effect of these differences on component costs and sizes.

GENERAL:

The OTEC power cycle is supplied with warm and cold water via sea water pumps. The configuration consists of a module of 4 sea water pumps mounted in parallel. There is one pump module each for the evaporator and condenser. Because the seawater pressure and flow requirements are nearly identical, sea water pumps modules will be designed for the more stringent requirement for the condenser. A small efficiency penalty will be incurred on the evaporator pump module because of this design philosophy.

A trade study will be undertaken on the relative merits of variable pitch-fixed rpm, fixed pitch-variable rpm and fixed pitch-fixed rpm. Part of the trade study will consider a requirement for obtaining reverse flow equal to or greater than 40 percent of the design forward flow. A performance map for evaluating off-design conditions is desired.

The pump shall operate with the rotating shaft horizontal. The bearing system shall be designed to allow transport at sea in the vertical position. The pumps shall be axial flow with a streamlined "pod" as indicated in Sketch CWP-1. The pump pod shall include a drive motor, reduction gear, necessary accessories, and it shall carry the impeller.

The pump pod will be horizontally mounted on struts in a pump shroud having an inlet bellmouth section and an outlet diffuser section. Straightening vanes shall be part of the pod/shroud assembly and may double as pod mounting struts. Exterior water pressure may vary from 300 feet to 40 feet. The interior of the pod will be pressurized to prevent leakage. Dynamic seals at the propellor shaft and static seals at any other pod penetrations shall prevent in-leakage with the head of water external to the pod as given above.

PUMP DESIGN PARAMETERS:

Design parameters are tabulated in Table I for the 5 and 25 MW(e) net power output cycle plant. Included in the Table is a list of desired physical characteristics and performance data.

SCOPE OF SUPPLY:

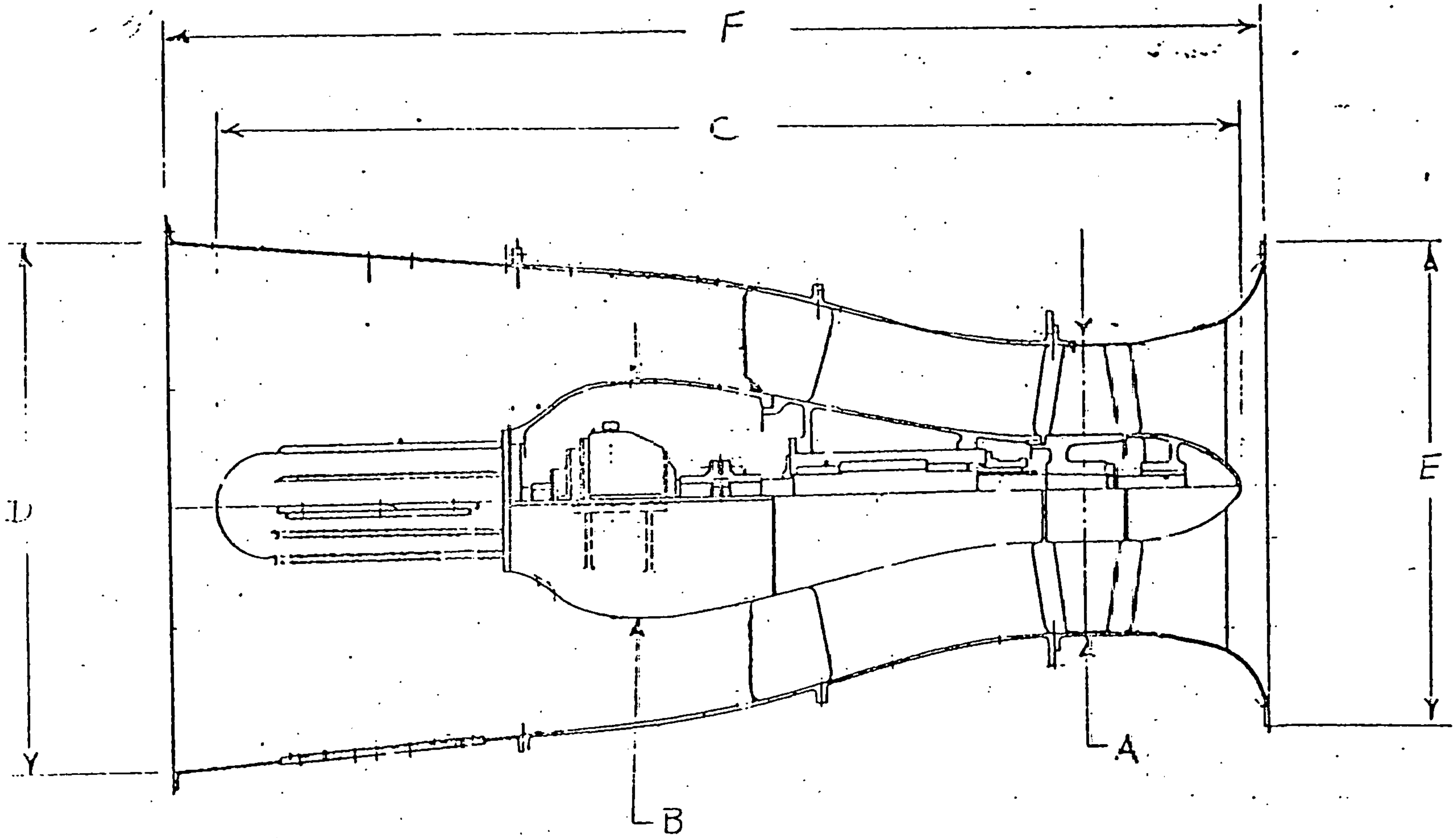
Each warm and cold circulating water pump package shall include: pump pod with impeller, drive motor, reduction gear, bearings, lubrication system, and seals; pump shroud with inlet bellmouth section, outlet diffuser section, inlet strainghtening vanes, outlet guide vanes, and pod mounting struts.

MATERIALS:

The pump impellors shall be of cast steel with any epoxy or other synthetic coating for water service. Other pump materials shall be of a material suitable for seawater service.

SAFETY:

Each pump and motor shall have adequate "built-in" instrumentation to preclude the possibility of pump or motor loss due to such factors as bearing or gear failure or motor winding burn-out. The incipient warning of problems of this nature should permit rapid shutdown (automatic or manual) to prevent major equipment damage.



TYPICAL SEA WATER CIRCULATING PUMP

TABLE I

PUMP DESIGN AND PERFORMANCE PARAMETER

Parameter	5MWe	25MWe	
	(Surface)	(Surface)	(Submerged)
Number of Pumps	1	4	4
Design Flow (cfs)	650	812	1000
Head (ft)	10	10	6
Power Supply	← 4160v, 60H, 3-phase →		

DATA REQUIRED

Pump Design Efficiency
Overall Design Efficiency (Pump, Gear & Motor)
Design Power Required
Pump Dimension (See Sketch)
- A - Impeller Diameter
- B - Pod Diameter
- C - Pod Length
- D - Diffuser Outlet Diameter
- E - Bellmouth Inlet Diameter
- F - Overall Length

Estimated Weight
NPSH or Submergence Required

Blank

KSB TECHNICAL SALES CORPORATION

1 HUNTINGTON QUADRANGLE, HUNTINGTON STATION, NEW YORK 11746 • PHONE (516) 293-7878 • TELEX 14 3111

REPLY TO: 2555 E. CHAPMAN AVE.
SUITE 500
FULLERTON, CALIF. 92631
PHONE: (714) 871-3200

Nov. 18, 1977

Lockheed Missiles & Space
Co. Inc.

1111 Lockheed Way

Sunnyvale, Calif. 94 088
Attn. Mr. B. Jean Bartel,
Subcontract Administrator

Re.: Request for Information - Seawater Pumps
Assemblies -
Your # 50-43/7902
our TSC-LA 02/145/77

Gentlemen:

We confirm the receipt of your letter dated Oct. 11, 1977. Some of the questions mentioned herein, we have already discussed with Mr. M. Leitner on Nov. 14 and 15. In the following, we summarize the requested information.

Design Flow	CFS	650	812	1000
Pump Model	PUZ	3400-2330	3800-2580	4200-2790
TDH	FT	10	10	6
Pump stage efficiency	%	86	86	86
Pump efficiency	%	79	79	76
overall effic.	%	71	71	69
Motor rating	KW	820	1000	780
Pump speed	RPM	188.5	170	158
Motor speed	RPM	1180	1180	1180
NPSHR at centerline shaft	FT	32	32	32
Pump weight	LBS	59000	78500	88000
Motor weight	LBS	15500	15500	17000
Gear weight	LBS	4000	5500	7500
Unit price for pumps	$	760,000	835,000	935,000
Unit price for motor and gear	$	260,000	270,000	285,000

Above prices are estimated, fob German seaport, seaworthy crated for delivery by the end of 1979.

Lockheed Missiles & Space
Co. Inc.
Sunnyvale, Calif. 94 088

Dimensions: Pls. refer to your drawing on page PSD-2.1 of your RFQ.

Design Flow:	650	812	1000
A	92	102	110
B	67	75	79
C	284	319	312
D	134	150	166
E	138	154	166
F	292	319	343

All dimensions are in inches.

The delivery of any size and number of pumps will be approx. 75 weeks.

We trust that this information will answer your questions.

We confirm our continued interest in this project and ask you to consult us if further data are needed.

Thank you for this opportunity to be at your service.

Very truly yours,

KSB TECHNICAL SALES CORPORATION

Hans R. Niemes
Western Regional Sales Manager

HRN:NS

RE: SEAWATER PUMP ASSEMBLIES
WITH REFERENCE TO YOUR LETTER NO. 50-43/7902 DATED
OCTOBER 11, 1977, WE ARE PLEASED TO INFORM YOU AS FOLLOWS,
AA) TECHNICAL DATA

	650 CFS PUMP	812 CFS PUMP	1000 CFS PUMP
PUMP DESIGN EFF	86.2	86.5	86.2
MOTOR EFF	93.8	94.2	93.7
GEAR BOX EFF	97.0	97.0	97.0
OVERALL EFF	78.4	79.0	78.3
REQ POWER (KW)	656	815	605
DIMENSION A	2430	2710	3240
B	2300	2500	2700
C	8200	9300	8400
D	2800	3120	3470
E	3890	4340	5180
F	9500	10600	10800
PUMP WT (TON)	72	96	120
IM WT (TON)	13	15	12.5
G.BOX WT (TON)	4.5	6	8.5
NPSH REQ (M)	8.2	8.2	7.4
MOTOR (KW)	780	970	720
MOTOR POLES	10	10	10
PUMP SPEED (RPM)	132	118	88

BB) PRICE INCLUDNG PUMP, GEAR AND INDUCTION MOTOR : ON
FOB SUNNYVALE (DUTY PAID) IN US DOLLARS

ITEM	DESCRIPTION	QTY	UNIT	TOTAL
1	650 CFS PUMP	1	707,950	707,950
2	812 CFS PUMP	4	903,900	3,615,600
3	1000 CFS PUMP	4	1,098,240	4,392,960
	GRAND TOTAL FOB SUNNYVALE		USD	8,716,510

CC) REMARKS
1) PRICES QUOTED ARE SUBJECT TO PRICE VARIATION
AND VALID UNTIL FEBRUARY 1, 1978. AND THEY
ARE BASED ON FOB LOCKHEED MISSILES AND
SPACE COMPANY, INC., SUNNYVALE, CALIF., USD.,
INCLUDING OCEAN FREIGHT, INSURANCE, IMPORT DUTY,
CUSTOM CLEARANCE AND INLAND TRANSPORTATION.
2) DELIVERY
SHIPMENT IN TERMS OF FOB YOKOHAMA, JAPAN
WILL BE MADE WITHIN 13 MONTHS AFTER RECEIPT OF ORDER.
3) TECHNICAL COMMENTS
THE PUMPS QUOTED ARE OF FIXED BLADE- CONSTANT
SPEED TYPE.
WE WOULD LIKE TO SUBMIT ADDITIONAL QUOTATION
FOR FIXED BLADE-VARIABLE SPEED PUMPS AND
VARIABLE PITCH BLADE-FIXED SPEED PUMPS LATER

BEST REGARDS
YAMAMOTO HITACHI AMERICA LTD

1059 EST

LMSC SU CAL

P
LMSC SV CAL

WU INFOMASTER 1-015346N314 11/10/77
TLX MITSUBIS B SFO
01 SAN FRANCISCO CALIF 11/10/77
TWX 9103399221 LMSC SV CAL

ATTN: MRS. JEAN BARTEL, SUBCONTRACT ADMINISTRATOR,
DEVELOPMENT PROGRAMS MATERIALS
SUBJECT: COMPOSITE OTEC PROGRAM- REQUEST FOR INFORMATION
SEAWATER PUMP ASSEMBLIES
FILE NO. 50-43/7902
IN RESPONSE TO YOUR OCTOBER 11, 1977 RFI WE ARE PLEASED TO SUBMIT OUR BUDGETARY ESTIMATE FOR YOUR CONSIDERATION.
111. PRICE:

ITEM	DESCRIPTION	Q'TY	TOTAL PRICE (FOB JAPAN)
1.	SEAWATER CIRCULATING PUMP FOR 5 MW PLANT (SURFACE)	1	YEN 629,000,000 (629,000,000)
2.	DITTO FOR 25 MW PLANT (SURFACE)	4	2,665,200,000 (2,665,200,000)
3.	DITTO FOR 25 MW PLANT (SUBMERGED)	4	3,294,800,000 (3,294,800,000)
		TOTAL	6,589,000,000 (6,589,000,000)

222. TERMS OF PAYMENT:
100 PCT AT SHIPMENT
333. TERMS OF DELIVERY: FOB JAPANESE PORT
444. TIME OF DELIVERY: FOB 25 (25) MONTHS
AFTER RCPT OF P/O
555. VALIDITY: UNTIL DEC 31, 1977
666. REMARKS.
(1) ABOVE PRICE INCLUDES MOTOR AND REDUCTION GEAR AS WELL
(2) ABOVE PRICES OF ITEM 2/3 HHAVE BEEN ESTIMATED ONE THE ASSUMPTION THAT WE RCV BLANKET ORDER FOR 4 SETS EACH.
UNDER OUR TRANSMITTAL SF-G-1312 OF NOV. 7, 1977, WE HAD FORWARDED THE SUPPORTING SPECS./DATA FOR OUR BUDGETARY ESTIMATE. IF THERE ARE ANY QUESTIONS OR COMMENTS, PLEASE FEEL FREE TO CONTACT US.

REGARDS,
BILL TANAKA (34-334)
MITSUBISHI SFO

1247 EST

LMSC SV CAL

[redacted]

PROPOSAL SPECIFICATION

OF

SEAWATER PUMP

FOR

Ocean Thermal Energy Conversion

(OTEC) Program

(SPEC. NO. 36.E-1949-1)

MITSUBISHI HEAVY INDUSTRIES, LTD.
OCT. 26. 1977.
PUMP DESIGNING SECTION
TAKASAGO MACHINERY WORKS

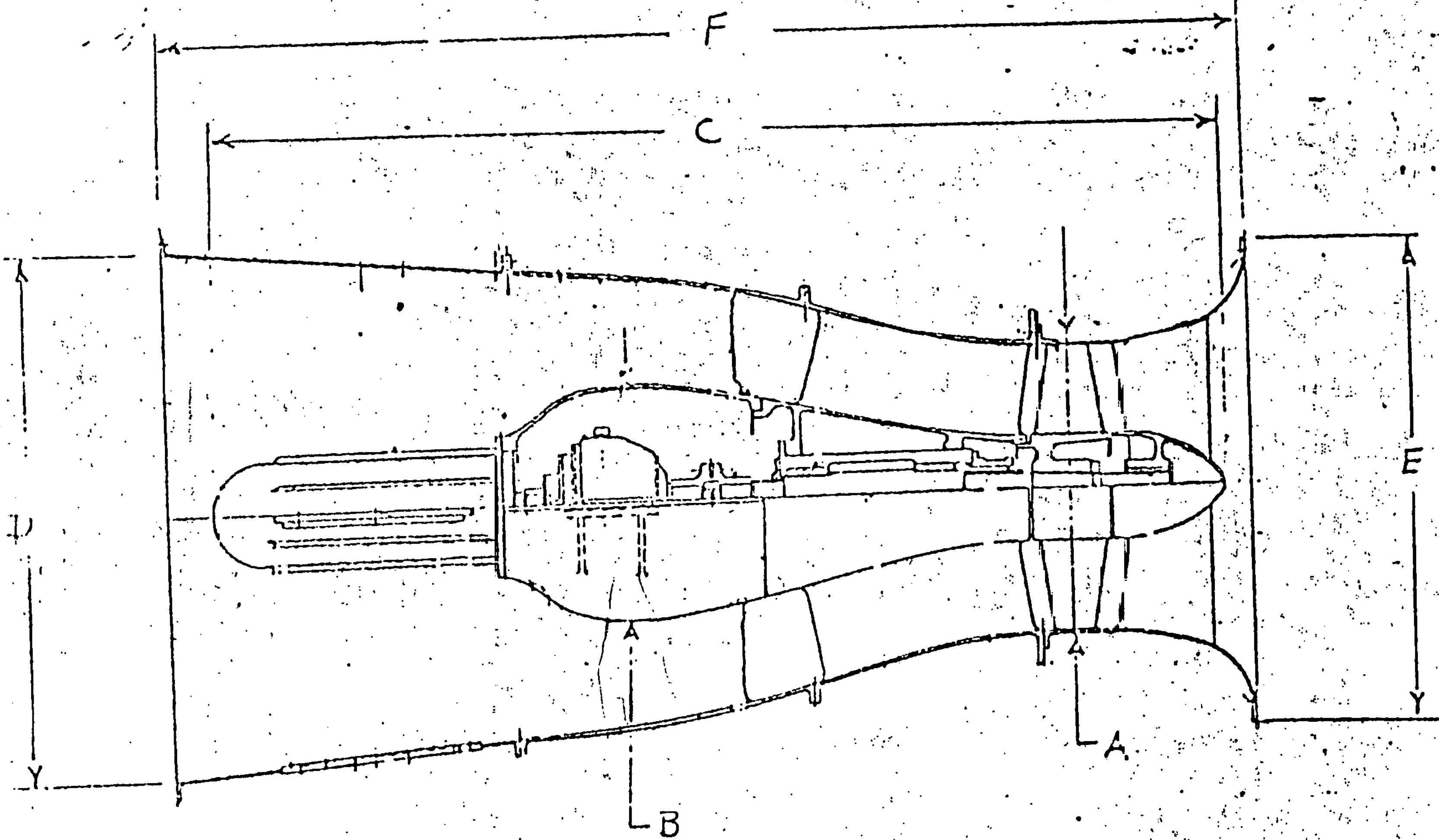
OCT. 1977

MITSUBISHI HEAVY INDUSTRIES, LTD.

OTEC - SEAWATER CIRCULATING PUMP

OUTPUT OF TEST PLANT		5 MW (Surface)	25 MW (Surface)	25 MW (Submerged)
NUMBER OF PUMPS		1	4	4
DESIGN FLOW (cfs)		650	812	1000
HEAD (ft)		10	10	6
PUMP TYPE		VERTICAL SHAFT AXIAL FLOW	→	→
TYPE OF IMPELLER		VARIABLE PITCH	→	→
PUMP SPEED (rpm)		108	96.5	59
MOTOR	SPEED (rpm)	590	→	390
	OUTPUT (kW)	750	940	690
	V, CYCLE, PHASE	4160 V, 60 Hz, 3-PHASE		
GEAR	TYPE	PLANETARY GEAR	→	→
	REDUCING RATIO	5.46	6.11	6.61
APPROX. WEIGHT	PUMP (t)	120	140	200
	MOTOR (t)	9	12	14
	GEAR (t)	6	7	7
	TOTAL (t)	135	159	221

EFF. AND POWER	PUMP DESIGN EFF.(%)	85.6	→	→
	OVERALL DESIGNING EFF. (P,G8M) (%)	79	→	→
	DESIGN POWER REQUIRED (kW)	693 (MOTOR IN-PUT)	865 (MOTOR IN-PUT)	640 (MOTOR IN-PUT)
	NPSH req. (ft)	8	8	5
APPROX. DIMENSION (ft)	A-IMPELLER DIA.	8.6	9.6	12.2
	B-POD DIA.	6.9	7.9	9.8
	C-POD LENGTH	36.5	43.5	53.5
	D-DIFFUSER OUTLET DIA.	15	17	22
	E-BELLMOUTH INLET DIA.	14	16	20
	F-OVERALL LENGTH	39	46	56



TYPICAL SEA WATER CIRCULATING PUMP

PSD - 2.1

36-E-1944-1

ALLIS-CHALMERS

MILWAUKEE, WISCONSIN 53201

December 5, 1977

RECEIVED DEC 7 1977

Lockheed Missles & Space Company, Inc.
1111 Lockheed Way
Sunnyvale, California 94088

Attention: Ms. B. Jean Bartel, Subcontract Administrator
Development Programs Material

SUBJECT: OTEC CONTRACT NUMBER EG-77-C-03-1568
YOUR REFERENCE FILE NO. 50-43/7902
OUR REFERENCE I08E1086A

Dear Ms. Bartel:

We refer to your letter of October 11, 1977 and your telex of November 17, 1977, concerning sea water pumps for OTEC Power System Development. At this time, we are pleased to submit preliminary information.

Although we have been building pumps for over one hundred years, we have not had the occasion to design and build the exact "bulb type" arrangement in which you are interested. From the standpoint of hydraulic design, we have however, previously built many pumps within the specific speed range of your requirement and our preliminary selections are based on existing designs.

Eventually, when pump requirements including capacity, head, range of operation, etc., have been firmly fixed, we would recommend the construction and complete testing of a model pump to obtain optimum performance. The one time cost for such a model test and including mechanical considerations of a bulb type arrangement would be approximately $300,000. The resulting model design would be "stepped up" to meet the three different pump rating requirements. It would amount to the same specific speed design, but three different physical sizes.

We attach our tabulation of preliminary specifications for the three different ratings. This tabulation includes performance data, and very preliminary pricing. In addition, we are attaching copies of your inquiry sketch on which we have indicated preliminary dimensions, weights, and other technical data you requested.

We have indicated 86% rated pump efficiency in each case. This is a very conservative figure and we would certainly design for several points higher.

Proper pump materials for handling seas water, is of course a serious consideration. In this instances, we have figured on using 316 stainless steel for all wetted parts. Other materials, such as, nickel aluminum bornze, Ni-resist, monel, etc., might be considered. We do not recommend the use of epoxy coated carbon steel.

The pump motors for this bulb type arrangement, should of course, be as small as possible. Working with our Motor Division, we foresee the use of 1800 RPM water cooled induction motors. In our performance data, we have used 94% motor efficiency, which is again, very conservative.

Obviously, we will need to utilize a reduction gear between the high speed motor and the low speed pump. Undoubtedly, we would use a "epicyclic" gear to obtain shaft concentricity and mimimum physical size. Again, our data includes 94% gear efficiency which should be conservative. Gear information was obtained from the Philadelphia Gear Corporation and it seems there are not many concerns making suitable epicyclic gears in the USA. We are in touch with several European concerns where this type of gearing is commonly used and we will pass along to you any important information we may receive.

Time to accomplish model testing mentioned above, would be from 16 to 20 months. After completion of the model work, we anticipate a time requirement of 20 months to build the first prototype pump. Thereafter, we should be able to complete one pump a week to meet your requirements of five sets or forty pumps per year.

Hopefully, this preliminary information will be of help to you and we would be happy to provide any additional data you might need.

Sincerely,

S. M. Austin

S. M. Austin, Manager
Industrial Pump
Custom Pump Division

SMA/gr
attachment

cc: Mr. J. L. Bertoli, San Francisco Sales Office (VV)
555 California Street
Bank of America Center, Suite 4730
San Francisco, California 91776
(206)883-4000

ALLIS-CHALMERS
CUSTOM PUMP DIVISION

PRELIMINARY **Pump Specification**

For LOCKHEED MISSILES & SPACE COMPANY Date DECEMBER 5, 1977

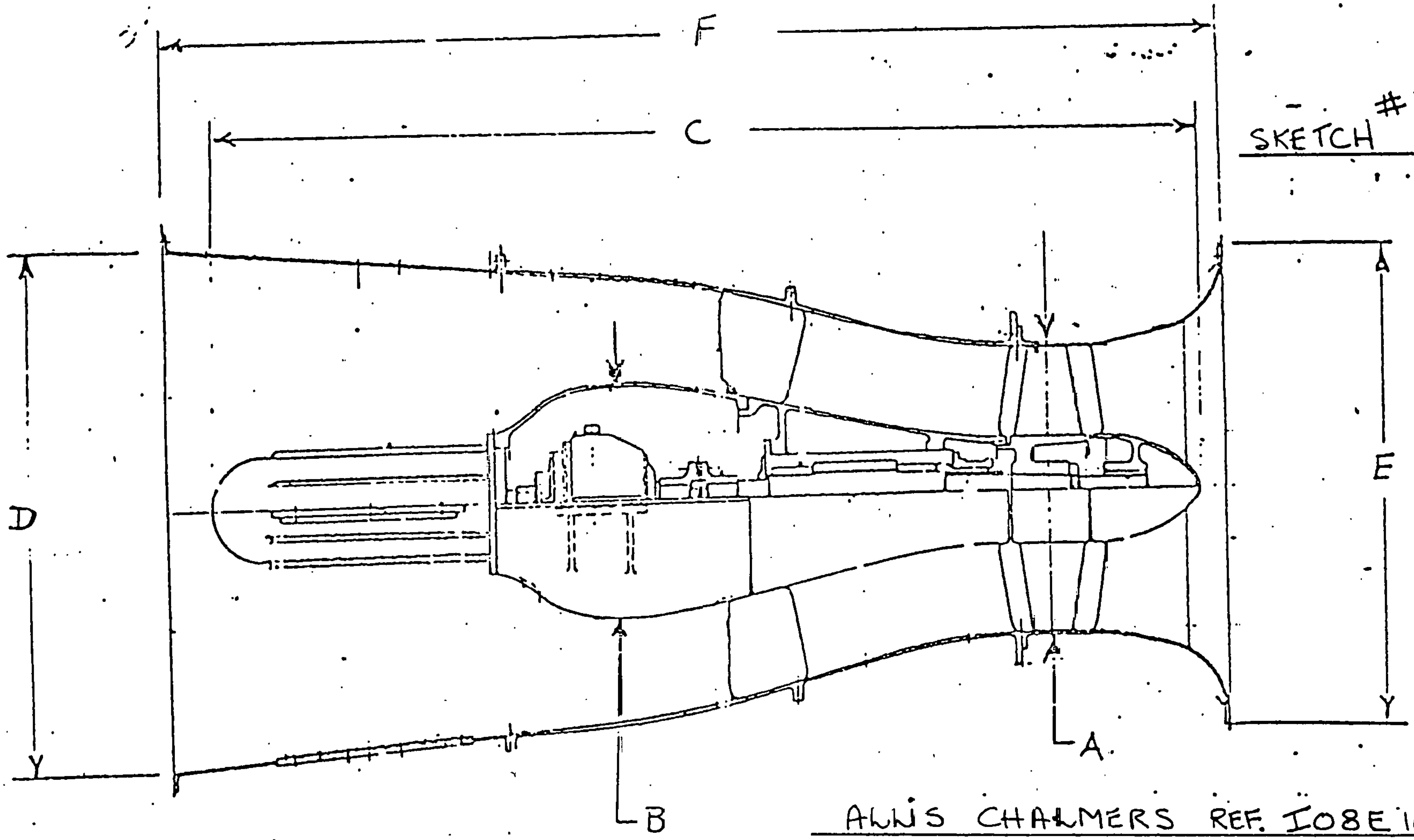
Your No. EG-77-C-03-1568 Our No. Factory No. I08E1086A

Item No.	5MW	25MW-WARM	25MW - COLD
Number of Units	TWO (2)	EIGHT (8)	EIGHT (8)
Service	EVAP./COND.	EVAPORATOR	CONDENSER
Liquid	SEA WATER		
Model	AX(K=7.32)	AX(K = 8.18)	AX(K = 10.25)
Size (Suction	156"	174"	192"
Type of Pump	AXIAL FLOW		
Materials of Construction	"SPECIAL"		
IMPELLER DIAMETER	96"	108"	134"
Stufiing Box Arrangement	PACKING		
Base Plate, Fabricated Steel			
Coupling Type			
Capacity—U. S. Gpm .... Water	291,720(650 CFS)	364,450(812 CFS)	448,800(1000 CFS)
Discharge Head (ft)			
Suction Head (ft)			
Friction Head (ft)			
Total Head (ft) .... Water	10	10	6
NPSH (ft) .... Required	22.5	22.5	14
Speed (Rpm)	153	137	85
Rotation (Viewed from Drive End)			
Pumping Temperature (°F)	AMBIENT		
Specific Gravity at P. T.	1.03	1.03	1.03
Viscosity at P. T. (SSU)			
% Solids or % Consistency			
Max. Sphere Impeller will pass			
Pump Efficiency % .... Water	EST. 86	EST. 86	EST. 86
Brake Horsepower	882.3	1102.2	814.4
Driver Hp Required at Rating	1250	1250	1250
Driver Hp Required for Non Overloading			
Phase-Cycle-Voltage	3/60/4160		
Type Motor or other Driver	SQUIRREL CAGE INDUCTION		
Frame Motor or other Driver			
Curve No.			
Bulletin			
Specification Sheet			
Preliminary Dimension Print	SKETCH 1086A		
Weight (lbs.) Pump, Base and Coupling			
Weight (lbs.) Driver			
Weight (lbs.) Complete Unit EACH	125,000	185,000	264,000
Pump Frame Size			
PRELIM. Net Price EACH Pump, Base and Coupling	$955,275.00	$1,337,775.00	$1,840,275.00
PRELIM. Net Price EACH GEAR			
PRELIM. Net Price EACH Driver... 1250 HP			
*Shipment (weeks).. FIRST UNIT	80		
SPECIFIC SPEED (N_s)	14696	14709	14854
SUCTION SPEC. SPEED (S_s)	8000	8006	7868

*Shipping Schedules are not effective until receipt of complete specifications and required approvals at the factory.

Please return one copy with your order.

SKETCH # 1086 A



ALLIS CHALMERS REF. I08E 1086A

I-22

PSD - 2.1

VERY PRELIMINARY

TYPICAL SEA WATER CIRCULATING PUMP

PUMP	RATING	PUMP EFFICIENCY	WIRE TO WATER EFFICIENCY	MOTOR HP	NPSH REQ'D	WEIGHT	A	B	C	D	E	F
5 MWe SURFACE	650 cfs 10 FT.	86%	* 76%	1250	22.5'	125,000#	96"	90"	29'	144"	156"	33.5'
25 MWe SURFACE	812 cfs 10 FT.	86%	76%	1250	22.5'	185,000#	108"	96"	30'	156"	174"	35'
25 WWe SUBMERGED	1000 cfs 6 FT.	86%	76%	1250	14'	264,000#	134"	108"	35'	174"	192"	40'

* GEAR 94%
* MOTOR 94%

ALLIS-CHALMERS

CUSTOM PUMP DIVISION

PRELIMINARY **Pump Specification**

For LOCKHEED MISSILES & SPACE COMPANY Date DECEMBER 5, 1977

Your No. EG-77-C-03-1568 Our No. Factory No. I08E1086A

Item No.	5MW	25MW-WARM	25MW - COLD
Number of Units	TWO (2)	EIGHT (8)	EIGHT (8)
Service	EVAP./COND.	EVAPORATOR	CONDENSER
Liquid	SEA WATER		
Model	AX(K=7.32)	AX(K = 8.18)	AX(K = 10.25)
Size (Suction	156"	174"	192"
Type of Pump	AXIAL FLOW		
Materials of Construction	"SPECIAL"		
IMPELLER DIAMETER	96"	108"	134"
Stuffing Box Arrangement	PACKING		
Base Plate, Fabricated Steel			
Coupling Type			
Capacity—U. S. Gpm Water	291,720(650 CFS)	364,450(812 CFS)	448,800(1000 CFS)
Discharge Head (ft)			
Suction Head (ft)			
Friction Head (ft)			
Total Head (ft) Water	10	10	6
NPSH (ft) Required	22.5	22.5	14
Speed (Rpm)	153	137	85
Rotation (Viewed from Drive End)			
Pumping Temperature (°F)	AMBIENT		
Specific Gravity at P. T.	1.03	1.03	1.03
Viscosity at P. T. (SSU)			
% Solids or % Consistency			
Max. Sphere Impeller will pass			
Pump Efficiency % Water	EST. 86	EST. 86	EST. 86
Brake Horsepower	882.3	1102.2	814.4
Driver Hp Required at Rating	1250	1250	1250
Driver Hp Required for Non Overloading			
Phase-Cycle-Voltage	3/60/4160		
Type Motor or other Driver	SQUIRREL CAGE INDUCTION		
Frame Motor or other Driver			
Curve No.			
Bulletin			
Specification Sheet			
Preliminary Dimension Print	SKETCH 1086A		
Weight (lbs.) Pump, Base and Coupling			
Weight (lbs.) Driver			
Weight (lbs.) Complete Unit EACH	125,000	185,000	264,000
Pump Frame Size			
PRELIM. Net Price EACH Pump, Base and Coupling	$955,275.00	$1,337,775.00	$1,840,275.00
PRELIM. Net Price EACH GEAR			
PRELIM. Net Price EACH Driver *1250 HP			
*Shipment (weeks) FIRST UNIT	80		
SPECIFIC SPEED (N_s)	14696	14709	14854
SUCTION SPEC. SPEED (S_s)	8000	8006	7868

*Shipping Schedules are not effective until receipt of complete specifications and required approvals at the factory.

Please return one copy with your order.

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WORTHINGTON PUMP CORPORATION (U.S.A.)

901 Sneath Lane, San Bruno, California 94066 / Telephone (415) 871-6455

Lockheed Missiles and Space Company Inc.
1111 Lockheed Way
Sunnyvale CA 94088

Attention: Mrs. Jean Bartel

Subject: Request for Information - Seawater Pump Assemblies

Reference: (A) LMSC, RFI 50-40/BJB#004, dtd Jan. 19, 1977
(B) LMSC, RFQ No. BJB:2476, dtd June 24, 1977
(C) Prime Contract No. EG-77-C-03-1568

Dear Mrs. Bartel:

In accordance with your subject request for information for pumping equipment on the OTEC-1 and Power System Development project, Worthington is pleased to offer the preliminary information requested for your review. The attached letter from Mr. W. C. Krutzsch at our Engineered Pump Division references the conversation with your Mr. M. I. Leitner and T. M. Robieart regarding this RFI.

Attached you will find Table I including information requested under the heading "Pump Design and Performance Parameter". The information includes physical characteristics and cost estimates of pumps required for the 5 MWe and 25 MWe power module.

We trust the information included in the attached letter is sufficient for your evaluation. Should there be any further questions, please do not hesitate to contact me at (415) 871-6455.

Sincerely,

WORTHINGTON PUMP CORPORATION (U.S.A.)

Duane A. Gettman /ih

Duane A. Gettman, P. E.
District Sales Manager
San Francisco Sales Office

DAG:ih
Attach

A 635 1/2 - D

WORTHINGTON

Date October 20, 1977

SUBJECT

OTEC Pumps
Lockheed

TO	D. Gettman	San Francisco
	Name	Location
FROM	W. C. Krutzsch	Harrison - 13
	Name	Location
	C. H. Thaw	Mountainside
COPIES TO	L. J. Karassik	Mountainside
	B. Neumann	Mountainside
	A. Agostinelli	Harrison - 13
	J. P. Fenlon	Harrison - 32
	J. E. C. Valentin	Harrison - 32
	P. D. Gold	Washington - S. O.
	W. H. Fraser	Harrison - 13

Division Quotation No.
S. O. Quotation No.
S. O. Order No.
Works Order No.
Invoice No.
If Quotation Requested, It Is To Be:
☐ FINAL ☐ ESTIMATING

REPLY REQUIRED BY

Subsequent to our visit of 10/6/77 to Lockheed I received a telephone call from M.I. Leitner, Senior Staff Engineer, who had been unable to be present during our meeting. The purpose of this call was to ask if I would be in a position to provide him with a minimal amount of preliminary information for pumps which might ultimately be required for the 5MWe demonstration plant and the 25MWe prototype modules, in accordance with their preliminary specification PSD-2.1 which, you may recall, was unofficially handed to us during our visit, but stamped 'Not Released". In addition, he advised that he expected to be at Foster-Wheeler in Livingston on 10/17, and if time allowed, would like to stop at Harrison while in the area.

I agreed to furnish the information requested, but was unable to get it together before 10/17, when he arrived in Harrison together with C.M. Robidart, who was present at our meeting in California. Some of this material was compiled while these two gentlemen were here at that time, and I have now worked up the balance of it, and have put it all together in the form of Table I attached. Mr. Leitner indicated that this should be transmitted to him through J. Bartel, and I would like to request that you furnish to her not only Table I, but a copy of this letter as well, because of the limitations expressed herein regarding the values shown in the Table.

All the information provided must be considered as approximate, and as having been furnished gratuitously for preliminary estimating purposes only. Costs shown are not to be construed as quotations.

The dimensions tabulated have been based on pumps having in all cases a design specific speed of approximately 13,000, and for the conditions of service shown, are probably accurate within plus or minus 3%. Note, however, that they do not include the suction bell, discharge conduit, or drive trains or supports therefor. These additional components will add considerably to the overall dimensions, but by proportioning them reasonably in scale with the impeller and diffuser, some estimate of overall size could be arrived at.

Weights have been based on previous estimates prepared for pumps of similar specific speeds, and factored up or down in accordance with anticipated size ratios. They are considerably less accurate than the dimensions, probably no better than plus or minus 20%, and are probably on the high side for the three small pumps and on the low side for the two large ones. Again, they pertain essentially to the impeller, diffuser, and pump shaft and bearings contained within the impeller and diffuser envelope. Addition of the remaining components required for a complete pumping unit would undoubtedly at least double the weights shown.

Costs have also been estimated from previous jobs, but because of uncertainties associated with differences in materials of construction and escalation over periods ranging from 5 to 15 years, are even less accurate than the weights, probably no better than plus or minus 50%. These figures pertain to the same items as the weights, and as in that case, addition of the components not included in these figures would at least double the costs shown.

This is intended only as a partial response to Lockheed's preliminary specification, based on the specific request made to me by Mr. Leitner. At such time as Lockheed officially releases its specification, we will then consider what additional information we can provide at that time.

W. C. Kratzsch

WCK/ds

TABLE I

	5 MWe	25MWe			
	(Surface)	(Surface)		(Submerged)	
Number of Pumps	1	4	1	4	1
Design Flow (cfs)	650	812	3248	1000	4000
Head (ft)	10	10	10	6	6
Impeller Dia. (in.)	96	108	216	136	272
RPM	131	117	59	72	36
Combined Length of Impeller & Diffuser (in.)	96	108	216	136	272
Estimated Weight (000 lbs.)	50	64	250	100	400
Estimated Cost (000 $)	300	360	1400	450	1800

Note: All values above are approximate and are subject to limitations stated in my letter attached.

W. C. Krutzsch
10-20-77

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TO906 12/08
LMSC SUVL

213274 PLE D
TELEX-NO. 3172/77 DECEMBER 8, 1977

DEC 8 7 09 AM '77

ATTN. MRS. B.J. BARTEL

REF.: OTEC SEAWATER PUMP - FILE NO. 50-43/7902 AND 50-43/7967
OUR PROJECT OP 64 (US)

WE REFER TO OUR TELEX NO. 3158/77 OF DECEMBER 8TH, 1977. FOR THE THREE DIFFERENT PUMP TYPES WE HAVE CALCULATED THE ROM PRICES BASED ON A DELIVERY END 1978. YOU WILL UNDERSTAND THAT IT IS NEARLY IMPOSSIBLE TO GIVE PRICES FOR DELIVERY IN 1985 TO 1990 AS NOBODY CAN FORESEE THE FUTURE INFLATION RATES. FOR THE NEXT 3 - 4 YEARS WE EXPECT AN INFLATION RATE OF ABT. 6 PER CENT PER YEAR.

OUR ROM PRICES ARE BASED ON FOLLOWING EXTENT OF DELIVERY:

- AXIAL FLOW PUMP (1 STAGE) WITH SUCTION BELL AND GUIDE VANE/DIFFUSER ARRANGEMENT
- PLANETARY REDUCTION GEAR
- SUBMERSIBLE ELECTRIC MOTOR
- FOUNDATIONS

ROM PRICE FOR PUMP TYPE S 2183-1"VRZ 30-130-8	DM 750.000,--
ROM PRICE FOR PUMP TYPE S 2753-1"VRWZ 40-160-8	DM 1.150.000,--
ROM PRICE FOR PUMP TYPE S 3302-1"VRZ 30-120-8	DM 1.150.000,--

THE PRICES ARE BASED ON FOB HAMBURG DELIVERY END 1978 OF ONE UNIT. DEPENDING ON THE NUMBER OF PUMP SETS ORDERED SIMULTANEOUSLY A QUANTRITY DISCOUNT WILL BE GIVEN.

AS ALREADY MENTIONED IN OUR TELEX 3158/77 PUMP CURVES AND DIMENSIONS WILL BE MAILED SOON.

REGARDS
BARTHOLLY
MARINE DIVISION (PLEUGER PUMPS)
213274 PLE D
LMSC SUVL

ANSWER VIA TRT 605

9

1. 5 MWE UNIT

DESIGN FLOW:	650 CFS
HEAD:	10 F
PUMP TYPE:	S2183-1"VRZ 30-130-8
MOTOR OUTPUT:	960 HP
IMPPELLER SPEED:	192 RPM
IMPELLER DIAMETER:	2180 MM
PUMP EFFICIENCY	84 PER CENT
PUMP EFFICIENCY INCLUDING EFFICIENCY OF PLANETARY GEAR:	80 PER CENT
MOTOR SPEED:	870 RPM
VOLTAGE:	4160 V
FREQUENCY:	60 HZ

2. 25 MWE UNIT (SURFACE)

DESIGN FLOW:	812 CFS
HEAD:	10 F
PUMP TYPE:	S2753-1"VRWZ 40-160-8
MOTOR OUTPUT:	1615 HP
IMPELLER SPEED:	140 RPM
IMPELLER DIAMETER:	2750 MM
PUMP EFFICIENCY:	84,5 PER CENT
PUMP EFFICIENCY INCLUDING EFFICIENCY OF PLANETARY GEAR:	80 PER CENT
MOTOR SPEED:	870 RPM
VOLTAGE:	4160 V
FREQUENCY:	60 HZ

3. 25 MWE UNIT (SUBMERGED)

DESIGN FLOW:	1000 CFS
HEAD:	6 F
PUMP TYPE:	S3302-1"VRZ 30-120-8
MOTOR OUTPUT:	875 HP
IMPELLER SPEED:	100 RPM
IMPELLER DIAMETER:	3300 MM
PUMP EFFICIENCY:	85 PER CENT
PUMP EFFICIENCY INCLUDING EFFICIENCY OF PLANETARY GEAR:	80,5 PER CENT
MOTOR SPEED:	870 RPM
VOLTAGE:	4160 V
FREQUENCY:	60 HZ

PUMP CURVES AND DIMENSION SKETCH ARE UNDER PREPARATION AND WILL BE SENT WITHIN THE NEXT DAYS. THE ROM PRICE ACCORDING TO YOUR TELEX TO PLEUGER STATESVILLE ON 11-17-77 (FILE NO. 50-43/7967) WILL BE TELEXED TO YOU TOMORROW.

