428
121/80
1

Lh. 628

SAN-1570-1-T1

MASTER

**OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM
DEVELOPMENT UTILIZING ADVANCED, HIGH-PERFORMANCE
HEAT TRANSFER TECHNIQUES**

Volume 1: Conceptual Design Report

May 12, 1978

Work Performed Under Contract No. EG-77-C-03-1570

**TRW Systems and Energy
Redondo Beach, California**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

U.S. Department of Energy

Solar Energy

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

This report has been reproduced directly from the best available copy.

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161.

Price: Paper Copy \$10.75
Microfiche \$3.00

**OCEAN THERMAL ENERGY CONVERSION (OTEC)
POWER SYSTEM DEVELOPMENT
UTILIZING ADVANCED, HIGH-PERFORMANCE
HEAT TRANSFER TECHNIQUES**

VOLUME I: CONCEPTUAL DESIGN REPORT

12 MAY 1978

CONTRACT NO. EG-77-C-03-1570

PREPARED BY

TRW
SYSTEMS AND ENERGY

**ONE SPACE PARK
REDONDO BEACH, CALIFORNIA 90278**

**PREPARED FOR
DEPARTMENT OF ENERGY
SAN FRANCISCO OPERATIONS OFFICE
133 BROADWAY
OAKLAND, CALIFORNIA 94612**

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

CONTENTS

	<u>Page</u>
1. INTRODUCTION	1-1
1.1 Summary	1-2
2. POWER SYSTEM CONCEPTUAL DESIGN	2-1
2.1 Conceptual Design Approach	2-1
2.1.1 Design Requirements and Guidelines	2-1
2.1.2 Modularity Options	2-9
2.1.3 Optimization Methodology	2-20
2.2 Baseline Power System Selection	2-22
2.3 Baseline Power System Description	2-27
2.3.1 Flow Diagrams	2-39
2.3.2 General Arrangement Drawings	2-47
2.3.3 System Performance Specifications and Characteristics	2-59
2.3.4 Equipment Itemization (size, weight, and envelope of major components and subsystems)	2-64
2.3.5 Critical Interfaces	2-70
3. TRADE STUDIES AND ANALYSES	3-1
3.1 Heat Exchanger	3-1
3.1.1 Design/Productibility Summary	3-1
3.1.2 Materials and Processes	3-8
3.1.3 Biofouling Control	3-10
3.2 Machinery and Subsystems	3-13
3.2.1 Turbine-Generator	3-13
3.2.2 Seawater Pumps and Drives	3-30
3.2.3 Ammonia Pump Requirements	3-44
3.2.4 Ammonia Systems	3-47
3.3 Instrumentation and Control	3-60
3.3.1 Power System	3-60
3.3.2 Water Supply and Distribution System Dynamics	3-68
3.3.3 Power Module Startup	3-77
4. OPERABILITY/MAINTAINABILITY	4-1
4.1 Availability	4-1
4.1.1 Requirements	4-1
4.1.2 Operational Considerations	4-2

CONTENTS (Continued)

	<u>Page</u>
4.1.3 Availability Prediction	4-3
4.1.4 System Starting Reliability Prediction	4-17
4.2 Safety Issues	4-24
4.3 Leak Detection and Tube Isolation	4-26
4.3.1 Ammonia Leakage Detection	4-26
4.3.2 Identification of Leaking Tubes	4-27
4.3.3 Tube Sealing	4-27
4.3.4 Tube Maintenance Time Line	4-29
5. COST ANALYSES AND PROJECTIONS	5-1
5.1 Base of Cost Estimates	5-1
5.2 Cost Comparison of Options	5-3
5.2.1 Modularity Options	5-3
5.2.2 Tube Material Options	5-3
5.2.3 Comparison of Titanium and Aluminum Heat Exchanger Costs on a Present-Worth Basis	5-11
5.3 Cost Analyses and Algorithms	5-19
5.3.1 Heat Exchanger Cost Model	5-19
5.3.2 Seawater Pump Costs	5-22
5.4 Sensitivities of Design to Key Design Requirements	5-26
5.5 Non-Recurring Costs	5-34
5.6 Production Cost Projections	5-35
6. RECOMMENDATIONS	6-1
6.1 Proof-of-Concept Scaled Pilot Plant	6-1
6.1.1 Test Objectives	6-1
6.1.2 Analysis of Scaled Testing Requirements	6-2
6.1.3 Rationale and Recommendations for Scaled System Selection	6-5
6.2 Proof-of-Concept Scaled Heat Exchanger	6-15
6.3 Technology Development Recommendations	6-17
6.3.1 Materials and Processes	6-17
6.3.2 Biofouling Control	6-19