REGARDS
BARTHOLLY
MARINE DIVISION

+
LMSC SUVL

213274 PLE D

213274 PLE D

TELEX-NO. 3308/77 DECEMBER 22, 1977

ATTN. MRS. B.J. BARTEL

REF.: OTEC SEAWATER PUMP
FILE NO. 50-43/79902 AND 7967
OUR PROJECT OP 64 (US)

WE REFER TO YOUR TELCON WITH MR. SCHUNKE, STATESVILLE AND ANSWER YOUR QUESTIONS AS FOLLOWS:

1. THE SPEED OF THE SUBMERSIBLE MOTOR CAN BE CONTROLLED IN THE RANGE OF ABOUT 10 PER CENT BY VARYING THE VOLTAGE AND KEEPING THE FREQUENCY CONSTANT.
   IF THE SPEED IS TO BE CONTROLLED OVER A WIDE RANGE DOWN TO ABOUT 20 PER CENT OF NOMINAL SPEED VOLTAGE AND FREQUENCY MUST BE VARIED. IN THIS CASE THE QUOTIENT VOLTAGE/FREQUENCY IS TO KEEP CONSTANT WHICH IS REACHED BY CONTROLLING THE SPEED OF THE ALTERNATOR.
   BOTH CONTROL METHODS CAN BE APPLIED WITHOUT MODIFYING THE DESIGN AND ELECTRICAL CHARACTERISTIC OF OUR SUBMERSIBLE MOTOR. THUS THE PRICE CAN BE KEPT UNCHANGED.
   IF THE SPEED OF THE MOTOR IS TO BE CONTROLLED IN A WIDE RANGE WITHOUT BEING ABLE TO VARY THE FREQUENCY PLEASE INFORM. IN THIS CASE THE CONTROL METHOD CALLS FOR A MODIFICATION OF MOTOR DESIGN.

2. THE SENSE OF ROTATION OF THE MOTOR RESPECTIVELY PUMP CAN EASILY BE REVERSED BY A REVERSING SWITCH IN THE SWITCH GEAR CABINET. THIS CREATES NO EXTRA PRICE FOR THE MOTOR.
   BY OPERATING THE PUMP IN THE REVERSE DIRECTION THE FLOW AND THE HEAD WILL BE DECREASED CONSIDERABLY.

REGARDS
BARTHOLLY
MARINE DIVISION

213274 PLE D
LMSC SUVL

VIA ITT
O

APPENDIX I.2
TURBINE/GENERATOR

SUMMARY

Conceptual ammonia turbine-generator frame sizes were established at 5 MW, 12½ MW, and 25 MW net ratings for the Ocean Thermal Energy Conversion Design (OTEC) Study.

Final results of the OTEC ammonia turbine-generator study evolved by consideration of the following key points:

- o Efficiency Trade Studies and Performance
- o Reliability and Availability
- o Safety
- o Producibility and Shop Limitations
- o Mechanical Design
- o Cost

This appendix presents a discussion of the process and rationale by which the conceptual ammonia turbine-generator designs were developed.

Also, performance and construction data for the turbine-generator alternates examined during the conceptual design study phase are presented.

back blank

APPENDIX I.2

TURBINE/GENERATOR

I.2.1 PERFORMANCE

In establishing turbine-generator performance and construction data for alternate frames at 5 MW, 12½ MW, and 25 MW net ratings, ammonia state points of 131.1 PSIA - 1% moisture at the turbine inlet and 94.26 PSIA at the turbine exhaust were used. Each turbine design used the inlet valve size as the basis for determining the turbine frame.

Having selected the turbine inlet size and calculating a pressure drop due to the inlet line and valves, ammonia flow to the turbine was obtained.

Available energy at the turbine wheel was obtained by subtracting the energy loss due to inlet piping and valve throttling and also the energy loss due to a 1% pressure drop in the turbine exhaust from the isentropic energy defined by the above-mentioned ammonia state points.

Knowing the wheel available energy and selecting a velocity ratio (W/V) = .56, the condition at which the best wheel efficiency

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O T E C   A M M O N I A   T U R B I N E   F R A M E   S I Z E S

5 0 0 0   K W   (N E T)
6 3 5 0   K W   (G R O S S)

131.1 PSIA - 1% Moisture - 94.26A

Turbine Frame	Turbine Type	Inlet Size Inches	Inlet Pressure Drop	Turbine/ Generator Speed RPM	Single Wheel Design: Flow Paths / Bucket Height / Pitch Diameter	NH_3 Rate $\frac{\text{Flow (G)}}{\text{KW (N)}}$
182A	Double Flow	2 x 36	1%	3600	2 / 5.17" / 32.53"	317 #/KWH
182B	Double Flow	1 x 42	2%	3600	2 / 5.34" / 32.02"	325 #/KWH
182C	Double Flow	2 x 36	1%	1800	2 / 2.84" / 65.06"	317 #/KWH
182C-1				1800/3600		323 #/KWH
182D	Single Flow	1 x 42	2%	1800	1 / 5.36" / 64.04"	318 #/KWH
182D-1				1800/3600		324 #/KWH

TABLE I.2-1

OTEC AMMONIA TURBINE FRAME SIZES

12500 KW (NET)
15875 KW (GROSS)

131.1 PSIA - 1% Moisture - 94.26A

Turbine Frame	Turbine Type	Inlet Size Inches	Inlet Pressure Drop	Turbine/ Generator Speed RPM	Single Wheel Design Flow Paths / Bucket Height / Pitch Diameter	NH3 Rate Flow (G) / KW (N)
183A	Double Flow	2 x 42	3%	1800	2 / 7.37" / 62.9"	328 #/KWH
183A-1				1800 / 3600		333 #/KWH
183B	Double Flow	2 x 48	2%	1800	2 / 6.56" / 64.04"	317 #/KWH
183B-1				1800 / 3600		322 #/KWH

TABLE I.2-2

OTEC AMMONIA TURBINE FRAME SIZES

25,000 KW (NET)
31,750 KW (GROSS)

131.1 PSIA - 1% Moisture - 94.26 PSIA

Turbine Frame	Turbine Type	Inlet Size Inches	Inlet Pressure Drop	Turbine/ Generator Speed RPM	Single Wheel Design: Flow Paths / Bucket Height / Pitch Diameter	NH_3 Rate Flow (G) / KW (N)
183A	Double Flow	2 x 60	3%	1800	2 / 14.55" / 62.9"	324 #/KWH
183A-1				1800/3600		327 #/KWH
183B	Double Flow	2 x 60	4%	1800	2 / 13.45" / 61.76"	335 #/KWH

TABLE I.2-3

applied for both the 1800 RPM and 3600 RPM applications. In addition, for the 1800 RPM turbine designs, arrangements using a 3600 RPM generator with a gear and an 1800 RPM turbine were also established. It's noted that the performance difference of a geared vs. a direct-drive arrangement is primarily due to gear-mechanical losses.

I.2.3 OFF-DESIGN PERFORMANCE

To support OTEC cycle thermodynamic analysis performance at off-design conditions, turbine performance off-design curves using 131.1 PSIA - 1% moisture inlet ammonia conditions and 94.26 PSIA for the turbine exhaust ammonia conditions were plotted as shown in Figure I.2-1. The first curve gives the variation of Turbine Shaft KW/Design Turbine Shaft KW vs. initial pressure at the turbine.

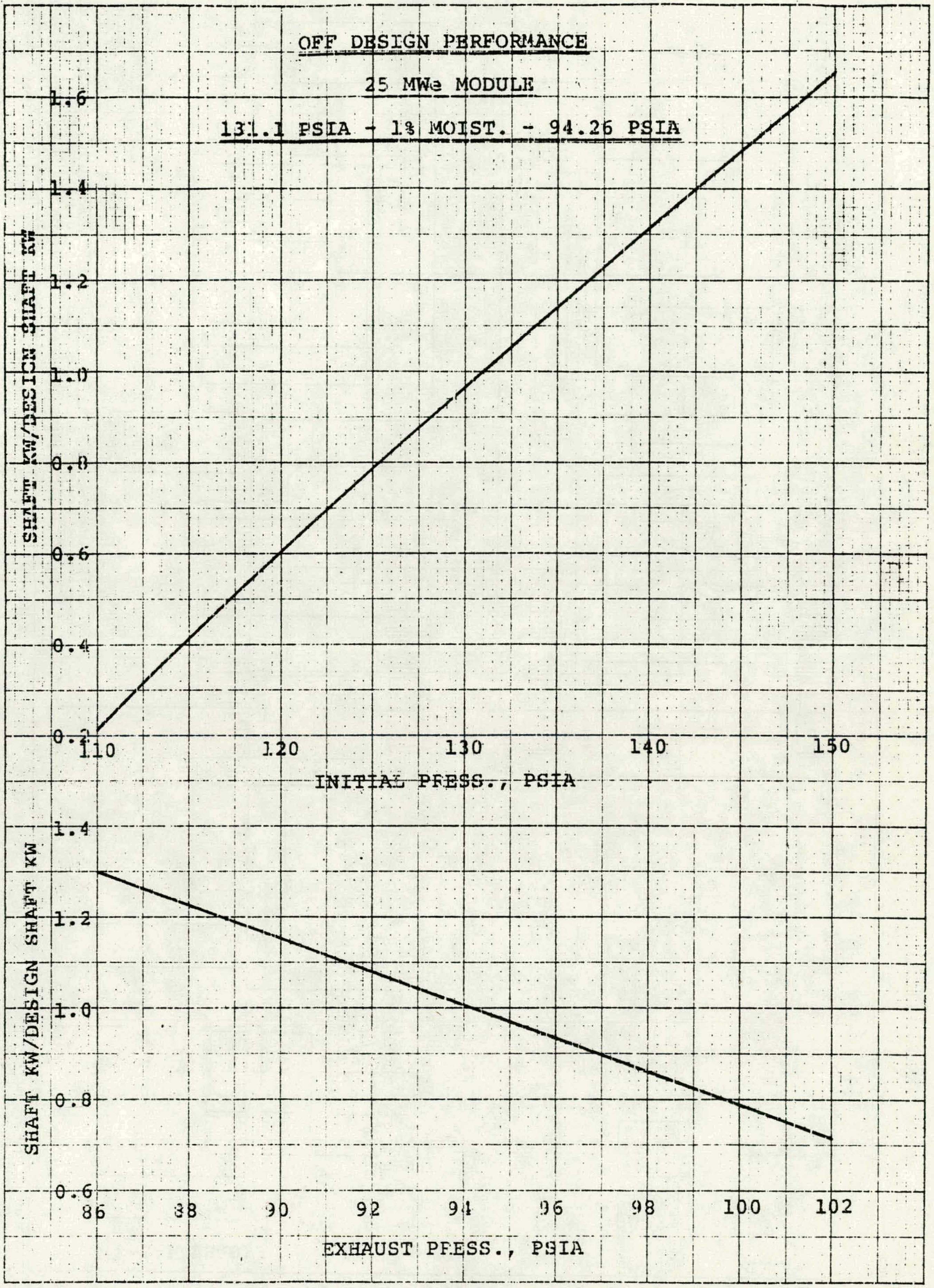
To correct for exhaust pressure variations, a curve of Turbine Shaft KW/Design Turbine Shaft KW vs. exhaust pressure is also given.

I.2.4 TURBINE-GENERATOR WEIGHTS AND DIMENSIONS

Arrangement of machinery outlines for the direct-drive, turbine-generator frames studied for 5 MW, 12½ MW, and 25 MW net turbine-generator ratings are presented in Figures I.2-2 through I.2-8.

Extracted from the illustrated outlines are turbine-generator weights and dimensions given in Tables I.2-4, I.2-5, and I.2-6 for the frames developed at 5 MW, 12½ MW, and 25 MW net turbine-generator ratings.

FIGURE I.2-1



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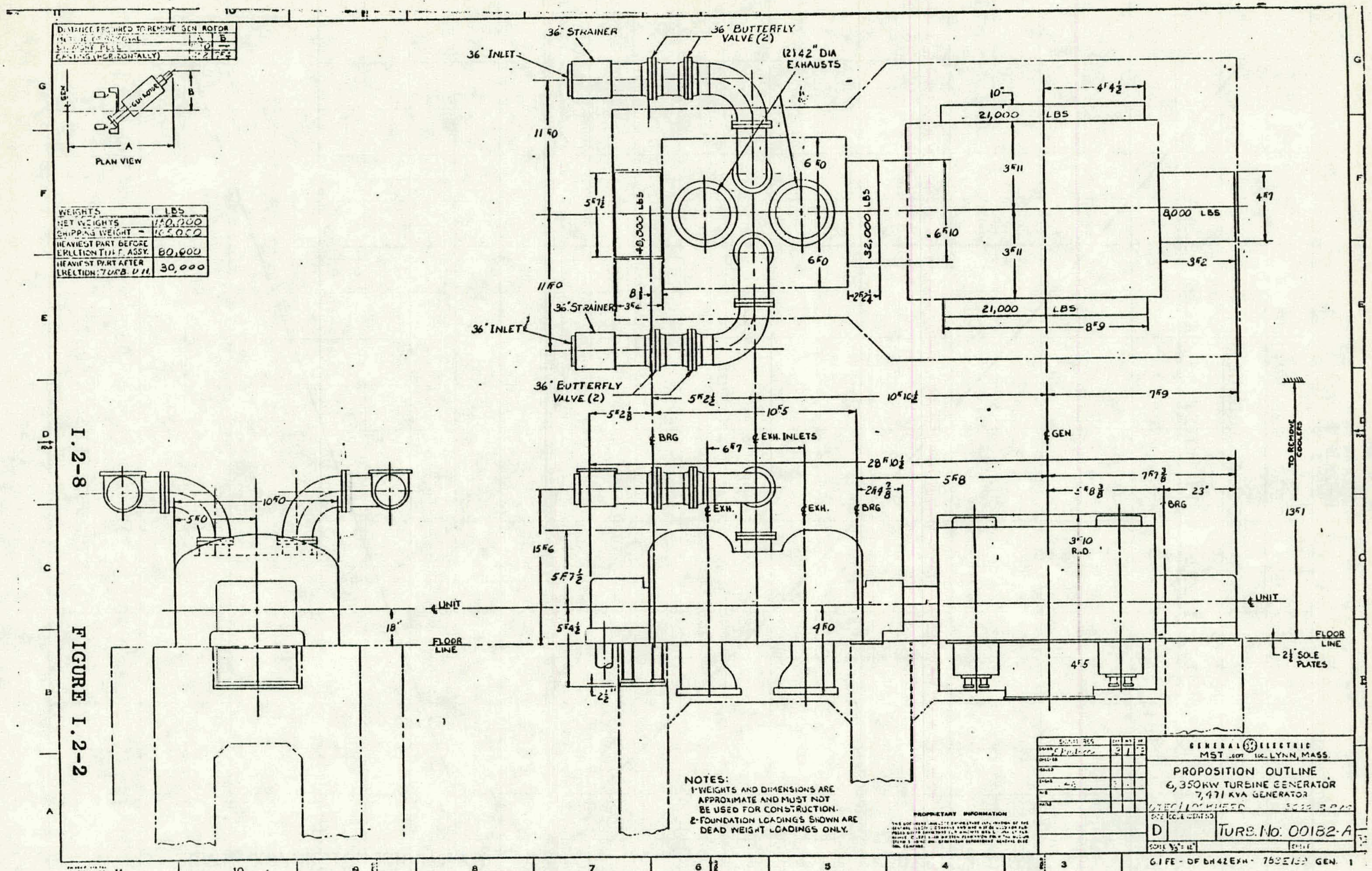
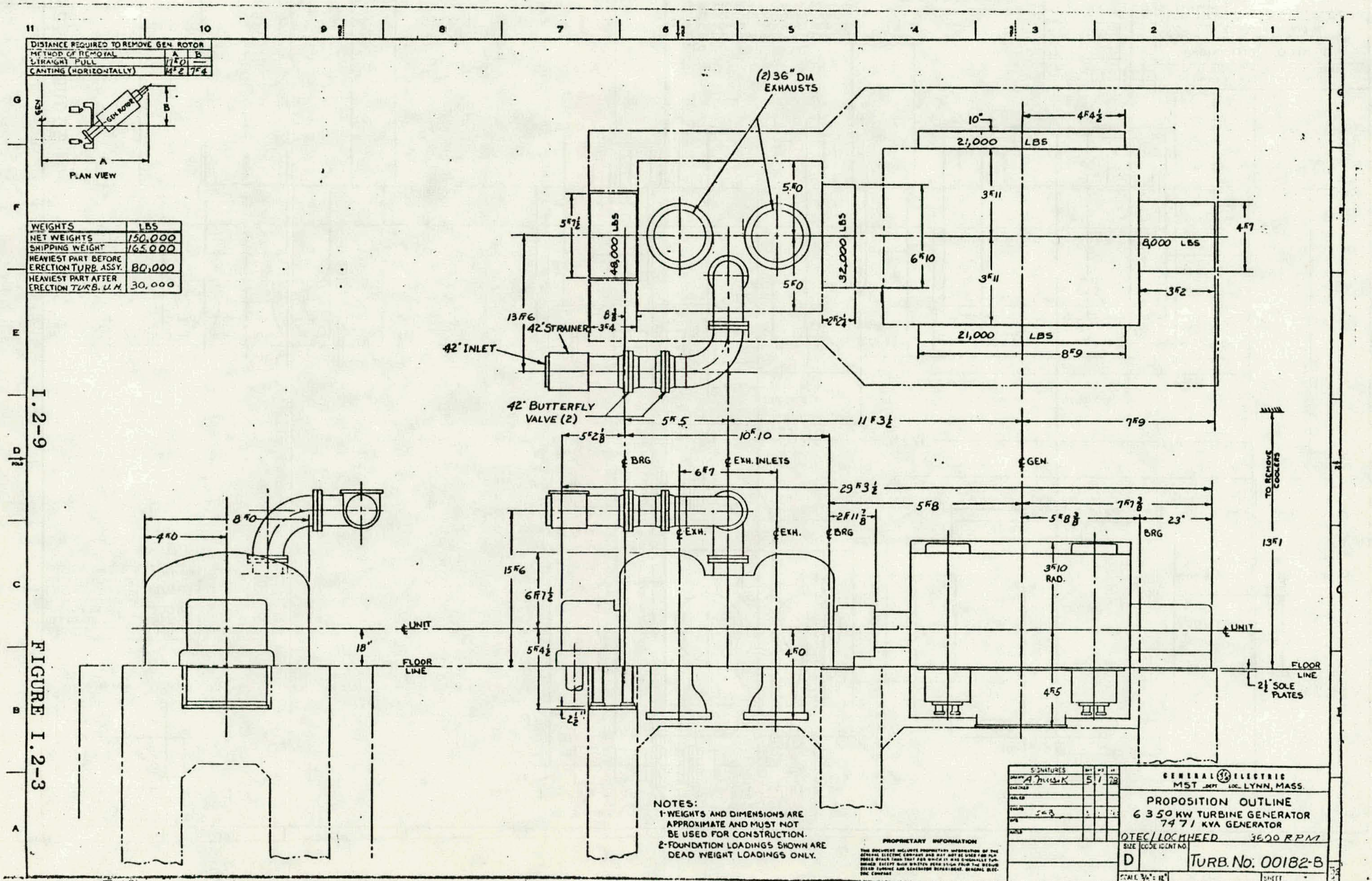
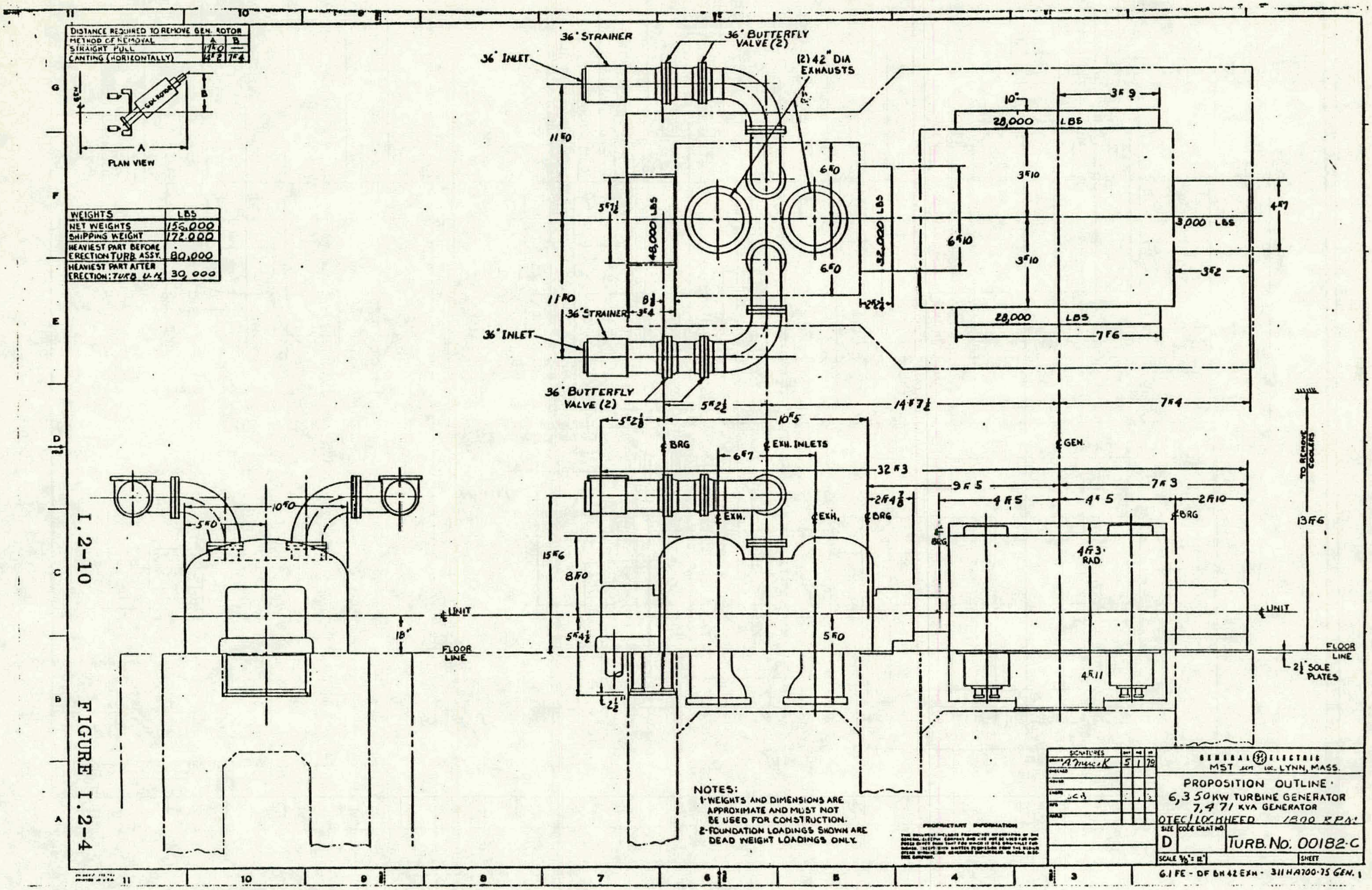

DISTANCE REQUIRED TO REMOVE GEN. ROTOR
PLAN VIEW
WEIGHTS | LBS
NET WEIGHTS | 110,000
SHIPPING WEIGHT | 165,000
HEAVIEST PART BEFORE ERECTION TURB. ASSY. | 80,000
HEAVIEST PART AFTER ERECTION: TURB. U.H. | 30,000
36" STRAINER
36" BUTTERFLY VALVE (2)
36" INLET
(2) 42" DIA EXHAUSTS
10"
4'-4½
21,000 LBS
11'-0
6'-0
5'-7½
40,000 LBS
32,000 LBS
6'-10
3'-11
8,000 LBS
4'-7
3'-2
2'-2¾
8½
36" STRAINER
36" INLET
21,000 LBS
8'-9
36" BUTTERFLY VALVE (2)
5'-2½
5'-2⅞
10'-5
10'-10½
7'-9
℄ BRG
℄ EXH. INLETS
6'-7
℄ GEN
28'-10½
5'-8
7'-7⅜
2'-4⅞
5'-8⅜
23"
TO REMOVE COOLERS
10'-0
5'-0
℄ EXH.
℄ EXH.
℄ BRG
℄ BRG
13'-1
15'-6
3'-10 RAD.
5'-7½
℄ UNIT
℄ UNIT
18"
FLOOR LINE
5'-4½
4'-0
FLOOR LINE
2½" SOLE PLATES
4'-5
2½"
NOTES:
1- WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2- FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
GENERAL ELECTRIC
MST LYNN, MASS.
PROPOSITION OUTLINE
6,350 KW TURBINE GENERATOR
7,471 KVA GENERATOR
D
TURB. No: 00182-A
61 FE - DF BH42 EXH - 753E159 GEN. 1

FIGURE I.2-2



DISTANCE REQUIRED TO REMOVE GEN. ROTOR
METHOD OF REMOVAL A B
STRAIGHT PULL 17F0 —
CANTING (HORIZONTALLY) 14F2 7F4
GEN. ROTOR
PLAN VIEW
WEIGHTS LBS
NET WEIGHTS 150,000
SHIPPING WEIGHT 165,000
HEAVIEST PART BEFORE ERECTION TURB. ASSY. 80,000
HEAVIEST PART AFTER ERECTION TURB. U.H. 30,000
(2) 36" DIA EXHAUSTS
42" STRAINER
42" INLET
42" BUTTERFLY VALVE (2)
48,000 LBS
32,000 LBS
21,000 LBS
8,000 LBS
21,000 LBS
℄ BRG
℄ EXH. INLETS
℄ GEN.
℄ EXH.
℄ EXH.
℄ BRG
BRG
3F10 RAD.
℄ UNIT
FLOOR LINE
℄ UNIT
FLOOR LINE
2½" SOLE PLATES
TO REMOVE COOLERS
NOTES:
1- WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2- FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
GENERAL ELECTRIC
MST DEPT LOC. LYNN, MASS.
PROPOSITION OUTLINE
6 350 KW TURBINE GENERATOR
7471 KVA GENERATOR
OTEC/LOCKHEED 3600 RPM
SIZE D
TURB. No. 00182-B
SCALE 3/4" = 1'0"

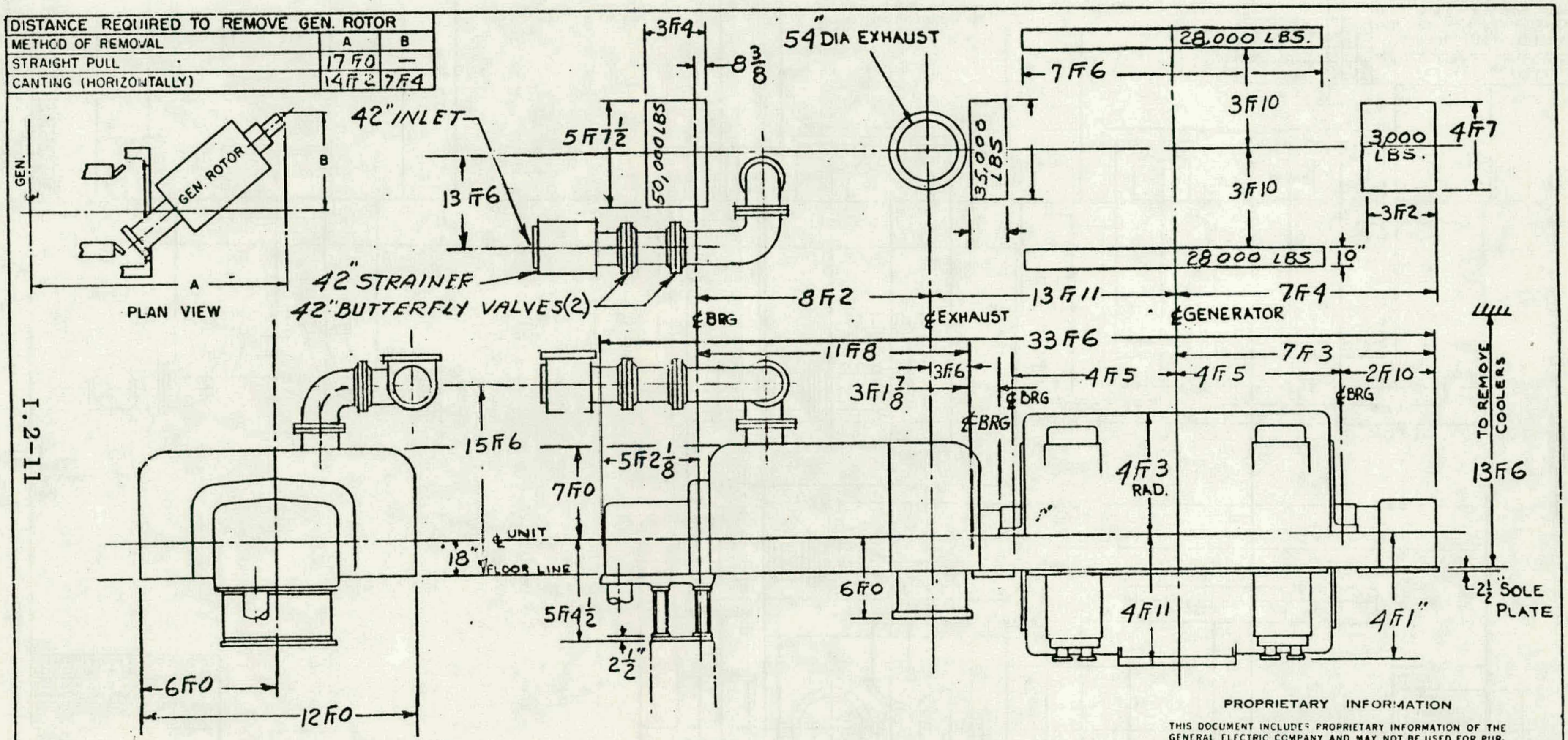
FIGURE I.2-3



DISTANCE REQUIRED TO REMOVE GEN. ROTOR
METHOD OF REMOVAL | A | B
STRAIGHT PULL | 17'-0 | —
CANTING (HORIZONTALLY) | 14'-2 | 7'-4
PLAN VIEW
GEN ROTOR
WEIGHTS | LBS
NET WEIGHTS | 156,000
SHIPPING WEIGHT | 172,000
HEAVIEST PART BEFORE ERECTION TURB. ASSY. | 80,000
HEAVIEST PART AFTER ERECTION: TURB. U.H. | 30,000
36" STRAINER
36" BUTTERFLY VALVE (2)
(2) 42" DIA EXHAUSTS
36" INLET
48,000 LBS
32,000 LBS
28,000 LBS
3,000 LBS
36" STRAINER
36" INLET
36" BUTTERFLY VALVE (2)
℄ BRG
℄ EXH. INLETS
℄ GEN.
℄ EXH.
℄ BRG
℄ UNIT
FLOOR LINE
RAD.
TO REMOVE COOLERS
2½" SOLE PLATES
NOTES:
1. WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2. FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
GENERAL ELECTRIC
MST LYNN, MASS.
PROPOSITION OUTLINE
6,350 KW TURBINE GENERATOR
7,471 KVA GENERATOR
OTEC/LOCKHEED 1800 RPM
D
TURB. No. 00182-C
6.1 FE - DF BH 42 EXH - 311HA700-75 GEN.

FIGURE I.2-4

I.2-11



DISTANCE REQUIRED TO REMOVE GEN. ROTOR		
METHOD OF REMOVAL	A	B
STRAIGHT PULL	17Ft0	—
CANTING (HORIZONTALLY)	14Ft2	7Ft4

PROPRIETARY INFORMATION

NOTES:
1- WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2- FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY

WEIGHTS	LBS.
NET WEIGHTS	160,000
SHIPPING WEIGHT	176,000
HEAVIEST PART BEFORE ERECTION TURB. ASSY.	85,000
HEAVIEST PART AFTER ERECTION	34,000

PROPOSITION OUTLINE

6,350 K.W. TURBINE GENERATOR
7,471 KVA GENERATOR

OTEC/LOCKHEED 1800 R.P.M

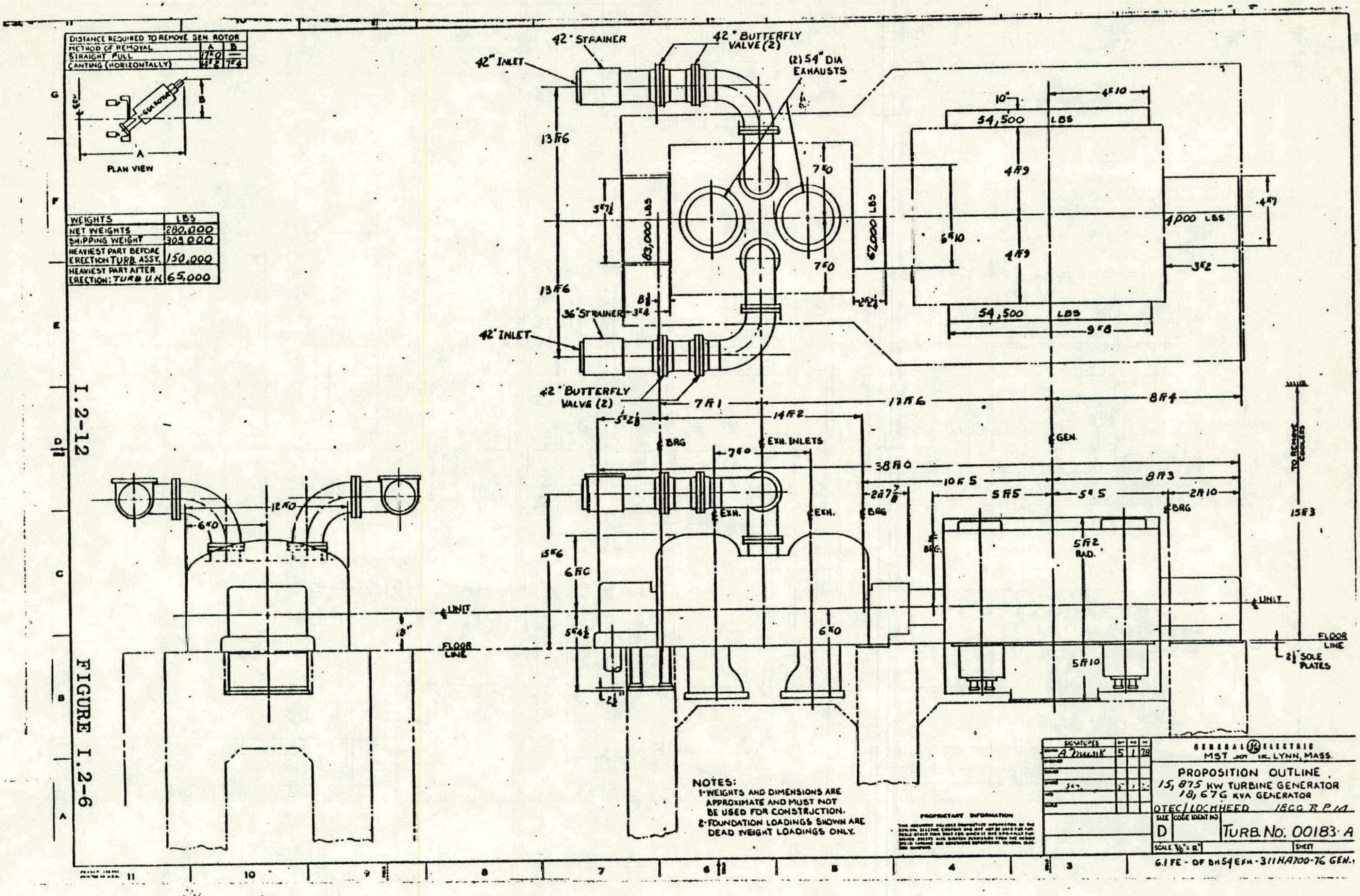
DRAWN BY A. Musik 1-4-78
APPROVED BY SCB 1-4-78

GENERAL ELECTRIC CO
LYNN, MASS.

TURB. NO. 00182-D

G.I. FE. BH 54 EXH. 311HA700-75 GEN.

FIGURE I.2-5



DISTANCE REQUIRED TO REMOVE GEN ROTOR
METHOD OF REMOVAL A B
STRAIGHT PULL
CANTING (HORIZONTALLY)
PLAN VIEW
WEIGHTS LBS
NET WEIGHTS 280,000
SHIPPING WEIGHT 305,000
HEAVIEST PART BEFORE ERECTION TURB. ASSY. 150,000
HEAVIEST PART AFTER ERECTION: TURB U.H. 65,000
42" STRAINER
42" BUTTERFLY VALVE (2)
42" INLET
(2) 54" DIA EXHAUSTS
36" STRAINER
54,500 LBS
83,000 LBS
62,000 LBS
4,000 LBS
EXH. INLETS
BRG
GEN.
EXH.
RAD.
UNIT
FLOOR LINE
SOLE PLATES
TO REMOVE COOLERS
NOTES:
1. WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2. FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
GENERAL ELECTRIC
MST LYNN, MASS.
PROPOSITION OUTLINE
15,875 KW TURBINE GENERATOR
18,676 KVA GENERATOR
OTEC/LOCKHEED 1800 RPM
TURB. NO. 00183-A
6.1 FE - OF BH54 EXH - 311HA700-76 GEN.

FIGURE I.2-6

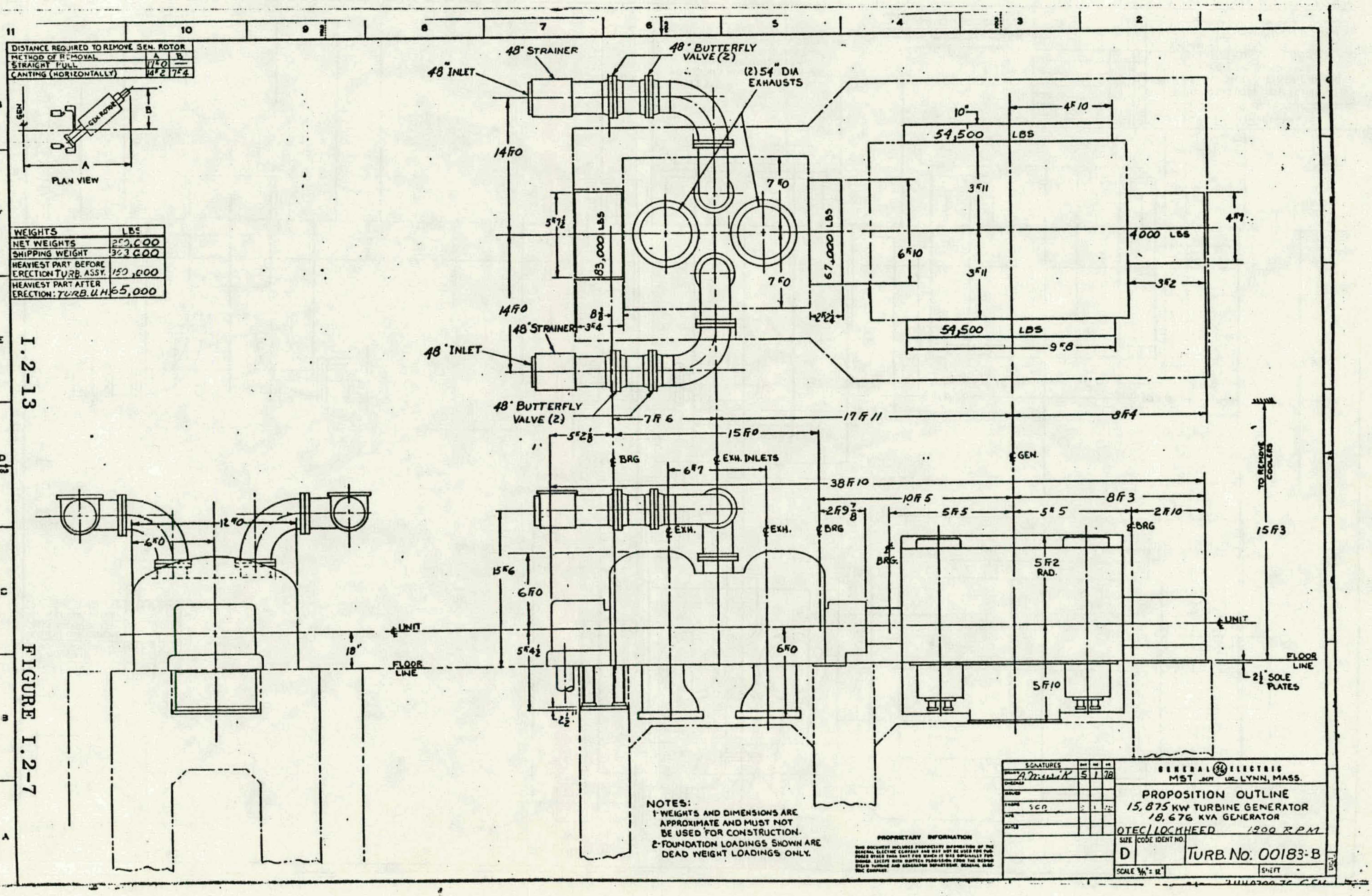
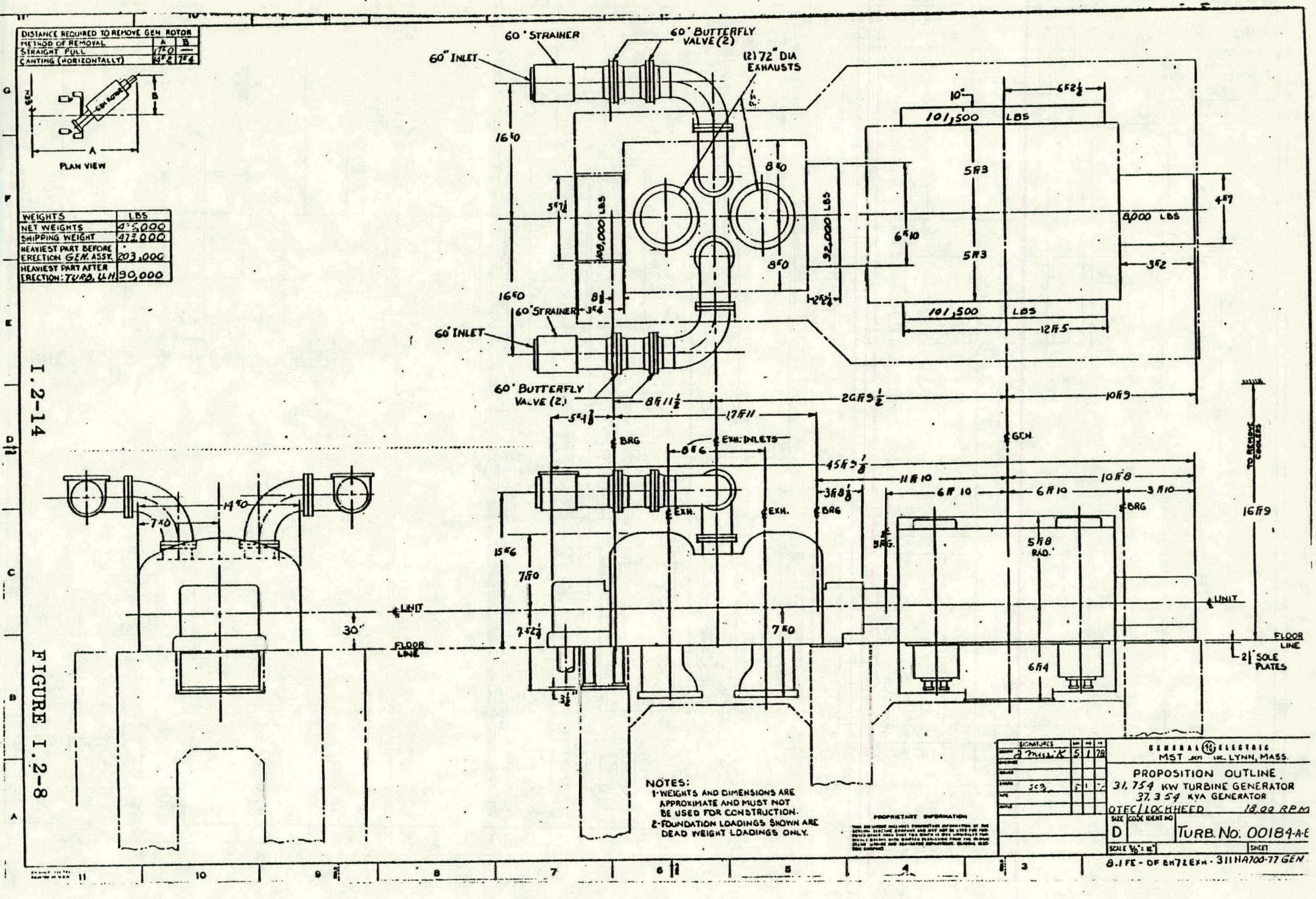

DISTANCE REQUIRED TO REMOVE GEN. ROTOR
METHOD OF REMOVAL | A | B
STRAIGHT PULL | 17'0 | —
CANTING (HORIZONTALLY) | 14'2 | 7'4
GEN ROTOR
PLAN VIEW
WEIGHTS | LBS
SHIPPING WEIGHT
HEAVIEST PART BEFORE ERECTION TURB. ASSY. | 150,000
HEAVIEST PART AFTER ERECTION: TURB. U.H. | 65,000
48" STRAINER
48" INLET
48" BUTTERFLY VALVE (2)
(2) 54" DIA EXHAUSTS
54,500 LBS
83,000 LBS
67,000 LBS
4000 LBS
14'0
7'0
6'10
3'11
4'10
3'2
9'8
BRG
EXH. INLETS
GEN.
EXH.
UNIT
FLOOR LINE
TO REMOVE COOLERS
15'3
5'2 RAD.
5'10
2½" SOLE PLATES
38'10
17'11
15'0
NOTES:
1-WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2-FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
SIGNATURES
A. Mieusik 5 1 78
SCD
GENERAL ELECTRIC
MST DEPT LOC. LYNN, MASS.
PROPOSITION OUTLINE
15,875 KW TURBINE GENERATOR
18,676 KVA GENERATOR
OTEC/LOCKHEED 1800 RPM
SIZE D
CODE IDENT NO.
TURB. No. 00183-B
SCALE 3/16" = 12"
SHEET

FIGURE I.2-7



DISTANCE REQUIRED TO REMOVE GEN ROTOR
METHOD OF REMOVAL | A | B
STRAIGHT PULL | 17F0 | —
CANTING (HORIZONTALLY) | 14F2 | 7F4
GEN ROTOR
PLAN VIEW
WEIGHTS | LBS
NET WEIGHTS | 475,000
SHIPPING WEIGHT | 472,000
HEAVIEST PART BEFORE ERECTION GEN. ASSY. | 203,000
HEAVIEST PART AFTER ERECTION: TURB. U.H. | 90,000
60" STRAINER
60" BUTTERFLY VALVE (2)
60" INLET
(2) 72" DIA EXHAUSTS
101,500 LBS
108,000 LBS
92,000 LBS
8,000 LBS
60" STRAINER
60" INLET
60" BUTTERFLY VALVE (2)
BRG
EXH. INLETS
GEN.
EXH.
UNIT
FLOOR LINE
2½" SOLE PLATES
TO REMOVE COOLERS
5F8 RAD.
NOTES:
1- WEIGHTS AND DIMENSIONS ARE APPROXIMATE AND MUST NOT BE USED FOR CONSTRUCTION.
2- FOUNDATION LOADINGS SHOWN ARE DEAD WEIGHT LOADINGS ONLY.
PROPRIETARY INFORMATION
SIGNATURES
GENERAL ELECTRIC
MST DEPT LOC. LYNN, MASS.
PROPOSITION OUTLINE
31,754 KW TURBINE GENERATOR
37,354 KVA GENERATOR
OTEC/LOCKHEED 1800 RPM
SIZE D
CODE IDENT NO
TURB. No: 00184-A-E
SCALE 3/16" = 12"
SHEET
8.1FE - DF 6H72EXH - 311HA700-77 GEN

FIGURE I.2-8

OTEC AMMONIA TURBINE GENERATOR
WEIGHTS & DIMENSIONS

5000 KW (NET)
6350 KW (GROSS)

Turbine Frame	Inlet Size Inches	Exhaust Size Inches	Turbine/Generator Speed RPM	Turbine/Generator			
				Length Ft.	Width Ft.	Height Ft.	Weight #
182A	2 x 36	2 x 42	3600	18	10	11	80,000
				11	10	8	55,000
182E	1 x 42	2 x 36	3600	19	8	12	80,000
				11	10	8	55,000
182C	2 x 36	2 x 42	1800	18	10	13	80,000
				15	10	9	56,000
182D	1 x 42	1 x 54	1800	20	12	13	85,000
				15	10	9	56,000

TABLE I.2-4

OTEC AMMONIA TURBINE GENERATOR WEIGHTS & DIMENSIONS

12,500 KW (NET)
15,875 KW (GROSS)

Turbine Frame	Inlet Size Inches	Exhaust Size Inches	Turbine/ Generator Speed RPM	Turbine/Generator			
				Length Ft.	Width Ft.	Height Ft.	Weight #
183A	2 x 42	2 x 54	1800	22	12	12	150,000
				16	12	11	109,000
183B	2 x 48	2 x 54	1800	23	12	12	150,000
				16	12	11	109,000

TABLE I.2-5

OTEC AMMONIA TURBINE GENERATOR
WEIGHTS & DIMENSIONS

25,000 KW (NET)
31,750 KW (GROSS)

Turbine Frame	Inlet Size Inches	Exhaust Size Inches	Turbine/ Generator Speed RPM	Turbine/Generator			
				Length Ft.	Width Ft.	Height Ft.	Weight #
184A&B	2 x 60	2 x 72	1800	27	14	14	200,000
				19	12	12	203,000

TABLE I.2-6

I.2.5 RELIABILITY AND AVAILABILITY

In presenting the reliability and availability data in Section 3.2.2 of the OTEC conceptual design report, it's noted that items which do not appear and are conspicuous by their absence with respect to causes of turbine-generator forced outages, are turbine-generator alignment changes, vibration, bucket and nozzle failures, and casing casting cracks. These are not errors in omission; but statistically and practically, these are simply not significant causes of forced outages. Current data shows that some type of nozzle or bucket deficiency related to design is reported for every 19,680,000 row hours of operation. Almost without exception, these do not cause forced outages but are observed and corrected during routine inspections.

To minimize the amount of turbine-generator maintenance time, features such as casing access covers, ease of bearing inspection, control valve tests, and borescope openings would be provided. With the borescope, the ammonia turbine flow path (buckets and nozzles) can be inspected in detail and without turbine disassembly. The borescope inspections can give the turbine operator a complete photographic record of the turbine internal condition; hence, the inspections prevent surprises in the normal maintenance process by giving a better idea of what to expect when a unit is opened for a scheduled maintenance.

I.2.6 SAFETY

The floating ring turbine shaft seal mentioned in Section 3.2.2 is shown in Figure I.2-9. The design consists of two rings around the shaft which are held in place in the seal casing by a spring. The rings have a close radial clearance to the shaft. Sealing fluid is pumped into the annulus between the rings at a pressure higher than that of the ammonia to be sealed. The floating ring seal has no metal-to-metal contact between rotating and stationary components; hence, there is no limit to seal life. This characteristic, coupled with the simplicity of the design and consequent ease of turbine maintenance, makes the floating ring seal the choice to provide the turbine shaft sealing function.

I.2.7 COST

Cost for the turbine-generator frame sizes at 5 MW, 12.5 MW, and 25 MW net ratings, which were established during the conceptual phase of the OTEC study, are given in Tables I.2-7, I.2-8, and I.2-9, along with turbine-generator expected performance.

The costs include the turbine, generator, and accessory items such as controls, protective valving, lubrication system, turbine shaft sealing system, and the generator excitation system. Also included in the cost is technical direction of installation and freight allowed to nearest rail siding (f.o.b. factory).

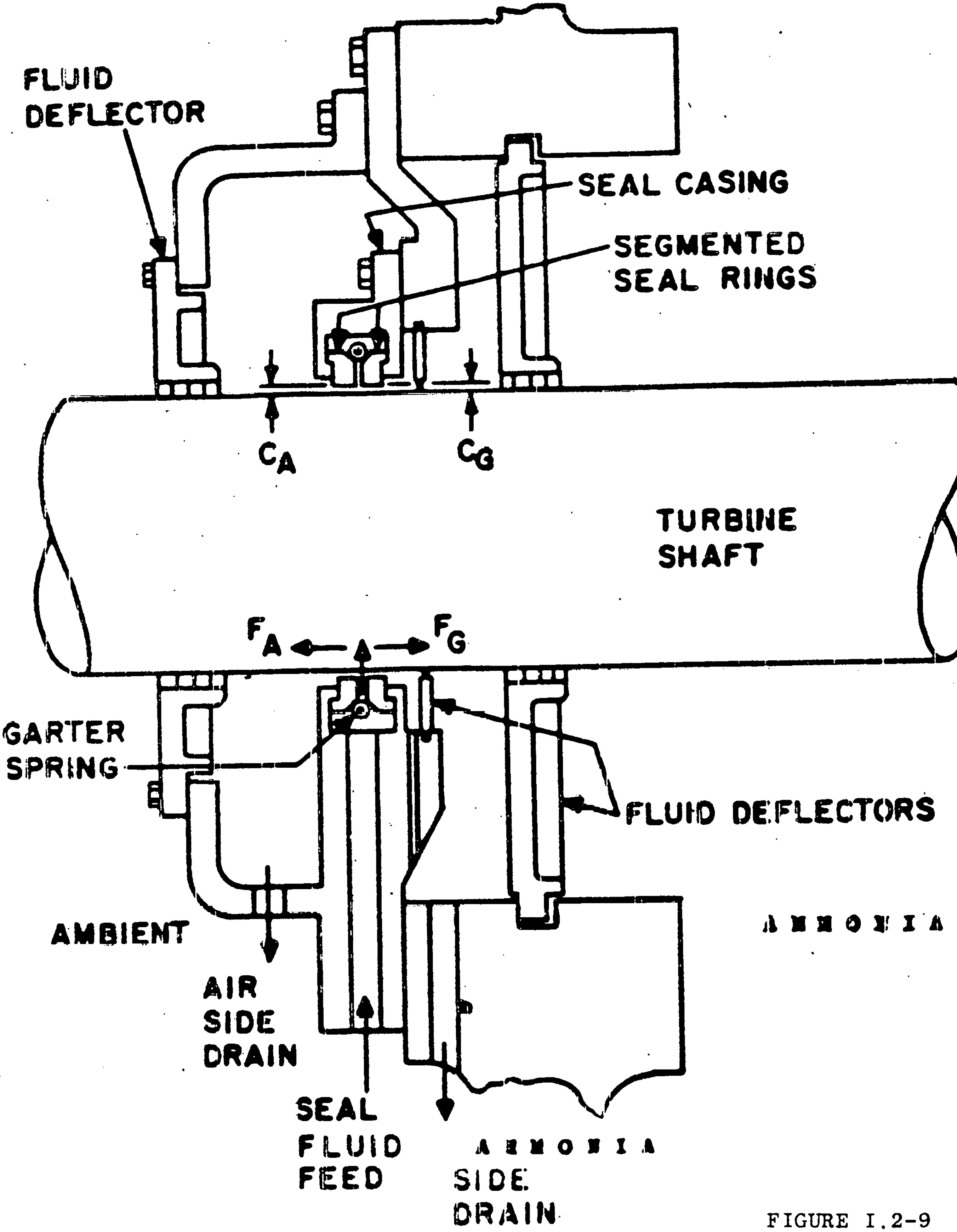

FLUID DEFLECTOR
SEAL CASING
SEGMENTED SEAL RINGS
C_A
C_G
TURBINE SHAFT
F_A
F_G
GARTER SPRING
FLUID DEFLECTORS
AMMONIA
AMBIENT
AIR SIDE DRAIN
SEAL FLUID FEED
AMMONIA SIDE DRAIN

FIGURE I.2-9

OTEC AMMONIA TURBINE COSTS

5000 KW (NET)
6350 KW (GROSS)

131.1 PSIA - 1% Moisture - 94.26A

Turbine Frame	Inlet Size Inches	Inlet Pressure Drop	Turbine/ Generator Speed RPM	Single Wheel Design: Flow Paths / Bucket Height / Pitch Diameter	NH_3 Rate Flow (G) / KW (N)	Turbine Generator Cost X 1000
182A	2 x 36	1%	3600	2 / 5.17" / 32.53"	317 #/KWH	$2,800
182B	1 x 42	2%	3600	2 / 5.34" / 32.02"	325 #/KWH	$2,300
182C	2 x 36	1%	1800	2 / 2.84" / 65.06"	317 #/KWH	$3,150
182C-1			1800/3600		323 #/KWH	$2,850
182D	1 x 42	2%	1800	1 / 5.36" / 64.04"	318 #/KWH	$2,750
182D-1			1800/3600		324 #/KWH	$2,450

TABLE I.2-7

OTEC AMMONIA TURBINE GENERATOR COSTS

12,500 KW (NET)
15,875 KW (GROSS)

131.1 PSIA - 1% Moisture - 94.26A

Turbine Frame	Inlet Size Inches	Inlet Pressure Drop	Turbine/Generator Speed RPM	Single Wheel Design Flow Paths / Bucket Height / Pitch Diameter	NH3 Rate Flow (G) / KW (N)	Turbine Generator Cost X 1000
183A	2 x 42	3%	1800	2 / 7.37" / 62.9"	328 #/KWH	$3,850
183A-1			1800/3600		333 #/KWH	$3,500
183B	2 x 48	2%	1800	2 / 6.66" / 64.04"	317 #/KWH	$4,300
183B-1			1800/3600		322 #/KWH	$3,900

TABLE I.2-8

OTEC AMMONIA TURBINE GENERATOR COSTS

25,000 KW (NET)
31,750 KW (GROSS)

131.1 PSIA - 1% Moisture - 94.26 PSIA

Turbine Frame	Inlet Size Inches	Inlet Pressure Drop	Turbine/ Generator Speed RPM	Single Wheel Design Flow Paths / Bucket Height / Pitch Diameter	NH_3 Rate Flow (G) / KW (N)	Turbine Generator Cost X 1000
183A	2 x 60	3%	1800	2 / 14.55" / 62.9"	324 #/KWH	$5,450
183A-1			1800/3600		327 #/KWH	$5,000
183B	2 x 60	4%	1800	2 / 13.45" / 61.76"	335 #/KWH	$5,450

TABLE I.2-9

Appendix J
PLATFORM INFLUENCES

The power system development project design requirements did not include the ocean system aspects of the OTEC plant. However, there are certain features of the ocean systems (platform, cold water pipe and water ducts, stationkeeping system) which have an influence on the performance and function of the power system module. These are platform motions, equipment orientation constraints, ocean systems, power required, and platform cost. Additional considerations, not evaluated, include site environment and location (fouling peculiarities, storm frequency and profile, current profile, distance to shipyard), on-board maintenance support capability, and power utilization.

J.1 PLATFORM MOTIONS

The platform motions imposed on the power system module results principally in performance degradation of the power module. As an example, the platform upward heave acceleration tends to strip liquid off vertical or horizontal tubes in the heat exchanger, conversely for downward heave acceleration. The net result is not precisely known; however, the overall effects are considered to be detrimental.

The effect of the weight of operational heat exchangers upon the platform structure caused by platform motions – heave, surge, or sway – is directly related to the magnitude of the platform motions. This load effect is transmitted directly into the platform structure and into the heat exchanger shell and thence to the tube baffles or support. Table J-1 summarizes four different platform motions for Sea State 6 (power system operating) and Sea State 9 (power system surviving).

Table J-1

PLATFORM MOTIONS SIGNIFICANT AMPLITUDE

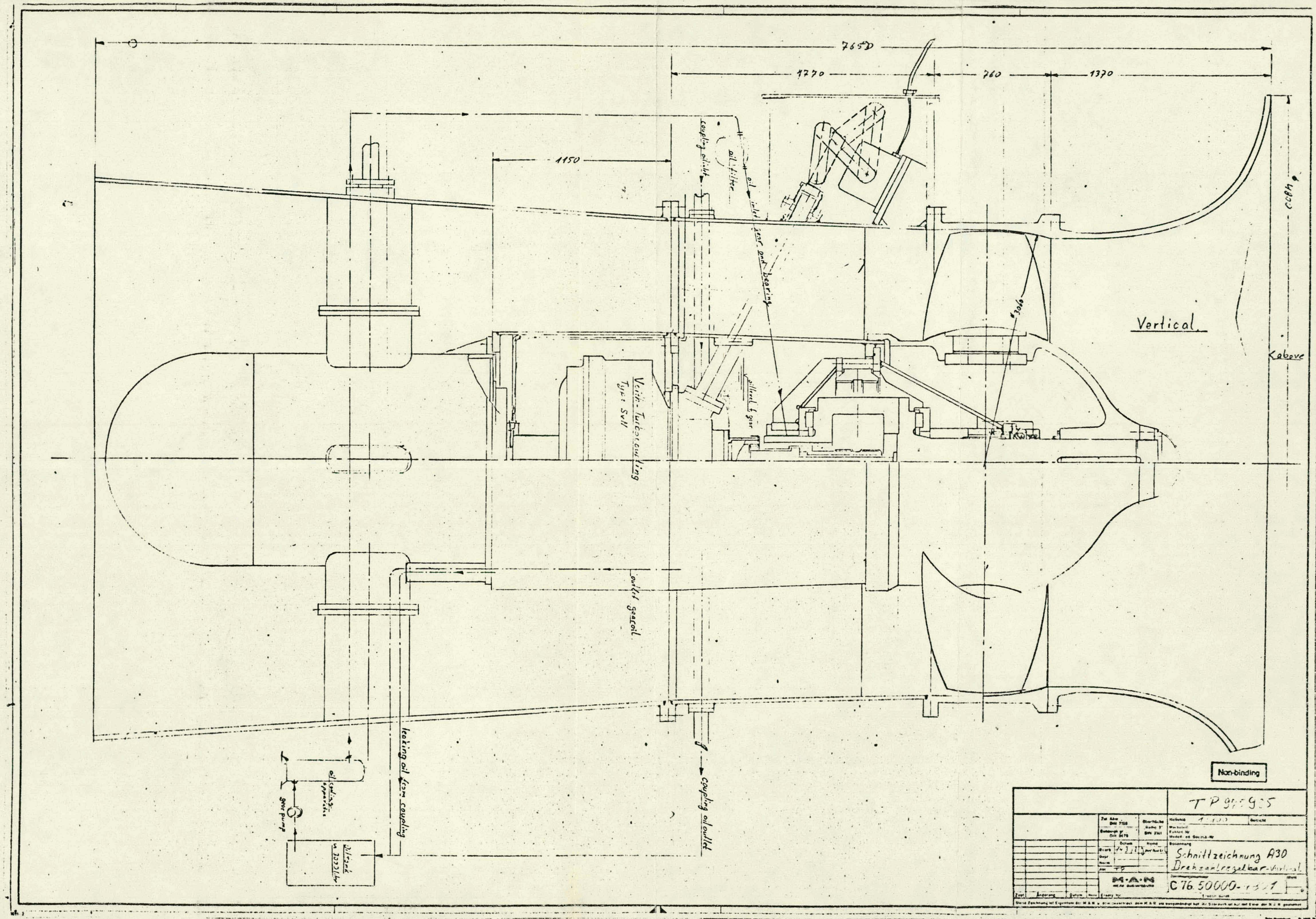
Item	Sea State 6 (Operate)					Sea State 9 (Survive)				
	Heave (ft)	Surge (ft)	Sway (ft)	Pitch (deg)	Roll (deg)	Heave (ft)	Surge (ft)	Sway (ft)	Pitch (deg)	Roll (deg)
Ship	3.7	1.6	2.6	0.6	2.2	32.2	14.6	15.3	6.2	14.8
Barge (Circular)	3.4	2.0	2.0	1.9	1.9	40.2	16.7	16.7	12.5	12.5
Submersible	1.6	1.1	1.4	0.6	0.7	11.2	9.5	10.2	3.4	10.5
Spar	0.7	0.8	0.8	0.3	0.3	8.2	8.7	8.7	2.5	2.5

J.3 POWER REQUIRED

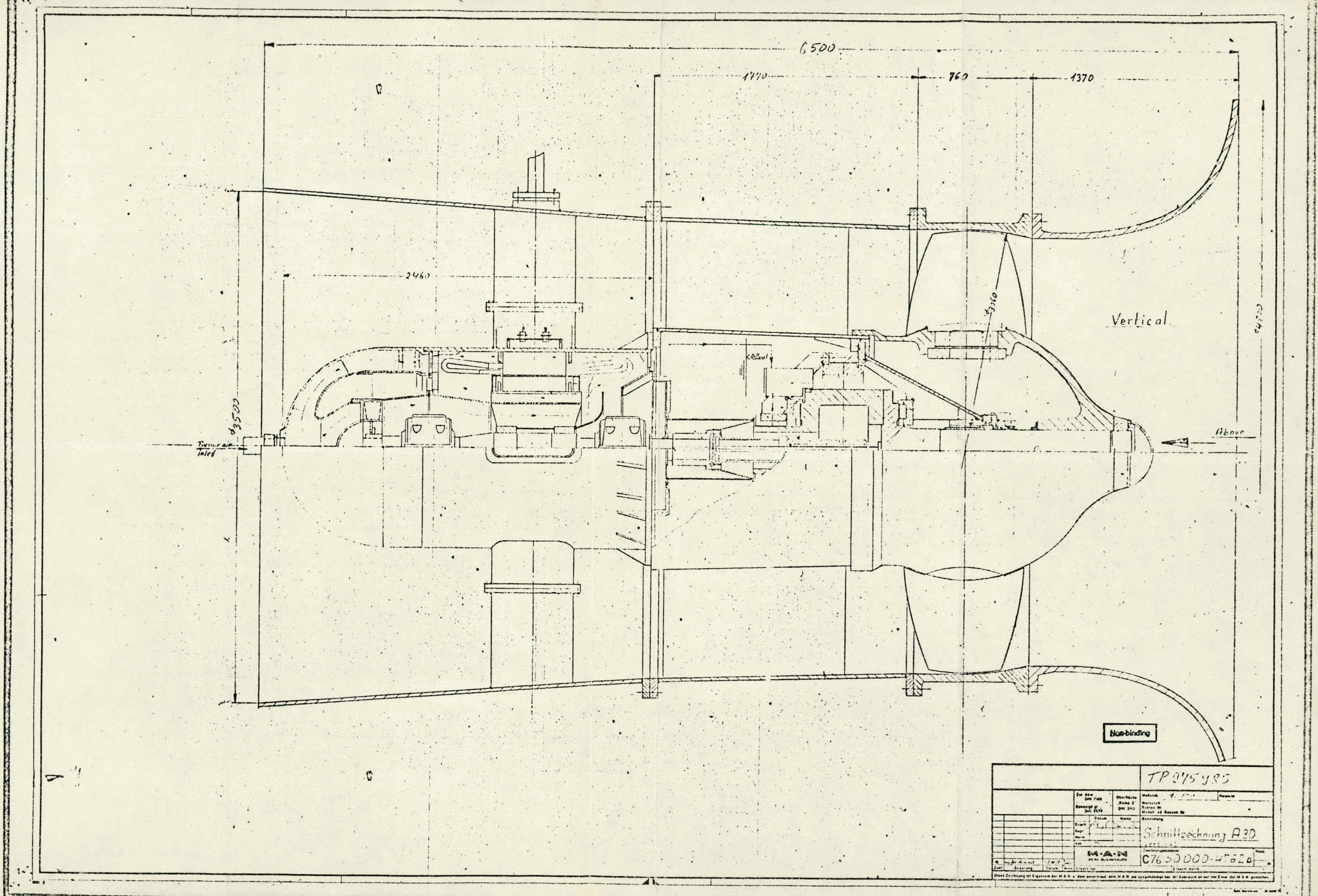
The power required for ocean systems operation such as hotel services, central control, platform stationkeeping, etc., were estimated. The platform to power system module power interface will be defined during preliminary design to ensure that the power system's gross power is large enough to provide a net 25 MW(e) at the busbar.

J.4 COST

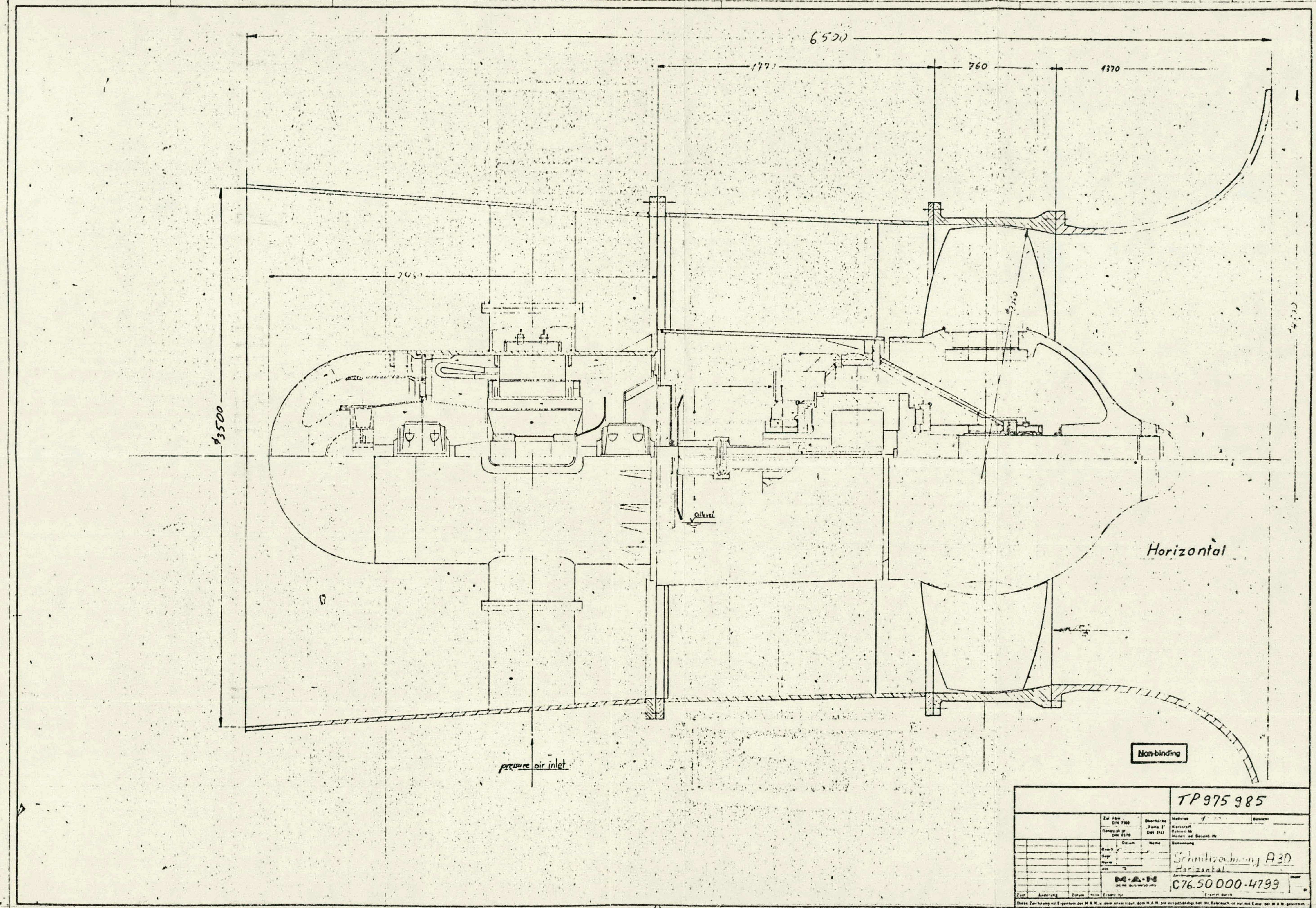
The impact on the platform due to power system module weight and volume were defined for several cases in order to determine the "best" power system module configuration. The result of this trade was inconclusive due to the wide range of platform cost partials. However, this does not negate the importance of integrating the platform and power system costs to arrive at the lowest plant cost.



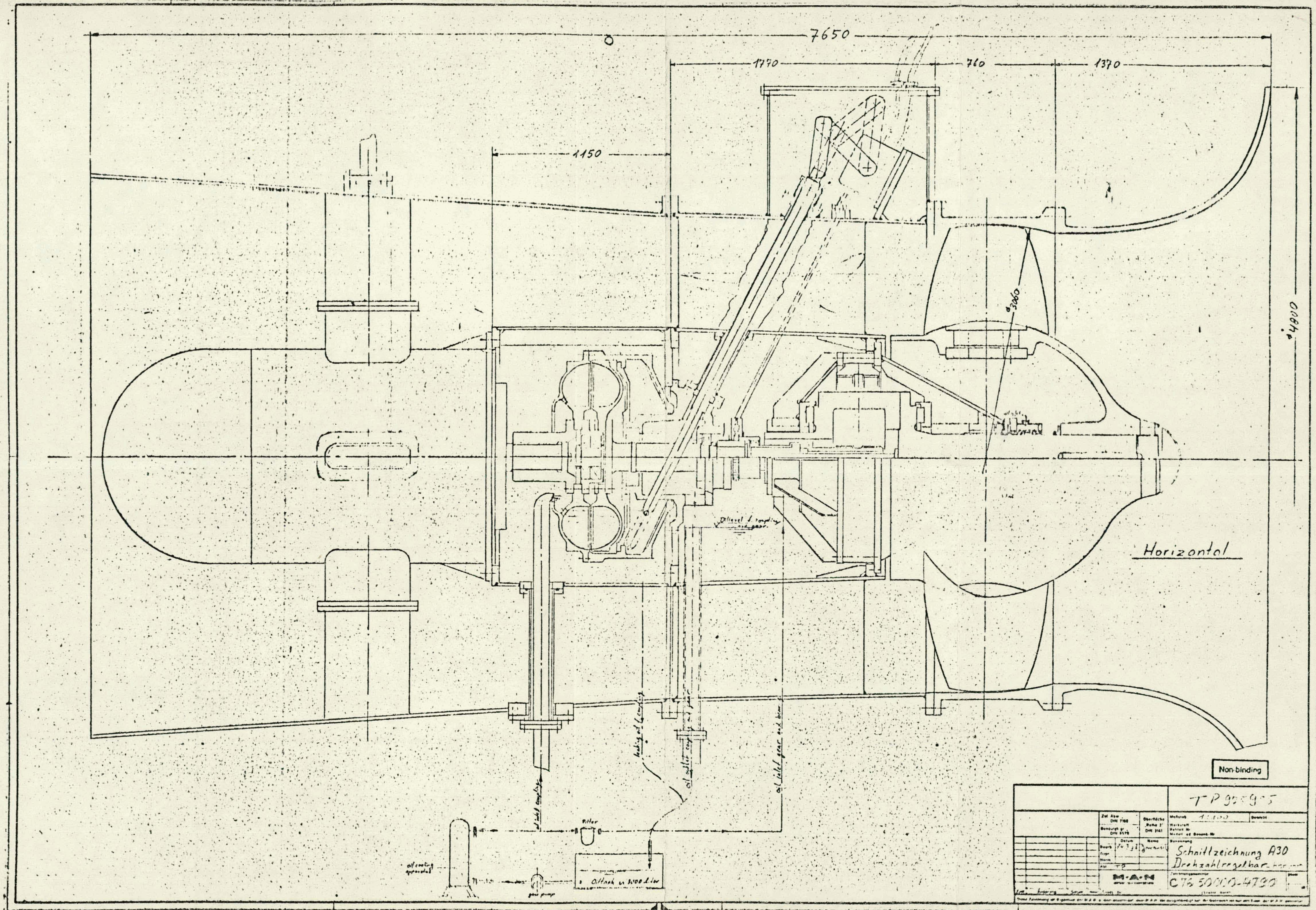
7650
1270
760
1370
1150
coupling oil inlet
oil filter
oil inlet gear and bearing
Voith-Turbocoupling
Type SVL
Vertical.
above
Non-binding
outlet gearoil
leaking oil from coupling
coupling oil outlet
gear pump
TP 9.. 9.5
1:100
Schnittzeichnung A30
Drehzahlregelbar-Vertical
M·A·N
C 76.50000-...1



6500
1770
760
1370
2460
⌀3500
⌀3140
Pressure air
Inlet
Ölablauf
Vertical
Above
Non-binding
TP 975 985
Maßstab 1:100
Schnittzeichnung A30
M·A·N
C76 50 000-4762a
Zust
Änderung
Datum
Ersatz für
Ersatz durch
Diese Zeichnung ist Eigentum der M A N



6500
1771
760
1370
2450
ø3500
Oilevel
Horizontal
pressure air inlet
Non-binding
TP 975 985
Schnittzeichnung A30
Horizontal
M·A·N
C76.50 000-4799
Diese Zeichnung ist Eigentum der M.A.N.



7650
1770
760
1370
1150
ø3060
ø4800
Horizontal
Non-binding
Maßstab 1:100
Benennung
Schnittzeichnung A30
Drehzahlregelbar
M·A·N
C76 50000-4790

APPENDIX E

APPENDIX E

Heat Exchanger Supporting Data

E1 Analyses/Configuration

E2 Contract Tooling

E3 Manufacturing Plan

E4 Specification

E5 Evaporator Ammonia Liquid Distribution System

APPENDIX E1

ANALYSES/CONFIGURATION

This appendix includes supplementary data pertaining to the 50 MW(e), 10 MW(e) and test article heat exchangers. The content of this appendix is summarized below.

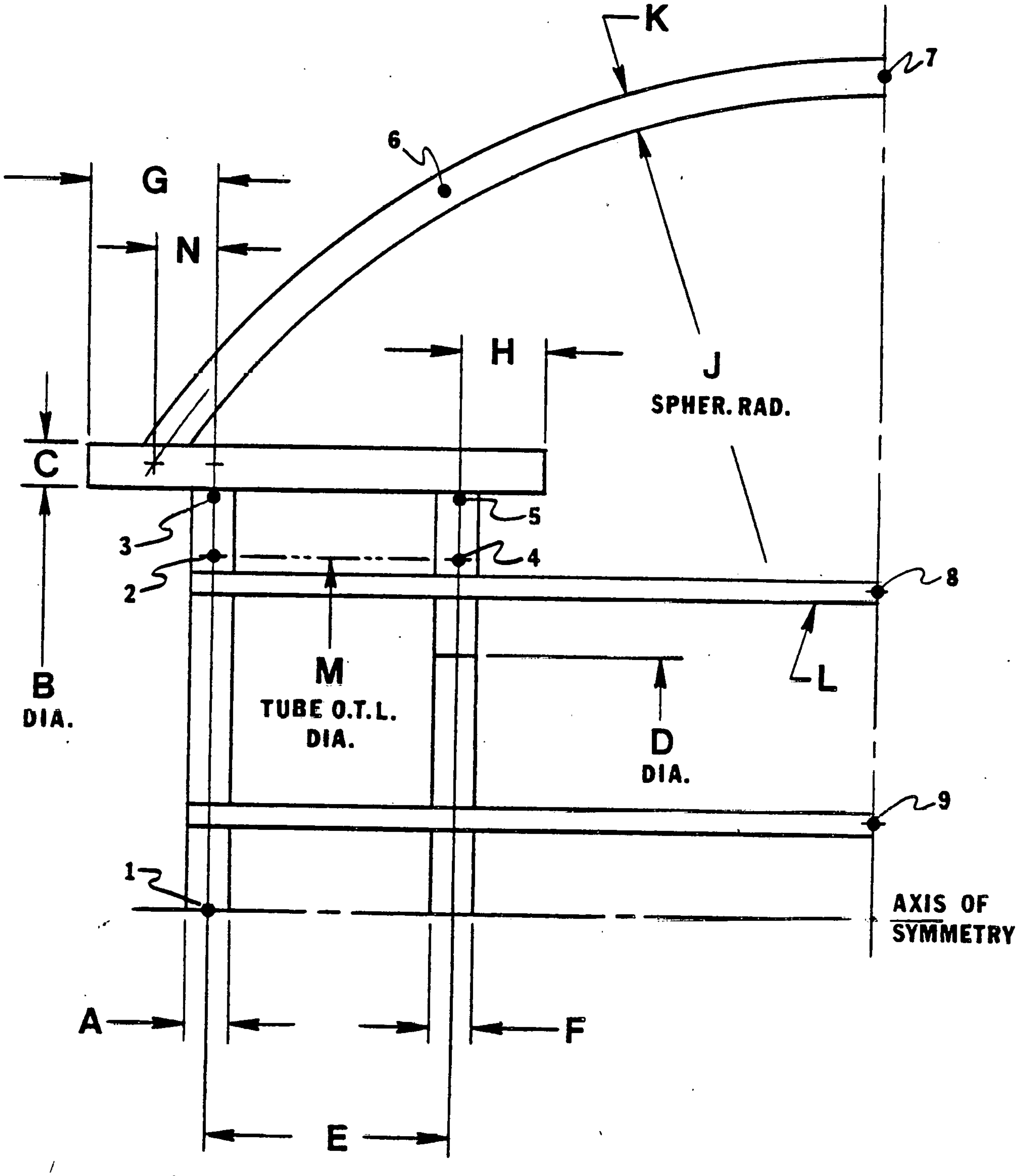
Figure E-1 and Tables E-3 to -6 show the major dimensions and critical stress results for the 50 MW and 10 MW computer stress analysis.

Tables E-1 and -2 summarize the additional cost trade studies which were made for the 50 MW(e) heat exchangers using the enhanced tubes.

Tables E-7, -8 and -9 list the tooling required for the three sizes of heat exchangers. The cost for material and labor to build each line item has been estimated. These costs are based on engineering judgement only from previous experience.

The Manufacturing Plan shows the sequence of major operations for fabricating the 10 MW heat exchangers.

The Specification is preliminary based on the results of the preliminary design study. Areas which need additional definition have been indicated.



FIXED TUBESHEET MODEL

FIGURE E-1

TABLE E-1

HEAT EXCHANGER COST COMPARISONS

50 MWe, Aluminum, 1 1/2" OD Tube with .1025" Wall

Case	Base Evap.	Base Cond.	86% Evap.	86% Cond.	84.5% Evap.	84.5% Cond.	83% Evap.	83% Cond.
Tube No.	89,000	103,000	89,000	103,000	89,000	103,000	89,000	103,000
Tube Length, ft	50.88	53.268	53.012	55.384	54.832	57.640	56.854	60.073
Tubesheet Dia., ft	55	55	55	55	55	55	55	55
Shell Sphere Dia., ft	75	76.5	76.4	78	77.7	79.7	79.1	81.4
Shell Wall in.	6.25	6.37	6.37	6.50	6.50	6.75	6.62	6.75
No. of Baffles	8	8	8	8	9	9	9	9
$ per MWe	213.2	240.6	220.6	249.5	229.4	261.5	236.7	270.7

Notes:

1. Tubing = $1.40/ft
2. Tubesheets = 1 1/2" thick, inner tubesheet = 10% area
3. Transition ring = 16" x shell thickness
4. Tube support ring = 1" x 12"
5. Tube supports = 1" thick

HEAT EXCHANGER COST COMPARISONS (Cont'd.)

50 MWe, Aluminum, 1 1/2" OD Tube with .1025" Wall

Case	N1 Evap.	N1 Cond.	N2 Evap.	N2 Cond.	N3 Evap.	N3 Cond.	N4 Evap.	N4 Cond	N5 Evap.	N5 Cond.
Tube No.	103,000	125,000	120,000	146,000	108,000	125,000	120,000	140,000	78,000	94,000
Tube Length, ft	45.888	45.369	37.724	38.903	40.45	43.07	35.82	38.63	62.45	62.8
Tubesheet Dia., ft	59	59	63	65	59	59	63	63	52	52
Shell Sphere Dia., ft	74.7	74.4	73.4	75.8	71.5	73.0	72.5	73.9	81.7	81.7
Shell Wall in.	6.25	6.25	6.25	6.37	6.0	6.12	6.12	6.25	6.75	6.75
No. of Baffles	7	7	6	6	7	7	6	6	10	10
$ per MWe	220.1	247.6	214.7	252.0	207.5	237.7	206.9	241.5	233.1	261.3

TABLE E-1 (cont'd.)

TABLE E-2

50 MW(e) Cost Trade Study - Evaporator

Case	1	2	3	4
Position	Immersed	Immersed	Submerged	Submerged
Tube OD, in.	1.5	2.0	1.5	2.0
Tube Min. Wall, in.	.084	.090	.078	.082
Tubes	89,350	60,567	87,049	58,998
Shell Dia., ft.	72.7	79.8	71.8	78.8
Shell Wall, in.	4.25	4.62	3.25	3.43
Tube Length, ft.	51.6	56.7	50.7	56.0
$/MW(t)	4105	4200	3760	3779
$/KW(e)	182	186	167	167

NOTE: Based on aluminum heat exchangers with spherical shell and enhanced tubes as shown in Table D of Lockheed LMSC-D566744, Conceptual Design report.