LIST OF TABLES

	<u>Page</u>
1-1 Comparative Option Cost Estimates (Titanium Tubes)	1-4
1-2 Compatibility of Heat Exchanger Design and Alternate Hull Concepts	1-12
A Baseline Power Module (12.5 MWe Nominal)	1-9
B Rotating Machinery (per Baseline Module of 12.5 MWe (net))	1-10
C Turbine Flow Rates and Conditions at Design Point Operation	1-11
D Evaporator Design Data	1-12
E Condenser Design Data	1-13
F Evaporator Thermal Data (6.25 MWe Nominal)	1-14
G Condenser Thermal Data (6.25 MWe Nominal)	1-15
H Power Plant Cost Breakdown	1-16
I-1 Evaporator Cost Breakdown	1-17
I-2 Condenser Cost Breakdown	1-18
2-1 Baseline Power Module (12.5 MWe Nominal)	2-29
2-2 Rotating Machinery (per Baseline Module of 12.5 MWe (net))	2-30
2-3 Turbine Flow Rates and Conditions at Design Point Operation	2-31
2-4 Evaporator Design Data	2-32
2-5 Condenser Design Data	2-33
2-6 Evaporator Thermal Data (6.25 MWe Nominal)	2-34
2-7 Condenser Thermal Data (6.25 MWe Nominal)	2-35
2-8 Power Plant Cost Breakdown	2-36
2-9 Evaporator Cost Breakdown	2-37
2-10 Condenser Cost Breakdown	2-38
2-11 Equipment List 12.5 MW _{Net} Power Module	2-65

LIST OF TABLES (Continued)

	<u>Page</u>
3-1 Recommended Design Materials	3-9
3-2 Turbine-Generator Requirements	3-15
3-3 A Typical 6.25 MWe Turbine's Parameters	3-16
3-4 Hot Gas Expanders	3-17
3-5 Centrifugal Ammonia Compressors	3-18
3-6 Seawater Flow Control	3-39
3-7 Seawater Pump Drives Requirements	3-41
3-8 Seawater Pump Drive Comparison	4-44
4-1 Mean-Time-Between-Failure (MTBF) and Mean-Time-To-Repair (MTTR)	4-9
4-2 Heat Exchanger Tube Failures	4-10
4-3 Repair Action Times (Hours): Titanium Heat Exchanger Tubes	4-12
4-4 Repair Action Times (Hours): Aluminum Heat Exchanger Tubes	4-13
4-5 Repair Action Times (Hours) of Heat Exchanger Connections (Nozzles)	4-14
4-6 Logic Structure	4-15
4-7 Availability Estimates - All Options	4-18
4-8 System Starting Failure Rate	4-23
5-1 Comparative Option Cost Estimates (Titanium Tubes)	5-7
5-2 Cost Breakdown (Aluminum Option)	5-8
5-3 Evaporator Cost Breakdown (Aluminum Option)	5-9
5-4 Condenser Cost Breakdown (Aluminum Option)	5-10
5-5 Aluminum and Titanium Heat Exchanger Characteristics	5-11
5-6 Total Evaporator and Condenser Present Value Costs for 30-Year Life	5-15
5-7 Cost Equations	5-21