TABLE E-3

50 MW(e) FIXED TUBESHEET MODEL DESCRIPTION

CASE NO.	MODEL GEOMETRY DIMENSIONS, INCHES (REF. FIGURE E-1)												
	A	B	C	D	E	F	G	H	J	K	L	M	N
1	1.5	617.2	6	450	10.5	1.5	2.5	2.5	434.52	6	*	594.951	0
2	1.5	617.2	6	450	10.5	1.5	12	2.5	434.52	6	*	594.951	0
3	1.5	617.2	6	517.5	10.5	1.5	2.5	2.5	434.52	6	*	594.951	0
4	1.5	617.2	6	517.5	10.5	1.5	2.5	12.78	434.52	6	*	594.951	0
5	1.5	617.2	6	571.94	10.5	1.5	2.5	2.5	434.52	6	*	594.951	0
6	1.5	617.2	6	571.94	10.5	1.5	2.5	22	434.52	6	*	594.951	0
7	1.5	606.95	6	571.94	10.5	1.5	2.5	2.5	430.89	6	*	594.951	0
8	1.5	617.2	6	571.94	10.5	2	2.5	2.5	434.52	6	*	594.951	0

Materials:

Shell and Tubesheets . . . . 5083-H321 Al Alloy
Tubes . . . . 5052-H32 Al Alloy

*1.5" O.D. x .1025 Min. Wall

TABLE E-4

50 MW(e) FIXED TUBESHEET ANALYSIS SUMMARY

Location, Ref. Fig. E-1	DESCRIPTION	COMPUTED RESULTS/CODE ALLOWABLE*							
		CASE NO.							
		1	2	3	4	5	6	7	8
1	Membrane Stress in Center of Tubesheet	9,230	7,990	9,420	8,840	9,525	8,890	9,310	9,730
		12,000	12,000	12,000	12,000	12,000	12,000	12,000	12,000
2	Bending Stress in Outermost Perforated Region	23,480	19,545	23,030	24,320	22,670	26,105	25,100	20,015
		36,000	36,000	36,000	36,000	36,000	36,000	36,000	36,000
3	Bending Stress at Ring Junction	17,940	20,580	17,645	16,700	17,560	15,360	16,670	15,170
		36,000	36,000	36,000	36,000	36,000	36,000	36,000	36,000
4	Bending Stress in Outermost Perforated Region	11,970	12,050	12,815	13,170	12,220	12,750	16,940	9,920
		36,000	36,000	36,000	36,000	36,000	36,000	36,000	36,000
5	Bending Stress at Ring Junction	14,340	15,730	14,910	12,850	15,045	11,105	15,515	14,710
		36,000	36,000	36,000	36,000	36,000	36,000	36,000	36,000
6	Maximum Bending Stress in Spherical Shell	10,400	10,490	10,400	10,345	10,405	10,310	10,315	10,320
		15,000	15,000	15,000	15,000	15,000	15,000	15,000	15,000
7	Membrane Stress in Spherical Shell at Centerline	8,900	8,880	8,895	8,905	8,895	8,915	8,795	8,880
		10,000	10,000	10,000	10,000	10,000	10,000	10,000	10,000
8	Maximum Compressive Stress on Outermost Tubes**	-714	-1,162	-508	-386	-531	-196	-2,169	-926
		-2,479	-2,479	-2,479	-2,479	-2,479	-2,479	-2,479	-2,479
9	Stress on Central Tubes	535	535	535	535	535	535	535	535
		10,300	10,300	10,300	10,300	10,300	10,300	10,300	10,300

*Rounded off to the nearest 5 psi - Case No. 5 is reference design

**Allowable stress given is the Euler Critical Buckling Stress, based on a support plate spacing of 6'-0" (72") and a safety factor of 2.

TABLE E-5

10 MW(e) FIXED TUBESHEET MODEL DESCRIPTION

CASE NO.	MODEL GEOMETRY DIMENSIONS, INCHES (REF. FIGURE E-1)												
	A	B	C	D	E	F	G	H	J	K	L	M	N
1	1	351.5	2.5	324.38	10.813	1	2.5	2.5	241.63	2.5	*	340.55	0
2	1	351.5	2.5	324.38	10.813	1	2.5	2.5	241.63	2	*	340.55	0
3	1	351.5	2.5	335.19	10.813	1	2.5	2.5	241.63	2	*	340.55	0
4	1	351.5	2.5	-	-	-	2.5	-	241.63	2	*	340.55	0
5	1	351.5	2.5	-	-	-	2.5	-	241.63	1.5	*	340.55	0
6	1.5	351.5	2.5	-	-	-	2.5	-	241.63	1.5	*	340.55	0
7	1	351.5	2.5	-	-	-	2.5	-	241.63	2	*	340.55	1
8	1	351.5	2.5	-	-	-	2.5	-	241.63	2	*	340.55	-1
9	1	351.5	2.5	-	-	-	2.5	-	241.63	2	*	340.55	2
10.	1	351.5	2.5	-	-	-	2.5	-	241.63	2	*	340.55	0

Materials:

Shell . . . . SA-516 GR. 70 Carbon Steel

Tubesheets . . . . Titanium Clad SA-516 GR. 70 Carbon Steel

Tubes . . . . SB-338 GR. 2 Titanium

*1.0" O.D. x .028 Min. Wall

TABLE E-6

10 MW(e) FIXED TUBESHEET ANALYSIS SUMMARY

Location Ref Fig. E-1	DESCRIPTION	COMPUTED RESULTS/CODE ALLOWABLE*									
		CASE NO.									
		1	2	3	4	5	6	7	8	9	***10
1	Membrane Stress in Center of Tubesheet	10,070	11,150	11,180	11,310	12,605	11,545	11,195	10,980	11,060	-3,295
		23,300	23,300	23,300	23,300	23,300	23,300	23,300	23,300	23,300	-12,300
2	Bending Stress in Outermost Perforated Region	22,395	26,785	26,780	29,345	36,075	36,660	28,147	33,370	27,985	-7,225
		34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	-12,300
3	Bending Stress at Ring Junction	22,521	24,710	24,640	34,910	40,070	27,600	31,270	37,245	27,180	-7,985
		34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	-12,300
4	Bending Stress in Outermost Perforated Region	9,795	10,995	11,755	-	-	-	-	-	-	-
		34,950	34,950	34,950	-	-	-	-	-	-	-
5	Bending Stress at Ring Junction	11,080	11,660	12,020	-	-	-	-	-	-	-
		34,950	34,950	34,950	-	-	-	-	-	-	-
6	Maximum Bending Stress in Spherical Shell	14,100	17,800	17,800	17,930	24,120	23,750	17,900	18,000	17,885	-4,560
		34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	34,950	-12,300
7	Membrane Stress in Spherical Shell at Centerline	12,050	15,040	15,040	15,030	19,995	19,785	15,060	15,010	14,235	-3,855
		23,300	23,300	23,300	23,300	23,300	23,300	23,300	23,300	23,300	-12,300
8	Maximum Compressive Stress on Outermost Tubes**	-1,215	-1,795	-1,805	-2,010	-2,930	-3,780	-1,680	-2,235	-1,290	-315
		-2,040	-2,040	-2,040	-2,040	-2,040	-2,040	-2,040	-2,040	-2,040	-2,040
9	Stress on Central Tubes	1,040	1,040	1,040	1,040	1,040	1,035	1,040	1,040	1,040	-270
		10,600	10,600	10,600	10,600	10,600	10,600	10,600	10,600	10,600	-2,040

*Rounded off to the nearest 5 psi - Case No. 4 is reference design

**Allowable stress given is the Euler Critical Buckling Stress, based on a support plate spacing of 66.5" and a safety factor of 2

***Stresses due to an external pressure of 43 psi

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TABLE E-7

Appendix E2

LIST OF CONTRACT TOOLING FOR FABRICATION OF

50 MW(e) CONDENSER AND EVAPORATOR

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
1	Checking fixture for formed & edge prepared segments	$ 10,000
2	Positioning fixture for set up of segments to a hemisphere	20,000
3	Holding clamps & fixture for welding positioners	20,000
4	Welding positioners (2) 350 ton capacity	900,000
5	Column type welding head positioners (2)	200,000
6	Turning rolls for circumferential welding (2)	100,000
7	Fabricated rings for rolling spheres	80,000
8	Turning trunnion components & journals	50,000
9	Turntable for vert- axis rotation (1)	125,000
10	Automatic plasma cutting equipment & set up for making approximately 55 ft. dia. cuts	100,000
11	Templates for checking contour & shrinkage at spherical seams	1,500
12	Braces for holding contour of the hemi-heads while welding	30,000
13	Fixture for positioning and aligning the two hemispheres during assembly	30,000
14	Templates for layout of openings in the spheres	2,000
15	Plasma hole cutting travel rings for nozzle openings	10,000
16	Scaffolding and staging for tube insertion and welding of tubes to tubesheets	132,000
17	Fixture for positioning flat tube and support plate segments for welding	10,000
18	Moline drilling tooling	12,000
19	Racks for shipping drilled tubesheet & support plate segment plates to assembly plant	15,000
20	Fixture for set-up & positioning the drilled tube and support plate segment plates for welding into full tube and support plates	12,000

TABLE E-7 (cont'd.)

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
21	Handling fixture for full tube and support plates	10,000
22	Checking templates for radius of rolled tubesheet bar ring segments	1,500
23	Assembly and set-up fixture for tubesheet subassembly	25,000
24	Support frame for automatic welding of the tubesheet subassembly	10,000
25	Contour cutting templates for internal bracing	4,000
26	Forming templates for support plate rings	1,500
27	Nozzle contour cutting templates	6,000
28	Reinforcing plates cutting and contour templates	6,000
29	Bundle assembly jigs and fixtures	120,000
30	Support structure for sphere used in tube assembly hydrotesting and in transport to the job site (2)	300,000
31	Tube end preparing tools	5,000
32	Tube protection in plant including covers, etc.	24,000
33	Tube-to-tubesheet welding equipment	200,000
34	Tube cut-back tools	10,000
35	Tube staking tools	5,000
36	Tube expanders	15,000
37	Cost of inhibitors for seawater hydrofill	120,000
38	Cost of a hydrofill pump	50,000
39	Test fixture for testing individual tube-to-tubesheet welds (2)	10,000
40	Special Radiographic Test Equipment	100,000
41	Aero-Go water casters for moving the final assembly to the barge	500,000
42	Miscellaneous Tooling	338,000
	TOTAL	$3,720,500

Note: Schedule requirements may affect tooling needs (quantity) for some items such as Nos. 2, 5, 6, 7, etc.

TABLE E-8

LIST OF CONTRACT TOOLING FOR FABRICATING FOR

10 MW(e) CONDENSER AND EVAPORATOR

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
1	Checking Template for Spherical Curvatures	$ 1,000
2	Checking and set up fixture for making hemispheres	50,000
3	Bracing for holding contours when welding each hemisphere	8,000
4	Checking template to measure controlled shrinkage at seams	1,000
5	Fixture for set up of 2 hemispheres	25,000
6	Construct "T" rail tires for assembly to each sphere - all around	80,000
7	Templates and fixture for layout and cutting openings in spheres	6,500
8	Turning trunnion components welded to each sphere	30,000
9	Beam and lifting slings and rigging for use in lifting and turning spheres	30,000
10	Insulated brick hearth with steel supports for stress relief of spheres	7,500
11	Bracing and holding racks for insulation during stress relief	4,000
12	Right angle plate holder for stripping back titanium-clad	20,000
13	Positioning fixture for holding plates for welding	5,000
14	Racks for shipping 15' x 30' plates from Dansville to Panama City	12,000
15	Fixture for setting up half plates for welding	6,000
16	Handling fixtures for tube and support plates	8,000
17	Checking template for radius of rolled bar ring segments	1,000
18	Set up and assembly fixture for tubesheet subassemblies	10,000
19	Support frame for automatic welding tubesheet assembly on weld positioner	7,500
20	Contour cutting templates for braces	3,000
21	Support plate rings forming templates	1,000

TABLE E-8 (Cont'd)

LIST OF CONTRACT TOOLING FOR FABRICATING FOR
10 MW(e) CONDENSER AND EVAPORATOR (cont'd.)

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
22	Nozzle contour cutting templates	$ 4,500
23	Reinforcing plates cutting and contour templates	4,500
24	Fixture for flame cutting tubesheet openings in spheres	18,000
25	Bundle assembly aligning rods, spacers, temporary bracing and work platforms	40,000
26	Stationary support for holding spheres during tube assembly, hydrotesting and shipping	60,000
27	Tube end prep tools and set up	3,000
28	Tube protection at plant, includes protective covers, etc.	6,000
29	Scaffolding for use during welding and assembly	24,000
30	Tube-to-tubesheet weld equipment	120,000
31	Tube cut-back tools and tube staking tools	6,000
32	Tube expanders	10,000
33	Cost of inhibitors for seawater hydro-fill	14,000
34	Cost of pump rental	6,000
35	Special tool for testing individual tube welds	6,000
36	Miscellaneous Tooling	63,800
	TOTAL	$702,300

TABLE E-9

LIST OF CONTRACT TOOLING FOR FABRICATING FOR TEST ARTICLE CONDENSER AND EVAPORATOR

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
1	Checking Template for Spherical Curvatures	$ 800
2	Checking and set up fixture for hemispheres	20,000
3	Bracing for holding contours when welding hemisphere	5,000
4	Checking template to measure controlled shrinkage at seams	800
5	Fixture for set up of 2 hemispheres	10,000
6	Construct "T" rail tires for assembly to each sphere - all around	20,000
7	Templates and fixture for layout and cutting openings in spheres	1,200
8	Beam and lifting slings and rigging for use in lifting and turning spheres	10,000
9	Positioning fixture for holding plates for welding	4,000
10	Tooling for Moline machine	12,000
11	Racks for shipping plates from Dansville to Panama City	4,800
12	Fixture for setting up half plates for welding	4,500
13	Handling fixtures for tube and support plates	6,000
14	Checking template for radius of rolled bar ring segments	800
15	Set up and assembly fixture for tubesheet subassemblies	8,000
16	Support frame for automatic welding tubesheet assembly on weld positioner	5,000
17	Contour cutting templates for braces	2,400
18	Support plate rings forming templates	800
19	Nozzle contour cutting templates	3,000
20	Reinforcing plates cutting and contour templates	3,000
21	Fixture for plasma cutting tubesheet openings in spheres	15,000
22	Bundle assembly aligning rods, spacers, temporary bracing and work platforms	20,000

TABLE E-9 (Cont'd)

LIST OF CONTRACT TOOLING FOR FABRICATING FOR

TEST ARTICLE CONDENSER AND EVAPORATOR (cont'd.)

ITEM NUMBER	TOOL DESCRIPTION	TOTAL
23	Tube end prep tools and set up	$ 1,000
24	Tube protection at plant, includes protective covers, etc.	1,800
25	Scaffolding for use during welding and assembly	8,000
26	Tube-to-tubesheet weld equipment	30,000
27	Tube cut-back tools and tube staking tools	1,000
28	Tube expanders	1,200
29	Cost for inhibitors for seawater hydro-fill	4,000
30	Cost of pump rental	3,000
31	Special tool for testing individual tube welds	6,000
32	Miscellaneous Tooling	18,300
	TOTAL	$231,400

Appendix E3

MANUFACTURING PLAN

10 MW - 40 FT. DIA. SHELL - APPROX. 500 TONS (EACH)
(STEEL SHELL - TITANIUM TUBES - TITANIUM CLAD TUBESHEET)

1. Procurement of shell plate

   1.1 FWEC purchase shell materials. FW QC inspect shell body material at mill for dimensions, markings, and finish. Review mill test certificates versus material specification and order requirements. Ship direct to shell/segment fabricator.

   1.2 Select vendor for making shell subassemblies made from segments of his manufacture. Full hemispheres or part hemispheres should be made dependent on vendor capacity to make and ship. Dimensional tolerance checks are to be made at vendor plant. Foster Wheeler Quality Control will over-check. Tolerances will be determined to maintain required fit-up tolerances. Excess is to be allowed for trimming at the tubesheet opening. Foster Wheeler will supply materials. Full X-ray of all joints and partial data reports per ASME VIII, Division 1. Foster Wheeler receive and inspect shell subassemblies for dimensions, finish and markings, and supplier Code certificates.

2. Fabricate 2 hemispheres and assemble sphere (1 required per unit)

   2.1 Set up and stitch weld subassemblies in positioning fixture to form hemispheres. Dress grind if necessary for a good fit-up. Brace to hold cross section and end contours.

   2.2 Remove from positioning fixture and weld the hemispheres. Position for best welding. Use controlled welding sequence to minimize distortions.

   2.3 X-ray all seams.

   2.4 Set up 2 hemispheres to make 1 sphere with axis vertical. Weld circle seam in horizontal position. Double weld with back gouge. Use controlled sequence welding to minimize distortion.

   2.5 X-ray the circle seam.

   2.6 Turn the unit 90° to axis horizontal. Assemble the "T" rails for rolling the unit. Roll for best position while working the nozzles.

   2.7 Lay out and flame cut the nozzle openings.

   2.8 Set up and weld nozzles to shell, set up and weld attachments to O.D. of shell, including turning trunnions.

   2.9 Turn unit 90° to axis vertical position using trunnions.

   2.10 Stress relieve unit. Brace and insulate for stress relief. Prepare for stress relief (gas-fired). Subcontract stress relief. Dismantle set-up.

2. Fabricate 2 hemispheres and assemble sphere - 1 required per unit (cont'd.)

2.11 Sand blast all surfaces of sphere, coat with preservative.

2.12 Hold for assembly of tube and support plates.

3. Tubesheet fabrication and drilling (2 titanium tubesheets per unit = 4 half plates per unit)

3.1 Receive and inspect titanium clad plates (maximum size = 92 x 295").

3.2 Lay out and plasma cut pieces.

3.3 Strip back titanium at weld seams to base metal at O.D.

3.4 Prepare edges.

3.5 Set up and weld plates into 15' x 30' plates SMAW/SAW. Back gouge and spot X-ray. Flatten if necessary.

3.6 Stack plates on Moline and strap.

3.7 Moline drill and ream.

3.8 Break stacks and deburr the plates.

3.9 Clean; preserve; pack; and ship to final assembly.

3.10 Set up half plates in positioning fixture.

3.11 Weld and turn over (autoweld) SMAW/SAW. Minimize distortion.

3.12 Back chip.

3.13 Weld other side.

3.14 Back gouge and final weld.

3.15 Grind if necessary.

3.16 Spot X-ray.

3.17 Hold for assmebly.

4. Support plates (6 per unit = 12 half plates)

4.1 Receive and inspect plates (grit blasted plates).

4.2 Lay out and flame cut large plates.

4.3 Prepare edges for welding.

4.4 Stack plates and strap.

4.5 Moline drill and ream.

4. Support plates (6 per unit = 12 half plates) (cont'd.)

   4.6 Break stack and deburr the plates.

   4.7 Pack, preserve, and ship to final assembly.

   4.8 Set up half plates in positioning fixture.

   4.9 Weld and turn over plate to minimize distortion SMAW/SAW.

   4.10 Back gouge.

   4.11 Weld other side.

   4.12 Grind if necessary.

   4.13 Hold for assembly.

5. Tubesheet transition rings (2 per unit)

   5.1 Receive and inspect plates.

   5.2 Flame cut to size.

   5.3 Prepare edges.

   5.4 Weld plate to form 3 segments of ring, SMAW/FCAW.

   5.5 Roll segments.

   5.6 Set up and weld segments into 1 ring SMAW/FCAW.

   5.7 Back gouge welds.

   5.8 Spot X-ray.

   5.9 Grind if necessary.

   5.10 Hold for assembly.

6. Tubesheet end assembly (2 required per unit)

   6.1 Position tubesheet and hold flat.

   6.2 Set up the transition ring and clamp to hold flat.

   6.3 Weld the tubesheet to ring. Minimize distortion by proper sequencing of weld beads. Turn over twice. Use SAW.

   6.4 Back gouge the weld.

   6.5 Check dimensions and distortion.

   6.6 Hold for assembly.

7. Bracing and other miscellaneous internals

   7.1 Set up and cut sets of bracing and other internal parts: demisters and baffles including upright supports, diagonal braces, and miscellaneous parts.

   7.2 Bevel the ends if necessary.

   7.3 Contour cut the ends of bracing and parts.

   7.4 Hold for assembly.

8. Support plate peripheral rings (6 per unit)

   8.1 Set up and cut segments in the flat.

   8.2 Roll bars.

   8.3 Weld bars together. Flux core welding FCAW.

   8.4 Hold for assembly.

9. Nozzles

   9.1 Lay out and cut nozzle material.

   9.2 Roll.

   9.3 Weld straight seam (back gouge and weld; X-ray if necessary) SMAW/FCAW/SAW.

   9.4 Hold for assembly.

10. Saddle plate for nozzle reinforcement

   10.1 Lay out and flame cut segments.

   10.2 Edge prepare segments.

   10.3 Form the segments.

   10.4 Weld segments into rings SMAW/FCAW; back gouge welds.

   10.5 Hold for assembly.

11. Assembly of tubesheet and support plates

   11.1 With unit set with axis vertical, flame cut 1 end opening to size for assembly of tubesheet subassembly.

   11.2 Set up and weld 1 tubesheet/ring subassembly at top of unit. Use tubesheet with ammonia headers.

   11.3 Local stress relieve the circle seam. Use electrical resistance heating.

11. Assembly of tubesheet and support plates (cont'd.)

11.4 Turn the unit 180° to axis vertical position using trunnions.

11.5 Set up all internals including support plates and bracing, install ammonia feed tubes in support plates, ammonia feed piping and supports, dryer supports, noncondensibles exhaust duct (condenser only)
Note: Tubesheet and support plates must be aligned for smooth passage of tubes.

11.6 Flame cut shell end opening to size for assembly of second tubesheet. Protect internals from flame cutting debris.

11.7 Set up and weld the second tubesheet/ring subassembly at top of unit.

11.8 Local stress relieve the circle seam. Use electrical resistance heating.

11.9 Turn the unit 90° to axis horizontal position for tubing using trunnion.

11.10 Roll the unit to best position for finish welding of bracing, ammonia spray tubes, baffles, headers, dryers, and supports.

11.11 Lift the unit to a stationary support designed to hold the unit for tubing, hydrotest, and for transport.

11.12 Remove the rolling rings.

11.13 Hold for tube assembly and testing.

12. Tubes - Titanium

12.1 Tubes to be fully inspected at vendor's plant; end preparation should be done at vendor's plant.

12.2 Inspect packaging at vendor's plant.

12.3 Receive and inspect for shipping damage and packaging. Review mill certifications.

12.4 Store to prevent physical damage or damage by weather.

13. Bundle Assembly

13.1 Stage tube boxes and work platforms inside and outside shell.

13.2 Tube the bundle.

13.3 Weld tubes to tubesheet (develop program for welding finned titanium tubes). Set and stake tube at one end, weld staked end, inspect visually, and expand the inlet end.

13.4 Visually inspect tube welds and repair as needed.

13. Bundle Assembly (cont'd.)

13.5 Cut back the outlet end of tubes if necessary.

13.6 Weld tubes to tubesheet at opposite end and inspect visually.

13.7 Expand the second end.

13.8 Visually inspect tube welds and repair as needed.

13.9 Air/freon test tube to tubesheet welds at 10/15 psi.

13.10 Repair as needed.

14. Final assembly - hydrotesting and shipping

14.1 Set the unit for hydrotesting. Assure even distribution of load through the support structure to the floor below.

14.2 Close all shell openings.

14.3 Fill the unit with rust-inhibited seawater. Inspect the tube/tubesheet welds as the level progresses upward and mark the tube welds that leak. (Make repairs after emptying and drying.)

14.4 After filling unit, allow to stand and vent completely. Then apply pressure in increments until the hydrotest pressure is reached. Reduce the test pressure to 2/3 to perform a complete visual inspection of all outside surfaces. Look for leaks and distress. Observe safe practices and any temperature limitations imposed. ASME inspector to witness test.

14.5 Drain the unit after a successful test and wash down with clean potable water (inhibited).

14.6 Dry the unit and check for internal distress due to hydrotesting. Mop up excess water, blow down droplets with air and vacuum dry if necessary to obtain visual dryness. Bleed-in relatively dry shop line air.

14.7 Repair leaking tube welds and retest locally using a special fixture for testing tube welds only, thus avoiding a refill.

14.8 Apply the ASME data plate and FW nameplates.

14.9 Clean and fill all titanium stripped back areas with metal filling epoxy or other suitable material to enhance corrosion resistance.

14.10 Apply shipping closures.

14.11 Clean for painting.

14.12 Apply exterior painting and interior preservatives.

14. Final assembly - hydrotesting and shipping (cont'd.)

14.13 Clear all documentation, QC data and ASME certifications.

14.14 Prepare to move the unit. Apply the necessary attachments for moving. Use 2 shop bridge cranes to lift onto barge.

14.15 Move the unit to the dock area outside the plant.

14.16 Load the units on the waiting barge. Photograph the unit.

14.17 Strap and secure the unit in place on the barge.

14.18 Release the unit for shipment.

14.19 Free the barge for movement.

14.20 Tug pull the barge to destination by a previously cleared shipping plan.

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Appendix E4

SPECIFICATION OF HEAT EXCHANGERS FOR

10 MW(e) POWER MODULE

Preface

This specification provides the parameters for the evaporator and condenser which would be incorporated in a 10 MW(e) power module of an OTEC power plant. The specification defines the applicable codes and standards, the requirements and characteristics of the heat exchangers, the constraints for design and construction and the necessary documentation and quality assurance.

Section 1 Introduction

1.1 Purpose

The OTEC power system operates as a Rankine cycle with ammonia as the working fluid. The evaporator supplies ammonia vapor to the turbine by evaporating liquid ammonia with warm water taken from near the surface of the sea. The condenser accepts the ammonia vapor from the turbine and condenser it to a liquid using cold water from the deeper portion of the sea. The evaporator and condenser combine with the turbine and the liquid return pump to form the closed power cycle. This document provides guidelines for the design and fabrication of the evaporator and condenser portions of the power cycle.

1.2 Scope

This specification is based on the evaporator and condenser requirements which were developed as part of Task 2, Preliminary Design, of the OTEC Power System Development. The heat exchangers are for the immersed option which places the top of the heat exchangers within 20 feet of the surface of the water. The interfaces between the heat exchanger and the balance of the power system can be identified, but not defined until the platform characteristic are better known.

Section 2 Codes/Standards

2.1 Introduction

The heat exchangers will be submerged in water next to a manned platform. Some in-service inspection/repair can be anticipated. The heat exchangers must conform to certain codes and standards to insure adequate service life and to minimize the danger to the operating personnel.

2.2 Specifications

The heat exchangers will conform to the specifications contained herein. These specifications will be expanded as a result of the detail design of the heat exchangers and a more complete definition of the balance of the power system and of the platform and other plant components.

Section 2 Codes/Standards (Cont'd)

2.3 Codes

The evaporator and condenser will be designed and fabricated in accordance with the American Society of Mechanical Engineers Boiler and Pressure Vessel Code, Section VIII, Division 1. The completed heat exchangers will have a Section VIII stamp which will show that they comply to all requirements.

2.4 Regulations

As a portion of a manned sea-going system, the heat exchangers must be approved by the Coast Guard. They will comply to the applicable regulations of the United States Coast Guard, Title 46, Code of Federal Regulations. For this purpose, the heat exchangers will be defined as ammonia tanks and will be designed for 110 percent of the vapor pressure corresponding to the temperature of ammonia at 85°F.

2.5 Standards

The heat exchangers will conform to the design practices as specified by the Standards of the Turbular Exchanger Manufacturers Association (TEMA).

2.6 Drawings

Drawings of the heat exchangers will be prepared in accordance with the recommendations of the American National Standard Engineering Drawing and Related Documentation Practices (ANSI Committee Y 14).

Section 3 Requirements

3.1 Definition

Both the evaporator and condenser are shell and tube type heat exchangers with the tubes in the horizontal direction for the evaporator and within 15° of horizontal for the condenser. The water flows through the straight, single pass tubes and the ammonia is on the shell side. The spherical heat exchanger shells are truncated at each end by the vertical tubesheets. The tubes are supported internally by vertical tube support plates.

The heat exchangers interface with water boxes which contain the water pumps and direct the water in to and out of the heat exchangers. The waterboxes will attach to the heat exchanger transition rings between the shell and tubesheets. The loads imposed by the waterboxes and the method of attachment must be specified.

The heat exchangers have an ammonia pump pod adapter ring welded to the bottom of the shell. The ring will interface with a pump pod which will contain the ammonia circulation/recirculation pumps and related piping. The diameter of the ring and the weld preparation must be specified.

The ammonia vapor and liquid connections at the shell are short cylinders which have heavy walls to provide reinforcement for the large openings in the shell. The weld preparations for the thinner wall connecting piping must be specified.

The attachment points and configuration must be defined between the heat exchangers and platform.

3.1.1 Evaporator

The liquid ammonia is evaporated using the irrigated film technique. The liquid ammonia enters the evaporator through the ammonia feed inlet and is distributed to horizontal manifolds behind one tubesheet. The manifolds supply perforated feed tubes which are positioned on the same tube spacing as the water tubes. There is one horizontal feed tube for each column of water tubes. Columns which are higher than 70 tubes have additional feed tubes. The ammonia flowrate is sufficient to insure all tubes are wetted with liquid ammonia. The excess ammonia from the lowest water tube returns to the sump which is inside the lower portion of the evaporator shell. The excess ammonia is combined with the liquid ammonia from the condenser and recirculated through the evaporator irrigation system.

The ammonia vapor passes through the tube bank and is collected inside the spherical shell. It passes through two hook and vane type demisters which are also contained within the shell. The dry ammonia vapor exits through the vapor outlet ports and goes to the turbine.

3.1.2 Condenser

The ammonia vapor discharge from the turbine enters the condenser through one or two inlet ports and is directed around the tube bundle by impingement baffles. The cold water condenses the ammonia on the tubes and the liquid collects in the sump inside the condenser shell. The condensate is pumped back to the evaporator.

The condenser has a shroud over a portion of the tubes in the center of the tube bundle. The non-condensible gases will tend to concentrate in the center of the tube bundle which is the coldest area. The mixture of non-condensible gases and ammonia will be piped to an auxiliary system to separate and recycle the ammonia which will be returned to the condenser sump.

3.2 Characteristics

The heat exchangers are designed for full load conditions. Transient, part load and other off-design conditions must be specified.

3.2.1 Performance Requirements

The heat exchangers are designed for the following design and operating conditions.

	Evaporator	Condenser
Ammonia pressure, psia, (design)	183	183
Ammonia pressure, psia, (operating)	129.15	89.47
Ammonia inlet temperature, $^{\circ}F$	49.76	50.16

	Evaporator	Condenser
Ammonia outlet temperature, ^{o}F	70.16	49.73
Ammonia Flowrate, lb/sec	760.03	760.03
Max. water depth at top, ft	20	20
Minimum internal pressure, psia	0	0
Water inlet temperature, ^{o}F	80.00	40.00
Water outlet temperature, ^{o}F	75.77	44.52
Water flowrate, lb/sec	101,389	91,690
Total heat flow, MW(t)	422	409

3.2.2 Physical Requirements

The shell diameter, volume and total heat exchanger weight should be minimized to reduce cost, handling problems, and platform requirements. The preliminary design characteristics are:

	Evaporator	Condenser
Shell outer diameter, ft	39.4	39.4
Overall length, ft	26.6	26.6
Total dry weight, tons	470	502

Access to the ammonia side of the heat exchangers must be made through the vapor inlet or outlet nozzles. A panel in the evaporator demister is easily removed for access.

3.2.3 Reliability

The plant availability requirement is 90%. In order to meet this, the following failure rates have been specified for the heat exchangers:

Component	Failure Rate (failures/year)	Mean Time to Repair (days)
Tube	.033	3
Shell	.033	6
Tubesheet, baffles, support	.066	4

3.2.4 Maintainability

The most probable failure for the heat exchangers is the tube wall or the tube-to-tubesheet weld. A failure at either place will be detected by ammonia sensors in the waterbox downstream of the heat exchangers. The water pumps will be stopped and a diver used to identify the leaking tube and place a plug at both ends of the tube. When the number of plugged tubes becomes sufficiently large that the heat exchanger can no longer meet the full load design requirements, the heat exchanger will be removed from service and returned to a repair facility. The tubes will be inspected to determine if the failed tubes should be replaced or if the heat exchanger should be scrapped.

The heat exchanger tube cleaning brushes must be replaced on a regular schedule which is to be determined. This can be done in place by a diver.

3.2.5 Environment

The heat exchangers will be immersed in the water next to the platform. They must be capable of normal operation in sea-state six conditions and must survive under sea-state nine conditions. The loads which will be imposed on the heat exchangers as a result of the interaction with the platform at these sea-states must be specified.

3.3 Design/Construction

3.3.1 Service/Life

The design life of the heat exchangers will be:

30 year life
1000 start/stop cycles
8760 hours/year operation

3.3.2 Materials

The water tubes will be welded, commercially pure titanium per SB-338 Grade 2. The outside surface of the tubes will be enhanced with circumferential fins. The fin spacing and height will be specified. The tubes will have smooth lands at the tubesheets and tube support plates. The minimum tube wall under the fin will be 0.028 in.

The tubesheets will be carbon steel per SA-516, Grade 70 with an explosively bonded titanium cladding per SB-265, Grade 1 on the water-side face.

The shell and all other pressure boundaries will be carbon steel plate per SA-516, Grade 70. The balance of the heat exchanger internals which contact only ammonia will be SA-36 plate or SA-105 or 106 pipe.

The titanium will not react with the water. The carbon steel will be protected by paint or some other type of coating which will be specified.

3.3.3 Fabrication

The shell, tubesheet and other pressure boundaries will be welded using qualified welders and procedures. The tubes will be welded to the tubesheet with either an end weld or a fillet weld around the tube. The tube support plates will be mechanically supported by rings and pipe spacers which are welded to the shell.

3.4 Support Equipment

The support equipment required for the heat exchangers is limited to a support saddle and lifting equipment.

A support will be used for holding the heat exchangers in the shop during tube assembly and hydrotesting. The same support will be used during shipping.

The evaporator has two vapor outlet nozzles which are located in the center of the shell at the horizontal midpoint. The nozzles will be designed to serve as trunnions for lifting and rotating the evaporator in the shop and during field assembly to the balance of the OTEC system. The condenser will have dummy nozzles at the same locations to permit handling with the same lifting equipment. Slings, spreader bars, etc. will be provided to lift the heat exchangers with either one or two cranes.

The heat exchangers will be transported by barge from the manufacturing facility to the OTEC assembly point. The nozzles will have shipping covers to prevent entry of water or other contaminents into the ammonia space during shipment. The type of covers must be specified to be either the hydro test caps which must be cut off in the field or loose covers which are bolted in place. The heat exchangers will contain shop air at one atmosphere.

3.5 Logistics

The only heat exchanger components to be replaced on a preset maintenance schedule are the brushes and cages for the tube cleaning system. The replacement time must be specified.

Tube plugs are required to plug leaky tubes while the heat exchangers are installed in the power module. Individual tubes can be replaced in a repair facility which has not been specified.

3.6 Documentation

The heat exchanger is one component of the OTEC power module. The documentation required to test and support the power module must be developed by the system integrator.

Procedures will be developed and supplied for tube plugging and tube replacement. No other assembly, disassembly or repairs are anticipated.

Section 4 Quality Assurance

The heat exchangers will be fabricated in accordance with ASME Section VIII and with all the quality control which is required by that document. No additional quality assurance requirements will be imposed.

4.1 Materials

All pressure boundary material will be purchased to an ASME or ASTM specification.

Material certifications will be required showing material chemical and physical properties and material heat number. All material will be inspected for proper marking by supplier and material identity will be maintained in the shop. Non-pressure boundary material will also be ordered to an ASME or ASTM specification and certification to that specification will be required.

4.2 Inspections

All in process inspections such as magnetic particle testing, radiographic testing, etc, will be done using procedures and inspectors which have been qualified per ASME requirements. Inspections will be spot checked by the cognizant ASME Code inspector.

4.3 Checkouts/Tests

The tube-to-tubesheet joints will be inspected visually and tested with an air/freon leak detector. The final test will be a hydrotest of the ammonia side. All tests will be done per a written procedure. The hydro test will be witnessed by the ASME Code inspector.

4.4 Records

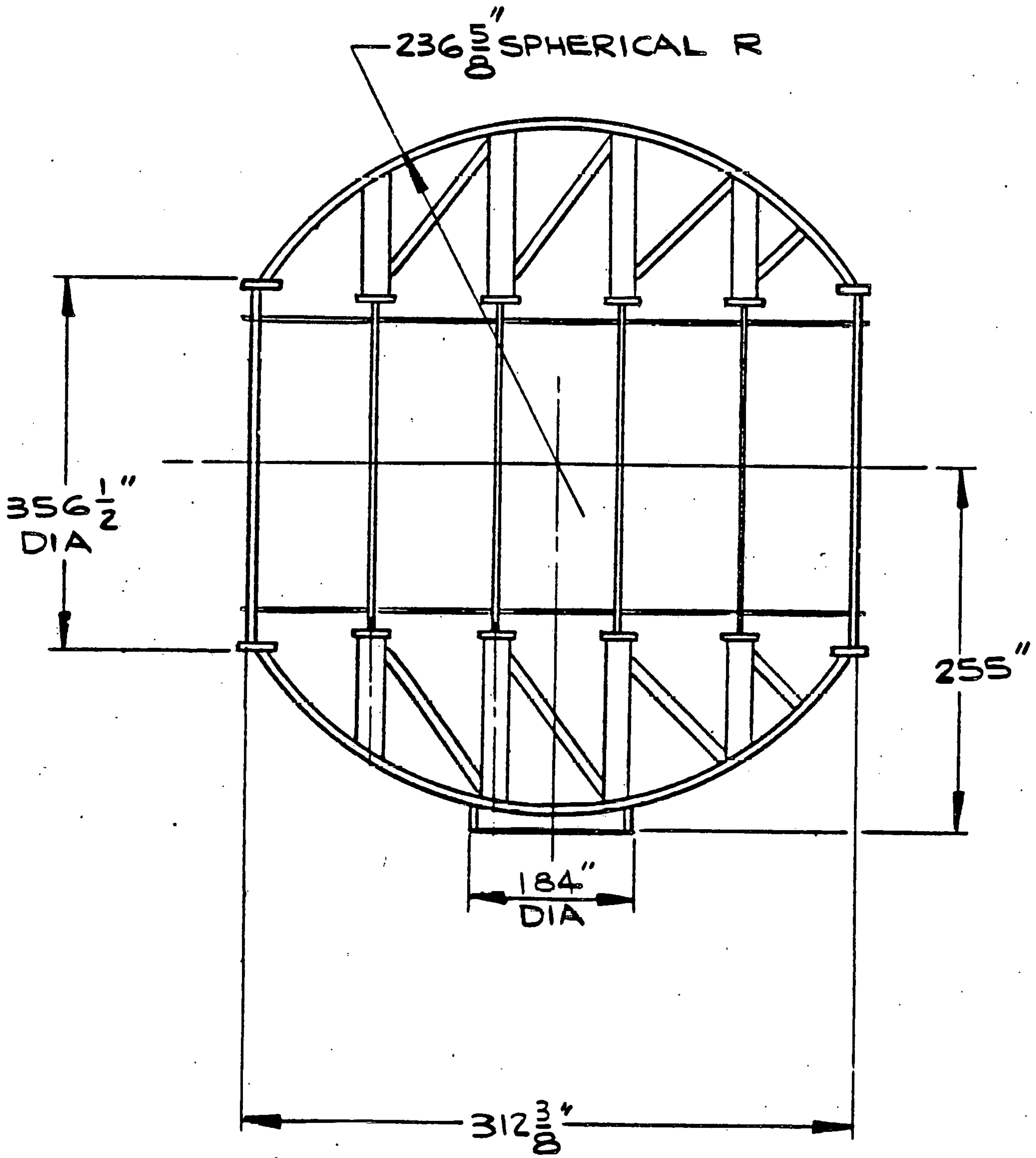
The shop will maintain the following records for each heat exchanger: material certifications, weld records, radiographic inspections, stress relief charts, hydro test, discrepant material reports and disposition, and code data report. All records will be retained by the shop for a minimum of 3 years. The code data report will be sent to the customer.

Section 5 Acceptance

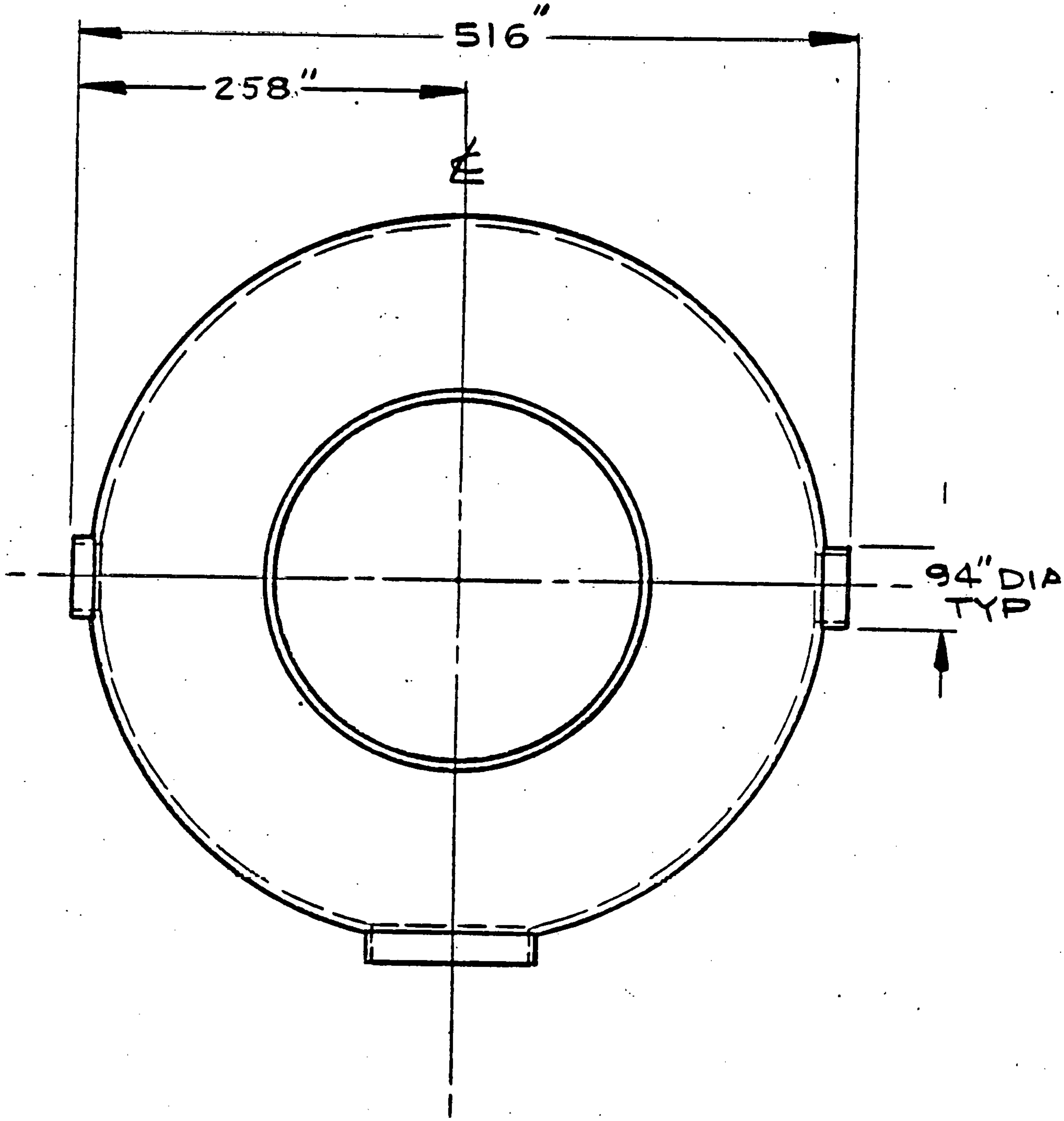
Acceptance of the heat exchanger will be based on successful completion of the hydrostatic test in the shop. The heat exchangers will be loaded on a barge and will become government property at that point. Heat exchanger vendor will provide installation and operational support on an as-needed basis.

Section 6 Personnel and Training

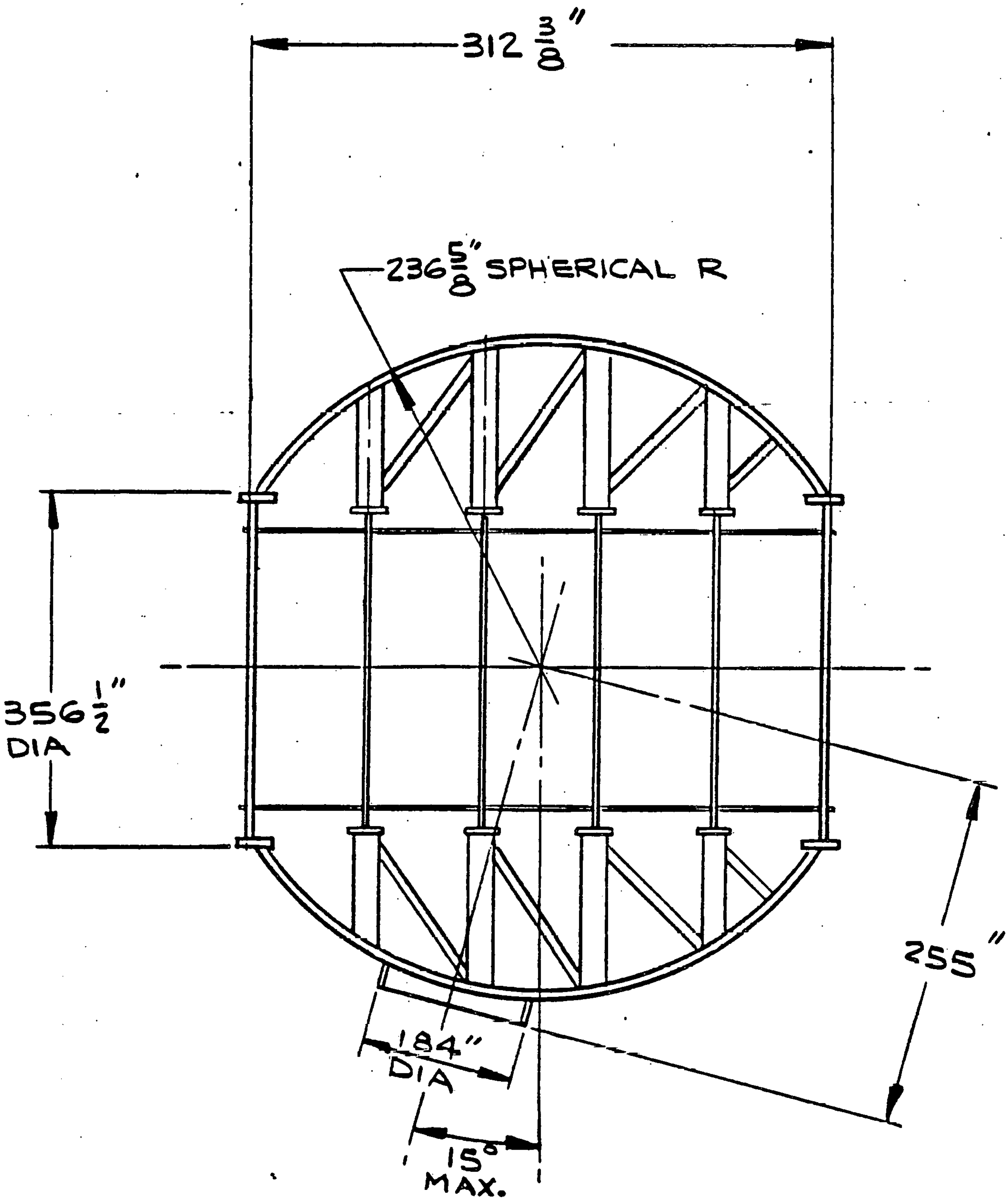
All installation and operating personnel will be supplied and trained by others.



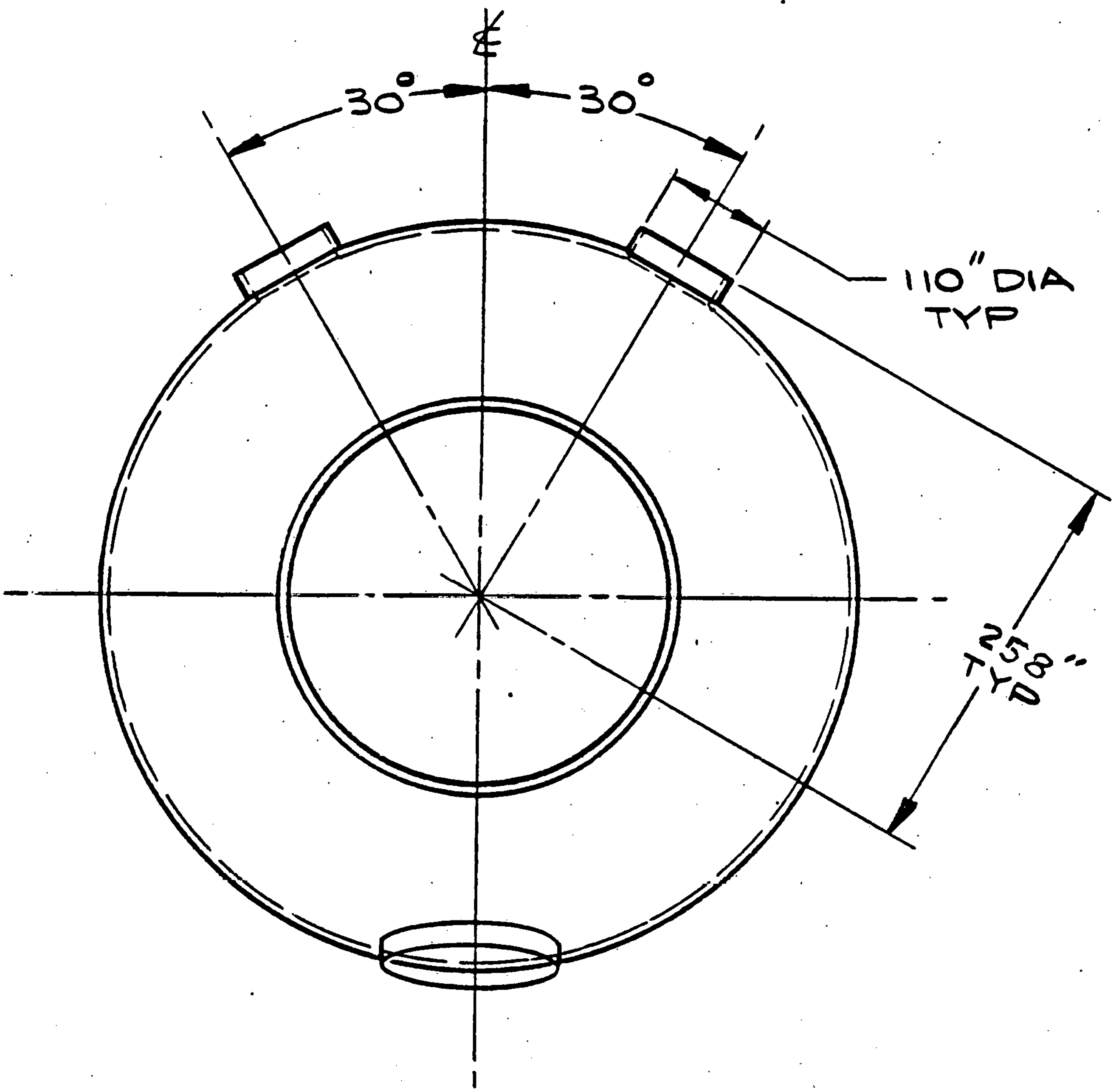
OTEC 10 MWe
EVAPORATOR



OTEC 10 MWe
EVAPORATOR



OTEC 10 MWe CONDENSER



OTEC 10 MWe
CONDENSER

Blank

Appendix E5

EVAPORATOR AMMONIA LIQUID DISTRIBUTION SYSTEM

The function of the evaporator liquid ammonia distribution system is to insure complete wetting of all of the evaporator tubes at the lowest possible flow rate. In addition, it is necessary to insure that liquid ammonia atomization is held to an acceptable minimum. Atomization will result in greater liquid ammonia flow rates and vapor-liquid separation requirements as a result of liquid ammonia particle entrainment by the vapor. Test performed by LMSC in 1977 for ammonia jet impingement on plain tubes showed sustantial breakup and atomization with the formation of a very fine mist. Tests were conducted for 0.030, 0.050, 0.070, and 01000 inch orifice diameters at 5, 10,and 15 psi pressure differentials. In contrast with plain tube impingement, experiments with impingement on thin (0.02 in) metallic felt pads on the tube surface produced almost no jet breakup. Because of the importance attached to jet atomization, the distribution tubes have been designed to avoid this problem by spraying upward into a spray deflector. A secondary benefit of this design is that by locating the orifice holes on the top of the tube, they are far less susceptible to plugging by particulate matter.

The LMSC liquid ammonia distribution system rather than being a spray system, as is sometimes used in desalination evaporators, is an irrigation system in that every column of tubes in the four tube-sections is fed by its own distribution tube. This is felt to be necessary to insure complete tube wetting inorder to avoid the consequences of dry evaporator tubes. A further restriction is the number of tubes in a column supplied by a single distribution tube, which has resulted in the evaporator tube bundle being broken into four sections as shown in FW Dwg. 58-3157-6-0210, (Appendix B).

Determination of the individual tube column irrigation rates is dependent on the tube evaporation rate and the minimum flow rate required to provide complete wetting. The minimum flow rate to any tube column is that which will be completely evaporated on the bottom tube of the column. This rate will be dependent on the average heat flux and degree of subcooling present in the liquid ammonia as well as the diameter and number of tubes in a column. The amount of excess liquid ammonia which must be supplied to insure complete tube wetting is not known at the present time although it is felt to be on the order of 50 percent. The present flow rates are based on approximately a 50 percent excess and are shown in the following table.

TABLE 1 10 MW DISTRIBUTION TUBE FLOW RATES (LBM/HR-FT)

	MIDDLE				SIDES			
SECT	EVAP	DIST	EXCESS	TOTAL	EVAP	DIST	EXCESS	TOTAL
1	105	165	---	165	---	---	---	---
2	135	135	60	195	135	195	---	195
3	135	135	60	195	135	135	60	195
4	105	105	60	165	---	---	---	---

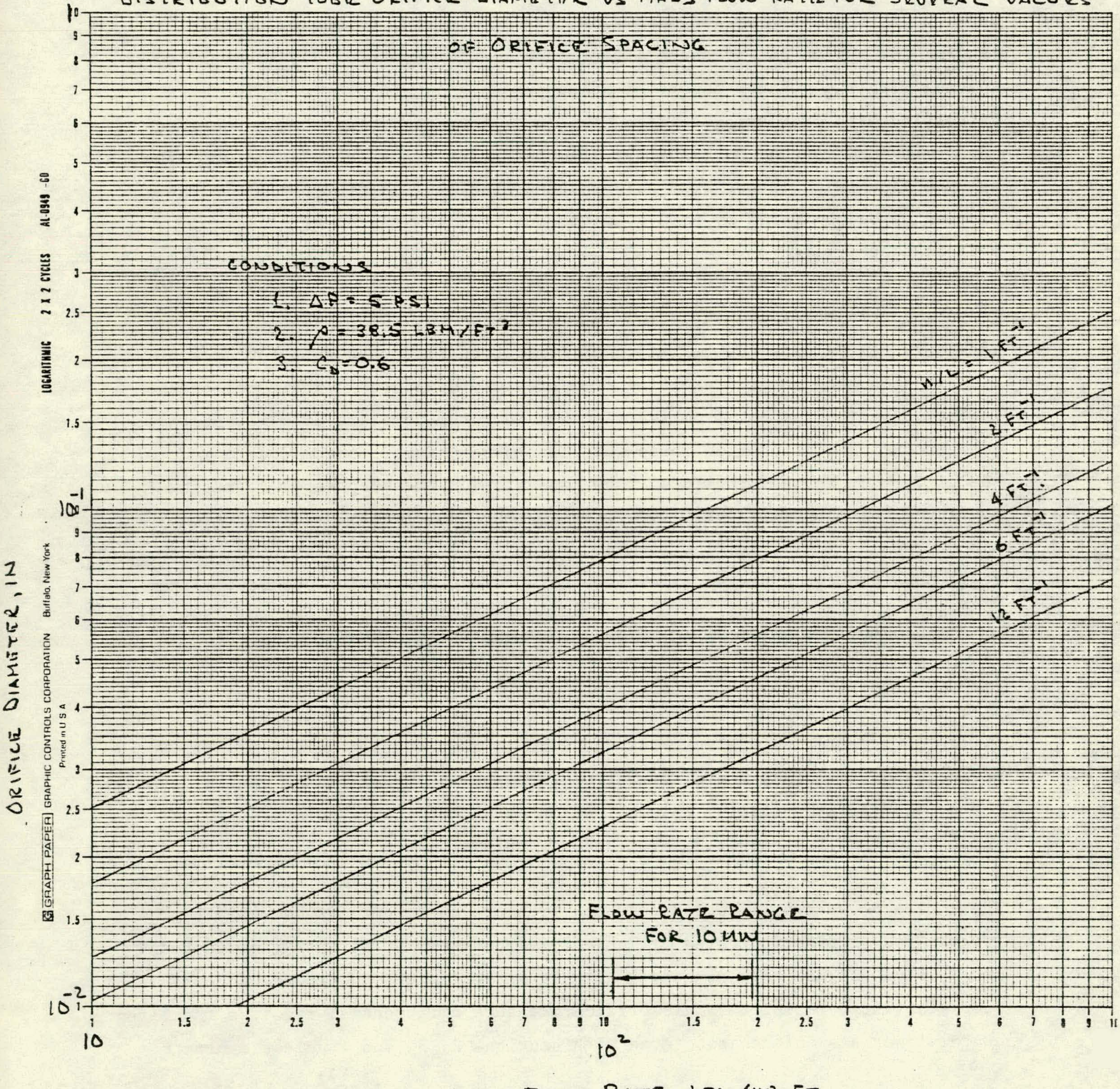
Section number refers to the four horizontal sections of 49 ½, 62 ½ 62½, and 49½ tube rows beginning at the top and shown in FW Dwg. 58-3157-6-0210. The middle portion covers the 100 columns on both sides of the centerline corresponding to the width of sections 1 and 4. The sides correspond to the thirty additional columns in sections 2 and 3 not covered by run off from section 1. The distribution tube flow rates shown under EVAP in Table 1 correspond to the amount to supply the evaporation requirement for 49½, 62½, 62½, and 49½ tubes respectively in the four sections. Shown under DIST is the distribution tube flow rate for the four sections where the exccess flow for the middle portion is applied through the distribution tubes to section 1 and that for the sides is applied through the outside 30 tubes of section 2. In this manner, an excess flow rate of approximately 60 lbm/hr ft will fall between sections and into the sump. Depending on subsequent experience, it may also be necessary to provide some additional flow at sections 2, 3, and 4.

Design of the distribution tube requires choice of an orifice diameter, pressure diffrential and spacing, which will provide the required flow rate without excessive flow variation along the tube. Figure 1 indicates the relationship between orifice diameter and mass flow rate for several values of orifice spacing for a five psi pressure differential. Uniformity of flow is related to spacing with the greater number of holes per unit length providing increased uniformity with decreasing orifice diameter. A tradeoff must be made between uniformity and cost as more smaller diameter holes means increased cost. The present assumption of two inch hole spacing with 0.04 in diameter is in the process of being tested by LMSC to determine the flow characteristics for uniform tube irrigation.

In addition to the local variation resulting from the hole spacing, it is necessary to evaluate the variation of irrigation flow rate as a function of distribution tube length resulting from pressure drop within the tube. The latter variation results from friction and momentum pressure drop within the tube producing orifice flow rates which are proportional to the square root of the difference between the local tube pressure and the evaporator operating pressure. In this particular flow problem, the frictional loss is offset by the momentum change resulting from decreasing tube flow velocity with length. It is possible for a pressure increase to occur at some point downstream from the inlet as shown in Fig. 2. Calculation of local tube pressure and irrigation rate are obtained by simultaneous solution of two nonlinear differential equations from the colebrook equation. The variation in irrigation rate is shown in Fig. 3 as a function of length for the four different tube sections. While the variation is relatively slight for the 0.87 in distribution tube I.D., it is advantageous to locate the distribution tube feed on the same end of the evaporator as the seawater inlet. This allows matching the greater evaporation rate which occurs at the seawater inlet and of the evaporator tubes.

The choice between orifice diameter and pressure difference is shown in Fig. 4 for the four irrigation rates of Table 1. A pressure differential of approximately five psi has been presently assumed as sufficient to provide acceptable flow stability. Using a simgle orifice pressure differential requires thottling at the lower elevations associated with sections 2, 3, and 4 unless seperate

DISTRIBUTION TUBE ORIFICE DIAMETER Vs MASS FLOW RATE FOR SEVERAL VALUES OF ORIFICE SPACING



MASS FLOW RATE, LBM/HR-FT

FIG 1

WLO, 21 SEP 78

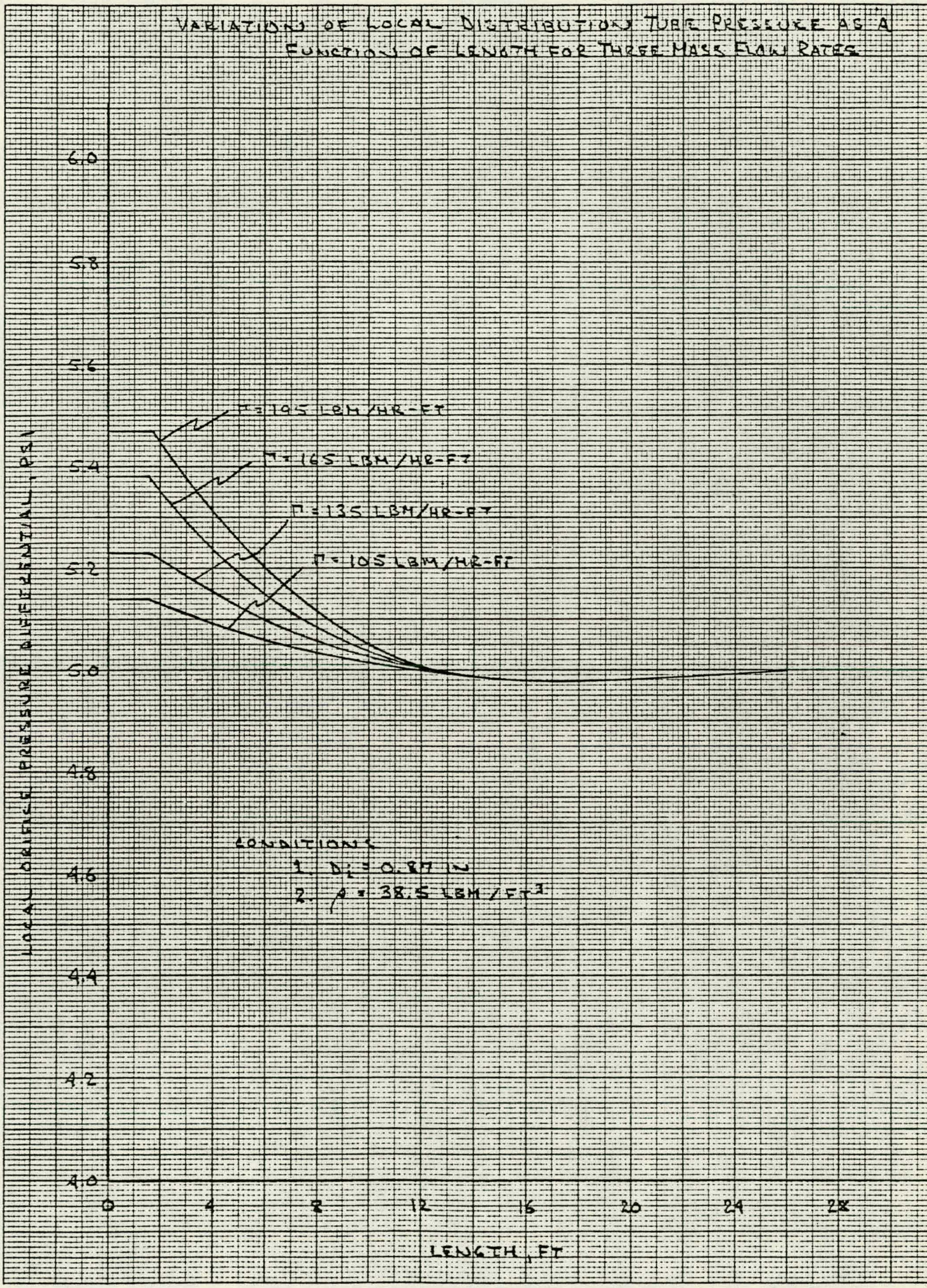


FIG. 2

WLO, 21 SEP 78

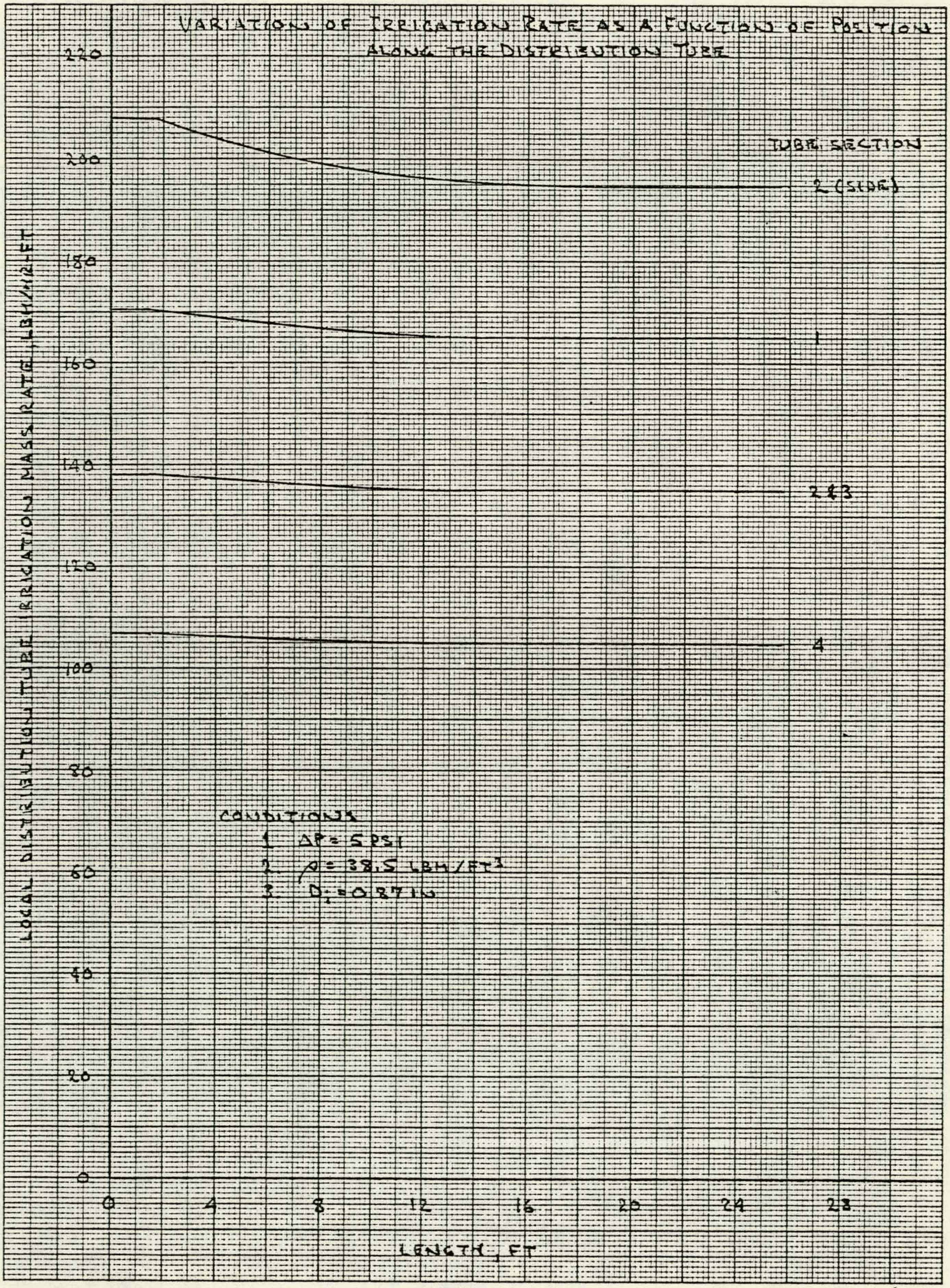
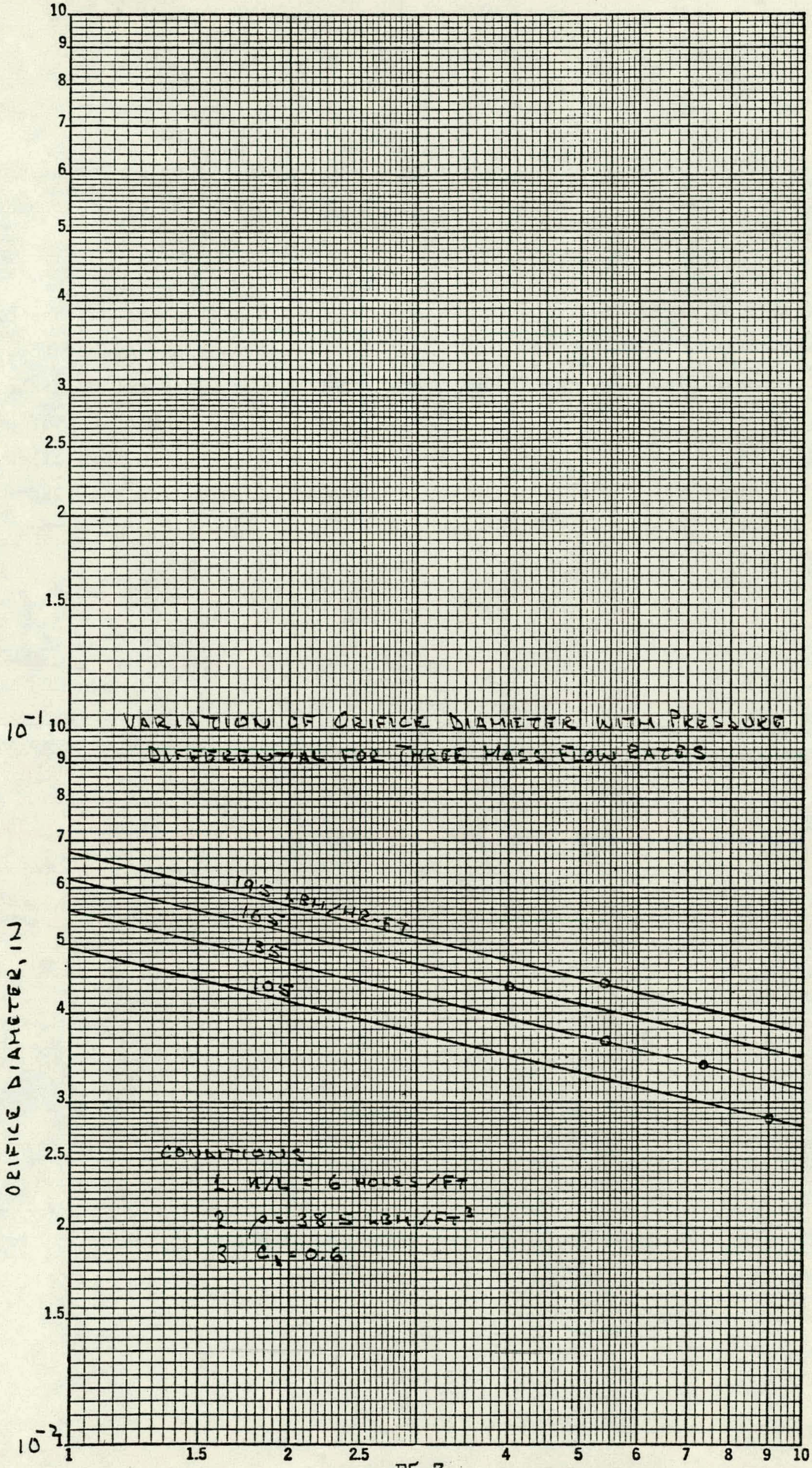


FIG. 3

WLO, 21 SEP 78



ORIFICE PRESSURE DIFFERENTIAL, PSI

WLD, 21SEP78

FIG. 4

pumps are used. A single pump can be used without thottling if one wishes to size the orifices of the various sections to accommodate the pressure head variation as indicated by the individual points of Fig. 4.

Sensitivity of irrigation rate to evaporator operating pressure has basically been eliminated with the present separation of condensate and evaporator pumping requirements. Since the evaporator operating prossure comprises both the liquid ammonia distribution system inlet and discharge pressure, variations of evaporator pressure have no effect on the distribution system flow rate. This allows lower orfice pressure differentials than would normally satisfy flow stability requirements. Details of the distribution tube are shown in Fig. FW Dwg. 58-3157-6-0211.

APPENDIX F

APPENDIX F1.1

Performance Characteristics - Radial Inflow Turbine

The turbine has a flange-to-flange efficiency of about 88%, guaranteed, with as much as 90% expected. The conditions are as follows:

Inlet Pressure	127.657 psia
Inlet Temperature	69.5°F
Discharge Pressure	89.42 psia
Discharge Temperature	50.13°F

This is a 40° day condition.

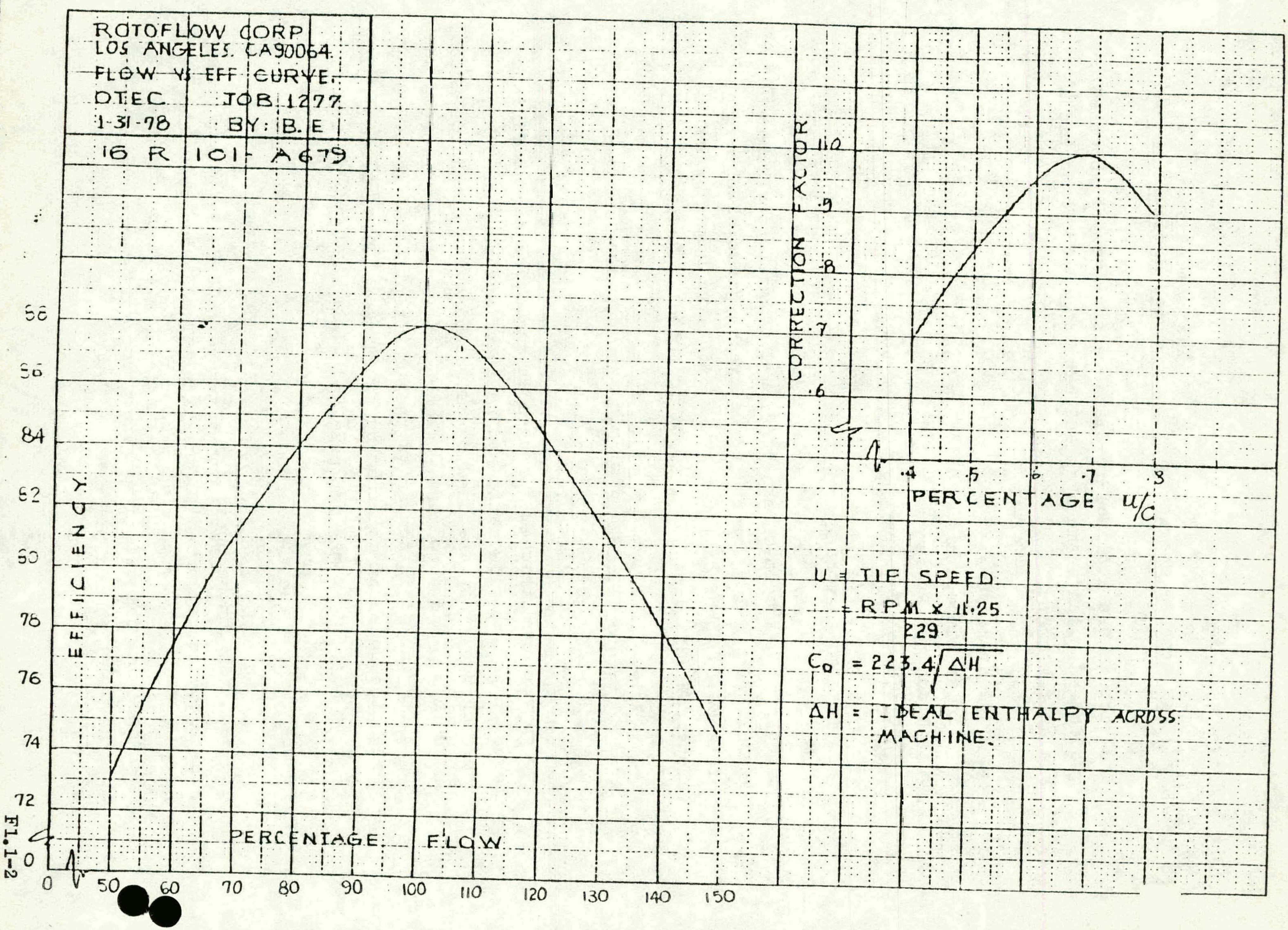
The casing has been chosen to flow 2.997 million lbs/hr of ammonia with a 30% margin, to deliver a gross power output of 14 MWe. Speed of rotation is 3,600 rpm with a wheel diameter of 43". The wheel specific speed is defined as:

$$\frac{\text{rpm}\sqrt{\text{Gall/min}}}{(\text{head in ft})\uparrow 3/4} = 1{,}980$$

Two curves are attached for calculating off-design efficiency based on volume flow ratio and relative enthalpy drops across the machine. They are used by calculating the discharge actual cubic ft/min and comparing with the design flow of 150,440 cubic ft/min. This number is then mulitiplied by an enthalpy correction constant obtained by dividing: $223.4\sqrt{\text{Delta H(Btu/lb)}}$ by 675, the design tip speed of the turbine.

Figure 1 shows the two curves.

The machines can be optimized about any design point and need not be considered fixed at the time. The design point may shift as final heat exchanger characteristics are obtained, and the wheel shape modified to suit. The only parts that change are the wheel and follower.


ROTOFLOW CORP
LOS ANGELES. CA90064.
FLOW VS EFF CURVE.
OTEC JOB 1277
1-31-78 BY: B. E
16 R 101- A679
EFFICIENCY
PERCENTAGE FLOW
CORRECTION FACTOR
PERCENTAGE U/C
U = TIP SPEED
= RPM x 11.25 / 229
Co = 223.4 √ΔH
ΔH = IDEAL ENTHALPY ACROSS MACHINE.

Fl.1-2

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/3
June 9, 1978

PERFORMANCE DATA

KVA: 43750 **KW:** 35000 **PF:** 0.8

VOLT: 13800 **FREQ:** 60 **PHASE:** 3

EFF: 4/4 97.4 3/4 97.1 1/2 96.3 1/4 -

TOTAL TEMPERATURE:

Generator Stator 125°C by detector
Generator Rotor 120°C by resistance

COOLING WATER REQUIRED: Temp 95°F gpm 750 Pressure 125 psig

OIL REQUIREMENTS:

Bearing-gpm/brg 25 psi 8-10 heat reject ______

Seals-gpm/seal - psi -

Oil type: 130-180 S.S.U. @ 100% and pour point below lowest expected ambient - mineral oil turbine type.

GENERATOR WINDING CONNECTION: Y

GENERATOR REACTANCE VALUES:

Direct-axis synchronous (at rated current)	Later	%
Transient saturated (at rated current)	Later	%
Subtransient (at rated voltage)	Later	%
Negative sequence (at rated current)	Later	%
Zero sequence (at rated current)	Later	%

WEIGHTS:

Total net weight of unit	196,000	lb.
Stator weight	150,000	lb.
Rotor weight	34,000	lb.

FRAME: 71120

MAXIMUM BASE WIDTH: 145.5"

GENERAL DATA – GENERATOR

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/3
June 9, 1978

BILL OF MATERIAL

KVA: 43750 KW: 35000 PF: 0.8 RPM: 3600

VOLT: 13800 FREQ: 60 PHASE: 3

SPECIFICATION: NEMA SM-12, ANSI C50.13

INSULATION: Class B

BEARINGS: Sleeve, ring oiled, forced lube by others.

COOLER: Side mounted, plate fin for sea water

WATER TEMP: 95°F PRESSURE: 125 psig

RTD: (12) in stator, (4) in air stream

CURRENT TRANSFORMERS: None

MAIN LEAD LOCATION: Bottom

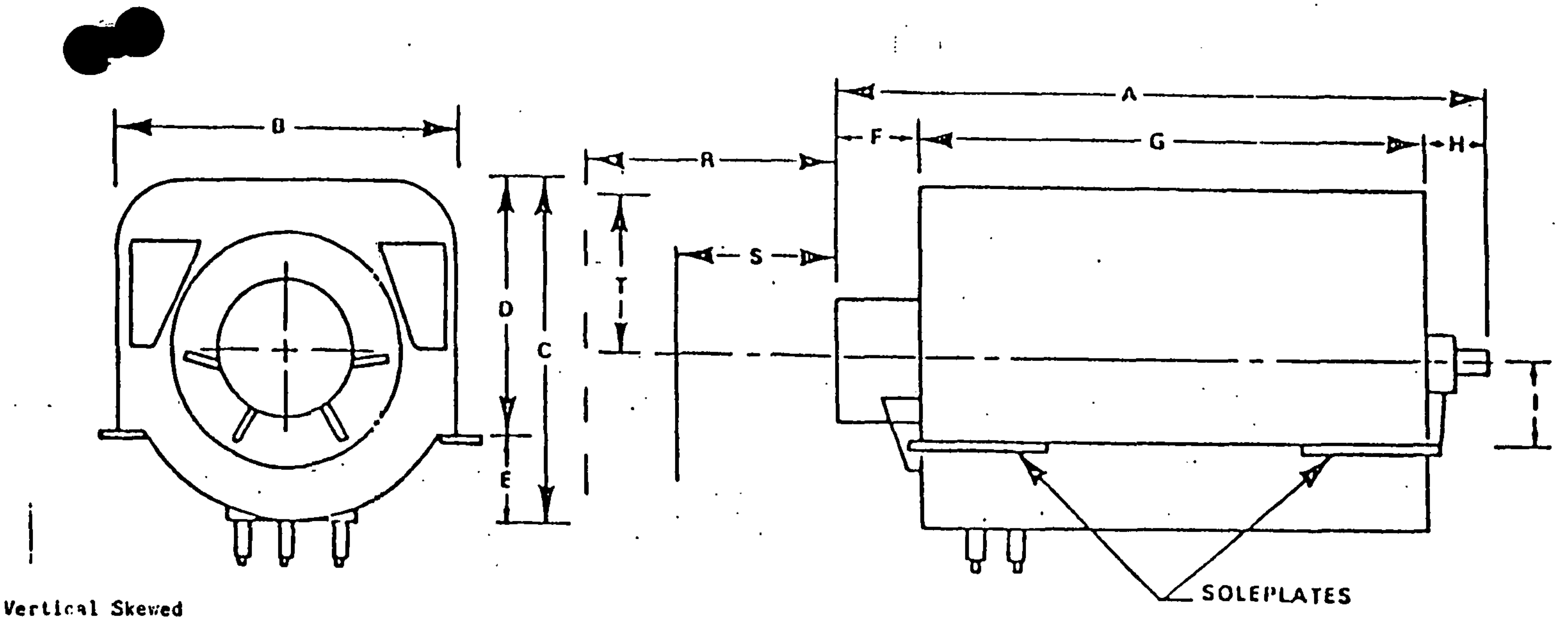
EXCITER: Brushless direct connected with PMG

VOLTAGE REGULATOR: (2) SCR Type - dual control for separate mounting.

VOLT REG ACCESSORIES: (2) volt adj. rheo, disc sw, transfer sw

START-UP: Not included - can be provided.