LIST OF FIGURES

	<u>Page</u>
1-1 Power Station Plan: 12.5 MWe Power Module	1-6
1-2 Elevation: 12.5 MWe Power Module	1-7
1-3 Surface Interacting Hulls	1-9
1-4 Surface Decoupled Hulls	1-10
2-1 Ship Type Platform	2-5
2-2 Hull Motion Limits, Velocity vs. Frequency	2-7
2-3 Unit Cost Trends of Heat Exchangers Versus Module/Shell Size	2-10
2-4 Logic For System Modularity Size Selection	2-13
2-5 Modularity Sizing Options	2-14
2-6 Sub-Option Modules — Turbine/Generator Arrangements	2-16
2-7 Series/Parallel Arrangement	2-17
2-8 Ammonia Loop Staging for Series Heat Exchanger Arrangement	2-18
2-9 Approach to Optimum Module Selection	2-21
2-10 Unit Capital Cost Variation With Module Size	2-23
2-11 Plant Availability and Delivered Energy Cost Versus Power Module Size	2-24
2-12 12.5 MWe Power Module Process Flow Diagram Selected System	2-41
2-13 Ammonia Storage & Recovery Subsystem	2-43
2-14 Liquid Nitrogen Storage	2-45
2-15 100 MWe Power Plant Surface Barge Platform General Arrangement	2-49
2-16 12.5 MWe Power Module General Arrangement Elevation	2-51
2-17 12.5 MWe Power Module General Arrangement Power Station Plan	2-53
2-18 12.5 MWe Power Module General Arrangement Pump Station	2-55

LIST OF FIGURES (Continued)

	<u>Page</u>
2-19 12.5 MWe Power Module General Arrangement Steel Framing and Elevation	2-57
2-20 Head Losses Due to Entrances, Exits, and Screens	2-63
2-21 100 MWe Power Plant Critical Interfaces	2-71
3-1 Evaporator	3-2
3-2 Condenser	3-3
3-3 Titanium Tubed Heat Exchanger Tube/Tube Sheet Detail	3-7
3-4 Effects of Fouling on Heat Transfer	3-11
3-5 Tube Cleaning Machine	3-12
3-6 Turbine Performance	3-15
3-7 Turbine-Generator Cost	3-19
3-8 Variable Stator Actuating Mechanism	3-20
3-9 Multi-Stage Between the Bearings Turbine	3-21
3-10 Turbine Tilting Shoe Bearing & Turbine Iso-Carbon Shaft Seal	3-22
3-11 Estimated Effect of Velocity Ratio Change on Turbine Static Efficiency	3-24
3-12 Estimated Effect of Ammonia Flow Rate Change on Turbine Static Efficiency	3-24
3-13 Variable Area Nozzle Arrangements With Pirated Inlet Vanes for Radial Inward Flow Turbines	3-26
3-14 Axial Compressor With Variable Stator Vanes	3-27
3-15 Modularity Size Effect	3-28
3-16 Modularity Performance Curve	3-29
3-17 Typical Pump Specific Speed Versus Efficiency	3-32
3-18 Seawater Pump Configurations	3-34
3-19 Cold Seawater Pump Performance	3-36

LIST OF FIGURES (Continued)

	<u>Page</u>
3-20 Warm Seawater Pump Performance	3-37
3-21 Typical Pump Specific Speed vs. Impeller Shape and Efficiency	3-46
3-22 Ammonia Systems: Charging and Purifying	3-50
3-23 Ammonia Systems: Rectification	3-51
3-24 Ammonia Storage and Recovery Subsystem	3-53
3-25 Ammonia Systems: Purging	3-56
3-26 Liquid Nitrogen Storage	3-57
3-27 Ammonia Systems: Leakage Control	3-59
3-28 Power Module Schematic	3-62
3-29 Nominal Module One Line Electrical Diagram	3-67
3-30 Transient Response of Trough and Heat Exchanger Liquid Levels	3-73
3-31 Shallow Water Standing Waves	3-75
3-32 Power Module Startup	3-78
3-33 Auxiliary Power Source Size (10 and 15 percent Maximum Voltage Drop)	3-79
3-34 Power Module: Power Output Versus Waterflow Characteristic	3-81
4-1 Baseline Availability Model	4-8
4-2 Availability Model Logic Diagram	4-16
4-3 Availability - Two Evaporator - Two Condenser Configuration	4-19
4-4 Availability - One Evaporator - One Condenser Configuration	4-20

LIST OF FIGURES (Continued)

	<u>Page</u>
4-5 Availability - Two Evaporator - Two Condenser Configuration	4-21
4-6 Availability - One Evaporator - One Condenser Configuration	4-22
4-7 Heat Exchanger Leaking Tube Isolation	4-28
5-1 100 MWe Power Plants General Arrangement Options	5-5
5-2 Cost of Warm/Cold Water Pumps Versus Flow Rate	5-23
5-3 Turbine Cost Versus Gross Generator Power Output	5-24
5-4 Generator Cost Versus Gross Power Output	5-25
5-5 Cost of Heat Exchangers Versus Tube Size for Range of Module Sizes	5-27
5-6 Cost of Heat Exchangers Versus Tube Size for Range of Module Sizes	5-28
5-7 Effect of Tube Diameter on Direct Cost	5-30
5-8 Direct Cost Versus Fouling Factor for Smooth and Enhanced Tubes	5-31
5-9 Direct Cost Variation with Design ΔT	5-32
5-10 Plant Output Variation with Available ΔT	5-33
6-1 6.25 MWe Power Module Process Flow Diagram Scaled Pilot Plant	6-7
6-2 6.75 MWe Power Module General Arrangement Elevation	6-9
6-3 6.25 MWe Power Module General Arrangement Power and Pump Stations	6-11
6-4 6.25 MWe Power Module General Arrangement Steel Framing and Elevation	6-13

FOREWORD

The TRW, Inc., conceptual design for the 100 MWe Ocean Thermal Energy Conversion (OTEC) Demonstration Plant includes:

- The full-sized, closed-cycle ammonia power system module that utilizes the high-performance heat transfer characteristics of enhanced, vertical falling film heat exchangers;
- A proof-of-concept power system scaled to the design of the power system modules; and
- A set of test heat exchangers that are representative of the power system design.

This work, begun on 1 August 1977, was performed for the Department of Energy under Contract EG-77-C-03-1570.

This report presents the conceptual designs and engineering data in the form requested by the Government Technical Representative on 6 January 1978. Subcontractor and other supporting engineering documentation are provided in Appendices A through I.

Acknowledgements

The Conceptual Design Phase of the 1/5/25 MWe Power System Development Project was conducted by a team consisting of:

- TRW Defense and Space Systems Group, the prime contractor, responsible for power systems design, materials technology, system integration, and project management.
- CF Braun & Co., subcontractor to TRW, responsible for engineering support in the areas of general arrangements, mechanical design, cost estimates of heat exchangers, and the design of turbines and ammonia support subsystems.
- Carnegie-Mellon University, subcontractor to TRW, responsible for support in the areas of system dynamics and control, system optimization, thermal design of heat exchangers, and scaling of test articles.
- Union Carbide Corporation, Linde Division, subcontractor to TRW, responsible for aluminum corrosion protection, particularly the assessment of protective coatings including Marston Excelsior Ltd's., technology.
- Dr. R. McLaughlin (ENWATS), consultant to TRW, who provided design support in the areas of seawater supply system design, seawater pumps, and dynamics.

1. INTRODUCTION

The objective of this project is the development of a preliminary design for a full-sized, closed cycle, ammonia power system module for the 100 MWe OTEC Demonstration Plant. In turn, this Demonstration Plant is to demonstrate, by 1984, the operation and performance of an ocean thermal power plant having sufficiently advanced heat exchanger design to project economic viability for commercial utilization in the late 1980's and beyond. Included in this power system development are the preliminary designs for a proof-of-concept pilot plant and test article heat exchangers which are scaled in such a manner as to support a logically sequential, relatively low-cost development of the full scale power system module.