TESTS: Standard commercial



T - Vertical Skewed
S - Horizontal Skewed
R - Horizontal Straight

DESCRIPTION			ESTIMATED DIMENSIONS												WEIGHTS		
CODE	KVA	KW	A	B	C	D	E	F	G	H	I	R	S	T	ROTOR	STATOR	TOTAL
5057	6,250	5,000	165.00	120	99.00	75.5	23.50	42	126.0	20	22	158	108	110	8,600	43,000	[illegible]
5742	7,500	6,000	175.00	126	106.00	79.0	27.00	42	104.0	26	22	145	105	110	8,700	53,000	63,400
5752	9,375	7,500	182.00	126	106.00	79.0	27.00	42	114.0	26	22	155	105	110	10,700	63,000	80,875
•5752	12,500	10,000	182.00	126	106.00	79.0	27.00	42	114.0	26	22	155	105	110	10,700	63,000	80,875
5765	15,625	12,500	195.00	126	106.00	79.0	27.00	42	127.0	26	22	168	110	115	13,600	76,500	93,700
6571	18,750	15,000	240.00	133	114.00	85.0	29.00	45	155.0	30	24	174	145	110	18,200	90,000	118,000
6571	21,875	17,500	240.00	133	114.00	85.0	29.00	45	155.0	30	24	174	145	110	18,200	90,000	118,000
6581	25,000	20,000	240.00	133	114.00	85.0	29.00-	45	165.0	30	24	184	150	115	21,200	98,000	128,500
7180	30,000	24,000	250.00	133	132.30	105.0	27.30	40	156.0	27	26	185	150	120	26,500	105,500	155,500
7195	35,000	28,000	250.00	133	132.30	105.0	27.30	40	172.0	27	26	200	150	125	30,000	120,000	160,000
71108	40,000	32,000	250.00	133	132.30	105.0	27.30	40	185.0	27	26	213	158	131	31,700	135,300	179,000
71120	45,000	40,500	250.00	133	132.30	105.0	27.30	40	197.0	27	26	225	166	137	34,000	150,000	196,000
71120	50,000	45,000	250.00	133	132.30	105.0	27.30	40	197.0	27	26	225	166	137	34,000	150,000	196,000

SOLE PLATE WIDTH - 145.5. -(6571)
" " - 145.5. -(71120)

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/3
June 9, 1978

DESCRIPTION

GENERATOR:

2 pole, non-salient pole totally enclosed water air-cooled bracket type with sleeve bearings, lube oil provided by others. All external generator wiring and piping by others.

EXCITER:

Brushless totally enclosed force ventilated from the main generator mounted outboard from bearing, parallel fused diodes providing 100% redundancy. PMG included for voltage regulator power.

VOLTAGE REGULATOR:

(2) SCR type dual control regulator for bumpless transfer, 100% redundance, disconnect switch on panel, (2) motor operated voltage adjusting rheostats on common shaft, and transfer switch all for separate mounting.

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/2
June 9, 1978

PERFORMANCE DATA

KVA: 17500 KW: 14000 PF: 0.8

VOLT: 13800 FREQ: 60 PHASE: 3

EFF: 4/4 96.9 3/4 96.7 1/2 95.7 1/4 --

TOTAL TEMPERATURE:

Generator Stator	125°C by detector
Generator rotor	120°C by resistance

COOLING WATER REQUIRED: Temp 95°F gpm 400 Pressure 125 psig

OIL REQUIREMENTS:

Bearing-gpm/brg 9 psi 8-10 heat reject ______

Seals-gpm/seal - psi -

Oil type: 130-180 S.S.U. @ 100% and pour point below lowest expected ambient - mineral oil turbine type.

GENERATOR WINDING CONNECTION: Y

GENERATOR REACTANCE VALUES:

Direct-axis synchronous (at rated current)	Later %
Transient saturated (at rated current)	Later %
Subtransient (at rated voltage)	Later %
Negative sequence (at rated current)	Later %
Zero sequence (at rated current)	Later %

WEIGHTS:

Total net weight of unit	118,000 lb.
Stator weight	90,000 lb.
Rotor weight	18,200 lb.

FRAME: 6571

MAXIMUM BASE WIDTH: 145.5"

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/2
June 9, 1978

BILL OF MATERIAL

KVA: 17500 KW: 14000 PF: 0.8 RPM: 3600

VOLT: 13800 FREQ: 60 PHASE: 3

SPECIFICATION: NEMA SM-12, ANSI C50.13

INSULATION: Class B

BEARINGS: Sleeve, ring oiled, forced lube by others.

COOLER: Side mounted, plate fin for sea water

WATER TEMP: 95°F PRESSURE: 125 psig

RTD: (12) in stator, (4) in air stream

CURRENT TRANSFORMERS: None

MAIN LEAD LOCATION: Bottom

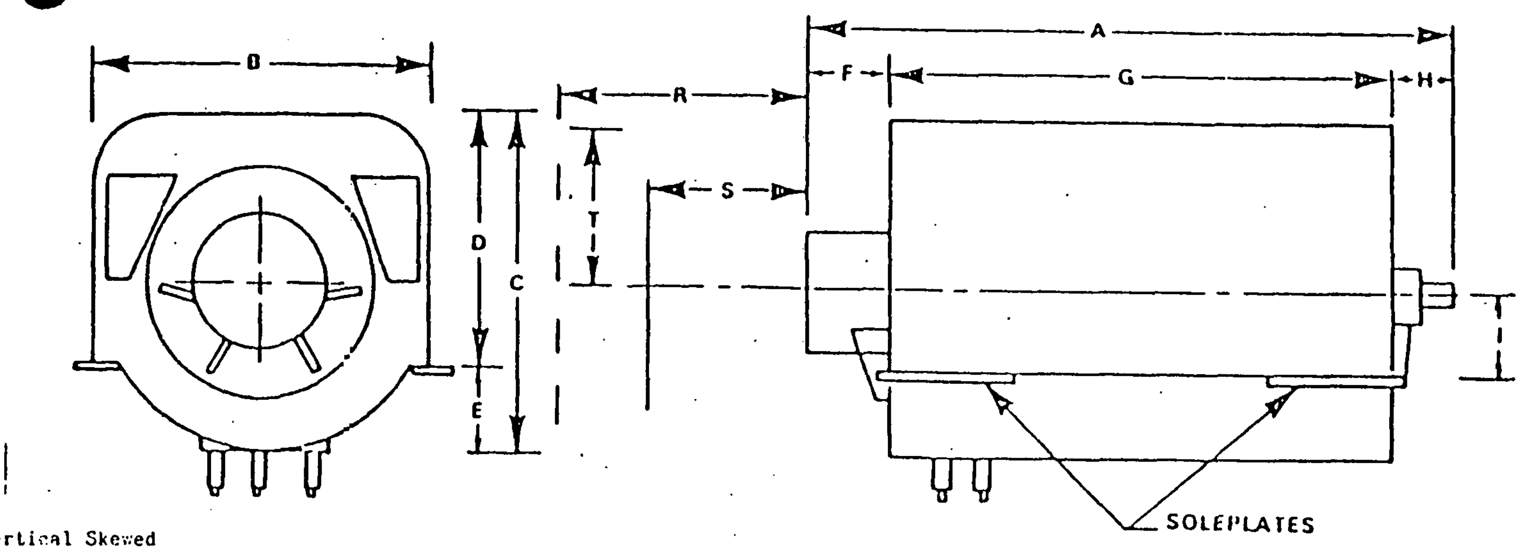
EXCITER: Brushless direct connected with PMG

VOLTAGE REGULATOR: (2) SCR Type- dual control for separate mounting.

VOLT REG ACCESSORIES: (2) volt adj. rheo, disc sw, transfer sw

START-UP: Not included - can be provided.

TESTS: Standard commercial



T - Vertical Skewed
S - Horizontal Skewed
R - Horizontal Straight

DESCRIPTION			ESTIMATED DIMENSIONS													WEIGHTS		
CODE	KVA	KW	A	B	C	D	E	F	G	H	I	R	S	T	ROTOR	STATOR	T[illegible]
5057	6.250	5,000	[illegible]	120	99.00	75.5	23.50	42	126.0	20	22	158	108	110	8,600	43,000	53,300
5742	7,500	6,000	172.00	126	106.00	79.0	27.00	42	104.0	26	22	145	105	110	8,700	53,000	63,400
5752	9,375	7,500	182.00	126	106.00	79.0	27.00	42	114.0	26	22	155	105	110	10,700	63,000	80,875
5752	12,500	10,000	182.00	126	106.00	79.0	27.00	42	114.0	26	22	155	105	110	10,700	63,000	80,875
5765	15,625	12,500	195.00	126	106.00	79.0	27.00	42	127.0	26	22	168	110	115	13,600	76,500	93,700
6571	18,750	15,000	210.00	133	114.00	85.0	29.00	45	155.0	30	24	174	145	110	18,200	90,000	118,000
6571	21,875	17,500	210.00	133	114.00	85.0	29.00	45	155.0	30	24	174	145	110	18,200	90,000	118,000
6581	25,000	20,000	240.00	133	114.00	85.0	29.00	45	165.0	30	24	184	150	115	21,200	98,000	128,500
7180	30,000	24,000	250.00	133	132.30	105.0	27.30	40	156.0	27	26	185	150	120	26,500	105,500	155,500
7195	35,000	28,000	250.00	133	132.30	105.0	27.30	40	172.0	27	26	200	150	125	30,000	120,000	160,000
71108	40,000	32,000	250.00	133	132.30	105.0	27.30	40	165.0	27	26	213	158	131	31,700	135,300	170,000
71120	45,000	40,500	250.00	133	132.30	105.0	27.30	40	197.0	27	26	225	166	137	34,000	150,000	196,000
71120	50,000	45,000	250.00	133	132.30	105.0	27.30	40	197.0	27	26	225	166	137	34,000	150,000	196,000

SOLE PLATE WIDTH – 145.5 = (6571)

SOLE PLATE WIDTH – 145.5 = (71120)

OVERALL WIDTH – 145.5 IN.

OVERALL LENGTH – 230 IN.

OVERALL WEIGHT – 114 IN.

Swearingen Brothers, Inc.
for Lockheed/OTEC

Quotation No. CBE-060978/2
June 9, 1978

DESCRIPTION

GENERATOR:

2 pole, non-salient pole totally enclosed water air-cooled bracket type with sleeve bearings, lube oil provided by others. All external generator wiring and piping by others.

EXCITER:

Brushless totally enclosed force ventilated from the main generator mounted outboard from bearing, parallel fused diodes providing 100% redundancy. PMG included for voltage regulator power.

VOLTAGE REGULATOR:

(2) SCR type dual control regulator for bumpless transfer, 100% redundance, disconnect switch on panel, (2) motor operated voltage adjusting rheostats on common shaft, and transfer switch all for separate mounting.

ELECTRIC POWER APPARATUS

Performance Engineered . . . Quality Built

Product Information

Two-Pole GENERATORS TURBINE DRIVEN

STATOR

CORE CONSTRUCTION

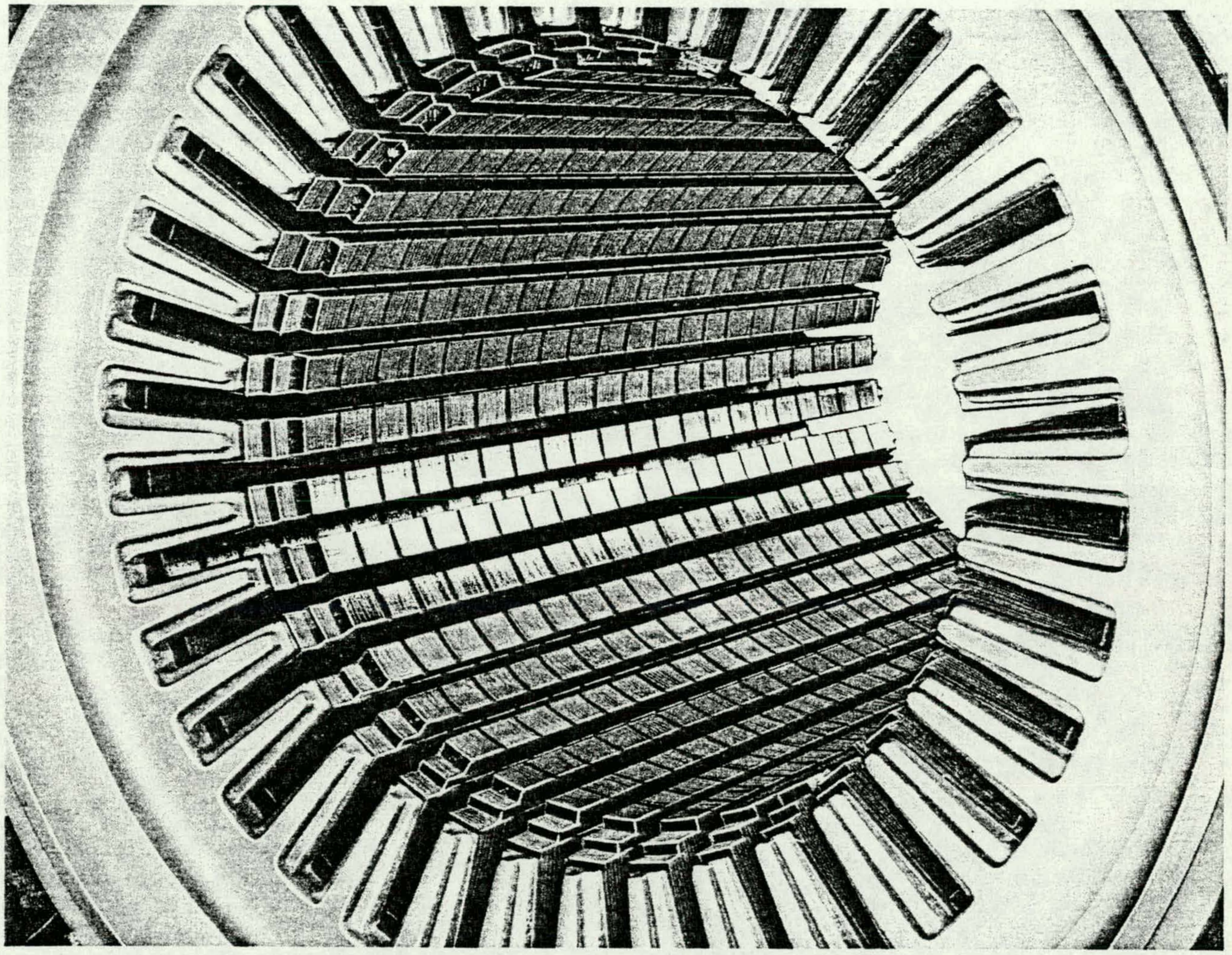


FIGURE 1—Heavy steel clamping rings hold the stator core laminations in place.

THE STATOR CORE is built up of high silicon electrical steel laminations. Each lamination segment is precision ground to remove burrs from punching. The segments are coated with insulating enamel and baked. This coating is regularly checked for thickness and resistivity.

STACKING—The lamination segments are stacked on dovetail keys which are fastened to the stator bore ribs. The keys are machined to close tolerances to assure a true and precise stacking circle. Radial ventilation ducts are spaced between segments at frequent intervals along the core.

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ELECTRIC POWER APPARATUS

Performance Engineered . . . Quality Built

Product Information

Two-Pole GENERATORS TURBINE DRIVEN

FLEXIBLE CORE MOUNTING

THIN CYLINDER SUPPORT

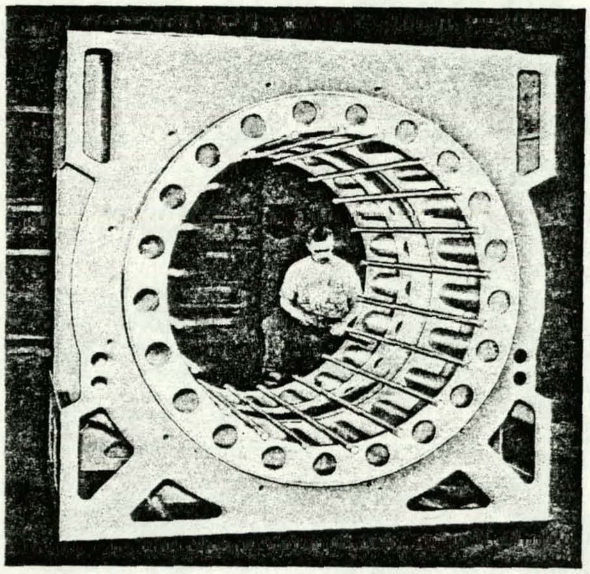

FIGURE 1 - Stator frame for turbine driven generator using thin cylinder flexible core support. Edges of cylinders show in illustration.

MAGNETIC ATTRACTION OF ROTOR FOR THE CORE —— The general configuration of the rotor results in a magnetic force on the stator at the two poles or at 180° intervals. This attraction tends to cause the core to go egg-shaped in spite of its strong construction.

As the rotor turns, the out-of-round wave of deformation follows the poles. For one complete revolution of the rotor, two pairs of high and low spots on the core pass any given point. At 3600 rpm this causes a 120 cycle vibration which can be transmitted to the frame support plates and wrapper plates causing a characteristic low frequency hum.

The vibration is limited by the rigid frame structure in small units and does not present a problem. However, in large machines, it is not practical to make the frame rigid enough to prevent transmission of the vibration in the core and prevent it from reaching the foundation. This is accomplished by making the core support members flexible in a radial direction. The flexible cylinder support method is most commonly used to pro-

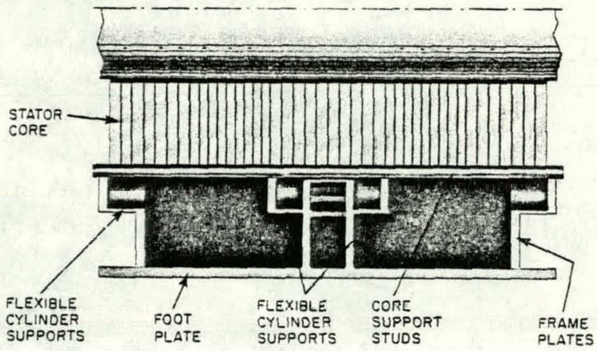

FIGURE 2 - Side view of stator frame with thin cylinder flexible core mounting.

THE FLEXIBLE CYLINDER DESIGN is based on the principle that a thin cylinder is flexible in the radial direction, yet able to support heavy loads in a tangential direction. The cylindrical mountings are sectioned providing support at each end of the stator and also in the center.

The core supports are fastened to the ends of the cylinders with steel plates. Frame plates connect the opposite end of each cylinder to the foot plate. The pulsation of the core is absorbed by the cylinders rather than being transmitted to the foundation.

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ELECTRIC POWER APPARATUS

Performance Engineered . . . Quality Built

Product Information

Two-Pole GENERATORS TURBINE DRIVEN (11kV and over)

STATOR COILS

HALF-COIL CONSTRUCTION

STATOR COILS are form wound into two half sections for ease of handling during the winding operation. The sections are precision shaped so that final connections between top and bottom legs can be made secure.

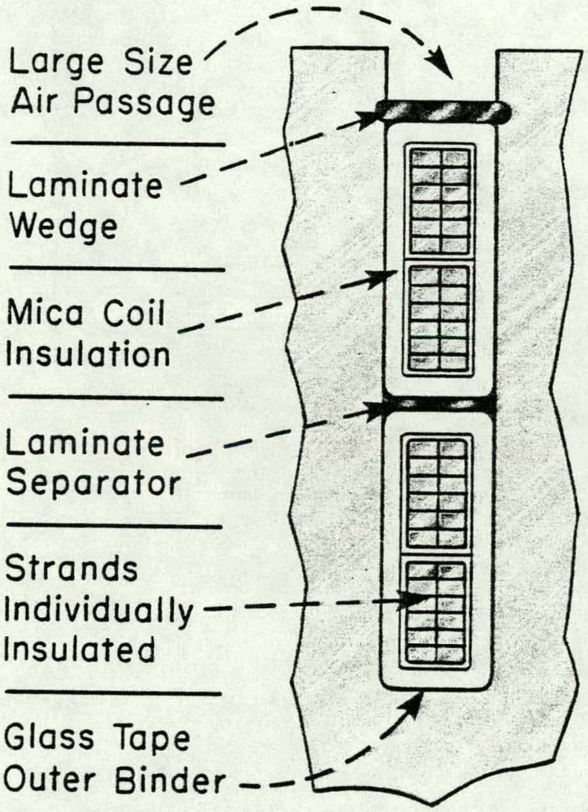

FIGURE 1 – Conductors and coils are individually insulated to comply with standards for Class B insulation.

COIL INSULATION (figure 1) – Class B insulation is used on all two-pole turbine driven generators. The individual strands in each conductor are double glass covered and transposed to provide higher efficiency. (In multi-turn coils, conductors are insulated with mica tape applied continuously over the entire length of the coil.) The slot portion of the coil is bound with glass fibre tape for mechanical protection during handling. The straight portion is thoroughly saturated with resin then hot pressed to size. Successive layers of mica tape applied over the entire coil provide the ground insulation. Coils are finished with a layer of glass tape which serves as a protective cover.

VACUUM-PRESSURE IMPREGNATION – Stator coils are vacuum-pressure impregnated using a modified polyester resin and are then placed in presses to insure that the coils conform uniformally to the proper dimensions and to consolidate the insulation while oven-curing the resin. This process produces a void-free mica insulation which is thoroughly saturated with a high quality resin, eliminating areas in which internal corona could occur.

The slot portions of the coils are painted with a conductive varnish to bring the glass outer binder to ground potential and thereby eliminate corona in air spaces between the coil sides and the stator laminations. This elimination of the corona is extended from the end of the slot onto the coil corners with a semi-conducting compound. This compound is covered with a protective layer of glass tape and then the entire end of the coil is brushed with varnish.

Coils are pressed into the slots with pressure applied evenly over the entire length of the coil. A semi-conducting laminate is used between the coils in each slot and between one side of the coils and the slot walls to insure tightness. Temperature detectors are placed between upper and lower coil sides in each phase to check stator coil temperature. Snug fitting laminated phenolic slot wedges hold the coils tight. A good connection is assured by pouring high conductive solder into copper sleeves which fit tightly over the turns and

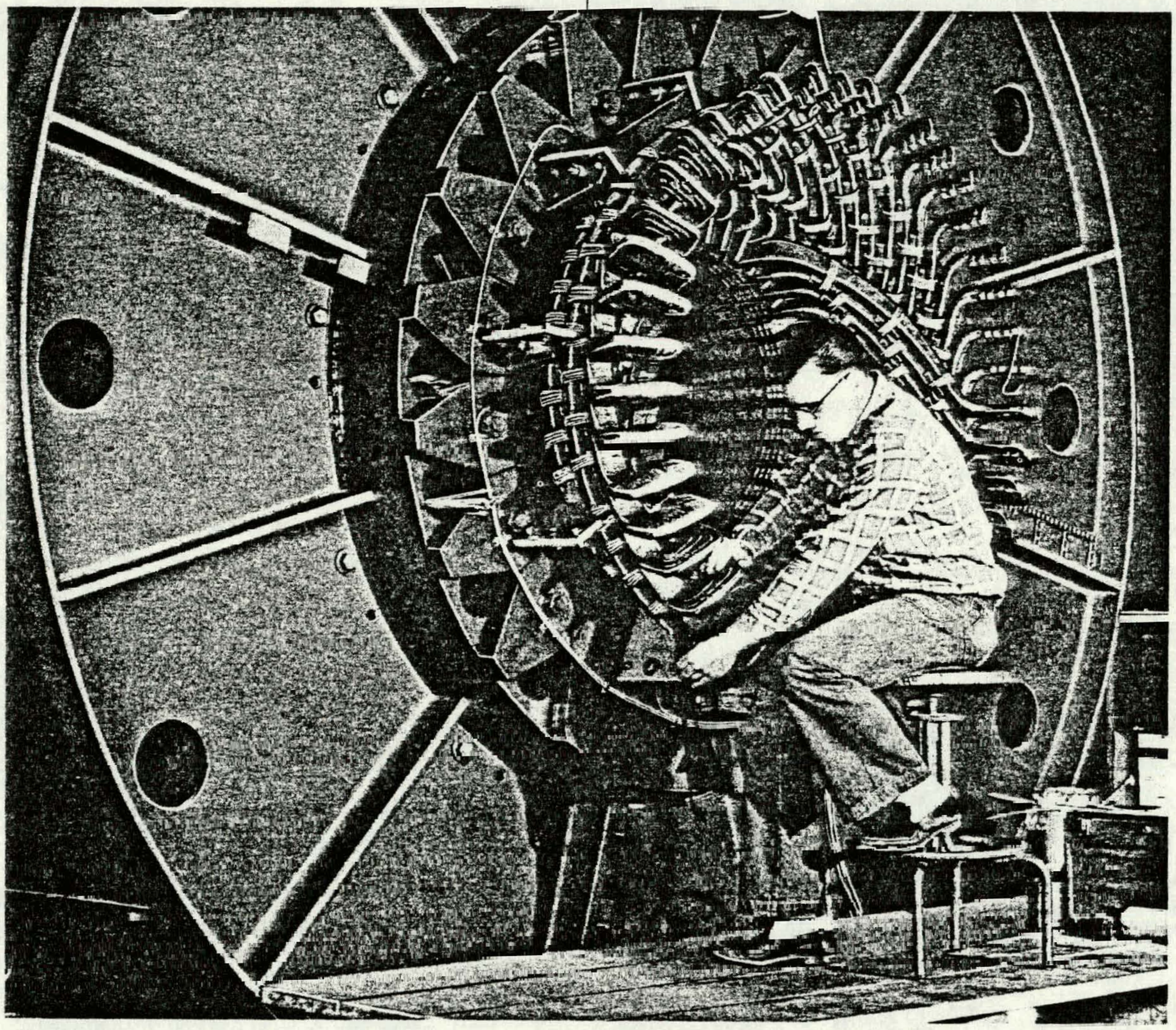

FIGURE 2 – Connections between top and bottom legs of half-coils are taped to maintain the high insulating qualities of the coil. A large wheel fixture around the frame allows core to be rotated to increase efficiency of the winders.

strand groups at both ends of the sections. The connections are then taped to retain the high insulation level of the individual sections.

COIL TESTING – Completed coils are tested to prove group-to-group and turn-to-turn insulation. Slot portions of coils are given a high voltage ground test before they are placed in the slots. A graduated test voltage is applied as the work on the windings progresses. An a-c dielectric test voltage of twice rated voltage plus 1000 volts is applied after the machine has completed all operating and electrical tests.

COIL BRACING AND LASHING (figure 2) The generator windings are braced to withstand the forces imposed on them by system surges and shorts circuits. The coils are subjected to two principal types of forces. One force tends to pull all the coils in the same phase together; it is resisted by placing blocks between adjacent coils and lashing in place with glass fibre cord. The other force tends to push the windings back from the rotor. This force is resisted by regular spaced insulating blocks which support the rings to which the coils are lashed with glass cords.

ELECTRIC POWER APPARATUS

Performance Engineered . . . Quality Built

Product Information

Two-Pole GENERATORS
TURBINE DRIVEN
BEARINGS
SPLIT SLEEVE BEARINGS

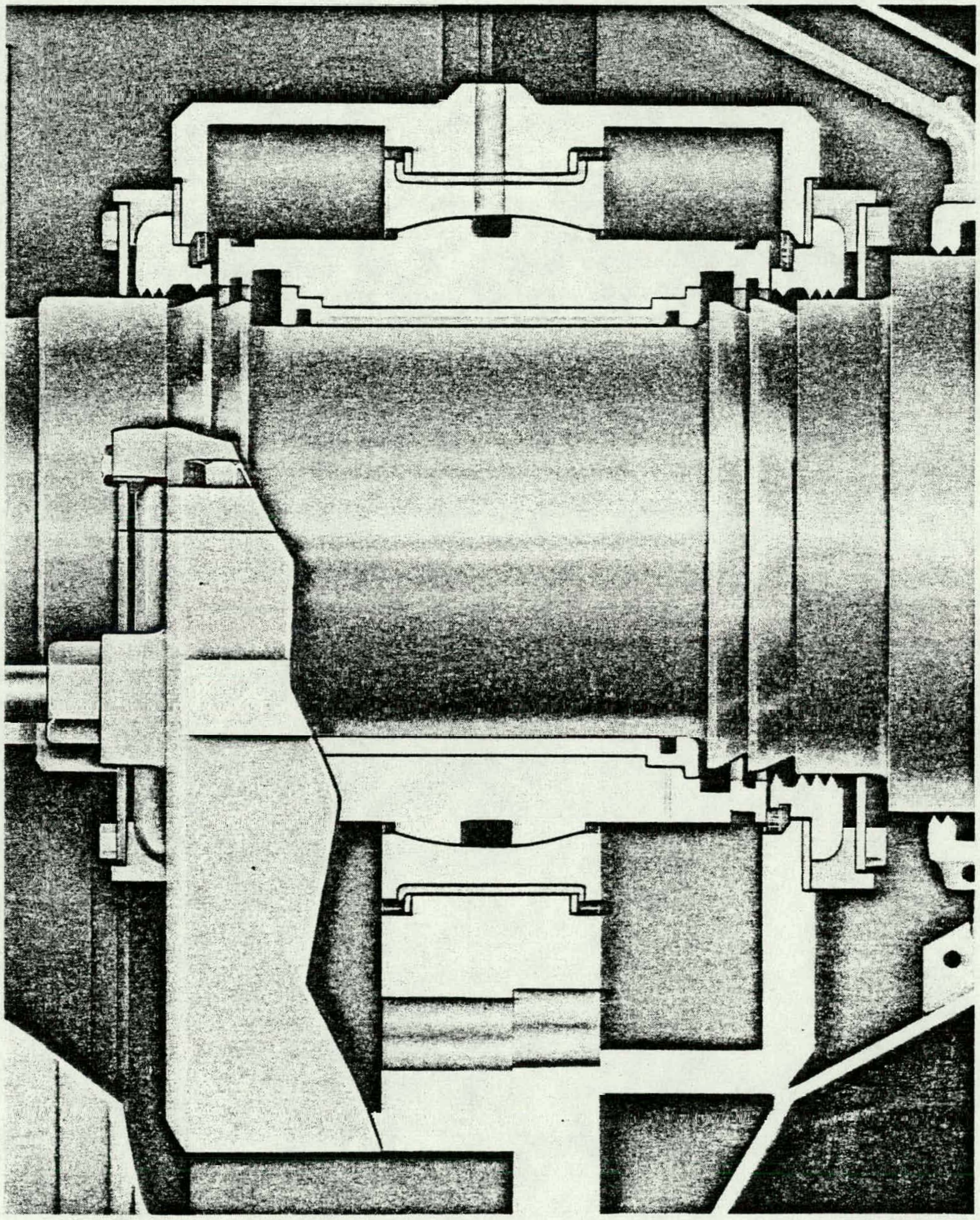

FIGURE 1—Cut-a-way of split sleeve bracket bearing.

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BEARING SLEEVES — Precision machined split type sleeve bearings are designed with a spherical seat for self alignment. After removing surface silicon and graphite, the cast iron bearing shells are electrolytically etched by the Kolene process. The shells are then tinned and babbitted by centrifugal casting which insures a strong metallurgical bond between the cast iron and the babbitt. Only tin base babbitt with high resistance to corrosion is used.

The bearings incorporate a generous load supporting area in the lower sleeve. Carefully designed clearances insure proper lubrication and cooling. Adequate drains are provided for end leakage. The upper sleeve is relieved to prevent oil whip.

LUBRICATION — The amount of oil entering the bearing is metered so that just the right quantity comes in contact with journal. The oil flows over the bearing surface in a fan-shaped pattern. Oil flows outward toward each end of the bearing where it is collected in drain grooves and discharged into the bearing housing. If any oil passes the shaft seals which are located outside the drain groove, it is thrown into a secondary drainage area by a shaft slinger. Metal seals prevent leakage of oil and vapor from the housing.

APPENDIX F1.2

Appendix Item h.

System Item Descriptions

Actuator: This is a hydraulic control device for controlling the rate of the flow and its application is based on adjusting the nozzles' inlet.

Accumulator: The pneumatic accumulator is essentially an oil pressure storage chamber in which the potential energy of oil under pressure can be stored against a compressible force of air. So when the pressure of the system is low or for some reason the pumps are shutdown it can provide a circulating lube oil for a few minutes for coastdown.

Water-Oil-Cooler: Cools the temperature of the circulating oil and is compatible with ammonia on the oil side and sea water on the water side.

Back Pressure Regulator: Back pressure regulator and relief valve provides automatic, continuous protection against high or low pressure. This valve dependably maintains a desired inlet pressure by discharging excess oil into oil tank.

Pressure Regulator: This regulator has been provided to control seal gas pressure after the filter.

Filter: Filtration is for removal of solid particles from the lube oil by passing oil through a filter on which the solids are deposited.

Heater: Heaters have been provided to keep the oil reservoir at suitable temperature or at the time when the system is shutdown during winter. Heaters are thermostatically controlled. The generator will also have built-in heaters to maintain the dryness of the insulation.

Pump: Two pumps, one main and one standby, have been provided to circuilate the lube oil.

Differential Pressure Gauge: This gauge is provided for measuring the pressure drop across the filter.

Electrical System: The electrical system incorporates the automatic synchronizer system. The basic adjustments are breaker closure time compensation and maximum slip frequency*. These adjustments enable the synchronizer to anticipate the optimum circuit breaker closure angle and close the breaker with minimal bus disturbance. The breaker compensator allows the operator to adjust for breaker closure time from .04 to .8 second. The slip frequency circuitry measures the difference in frequency between the generator and the bus. These differences are then compared to the slip frequency control setting on the autosynchronizer module. If the measured value exceeds this setting, synchronization will be inhibited.

Specifications:

Input Power	Oncoming (generator)	Bus	Synch.
Voltage	120/208	120/208	120/208
Frequency	50/60Hz	50/60Hz	50/60Hz
Phase	1	1	1
Burden (Volt Amperes)	5 VA	5 VA	25 VA
Burden Power Factor (Lagging)	.8	.8	.8

* Slip Frequency is the difference frequency, expressed in Hz between generator and station bus.

Appendix Item g. APPENDIX F1.3

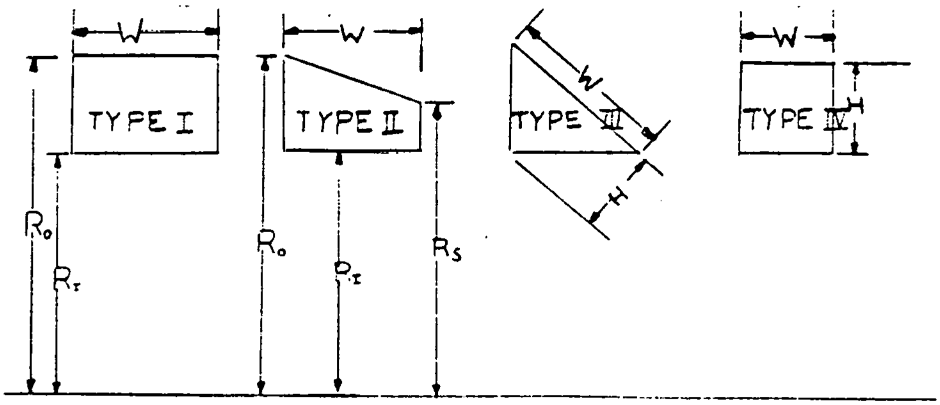
WEIGHT CALCULATION - ROTOR

Material: Aluminum (K01)

Density: .1 lbs/in^3

No. of Blades: 24

Blade Average Thickness: .515



Type I Cylinder
Type II Cone
Type III Plate
Type IV Plate

*	Ro	RI	Rs	W	H	V
1	16.4	15.4		2.0		200.
2	8.2	7.0		2.0		114.6
3	8.6	7.0		2.0		156.8
4	21.5	15.4		.7		495
5	21.5	13.6	16.0	1.3		690.6
6	13.6	12.0		.8		102.9
7	16.0	7.0	11.6	3.4		1527.6
8	11.6	7.0	8.6	4.0		675.2
9	8.6	6.0	7.4	5.2		458.9
10				7.2	2.5	222.5
11				11.2	2.8	387.6
12				9.2	2.9	329.7
13				7.8	1.7	163.9
14				9.6	3.4	201.7
15				1.4	10.0	173

$V_T = 5900$

Weight = (V_T) (.1)

= 590 lbs.

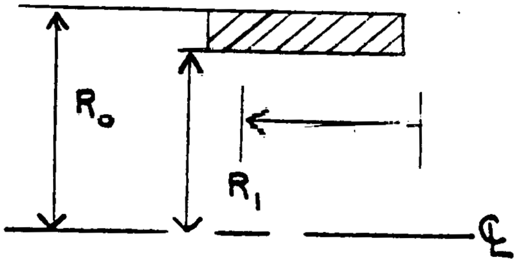
*Attached Figures 1 & 2 show the wheel and shaft sections and numbering for these sections.

WEIGHT CALCULATION - SHAFT

Material: Stainless Steel (17-4)

Density: .283 lbs/in^3

Type I



	Ro	RI	W	V
1	4.	3	11.	241.9
2	4.6	3	7	267.4
3	4.8	3	10.4	458.7
4	6.0	3	22.8	1933.9
5	7.0	3	11.6	1457.7

$V_{Total} = 6537$

W Shaft = V Total e x 2
= 3700 CBS

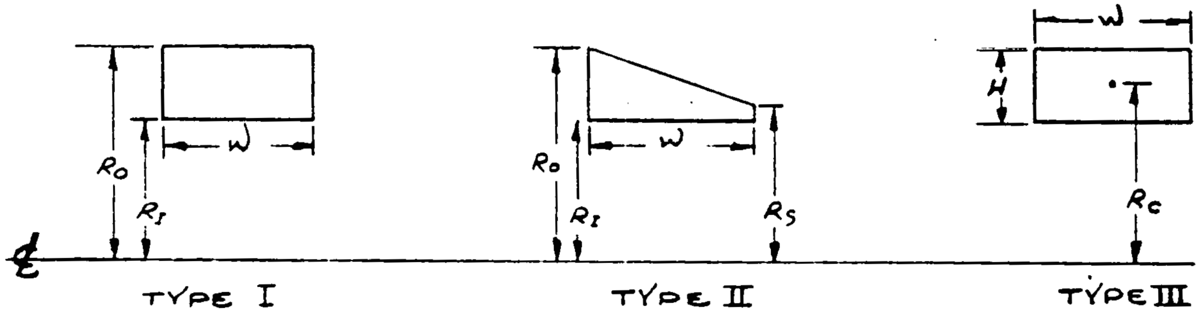
MASS MOMENT OF INTERIA - ROTOR

Material: Aluminum (Ko-1)

Density: .1 lbs/in^3

No. of Blades: 24

Blades averages thickness



Type 1 Cylinder
Type 11 Cone
Type 111 Plate

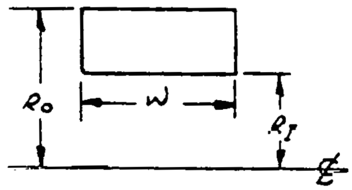
	R	RI	Rs	W	H	Rc	I
1	16.4	15.4		2.0			13
2	8.2	7.0		2.0			1.72
3	8.6	7.0		2.0			2.5
4	21.5	15.4		.7			44.8
5	21.5	13.6	16.0	1.3			.32
6	13.6	12.6		.8			2.9
7	16.0	7.0	11.6	3.4			46.0
8	11.6	7.0	8.6	4.0			9.86
9	8.6	6.0	7.4	5.2			3.3
10				7.2	2.5	20.0	23.0
11				11.2	2.8	17.8	31.8
12				9.2	2.9	15.0	19.3
13				7.8	1.7	13.2	7.4
14				5.8	3.4	12.6	10.1
15				1.4	10.0	10.4	5.2

One rotor = .253 lb-in-Sec^2

MASS MOMENT OF INERTIA - SHAFT

Material: Stainless Steel (17-4)

Density: .1 lbs/in^3



	Ro	RI	W	I
1	4	3	11	2.2
2	4.6	3	7	3.0
3	4.8	3	10.4	5.4
4	6	3	22.8	31
5	7	3	11.6	30

I Shaft - 72.6 lb + n - Sec^2

I total = 2xI Turbine +$_2$I½Shaft + IShaft (5)
= 60. lb-In-Sec^2

WEIGHT CALCULATION - ROTATING ASSEMBLY

W Total = W Shaft + W-Rotor

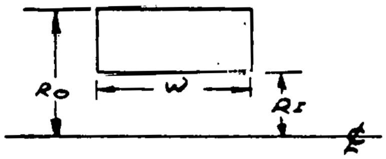
W Total = 3700 Qs + 1180 lbs

= 4880 lbs.