This report documents the work accomplished in the first step of this development, the Conceptual Design Phase. The conceptual designs are presented for the Demonstration Plant power module, the proof-of-concept pilot plant, and for a pair of test article heat exchangers. Costs associated with the design, development, fabrication, checkout, delivery, installation, and operation are included. The accompanying design and producibility studies on the full-scale power system module project the performance/economics for the commercial plant.

This section of the report briefly describes the full-size power system module, and summarizes the design parameters and associated costs for the Demonstration Plant module (prototype) and projects costs for commercial plants in production. The balance of the report expands this material, and discusses the pilot plant and test heat exchangers.

The material presented is directed primarily toward the surface platform/ship basic reference hull designated for use during conceptual design; however, other containment vessels were considered during the design effort so that the optimum power system would not be unduly influenced or restricted. TRW is prepared to modify the power plant design as the hull baseline is redirected by DOE.

1.1 SUMMARY

The conclusion of the Conceptual Design activity is that a 12.5 MWe (net) power module is the optimum cost-effective module size responsive to the design requirements. The power system module major components are:

- Two 6.25 MWe vertical tube evaporators
- Two 6.25 MWe vertical tube condensers
- Two 6.25 MWe turbines
- One 12.5 MWe electrical generator
- Four seawater pumps (one per heat exchanger)
- Three ammonia feed pumps (one standby redundant)
- Three ammonia recirculation pumps (one standby redundant)

The evaluation of module size, particularly with regard to costs due to heat exchangers' size, are summarized in Table 1-1. Cost savings accruing to design improvements, manufacturing methods, and the availability of a dedicated on-site facility will apply to heat exchangers of any size, will tend to maintain the cost per KWe constant out to larger sizes, but will, of course, involve large capital facilities investment which will certainly be justified for large scale production. It is questioned that such risk capital will be available for building the Demonstration Plant and, in fact, a design which requires this investment for the trial could be an impediment to its implementation.

Consequently, because there are no known compelling forces to go to larger modules, it is TRW's recommendation that the power module be approximately 12.5 MWe and that it utilize dual 6.25 MWe heat exchangers in the compact arrangement shown in Figures 1-1 and 1-2. There are no penalties seen in this approach and the advantage is the orderly generation of the building blocks for a logical progression from test articles, through pilot plant, and to the Demonstration Plant with a minimum of redesign and capital expenditure. Heat exchangers in the 6.25 to 12.5 MWe range can be manufactured, transported, and installed using existing facilities. Certain advantages of start-up auxiliary power, shut-down, load shedding, and system control result with the use of smaller modules. The major components of the full size power module (heat exchangers, turbines, and pumps) will

serve in both the pilot plant and in the Demonstration Plant. Test article heat exchangers can be conveniently taken as sections from the design of the full scale heat exchangers. This is appropriate to the vertical, falling film heat exchangers because, within limits of manufacturing tolerances and liquid/vapor distribution, each tube performs in a manner similar to all other tubes. The test heat exchangers will be used to verify the large scale performance design assumptions.

The capital cost of the power system module⁹ including seawater pumps and ammonia support subsystems was estimated as discussed in detail in Section 5 and summarized in Table H of this section. The estimate of the cost of the prototype is \$1451/KWe (net) and \$844/KW in full production (1977 dollars).

Significant features of the design are given below.

Heat Exchangers

The heat exchangers incorporate a vertical fluted tube, falling film, evaporator concept and a vertical fluted tube condenser concept. In the system the evaporator achieves an overall heat transfer coefficient (U) of 1200 B/hr-ft²-F and the condenser a U of 800 B/hr-ft²-F. The performance of these heat exchangers is based on experimental data taken at Carnegie-Mellon University and the Oak Ridge National Laboratory.