TORSIONAL STIFFNESS - SHAFT

Material: Stainless Steel

Modulus of Rigidity 11.2×10^6 psi

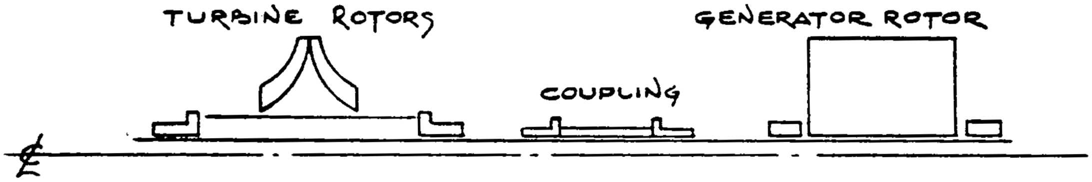


	Ro	RI	W	Stiffness
1	4	3	11	1.311 x 108
2	4.6	3	7	9.217 x 108
3	4.8	3	10.4	7.609 x 108
4	6	3	22.8	9.375 x 108
5	7	3	11.6	3.518 x 109

Stiffness equivelent = 8.7964×10^7 lb/in

TORSIONAL FREQUENCIES CALCULATION - POWER GENERATING ASSEMBLY

Power output 14 MW



Coupling stiffness 1×10^8 lb/in

Generator shaft stiffness = 5×10^8 lb/in

Stiffness equivelent = 4.28×10^2 lb/in

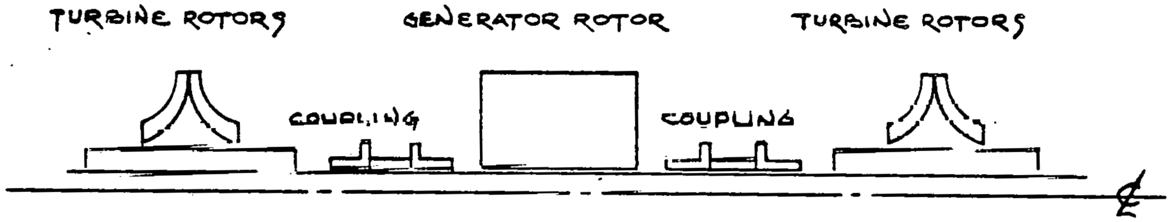
Mass moment of inertia Gen. Rotors assumes to be 5307 lb-in-Sec^2

$$W = \frac{60}{2\pi}\sqrt{\frac{(5307 + 610)}{(5307)(610)}\ 4.28 \times 10^7}$$

$$= 2671 \text{ rpm}$$

TORSIONAL FREQUENCIES - POWER GENERATING ASSEMBLY

Power output 35 Mw



Coupling stiffness 1. x 10^8 lb/in

Generator's Shaft Stiffness - 5 x 10^8 lb/in

Stiffness Equivalent = 4.28 x 10^7 lb/in

Mass moment inertia gen. rotors assume to be 5307 lb-in-sec^2

Solve for frequency of oscillation

$$W = \frac{60}{2\pi}\left(\frac{.1687 \sqrt{(.1687)^2 - (4)\ (1.078 \times 10^{-6} \times 6527}}{(2)(1.078 \times 10^{-6})}\right)^{\frac{1}{2}}$$

W = 2671 RPM

W_2 = 2808 RPM

Fig. F1.3-1 Sectioning of Wheel and Shaft for Weight, Inertia, and Stiffness Calculations

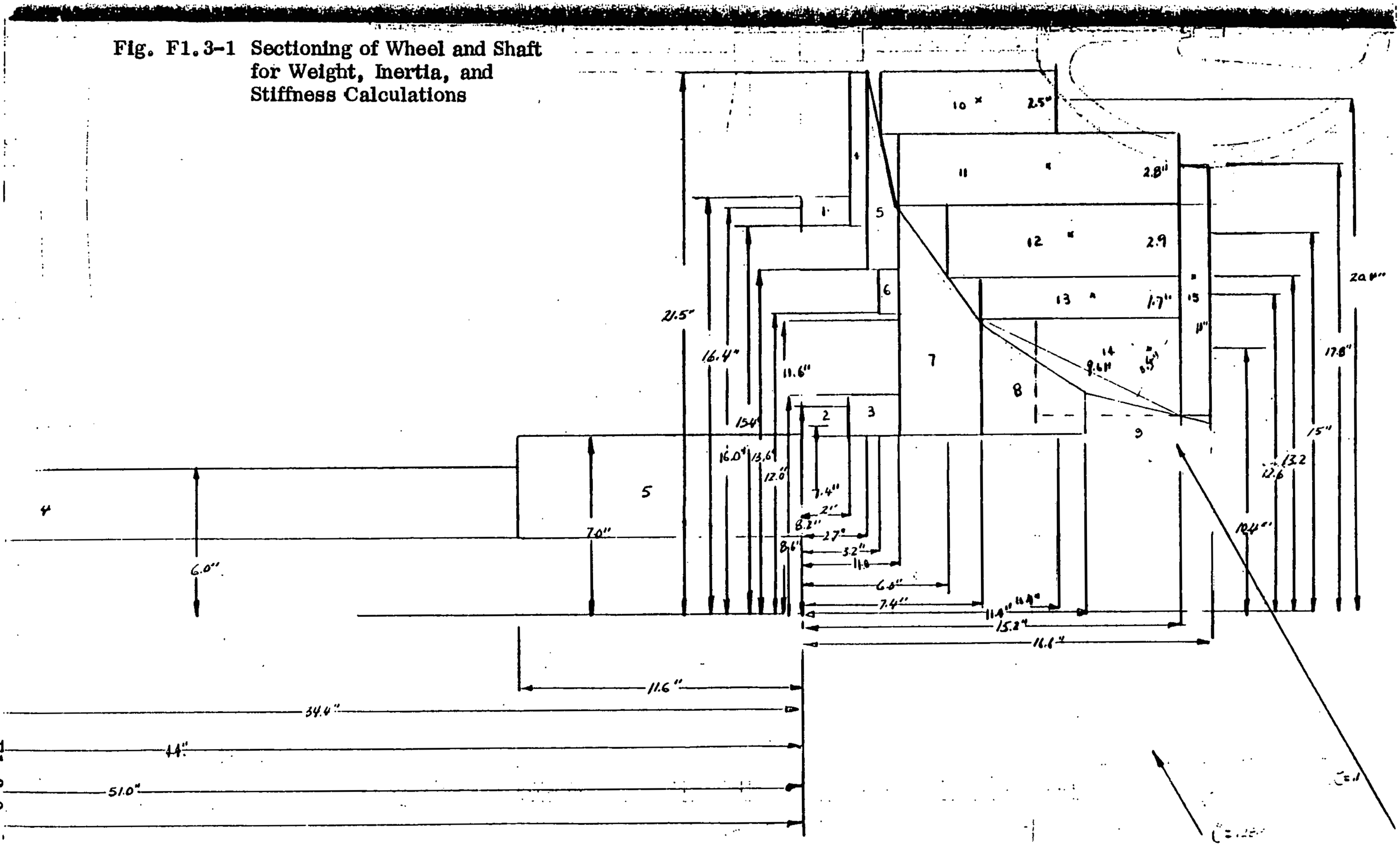


Fig. 1.3-2 Sectioning of Shaft for Weight, Inertia, and Stiffness Calculations

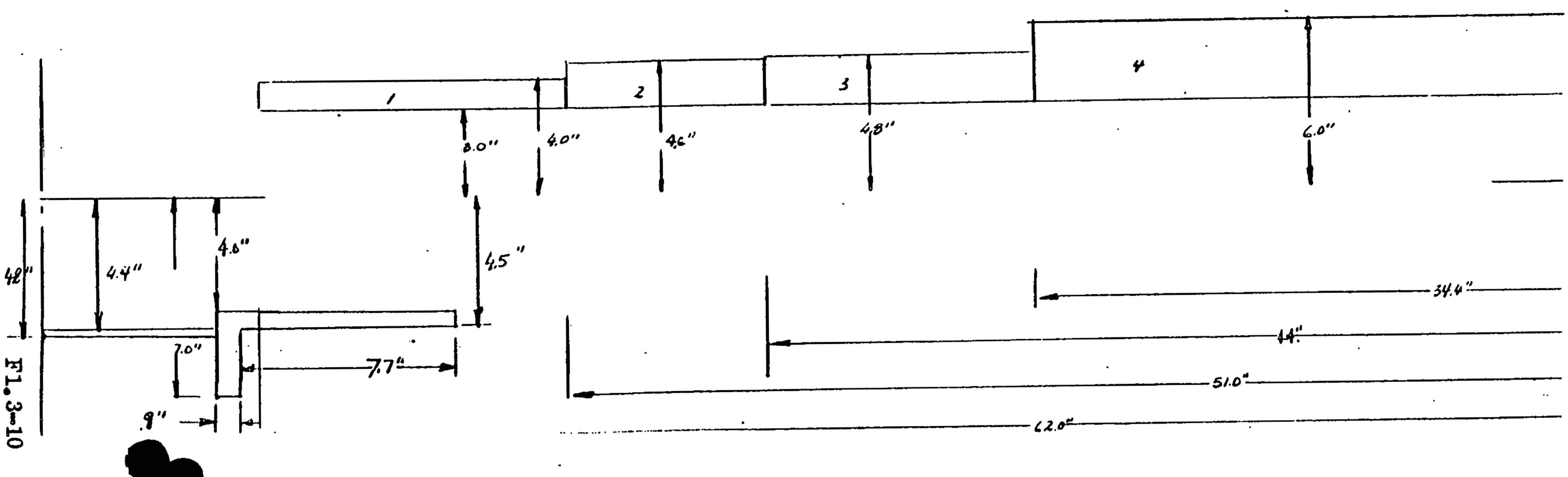


Fig. F1.3-3 Critical Speed Computer Run

```
(1) JOB NO. AND COMMENTS
1277  R.IHDE  9-19-78

(2) OIL VISCOSITY (CENTIPOISE)
    V.
   17.5

    BEARING DATA
(3)                 BEARING 1
    D.          L.          DC.         X.
    9.5         6.5          .009     21.562
(4)                 BEARING 2
    D.          L.          DC.         X.
    9.5         6.5          .009    102.438

(5)  SHAFT DATA
SEGM.
    X.          D.          DH.
  1
    8.062       8.           6.125
  2
   11.          8.           6.
  3
   18.125       9.25         6.
  4
   25.938       9.5          6.
  5
   27.5         9.875        6.
  6
   36.25       11.688        6.
  7
   44.25       11.688        6.
  8
   45.5        11.75         6.
  9
   47.25       12.           6.
 10
   50.5        11.938        6.
 11
   62.         14.           6.
 12
   73.5        14.           6.
 13
   76.75       11.938        6.
 14
   78.5        12.           6.
 15
   79.75       11.75         6.
 16
   88.5        11.688        6.
 17
   96.5        11.688        6.
 18
   98.062       9.875        6.
 19
 105.875        9.5          6.
 20
 113.           9.25         6.
 21
 115.938        8.           6.
 22
 124.           8.           6.125
 23
```

(6) ATTACHMENTS DATA

ITEM

XGC.	L.	D1.	D2.	W.
1				
5.5	111.	18.	8.	72.
2				
25.375	1.125	16.	~~9.5~~ 9.5	37.
3				
35.438	19.5	12.5	11.812	92.5
4				
55.438	16.5	29.	13.	694.
5				
68.562	16.5	29.	13.	694.
6				
88.562	19.5	12.5	11.812	92.5
7				
98.625	1.125	16.	9.5	37.
8				

(7) BEARING FUDGE FACTORS

FPS.	FV.
230.	1.

(8) SPEED

RPM.

3600.

SHAFT WEIGHT 2381.295

TOTAL WEIGHT 4100.295 LBS

PLEASE WAIT

SHAFT CRITICAL 12793.11 RPM

DIAM. MOMENT 2323856.0*

POLAR MOMENT 604879.000

	FORWARD	BACKWARD
BC1	11934.	11932.
BC2	23713.	18156.
BC3	5487.	5487.
BC4	10848.	9463.

BEARING STIFFNESS AT 3600. RPM

SB1 .52346E 07

SB2 .52346E 07

DAMPING

DB1 -.48758E 08

DB2 -.48758E 08

```
TYPE 9 FOR DEFLECTIONS

CHANGES ?
 ?
 9

   X           Y
  4.0310  -.0002147
  5.5000  -.0001964
  9.5310  -.0001464
 14.5625  -.0000847
 21.5620   .0000000
 22.0315   .0000056
 25.3750   .0000440
 26.7190   .0000582
 31.8750   .0001073
 35.4380   .0001389
 40.2500   .0001771
 44.8750   .0002090
 46.3750   .0002164
 48.8750   .0002250
 55.4380   .0002361
 56.2500   .0002368
 67.7500   .0002362
 68.5620   .0002353
 75.1250   .0002235
 77.6250   .0002147
 79.1250   .0002071
 84.1250   .0001723
 88.5620   .0001370
 92.5000   .0001022
 97.2810   .0000569
 98.6250   .0000429
101.9685   .0000054
102.4380   .0000000
109.4375  -.0000813
114.4690  -.0001401
119.9690  -.0002047

CHANGES ?
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APPENDIX F1.4

Appendix Item d.

PRODUCIBILITY ANALYSES

Material & Part Availability

All important materials such as Aluminum Alloy C.355 and C355 and Stainless Steel 17-4 PH are available.

The proposed supplier of the wheel castings, Morris Bean, Yellow Springs, Ohio, has very good experience in casting of C355 wheels iwth larger diameters and higher tip speeds than 575 ft/sec.

Special or unique Machinery, Facility or Processing Needs and Facility Availability

Facilities are in existance that can handle any size casing that we have conceived for the Ocean Thermal Machines.

Casings up to 50,000 lbs. can be realized at Standard Tool and Die, 1931 North Broadway, Los Angeles, CA or L & F Company, 2110 Belgrove, Huntington Park, CA.

Balancing and overspeed facilities for this size of rotor are available at companies such as Pacific Air Motive or Aviation Power Supply, and will be to an accuracy that allows a rotor assembly to be canged in the field without the need for re-balancing.

Facilities exist to manufacture the remaining parts within the Rotoflow organization.

APPENDIX F1.5

Long Lead-Time Items

This is a summary of the longer lead time deliveries and the present lead time available from normal suppliers.

These may change as the capacity of suppliers varies with demand, however, the longest lead time item the generator, is not expected to change significantly.

The list is for long lead time items for the 14 MW machine. For the 70 MW machine, the generator lead time is the same, and so are the other items, as the basic module size is the same for both. The 70 MW has four times the quantity of material.

Long Lead Items

Item	Qty	Time for Acquiring	Material	Remarks
A. Generator 14 MW	1	52 weeks after drawing approval	commercial	commercial
B. Expander Case built up	1	2 days	st steel 316L CF3M	
C. Shaft - (hollow)	1	16-20 weeks	st steel 17-4 Ph plate stock & centrifug-ally cast	similar to shaft on Job 1283
D. Rotor castings	2	8 weeks	st steel 316 ASTM-A-351	
E. Rotor followers	2	8 weeks	st steel CF3 316 ASTM-A-351	
F. Nozzle segments	50	8 weeks	st steel 17-4 Ph AMS 5355	
G. Nozzle adjusting crank	50	8 weeks	304 st steel ASTM-A-351	
H. Raco seals approx. 48" 0	4	8 weeks	teflon	Fluorocarbon Co.
I. RF flange 60"	1	15 weeks	st steel 316	Commercial
J. RF flange 72"	1	15 weeks	st steel 316	Commercial

APPENDIX F1.6

Appendix Item a.

Spares

The cost of spares has been estimated as follows:

Lube system spares:	
10 MW (14 MW gross)	$108,589.00
50 MW (70 MW gross)	$268,662.00
Machine spares:	
10 MW	$258,418.00
50 MW	$639,384.00
Generator spares:	
10 MW	$200,000.00
50 MW	$500,000.00

The component spares list is provisional at this stage, but gives an idea of the quantities involved. The larger quantities of bearings are those used to permit friction-free operation of the expander nozzle system and allow fine control of speed. Also, rapid shutdown in the event of load rejection thus protecting the unit against overspeed.

Spares Listing (Recommended)

Nomenclature	50 MW(70 gross)	10 MW(14 gross)
Expander Rotor	4	2
Nozzle Shaft	200	50
Nozzle Shaft Bearing	300	150
Crank Needle Bearing	200	100
Actuator Needle Bearing	4	2
Nozzle Thrust Bearing	100	50
Nozzle Shaft Seal	100	50
Main Shaft Seal	8	2
Main Bearing Seal	8	2
End Ring Seal	16	4
Crank Pivot Pin	96	48
Actuator Pivot Pin	8	2
Bearing	8	2
Thrust Washer & Pins	8	2
Shaft	4	1
Set of Gaskets	2	1
Set of O-Rings	2	1
Alarm Board	1	1
Control Board	1	1
Control Relay	10	10
Pilot Light 110V Green	2	2
Pilot Light 110V Amber	2	2
Pilot Light 110V Red	2	2
Level Switch	2	1
Magmeter 0-1000CPS 12VDC	1	1
Pressure Switch 1-25 psi	1	1
Pressure Switch High Range	1	1
Shutdown Board	1	1
Switching Tach Board	1	1
16R100D371 Wiring Board	1	1
Tachometer Pickup	2	1

Nomenclature	50 MW(70 gross)	10 MW(14 gross)
RPM Indicator with Dual Scale	1	1
Vibration Probe with Cable	1	1
Accumulator Charge Kit	1	1
Back Pressure Regulator	1	1
Pressure Reducing & Regulating Valve	1	1
Filter Element Main	2	1
Filter Element Strainer	2	1
Lube Oil Pump	2	2
Temperature Transmitter	2	2
Lube Oil Pump Repair Kit	1	1
Lube Oil Pump Motor	2	1
Filter Regulator	1	1
Hermetically Sealed Element	1	1
Current to Pneumatic Transducer	1	1
Heater with Thermostat	1	1
Watt Meter	1	1
Ammeter	1	1
Power Factor Meter	1	1
Vibration Indicator	1	1

APPENDIX F1.7

Appendix Item b.

Support Equipment

The following tools will be required if the power cartridge is to be broken down onboard the vessel.

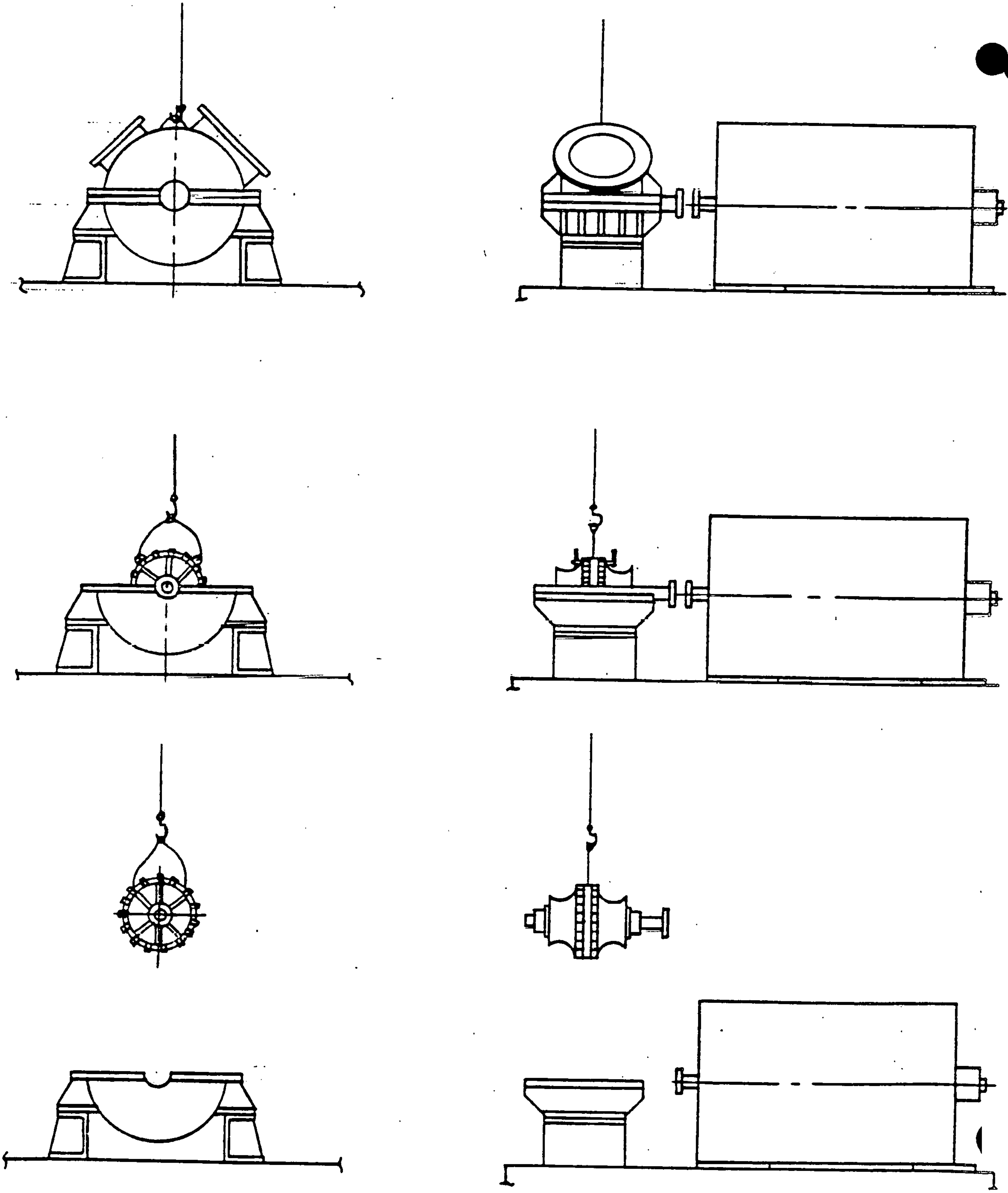
1. Power cartridge support frame.
2. Lifting means for the rotor cartridge.
3. Lifting means for the top half casing, which weighs approximately 12,800 lbs.

In the dissassembly area, the overhead lift must be able to raise 13,000 lbs. Figures 5.3 and 5.4 shows the sequence of disassembly.

Significant weights are as follows:

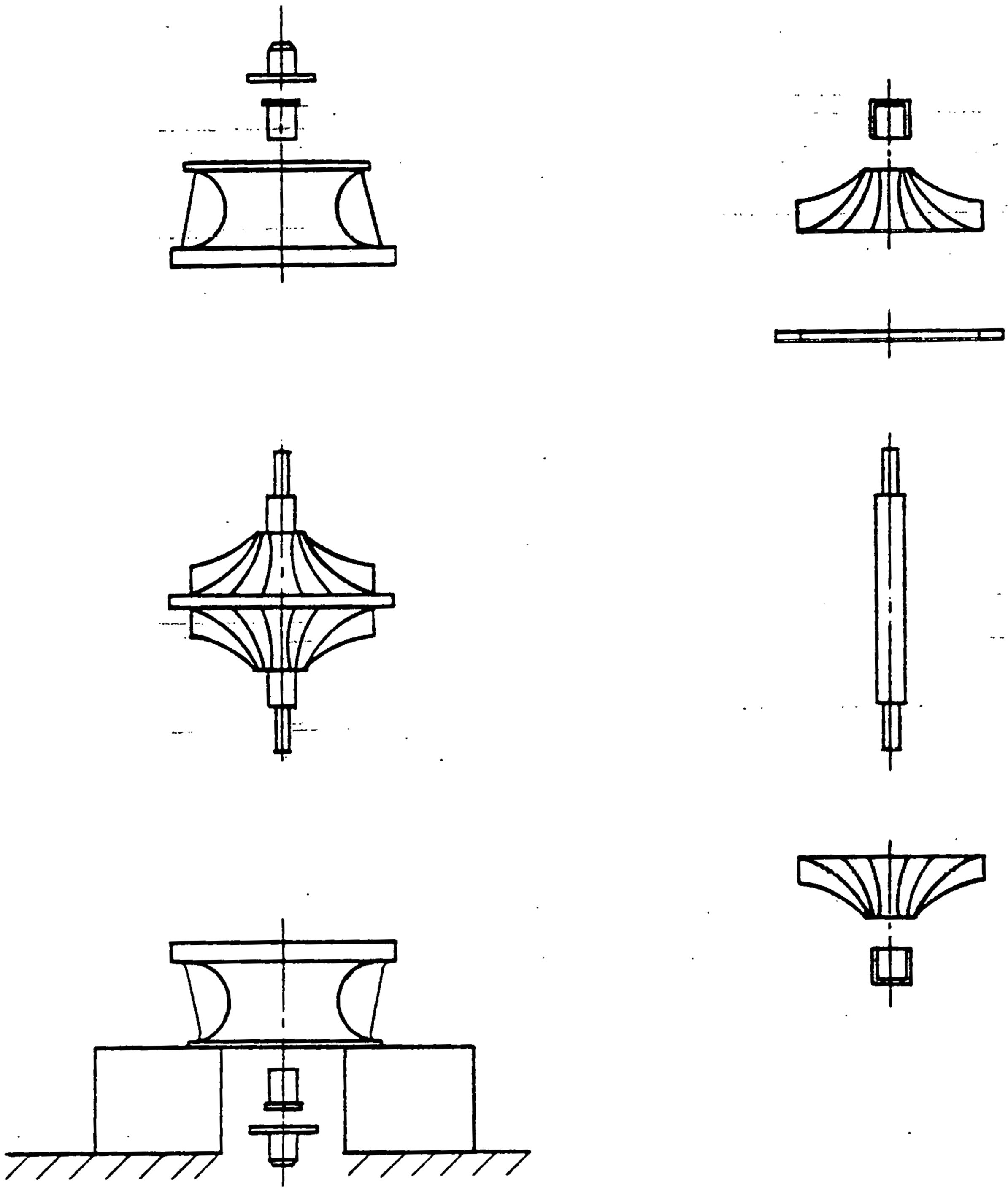
Part	14 MW Gross	70 MW Gross
Turbine	65,000	65,000 x 4*
Generator	118,000	196,000
Turbine Case Cover	128,000	128,000
Power Cartridge	28,900	28,900
Shaft	2,380	2,380
Turbine Wheel	694	730
Rotating Assembly	4,100	4,100
Seal & Bearing Housing (Incl. nozzle assy)	10,600	10,600
Lube Consoles	15,000	18,000

* The 70 MW unit has four turbine casings.



Power Cartridge Removal

F1.7-2



Servicing Power Cartridge

Sizes are summarizes as follows in inches:

Part	14 MW Gross			70 MW Gross		
	L	W	H	L	W	H
Generator	230	145.5	116	250	145.5	132.3
Turbine + Main Oil Pump	194	128	136	194	128	136
Lube Console	370	142	90	370	142	90

Special instrumentation and equipment will be needed for the following operations:

(a) Shaft Alignment - jacking devices, indicators.
(b) Evacuation of turbine - vacuum pumps, hoses.
(c) Leak Test - pressure gauges, leak test.
(d) Electrical Connections - Low-high voltage standard test equipment.
(e) Setting of Controls - Recording apparatus.
(f) Assembly & Disassembly - lifting equipment, cleaning facilities.

For maintenance, tools and equipment will be needed for changing oil, filters, lifting electrical pumps and motors.

Assembly of the turbine is followed by injection of a suitable sealant between two seal rings. A suitable sealant pump will be provided.

APPENDIX F1.8

Non Recurring Costs

$250,000.00 14 MW 2 wheels 9.5"

$701,000.00 70 MW 8 wheels 9.5"

	14 MW	70 MW
HP/wheel	9,700	12,000
Eng. rate $/hr	$12.00	$12.00
Shop rate $/hr	$6.50	$6.50
All overhead	100%	100%
G & A	18%	18%
Total Eng. hours	6,862	20,246
Total Eng. cost hourly, burden, GNA & fee	$213,780	$630,715
Pattern costs	$22,721	$28,100
1st article test labor	6.50 x 2 x 1 x 1.18 x 800 = $13,499	6.50 x 2 x 1.18 x 1.1 x 2500 = $42,185
Total Cost	$250,000	$701,000

APPENDIX F2

Appendix Item a

Performance Characteristics

The axial flow turbine temperature design conditions are specified in Table F2-1. Detailed aerodynamic analysis resulted in a fixed geometry, high degree of reaction design with a maximum efficiency of 90 percent. For the 10-MW(e) module, the turbine is providing 14-MW(e) gross power and for the 50-MW(e) module, 70-MW(e) gross power. The high efficiency is achievable due to the optimized aerodynamic design of a specific speed of $N_s = 126$, and the attention given to minimize inlet and discharge losses. For example, a high performance inlet scroll is provided to assure uniform flow distribution prior to entering the stator vanes and a center body and diffuser are placed downstream of the turbine in order to recover 75 percent of the discharge velocity head of the flow.

The aerodynamic and thermodynamic analysis conducted is based upon the fact that saturated vapor enters the turbine scroll, and liquid droplets larger than 10 micron diameter have been removed from the stream by the separator placed upstream of the turbine. In view of the clean flow path and the generous acceleration through the scroll and turbine, it is realistic to consider the expansion process based on a 99 percent quality flow.

Table F2-1

TURBINE DESIGN CONDITIONS (OTEC)

Alternator $\eta_e = 0.98$

Inlet Temperature	Condenser Temperature	Temperature Differential
76.25°F Inlet	58.75°F Outlet	17.50°F
74	52	22
70	50	20
72.50	57.50	15
62.75	48.25	14.50
60.50	41.50	19

The assumptions for the turbine analysis are summarized in Table F2-2. Table F2-3 summarizes the turbine parameters and Table F2-4 presents the geometry of the preliminary design.

In order to arrive at the preliminary design, the effect of degree of reaction and the velocity ratio μ/c_o have been studied in detail together with the resulting stator and rotor exit angles. In addition, 3 percent and 5 percent exit energy values have been considered. The result is presented in Figure F2-1 and Figure F2-2. The off design performance trend of the axial turbine is presented in Figure F2-3. The power output ratio KW'/KW_o, (where $KW_o = 14{,}000$ for the 14 MW(e) turbine and KW' is the output at off design point), is strictly a function of the temperature differential across the turbine. The following equation represents the off design performance of the OTEC axial turbine with fixed geometry.

$$KW'/KW_o = 1 + 0.015\ (T_c' - 50) \times 0.0625\ (T' - 0.25)$$

where

KW_o = plant design power output

T' = temperature differential across turbine

T_c' = turbine exhaust or condenser temperature

Table F2-2

ASSUMPTIONS USED FOR TURBINE DESIGN

Ammonia Conditions:	P_o	128.79	psia
	T_o	70	^oF
	Q_o	0.99	
	P_3st	88.00	psia
	P_est	89.18	psia
	C_e	30.	ft/sec
	Equilibrium during expansion and diffusion (fog flow).		

Turbine: Cylindrical end walls.

Continuity at tip and root.

Radial equilibrium at No. 2 exit.

Subsonic flow at all stations.

Rotor blades with aspect ratio 1:6.

Uniform meridianal velocity at rotor exit of 224 ft/sec.

Diffuser efficiency of 75 percent.

Table F2-3

TURBINE PARAMETERS

	Units	70 MW(e)	14 MW(e)
AMMONIA TURBINES PERFORMANCE (OTEC)			
Parameters			
N	RPM	1200	3600
ω	lb/sec	3756	778
N_s	RPM	94	126
SHP	HP	93834	18767
η_o	-	0.901	0.905
Ammonia Supply			
P_o	psia	128.79	128.79
T_o	F	70.00	70.00
Q_o	-	0.99	0.99
Turbine			
P_3 Static	psia	88.00	88.00
T_3 Static	°F	49.31	49.31
Q_3 Static	-	0.962	0.962
P_o/P_3	-	1.464	1.464
ΔH^1	Btu/lb	20.30	20.30
C_o	Ft/Sec	1008	1008
η_t	-	0.870	0.870
Turbine Plus Diffuser			
P_e Static	psia	89.18	89.18
T_e Static	F	50.00	50.00
Q_e Static	-	0.963	0.963
P_o/P_e Static	-	1.444	1.444
ΔH^1	Btu/lb	19.61	19.61
C_o	Ft/Sec	991.0	991.0
η_o	-	0.901	0.905

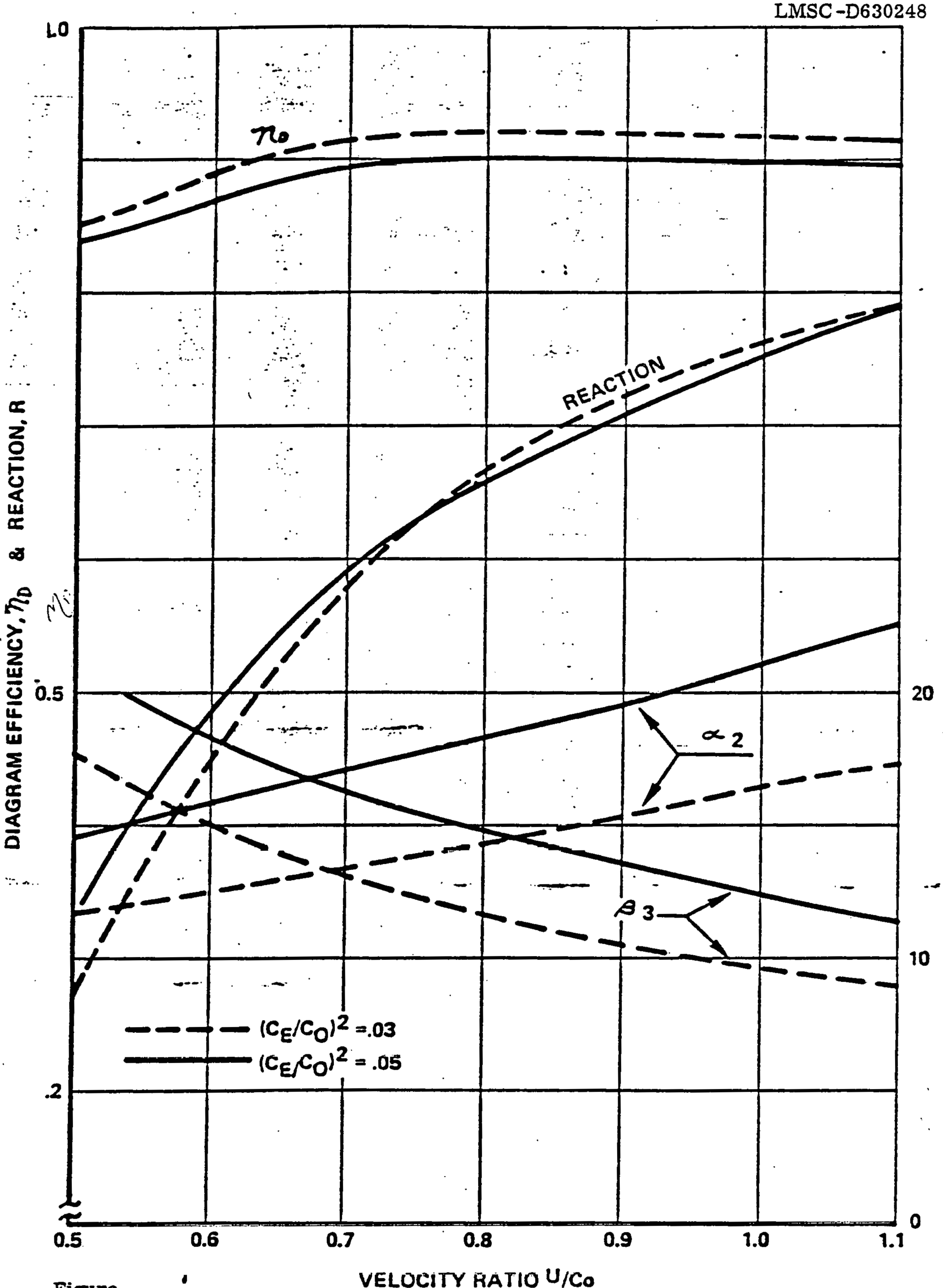


Figure
~~5.4.3.1-~~1 Ammonia Turbine Design(OTEC) Optimum Reaction and Gas Angles

Table F2-4
Ammonia Turbine Geometry

	70 MW(e)		14 MW(e)	
	TIP	ROOT	TIP	ROOT
D	173.3	133.1	64.2	40.7
U	907.5	696.9	1008.3	639.3
R	0.730	0.570	0.770	0.435
α_2	20.28	16.86	21.74	15.35
β_3	13.43	16.86	12.29	19.50
$\Delta\beta$	8.52	47.62	4.41	88.74
C_v rotor	0.984	0.980	0.984	0.965
C_3	226	228	224.8	224.0
η diagram	0.900	0.893	0.898	0.875

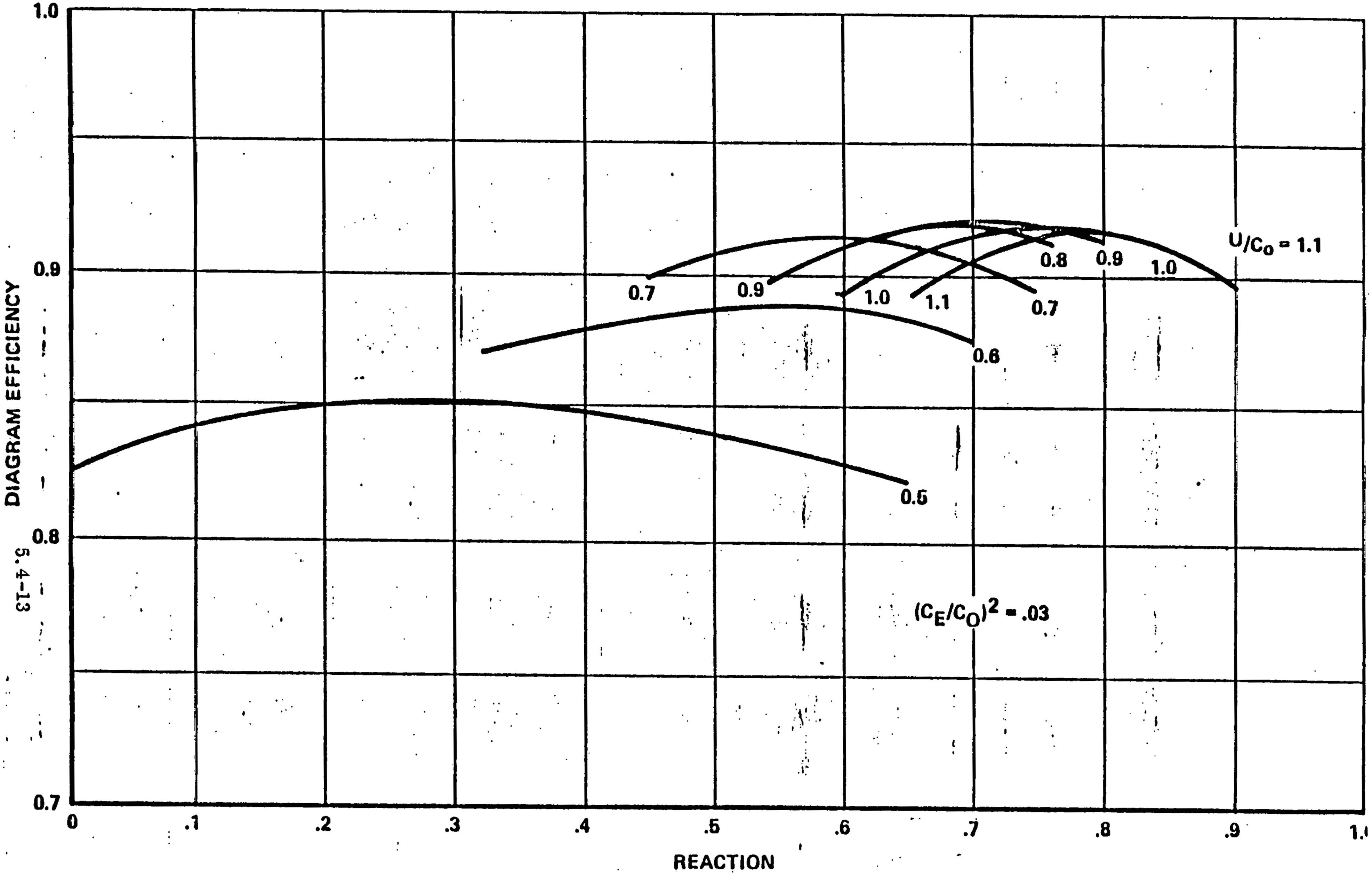


Figure F2-2 Ammonia Turbine Design (OTEC) Optimum Velocity Ratio

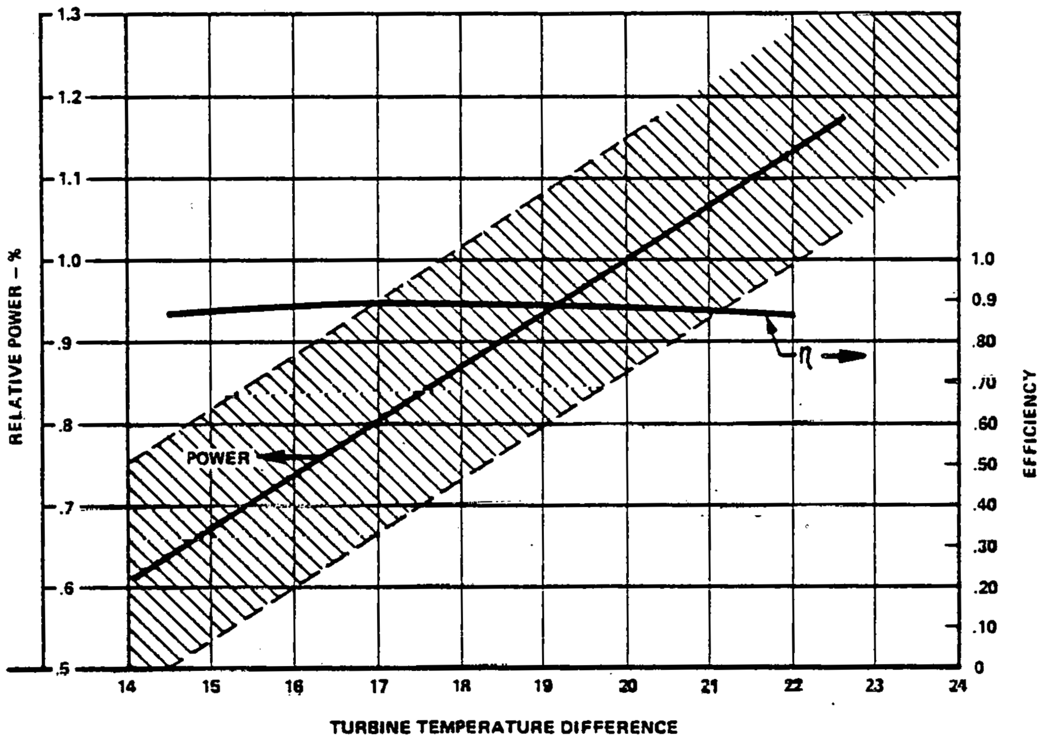


Fig. F2-3 Axial Turbine (Fixed-Geometry) Off-Design Trend for OTEC

APPENDIX F3.1

Worthington Pump Corporation Data

WORTHINGTON PUMP CORPORATION (U.S.A.)
401 WORTHINGTON AVENUE, HARRISON, NEW JERSEY 07029 Tel. (201) 484-1234

June 19, 1978

Lockheed Missiles & Space Co., Inc.
Ocean Systems Research & Development Div.
1111 Lockheed Way
Sunnyvale, California 94086

Subj: LMSC P/O YFV9P6610F
Pump Specification
PSD-2.1 Rev. 3/9/78
Worthington Order
HRD-6061

Attn: Mr. M.I. Leitner,
Senior Staff Engineer
O/57-20, B150

Gentlemen:

In accordance with your above subject purchase order, we are forwarding herewith the following:

1. One copy each of ten different pump data sheets, each of which covers one of the pump selections required by the subject purchase order. These data sheets provide essential performance information, physical dimensions, and estimated weights applicable to the pumps only. Similar information on motors and gears has been solicited from others, and will be added when available, which we expect to be by June 30.

2. Copy of pump unit cross-section, identified as figure 1, which has been marked-up from page 3 of your specification to show additional dimensions which we are providing.

3. Copy of pump performance curve, figure 2, which applies to all of these pumps. Note that this curve is plotted in percentage values for all variables.

4. Copies of five pages on the subject of materials and cost for OTEC pumps, extracted from a report which we recently prepared for another client, and which are equally pertinent here.

In addition to the foregoing we would like to provide information on pump selling prices as follows:

1. Basic selling prices in 1978 dollars for all stainless steel construction may be taken as $7.00 per pound.

2. By substitution of lower grade materials for some components, such as carbon steel for the suction bell and discharge column, and stainless steel clad carbon steel for the suction case, basic selling prices may be reduced to $5.25 per pound (Please note

2. (cont'd)

that we do not consider carbon steel with epoxy coating an acceptable impeller material, and have quoted on the basis of 316 L stainless steel only. Tip speeds for the impellers required here will range from 40 to 60 ft. per second, and it is improbable that the coating would remain intact for very long at these velocities. Exposure of the underlying carbon steel to sea water would then result in unacceptably high rates of corrosion and erosion.)

3. For the prototype units for the 25 and 50 MW applications, and for the first production set for the 10 MW demonstration plant, add to either 1 or 2 above the amounts of $50,000 for engineering and $500 per inch of impeller diameter for patterns and tools.

4. For the eighth production set of any one of these pump selections, deduct 5% from 1 or 2 above.

Please note that the foregoing are preliminary figures for estimating purposes only, and are not to be construed as quotations.

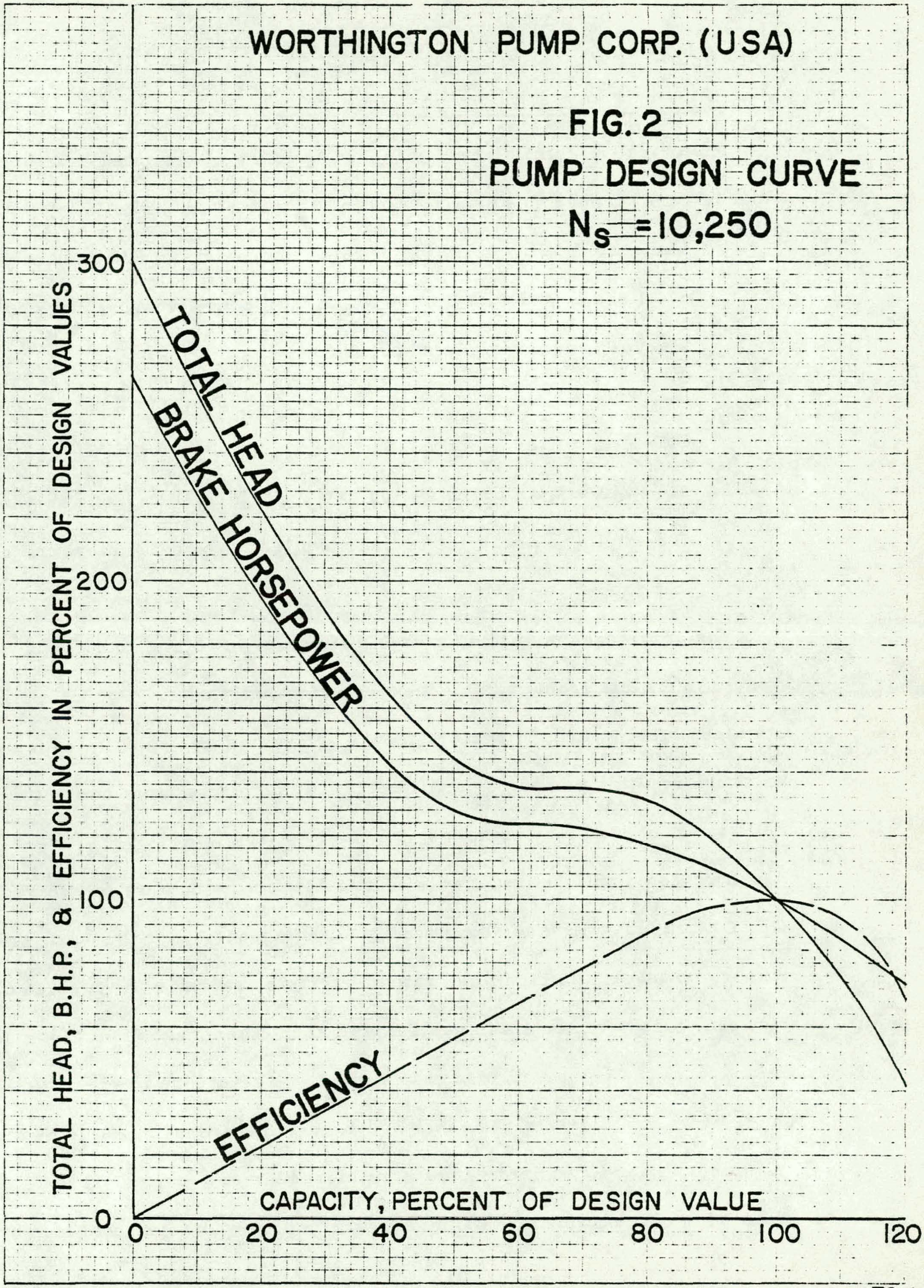
As soon as we have the required information on motors and gears we shall pass it along. In the meantime, should you have any questions on the foregoing please do not hesitate to call us.

Very truly yours,

W. C. Krutzsch
Senior Consulting Engineer

WCK/ds

cc: Ms. B. J. Bartel
Mr. A. Agostinelli
Mr. H. D. Allan
Mr. C. H. Thaw
Mr. I. J. Karassik
Mr. B. Neumann
Mr. R. J. Sirico
Mr. L. Mazzarone
Mr. T. Layne
Mr. J. E. C. Valentin
Mr. D. Gettman



WORTHINGTON PUMP CORP. (USA)
FIG. 2
PUMP DESIGN CURVE
N_S = 10,250
TOTAL HEAD
BRAKE HORSEPOWER
EFFICIENCY
TOTAL HEAD, B.H.P., & EFFICIENCY IN PERCENT OF DESIGN VALUES
CAPACITY, PERCENT OF DESIGN VALUE
0
100
200
300
0
20
40
60
80
100
120

F3-1-3

WORTHINGTON PUMP CORP. (USA)

PUMP MATERIALS AND COSTS

Historically, pumps intended for the handling of seawater have been manufactured from three primary groups of materials, as follows:

1. Standard fitted, consisting of cast iron casing or diffuser, carbon steel shaft, and with bronze for the impeller, wearing rings, shaft sleeves, and other wetted parts.
2. All bronze, usually with Monel or K Monel for shafting, and often with K-Monel or S-Monel shaft sleeves.
3. All stainless, in which a wide variety of high alloy irons and steels are employed according to their availability and chemical and physical properties.

The above groups are not always maintained inviolate, and combinations of elements from pairs of them are frequently used. For example, stainless steel wearing rings or sleeves are sometimes substituted for bronze on standard fitted pumps, or some parts may be furnished in one of the Monel's in an all stainless pump.

Such combinations might well be suitable for the cold water pumps for OTEC plants if otherwise economically advantageous. For the warm water pumps, however, the increased rate of galvanic reactions which could occur between various materials in contact with each other makes them less desirable. Since the pumps could well turn out to be otherwise identical for these two services, it is desirable to consider the use of the same materials for both, and this preference is reinforced in the absence of complete knowledge of the properties of the seawater which will be encountered in various locations at the depths from which the cold water is pumped. It is entirely possible, for example, that unanticipated properties of this water, such as higher dissolved gas content, could offset the otherwise beneficial effect of the low temperature.

Thus the first limitation which appears desirable to impose on material selections is that they be galvanically compatible. This tends to eliminate standard fitted construction (which is normally used only for relatively cold water applications, anyway, and where the life cycle of the pump or most of its components can be shorter than here) and narrows our choices in conventional materials to all bronze or all stainless steel.

In terms of corrosion resistance to seawater the all bronze choice would be excellent. Because of the size and configuration of the pumps required for this application, however, its use becomes undesirable. It is far less suitable for fabricated structures, being more difficult (and in some alloys impossible) to weld, and is not as readily available in plate form as are the ferrous materials. Even for castings, as these become very large, availability is lower than for ferrous materials, maintaining material soundness is often difficult, and the upgrading of casting quality to required standards can become time consuming and very costly. In addition, based on current thinking regarding the materials of construction for other major components of OTEC power plants, it would probably be less compatible galvanically than the remaining possibilities.

In view of the foregoing, and based on demonstrated suitability in numerous previous applications, our present recommendation for pump materials is for all stainless construction, except for shafting of low alloy steel. The choice of particular alloys which might be employed for various individual components of the pumps need not be belabored at this point since there are a considerable variety of possibilities, many of which are nevertheless similar in chemical analysis, physical properties, availability, cost, and manufacturability. It is therefore possible to assure compliance with the forty year design life requirement, and to predict pump costs, without the necessity for final material selection.

Because of the large size of these pumps, and the obvious high cost of producing them in these materials, it may become necessary in a later phase of plant development to consider certain substitute materials which cannot presently be recommended because of lack of prior experience and insufficient research performed to assure their suitability. Some of the possibilities for future consideration, not necessarily in order of preference, would be the following:

1. Carbon steel. For certain components which would be subject only to predictably low velocities, such as the suction bell, the service requirements would not be materially different from those to which the platform itself will be exposed, and similar materials could be employed

2. Aluminum alloys. Within a limited range of velocities, between approximately 5 feet per second and 20 feet per second, several of the 5000 series of aluminum alloys have demonstrated good corrosion resistence to seawater at temperatures which will be encoun-

tered by the warm water pumps. We have found little specific information regarding corrosion resistance at the cold water temperature, but it is assumed that this will also be satisfactory in view of the experience at higher temperatures. Beyond 20 feet per second, the rate of corrosion increases quite rapidly, to the extent that these materials would be unsuitable for pump impellers, where the tip speeds will be from two to four times this value.

3. Non-metallics. Filament wound fiberglass reinforced epoxies, industrial plastics, and even light aggregate concretes may be suitable for some of the low stressed components, but the absence of any solid experience with these materials in the manufacture of large pumps requires that they be considered experimental at the present time. Nevertheless, some of these may have enough to offer in terms of light weight, low cost, and/or corrosion resistance that they should not be overlooked for study as candidates for use in future generations of OTEC power plants.

For the immediate future, however, since the viability of the entire OTEC project could well depend on the reliability of operation of the first few demonstration plants, we must recommend the conservative approach and urge the use of stainless steel pumps.

Current estimates indicate that the cost of such pumps should be taken as $7.00 per pound, in 1978 dollars, for the full range of heads and capacities covered in this analysis. Adoption of some alternate materials, such as carbon steel or austenitic cast irons, where suitable, could reduce this cost up to 20 to 25 per cent.

The utilization of cast iron and carbon steel material throughout, such as is done for fresh water applications, might lower costs by as much as 50%, to a level approaching $3.50 per pound, but pumps fabricated of these materials would be totally unsuitable for the OTEC environment.

WORTHINGTON PUMP CORPORATION (U.S.A.)
401 WORTHINGTON AVENUE, HARRISON, NEW JERSEY 07029 Tel. (201) 484-1234

July 10, 1978

Subject: LMSC P/O YFV9P6610P
Pump Specification
PSD-2.1 REV. 3/9/78
Worthington Order
HRD-6061

Lockheed Missiles & Space Co., Inc.
Ocean Systems R & D Div.
111 Lockheed Way
Sunnyvale, California 94086

Attention: Mr. M.I. Leitner
Senior Staff Engineer
0/57-20, B150

Dear Murray;

This will confirm our phone conversations of July 6 & 7, 1978 regarding information on gear units and motors.

We are forwarding herewith the following:

1. One copy each of ten different data sheets revised to add information on gear units and motors.
2. A copy of pump unit cross-section, identified as Figure-1, revised to indicate required dimensional changes to accomodate rectangular cross-section open drip proof motors.

The following will give further details on gear units and motors.

Gear Units

a) Information on dimensions, weight and price is based on 'VOITH' planetary gear units.

b) Two gear units are mounted in series to get required gear ratio.

c) Specified information on the revised data sheets is based on two gear units mounted in series.

d) Per 'VOITH' most of these gear units have been previously designed and built.

e) Approximate lead time will be 10 to 12 months.

Motors

a) Information on dimensions, weight and price is based on the Reliance Electric motors.

b) All motor frames are open drip proof requiring pad mounting.

c) For submercible motors new frame designs will be required which will increase selling price approximately 50%.

d) We were unable to obtain quote on motor for condition LMSC #3 due to high required motor horsepower.

e) Approximate lead time will be 14 to 16 weeks.

In addition to the foregoing we would like to provide information on gear units and motor selling prices as follows:

1. Basic selling price in 1978 dollars for all gear units may be taken as:

   a) $11.00 per pound if motor rated horsepower is greater than 2000 HP.

   b) $19.00 per pound if motor rated horsepower is less than or equal to 2000 HP.

2. Basic selling price in 1978 dollars for all except 7204 HP motor may be taken as $15.00 per rated motor horsepower. This price is based on drip proof frames.

Please note that the above are preliminary figures for estimating purposes only, and are not to be constructed as quotations.

We hope this will complete all the requirements on the subject purchase order.

We are looking forward for an opportunity to develop pumping systems for this important energy development program.

If you have any further questions regarding the subject matter please do not hesitate to contact us.

Sincerely yours,

R.H.Javia

R.H. Javia
Project Manager

RHJ/ds

cc: Ms. B.J. Bartel
Mr. A. Agostinelli
Mr. W.C. Krutzsch
Mr. H.D. Allan
Mr. C.H. Thaw
Mr. I.J. Karassik
Mr. B. Neumann
Mr. R.J. Sirico
Mr. L. Mazzarone
Mr. T. Layne
Mr. J.E.C. Valentin
Mr. D. Gettman

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #1

Operating Conditions

Total Head (ft.)	15	BHP (Sp Gr 1.03)	3622
Capacity (USGPM)	817000	Motor HP	4527
RPM	86	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	18	Unit Efficiency (%)	78
Submergence (ft.)	5		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				
		Dia.		Length	Est'd Wt. (lbs)
Suction Bell	E =	237	H =	39	29000
Impeller	A =	160	G =	65	23000
Impeller Case	A =	160	G =	65	27000
Diffuser	L =	194	J =	86	72000
Diffuser Extension	M =	222	K =	100	59000
Discharge Column	D =	237		137	34000
Shaft (@ Coupling)		13		194	12000
Sub - Total					256000
Motor	N =	* 87 x 74	P =	89	14700
Gear	R =	44		96	16500
Sub - Total					31200
Unit Total	B =	107	C =	388	
			F =	427	287200

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 nos. Manufacture 18 nos.

* Rectangular cross-section

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #2

Operating Conditions

Total Head (ft.)	15	BHP (Sp Gr 1.03)	1813
Capacity (USGPM)	409000	Motor HP	2266
RPM	122	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	18	Unit Efficiency (%)	78
Submergence (ft.)	5		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.) Dia.	Length	Est'd Wt. (lbs)
Suction Bell	E = 167	H = 39	11600
Impeller	A = 113	G = 46	9100
Impeller Case	A = 113	G = 46	10600
Diffuser	L = 137	J = 61	28300
Diffuser Extension	M = 157	K = 71	23500
Discharge Column	D = 167	83	13400
Shaft (@ Coupling)	9	137	4100
Sub - Total			100600
Motor	N = *74x65	P = 87	11900
Gear	R = 33	72	6160
Sub - Total			18060
Unit Total	B = 76	C = 272	
		F = 300	118660

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 18 mos.

*Rectangular cross-section

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #3

Operating Conditions

Total Head (ft.)	15	BHP (Sp Gr 1.03)	5763
Capacity (USGPM)	1300000	Motor HP	7204
RPM	69	Motor Efficiency (%)	
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	18	Unit Efficiency (%)	
Submergence (ft.)	5		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				Est'd Wt. (lbs)
		Dia.		Length	
Suction Bell	E =	298	H =	50	55000
Impeller	A =	201	G =	81	43000
Impeller Case	A =	201	G =	81	51000
Diffuser	L =	244	J =	108	135000
Diffuser Extension	M =	280	K =	127	112000
Discharge Column	D =	298		170	64000
Shaft (@ Coupling)		16		244	23000
Sub - Total					483000
Motor	N =	Info. not available yet	P =		
Gear	R =	44		96	16500
Sub - Total					
Unit Total	B =	135	C =	486	
			F =	536	

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 21 mos.

Revision 1
July 7, 1978

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #4

Operating Conditions

Total Head (ft.)	15	BHP (Sp Gr 1.03)	2882
Capacity (USGPM)	650000	Motor HP	3602
RPM	97	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	18	Unit Efficiency (%)	78
Submergence (ft.)	5		

Approximate Physical Characteristics (See Fig. 1)

		Dimensions (in.)			
		Dia.		Length	Est'd Wt. (lbs)
Suction Bell	E =	211	H =	35	22000
Impeller	A =	142	G =	58	17000
Impeller Case	A =	142	G =	58	20000
Diffuser	L =	173	J =	77	53000
Diffuser Extension	M =	198	K =	89	44000
Discharge Column	D =	211		121	25000
Shaft (@ Coupling)		11		173	8000
Sub - Total					189000
Motor	N =	*87 x 74	P =	89	14700
Gear	R =	44		84	11000
Sub - Total					25700
Unit Total	B =	95	C =	345	
			F =	380	214700

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 nos. Manufacture 18 nos.

*Rectangular cross-section

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #5

Operating Conditions

Total Head (ft.)	10.2	BHP (Sp Gr 1.03)	1935
Capacity (USGPM)	642000	Motor HP	2419
RPM	73	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	12	Unit Efficiency (%)	78
Submergence (ft.)	4		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.) Dia.		Length	Est'd Wt. (lbs)
Suction Bell	E = 231	H =	38	28000
Impeller	A = 156	G =	63	22000
Impeller Case	A = 156	G =	63	25000
Diffuser	L = 189	J =	84	67000
Diffuser Extension	M = 217	K =	98	56000
Discharge Column	D = 231		133	32000
Shaft (@ Coupling)	11		189	8000
Sub - Total				~~238000~~
Motor	N = *74 x 65	P =	87	11900
Gear	R = 44		84	11000
Sub - Total				22900
Unit Total	B = 105	C =	378	
		F =	416	260900

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 18 mos.

*Rectangular cross-section

Revision 1
July 7, 1978

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #6

Operating Conditions

Total Head (ft.)	10.2	BHP (Sp Gr 1.03)	968
Capacity (USGPM)	321000	Motor HP	1210
RPM	103	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	12	Unit Efficiency (%)	78
Submergence (ft.)	4		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				
		Dia.		Length	Est'd Wt. (lbs)
Suction Bell	E =	163	H =	27	10900
Impeller	A =	110	G =	44	8600
Impeller Case	A =	110	G =	44	10000
Diffuser	L =	133	J =	59	26600
Diffuser Extension	M =	153	K =	69	22000
Discharge Column	D =	163		94	12600
Shaft (@ Coupling)		8		133	3100
Sub - Total					93800
Motor	N =	*30 x 41	P =	63	5100
Gear	R =	30		65	4620
Sub - Total					9720
Unit Total	B =	74	C =	266	
			F =	293	103520

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 nos. Manufacture 18 nos.

*Rectanuglar cross-section

Revision 1
July 7, 1978

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #7

Operating Conditions

Total Head (ft.)	10.2	BHP (Sp Gr 1.03)	484
Capacity (USGPM)	161000	Motor HP	605
RPM	146	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	12	Unit Efficiency (%)	78
Submergence (ft.)	4		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				Est'd Wt. (lbs)
		Dia.		Length	
Suction Bell	E =	116	H =	19	4300
Impeller	A =	78	G =	32	3400
Impeller Case	A =	78	G =	32	3900
Diffuser	L =	95	J =	42	10400
Diffuser Extension	M =	109	K =	49	8700
Discharge Column	D =	116		67	5000
Shaft (@ Coupling)		6		95	1300
Sub - Total					37000
Motor	N =	*30	P =	49	2800
Gear	R =	24		53	2460
Sub - Total					5260
Unit Total	B =	52	C =	190	
			F =	209	42260

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 18 mos.

*Does not include conduit box

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #8

Operating Conditions

Total Head (ft.)	6.4	BHP (Sp Gr 1.03)	1358
Capacity (USGPM)	718000	Motor HP	1698
RPM	49	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	8	Unit Efficiency (%)	78
Submergence (ft.)	3		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.) Dia.	Length	Est'd Wt. (lbs)
Suction Bell	E = 274	H = 46	44000
Impeller	A = 185	G = 75	35000
Impeller Case	A = 185	G = 75	41000
Diffuser	L = 224	J = 100	108000
Diffuser Extension	M = 258	K = 116	90000
Discharge Column	D = 274	156	51000
Shaft (@ Coupling)	11	224	10000
Sub - Total			388000
Motor	N = *37 x 53	P = 68	8500
Gear	R = 33	72	6160
Sub - Total			14660
Unit Total	B = 124	C = 448	
		F = 493	402660

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 21 mos.

* Rectangular cross-section

Revision 1
July 7, 1978

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #9

Operating Conditions

Total Head (ft.)	6.4	BHP (Sp Gr 1.03)	679
Capacity (USGPM)	359000	Motor HP	849
RPM	69	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	8	Unit Efficiency (%)	78
Submergence (ft.)	3		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				Est'd Wt. (lbs)
		Dia.		Length	
Suction Bell	E =	194	H =	32	17000
Impeller	A =	131	G =	53	14000
Impeller Case	A =	131	G =	53	16000
Diffuser	L =	159	J =	71	42000
Diffuser Extension	M =	182	K =	82	35000
Discharge Column	D =	194		111	20000
Shaft (@ Coupling)		8		159	4000
Sub - Total					148000
Motor	N =	*30 x 91	P =	56	4400
Gear	R =	24		53	2460
Sub - Total					6860
Unit Total	B =	88	C =	317	
			F =	349	154860

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 18 mos.

*Rectangular cross-section

WORTHINGTON PUMP CORP. (USA)

OTEC Sea Water Pump Design Information

Identification LMSC #10

Operating Conditions

Total Head (ft.)	6.4	BHP (Sp Gr 1.03)	340
Capacity (USGPM)	180000	Motor HP	425
RPM	97	Motor Efficiency (%)	96
Efficiency (%)	88	Gear Efficiency (%)	93
NPSHR (ft.)	8	Unit Efficiency (%)	78
Submergence (ft.)	3		

Approximate Physical Characteristics (See Fig. 1)

	Dimensions (in.)				
		Dia.		Length	Est'd Wt. (lbs)
Suction Bell	E =	137	H =	23	6800
Impeller	A =	93	G =	37	5400
Impeller Case	A =	93	G =	37	6300
Diffuser	L =	112	J =	50	16600
Diffuser Extension	M =	129	K =	58	13800
Discharge Column	D =	137		79	7900
Shaft (@ Coupling)		6		112	1500
Sub - Total					58300
Motor	N = *	30	P =	44	2500
Gear	R =	21		44	1320
Sub - Total					3820
Unit Total	B =	62	C =	224	
			F =	247	62120

Approximate Lead Times (Pump only, excluding motor and gear)

Design 4 mos. Manufacture 18 mos.

*Does not include conduit box

A 635 1/2 - D

WORTHINGTON

Date July 7, 1978

TO T. J. Anderson — H - 13
Name — Location

FROM A. Agostinelli — H - 32
Name — Location

COPIES TO

SUBJECT

Material Requirements
API-610

Division Quotation No.
S. O. Quotation No.
S. O. Order No.
Works Order No.
Invoice No.
If Quotation Requested, It Is To Be:
☐ FINAL ☐ ESTIMATING

REPLY REQUIRED BY

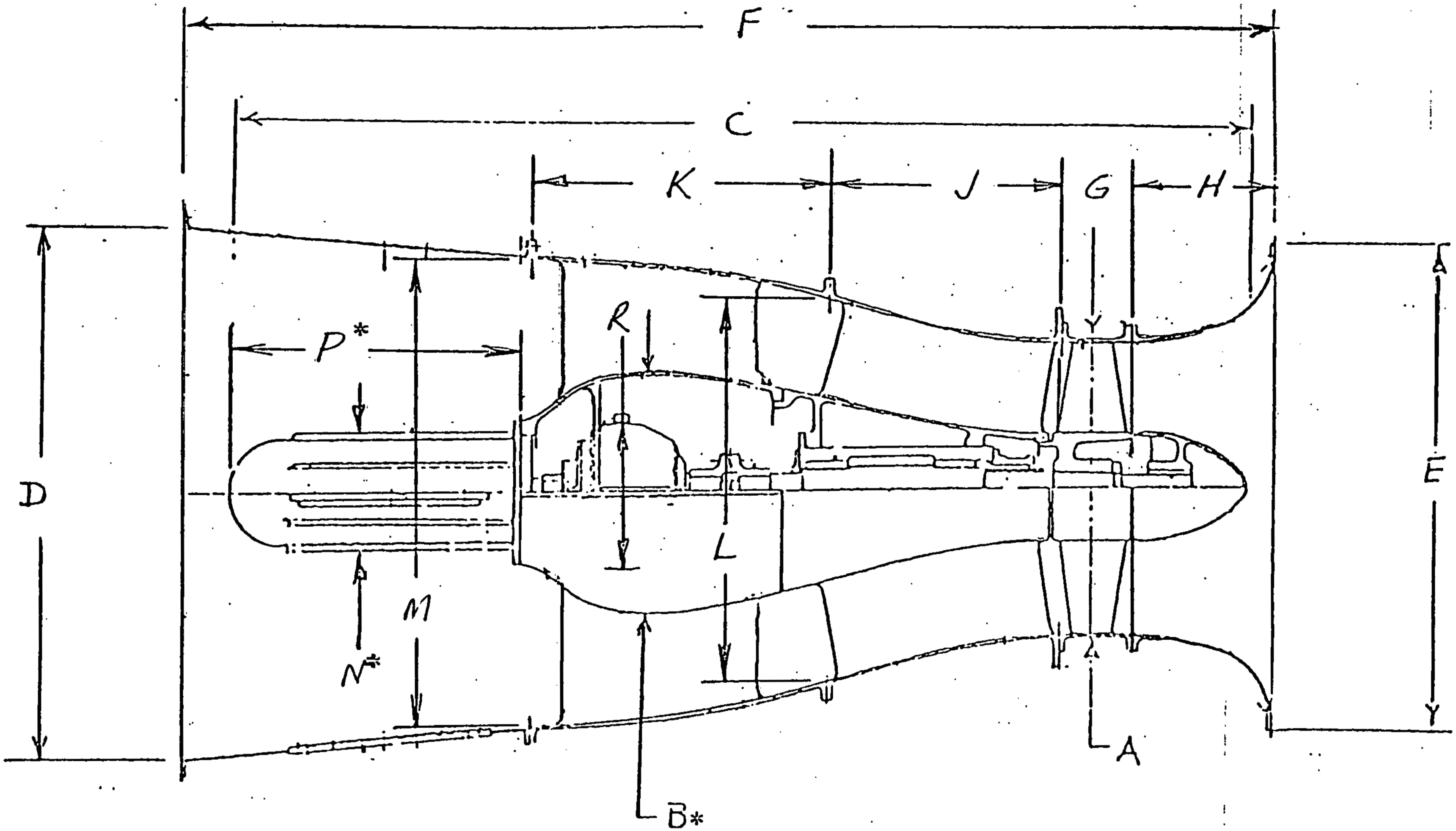
Tom,

We shall have to address Section III of API-610 for material requirements on the new HQ pump. This section is very stringent when you read the requirements and it behooves us to make sure that we are in full compliance. When you return from vacation give me a call and let's discuss this a little further.

A. Agostinelli

A. Agostinelli

AA:grs



TYPICAL SEA WATER CIRCULATING PUMP (FIG. 1)

*Adjustment in these dimensions required to accomodate rectangular cross-section open drip proof motors.

APPENDIX F3.2

American M.A.N. Corporation Data

M·A·N

AMERICAN M.A.N. CORPORATION

WEST COAST OFFICE 50 CALIFORNIA STREET SAN FRANCISCO, CA 94111 TELEPHONE (415) 391-2935

October 17, 1978

Lockheed MSL Space Company
Post Office Box 504
Sunnyvale, CA 94086

OTEC Studies
LSMC-D62358

Gentlemen:

With our letter of September 8, we submitted to you our proposal for our pump A 30 for a flow rate of 408,500 GPM. During our meeting on September 11, you requested that we submit also budget proposals for our models A 37.5, A 42 and A 53 for fixed-blade design, with variable speed control and, as an alternative, impeller pitch control. Today we would like to submit these alternatives as follows:

Pump Model	A 42	A 30	A 53	A 37.5
Flow rate (GPM)	817,000	408,500	1,300,000	650,000
Unit weight (kg)	131,300	56,000	247,600	97,000
Unit price (DM) non-controllable pitch	2,650,000	1,550,000	4,200,000	2,200,000
add cost for variable speed control by means of fluid coupling	360,000	240,000	455,000	310,000
alternatively add cost for impeller pitch control	350,000	250,000	500,000	300,000
add cost for packing and delivery FOB German port of export	130,000	40,000	220,000	102,500
add cost for delivery CIF East Coast USA	110,600	35,000	205.000	82,000

We also take the opportunity to include the original technical description by M.A.N., TP 975 985, which supersedes the one attached to our letter of September 8, 1978.

./.

cc: Mr. D. Meier-Althaus
TP, Gustavsburg

HEAD OFFICE 1114 AVENUE OF THE AMERICAS NEW YORK, NY 10036 TELEPHONE (212) 221-3340

As briefly discussed during our recent telephone conversation with you, M.A.N. has the shop facilities to manufacture these pumps, however, in order to justify setup costs an order should consist of a minimum of four pumps of one model.

Above budget prices are based on cost factors as of October 1, 1978 and are subject to escalation.

We trust that today's additional information will be useful for your finalizing a proposal to the Department of Energy. We do hope that you will be successful and that we can continue to work together.

Very truly yours,

AMERICAN M.A.N. CORPORATION

Hans-Peter Feddersen
V.P. - West Coast Office

HPF:ct
Enclosure

M·A·N TP 975 985

Gustavsburg, 4.9.1978
TP-bru/gu/bi/lor

Technical Description

M.A.N. Pump Set
Single-stage, vertical or horizontal axial-flow propeller pump of the closed tubular casing type;
impeller blades firmly set in the impeller body;
driven by a submersible electric motor with reduction gear.

Mounting and overall dimensions for pump type A 30 as per enclosed drawing C 76.50 000-4762.

Pump characteristics:

Specific gravity: 1020 kg/m^3

	25 MW (e)		50 MW (e)		
Rate of flow:	186 000	93 000	296 000	148 000	m^3/h
	817 000	408 500	1 300 000	650 000	gpm
Pump set, type:	A 42	A 30	A 53	A 37.5	
Impeller dia.:	4200	3000	5300	3750	mm
Pipe outlet dia.:	4900	3500	6200	4400	mm
Discharge head of pump as measured at centre of pipe bend, related to inlet water level:	4.60	4.60	4.60	4.60	mWC
Discharge head of pump stage:	4.80	4.80	4.80	4.80	mWC
Efficiency of pump stage:	88.00	88.00	88.00	88.00	%

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G 0002

Efficiency of reduction gear:	98.00	98.00	98.00	98.00	%
Efficiency of pump set:	86.24	86.24	86.24	86.24	%
Power input of pump set as measured at the motor coupling:	2876	1438	4576	2288	kW
Pump speed:	91	127	72	101.5	r.p.m.
Motor speed:	1200	1200	1200	1200	r.p.m.
Minimum motor output:	3170	1590	5040	2520	kW

Other pump characteristics as per enclosed diagrams M 2024, M 2025, M 2026, M 2027.

Torque curve as per diagram D-Z-32/A.

M a t e r i a l s

	Material Code	DIN	Material No.
Intake section	Ni-Resist, type 1		
Wearing ring	Ni-Resist, type D2		
Diffuser	R-St 37.2	17 100	1.0114
Jacket tube	R-St 37.2	17 100	1.0114
Impeller body	Ni-Resist D2		
Impeller blades	G-X 8 CrNi 26 6		
Shaft	C 45 V		
Shaft protecting sleeves	St 50.2)	17 006	1.0663
Bearing bushes	Steel with ergon		
Bolts in contact with the water, up to M 10	X-10 CrNiTi 18 9	17 006	1.4541

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	Material Code	DIN	Material No.
Bolts in contact with the water, larger than M 10	X-22 CrNi 17	17 006	1.4057
Nuts in contact with the water, up to M 10	X 10 CrNiTi 18 9	17 006	1.4541
Nuts in contact with the water, larger than M 10	X-12 CrMoS 17	17 006	1.4104
Washers in contact with the water	X-12 CrNi 17 7	17 006	1.4310

The pump set comprises:

1 single-stage axial-flow propeller pump, with wearing ring in impeller blade area,

bolted to:

1 jacket tube

1 intake section flange-mounted on the pump,

1 impeller shaft flange-mounted in the reduction gear,

2 oil-lubricated plain thrust bearings,

1 planetary reduction gear with built-on oil circulating pump,

1 submersible drive motor,

1 toothed coupling to connect the gear and motor shafts,

2 mechancial seals.

All parts factory-assembled prior to despatch, including bolts, seals, fittings and hardware, but excluding oil and grease fillings.

A number of individual pump components are described below in more detail.

Non-adjustable axial-flow impeller

The blades are inserted in the impeller body and fixed in position. Subsequent variation of the blade angle and the resulting change in the delivery characteristics (rate of flow, discharge head) is possible if due consideration is given to the motor output and pump shaft diameter. To ensure smooth running, the impeller will be dynamically balanced.

Plain thrust bearing

This bearing takes the hydraulic axial thrust of the pump and the load of the rotating parts and provides additional radial guidance for the pump shaft.

To provide the facility of carrying the pumped liquid in both directions (at different flow rates though), we have included two plain thrust bearings acting in opposite directions.

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Planetary reduction gear.
A single-stage planetary gear with double toothed coupling on the high-speed side, double helical gearing, pinion and planet wheels in case-hardened and ground steel; outer centre wheel made of heat-treated special steel; casing of welded construction.

Submersible motor.
Plain bearing type with forced feed lubrication.
Enclosure IP 55 for sea water immersion, including standstill indicator and leakage water indicator.

Zinc dust coating system

All parts will be coated as follows:

The paint system,which is based on two-pack epoxy resin, is characterised by extremely high resistance to abrasion and chemical aggression. This system is the result of extensive series of tests and has been successfully used on a great number of cooling water pumps, low-lift pumps and sewage pumps.

The paint system will be applied as follows:

Metal surface

Steel or cast iron, sand-blasted to near white metal, rust removal grade 3 to DIN 18364.

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<u>Prime coat</u>

Two coats of EPOXY zinc dust paint, the first being applied immediately after sand-blasting.

<u>Top coat</u>

One coat of EPOXY varnish paint with chrome nickel steel pigments, colour: dark grey metallic.

MASCHINENFABRIK AUGSBURG-NÜRNBERG
Aktiengesellschaft

AMERICAN M.A.N. CORPORATION
WEST COAST OFFICE 50 CALIFORNIA STREET SAN FRANCISCO, CA 94111 TELEPHONE (415) 391-2935

October 27, 1978

Mr. Charles M. Robidart
Energy Systems/Ocean Systems
Research & Development Division
Lockheed Missiles & Space Company, Inc.
Post Office Box 504
Sunnyvale, CA 94086

OTEC Studies
M.A.N. Pumps

Dear Mr. Robidart:

We would like to confirm our telephone conversation of October 23, 1978 during which we informed you that

M.A.N.'s proposal includes all accessories, etc. for the hydraulic couplings quoted as an alternative in our letter of October 17, 1978;

The design criteria for these pumps are based on a life expectancy of at least 160,000 operating hours in terms of material selection, stresses, etc.;

The mean time between overhauls can vary between 20 and 50,000 operating hours, depending on the component under consideration and the expensiveness of its design. In other words, some components can be designed such that the time between overhauls can be extended, however, at the expense of making this component.

We trust that this information is of value for your current study. If you have any further questions, please feel free to call us.

Very truly yours,

AMERICAN M.A.N. CORPORATION

Hans-Peter Feddersen HPF:ct
V.P. - West Coast Office

PS: We enclose the pictures you lent us during our meeting of October 17. F3.2-9

M.A.N. Maschinenfabrik Augsburg-Nürnberg Aktiengesellschaft
6095 Ginsheim-Gustavsburg 1 · Ginsheimer Straße 1 · Postfach

Lockheed
Missiles & Space Comp. Inc.

Sunnyvale
California 94088
U.S.A.

By and copy to:

M.A.N. Corp. New York

GINSHEIM-GUSTAVSBURG 1

Ihre Zeichen	Ihre Nachricht vom	Unsere Zeichen	Fernruf	Ginsheimer Straße 1
		TP-oe/ba/lor	(0 61 43) 55-313	October 23, 1978

Betreff:

Gentlemen:

Power System Development
PSD-2.1, Rev. 3.9.78

We thank you for the above tender specifications, which were forwarded to us through our U.S. subsidiary, American M.A.N. Corporation, New York. As we gather from the specifications you are studying the feasibility of power plants requiring large-size circulating water pumps. These would be installed either horizontally or vertically. You also want to find out as to whether it is advisable to provide speed or impeller adjustment for these pumps, and therefore ask for submission of the corresponding documentation.

You will no doubt have been informed by our gentlemen of M.A.N.-Corporation that M.A.N. above all deal with large-size pumps and of these with Kaplan pumps, which can be adapted to changing operating conditions by means of modification of the impeller pitch. We

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have for instance installed such a pump for a water circulating tank for ship model measuring purposes conducted in Berlin,where the rate of flow can be controlled with such a control system within a range of from 0 to 60 cbm/sec. with a pump dia. of 3.50 m. We therefore think we can give you some useful proposals for the projects concerned.

Please understand that we are not in a position to process all required alternatives at the same time. The sizes concerned in the present case require completely new designs to be elaborated and corresponding negotiations to be held with the supplies concerned. We are, however, in a position today to offer you, as a main version, a pump without any adjustment control, both for vertical and horizontal installations, as well as a second version with speed control also for both vertical and horizontal installation. For these 4 cases we enclose some sketches. We shall furthermore quote you a budget price for the third version with blade pitch adjustment, but we are not yet in a position to send you herewith a project sketch with corresponding description. We shall, however, try our best to catch up on that within a short time.

You are requesting the above studies to be conducted again for various areas of rate of flow, i.e. for pumpsof various sizes. Please understand that we shall conduct the studies for one version only, and it is for reasons of practicality only that we have chosen the versions having the lowest rate of flow, but we see no difficulty in transferring the resulting design findings to the other versions with higher rates of flow. We are therefore quoting you, in the same letter, the budget prices for the latter versions.

./.

Versions 1a and b, non-adjustable pump

We would like to give you hereunder some explanations on the individual versions. Since the structure of the pump is basically the same for horizontal and vertical installation, we would like to describe the common features first and then talk about the different aspects of vertical and horizontal pumps.

The impeller, which is an axial wheel of suitable diameter, is mounted onto the sunwheel of a heavy-duty planetary gear direct. The distance of the driven side bearing has been chosen such that it is arranged at the centre of gravity of the impeller. The unit is driven by the electric motor via a gear coupling. A Michell bearing acting on both sides is mounted on either side of the sunwheel in the gearbox, especially in the case of vertical installation, to carry the hydraulic axle thrust and the weight of the impeller. The gearbox itself including the bearing is flanged into the pump diffuser hub. The drive motor is mounted to the diffuser hub of the axial pump direct and additionally to the discharge pipe supported by several ribs. For further details on the impeller, the sliding thrust bearing, the reduction planetary gear and the submerged motor please refer to our letter of 4th September, 1978.

For further design details see also our project sketch. According to your proposal, we have based our design on the feature that all mechanical components will be under air pressure, which will be above the under-water pressure to ensure that no seawater will enter due to minor leakage.

./.

We intend to provide the air at the lowest point to the mechanical elements so that all possible leakages will be covered by compressed air. This will, of course, require different arrangements for vertical and horizontal arrangements, including the impeller shaft seal.

Vertical installation

Since it may be assumed that it will be possible to fabricate the impeller body as an air-tight unit, and since furthermore minor air leakages can easily be compensated, there will certainly be no penetration of seawater at the areas where the shaft enters the gearbox and the electric motor, and therefore no sliding bearings will have to be provided there. Standard sleeve packings will be sufficient to ensure that water will almost completely be prevented from entering during erection. A pneumatic seal for standstill periods will be provided at these points to serve the same purpose.

We intend to feed the air to the motor itself through a stiffening rib of the electric motor and to supply it down to the lowermost point of same.

Horizontal installation

The impeller shaft will be sealed off against seawater by means of a double-acting rotating mechanical seal which will be supplied with sealing oil. The sealing air for lube oil and also the sealing oil of the mechanical seal will be fed to the electric motor to the lowermost point of a support rib in order to obtain a higher overal pressure than that of the seawater. As far as we gather from your

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information it will be possible to reach a depth at the level of the installed circulating water pumps through a hoistway. We therefore intend to arrange the necessary oil filter for the oil circuit, which will supply not only the gearbox and the rotating mechanical seal but also the two sliding bearings of the electric motor, outside the unit for maintenance purposes. We also intend to provide a settling tank for the oil in the gearbox space,, possibly inside a guide rib of the diffuser, so that impurities and condensation water are allowed to accumulate there to be removed at certain intervals by means of a hand-operated pump. This opening may also be used to draw off the complete oil for oil changes without having to disassemble the unit. The necessary sealing air and also the power supply line would pass through this hoistway to the units.

As we cannot obtain any information from the tender documents as to the way of installation of these units in such a hoistway, we have so far not worried about it for the time being, and the suspension facilities of the pumps have therefore not been taken into account when calculating the prices. We would ask you to bear this aspect in mind. For further details please refer to sectional drawings C 76.5000-4762 (vertical installation) and C 76.5000-4799 (horizontal installation).

<u>Versions 2a and b, speed controlled pump</u>

The structure of the unit is basically the same except that a Voith turbo coupling is provided between the gearbox and the electric motor, in order to reduce the desired nominal

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speed control by up to 30 %. You will know that this change of speed will be brought about by varying the oil volume of the hydraulic coupling. This change is by a pipe submerged more or less deep in the circulating oil and thus pumping oil into a reserve tank. Operation of this pipe is by an electric motor via a special gearbox and a crank mechanism.

This drive unit is arranged above the diffuser, and the pipe is led through a diffuser rib which as is well known is two-walled. This crank mechanism is again encapsulated outside the pump and supplied with compressed air. The servomotor requires 220 V power supplied outside the pump. An induction-type revolution counter is installed in the coupling itself. The measuring transducer required is to be arranged outside the pump, probably in the same room as the oil filter and oil cooler. The measuring information of the transmitter is fed into the pipe drive room via a separate tube to be integrated in the diffuser rib, and from there to the measuring transducer, which is to be supplied with 220 V.

It is intended to take the operating oil required for the coupling from the lube oil provided for the gearbox and motor in order to avoid the installation of a separate supply system. This will of course increase the volume of oil circulation that can no longer be handled by the impeller hub, which will therefore require provision of an extra tank. This shall be arranged either near the lube oil filter in the dry room, with a separate electrically

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driven oil circulating pump provided for the increased rate of flow, or below the electric motor thus creating the possibility of driving the oil pump off an additional shaft end. In our sketch we have roughly indicated the area where the tank is to be arranged in the pump in the dry room. Due to the power losses in the hydraulic coupling the oil must be re-cooled in this case. This is done by an oil cooler arranged in the dry room and supplied by others with seawater.

As regards the horizontal and vertical installation types there are minor differences as to the coupling section, which are shown on drawings No. C 76.5000-4801 (vertical) and No. C 76.5000-4790 (horizontal).

Hoping to have been of service to you, we remain,

Yours truly,
MASCHINENFABRIK AUGSBURG-NÜRNBERG
Aktiengesellschaft

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This Final Report includes the following:

1. Attachment 1, Conceptual Design Report LMSC-D566744
2. Attachment 2, consisting of
   - o Volume I — Preliminary Design Report, LMSC-D630248
   - o Volume II — Appendixes A, B, and C
   - o Volume III — Appendixes D, E, and F
   - o Volume IV — Appendix G
   - o Volume V — Appendixes H, I, J, and K