TRW has selected titanium over other materials for the tubes based upon its expected long life in a seawater environment. Life cycle cost analyses have indicated little advantage of any one material over others; we have therefore recommended the minimum risk approach. Fabrication of titanium into the desired flute configuration has been verified using an up-setting/rolling process. (Available data on aluminum, a primary alternative candidate material, do not support relatively thin-walled designs due to corrosion and erosion allowances. Therefore, the resulting thick wall tubes penalize both thermal/hydraulic performance and cost. Aluminum tubes can be easily incorporated in the design if technology can be developed to significantly extend the expected life of the aluminum in the seawater environment.)

The design includes a unique open top which simplifies the water box construction, reduces cost, and allows access to the waterside of the tubes

Table 1-1. Comparative Option Cost Estimates
(Titanium Tubes)

Table 5-1

Module Size (MWe)		25		12.5		6.25		3.125
Evaporators	Number	1	2	1	2	1	2	1
	Diameter (Ft)	35.2	24.9	24.9	17.6	17.6	12.4	12.4
Condensers	Number	1	2	1	2	1	2	1
	Diameter (Ft)	41.4	29.3	29.3	20.7	20.7	14.6	14.6
Power Plant Costs (\$/KW)								
Evaporator (Less Freight)		307	303	303	304	304	318	318
Condenser (Less Freight)		383	377	377	371	371	379	379
Turbine/Generator		104	124	136	142	160	160	172
NH ₃ Pumps		13	10	18	12	24	16	29
NH ₃ Piping/Valves		9	10	10	10	10	11	11
Cold Water Pumps		66	66	66	66	66	66	66
Warm Water Pumps		63	63	63	63	63	63	63
Misc. (Support, I&C, Elec)		93	93	98	101	103	113	118
Installation and Checkout		101	102	104	104	107	110	112
Subtotal (Direct)		1139	1148	1175	1173	1208	1236	1268
Indirect (Engr, Admin)		57	58	59	59	61	62	64
Power Plant Capital Cost (\$/KW)		1196	1206	1234	1232	1269	1298	1332
Experience Factor		0.7	0.7	0.7	0.7	0.7	0.7	0.7
Production Plant Cost		837	844	864	862	888	909	932
Availability		0.954	0.961	0.961	0.974	0.966	0.979	0.970
Relative Figure of Merit (Cost/Availability)		877	878	899	885	919	928	961

NOTE: Freight is included in exchanger costs in this table.
Freight is not included in exchanger costs in Table H.

for non-interruptive cleaning and repair. Incorporation of existing tube cleaning technology by designing a multiple spindle axial drive with brushes and using a low dosage rate of chlorination will result in economic control of fouling factors to approximately $0.0001^{\circ}\text{F-hr-ft}^2/\text{B}$ in the heat exchangers.

System Arrangement

The power module has been arranged in the conceptual design reference hull in an arrangement which minimizes containment volume while maintaining service accessibility. This design enables application of existing tools and procedures and precludes the necessity of developing undersea construction and maintenance operations.

As previously indicated, variations in containment vessel concept and design do influence the power plant design which will, in turn, most likely adversely affect the interface and the overall power system costs. Extremely close coordination between the power system and containment vessel design is essential for a successful design, and it is strongly felt that an optimized system design can only be achieved as an integrated design effort.

An open trough cold and warm water distribution system has been incorporated to eliminate water hammer effects inherent in closed systems, decouple water side hydraulics, minimize cavitation potential, and eliminate the requirement for expensive water side valves and piping. These open water troughs, coupled with the vertical orientation of the heat exchangers, result in a compact power plant configuration with shirtsleeve environment of all major power module components.

Turbine Control

By employing variable stator vanes for turbine control, TRW achieves a significant cost saving when compared to a throttling valve. The variable vanes increase the efficiency 2 to 3 percent when the turbine is operating at off-design-point conditions. These conditions occur when the seawater temperature difference is greater or less than 40°F . The efficiency improvement is achieved at comparatively small ammonia flow variations because the turbine has a less sharp efficiency drop off with variable

UPPER PLAN VIEW

Figure 1-1. Power Station Plan: 12.5 MWe Power Module

Figure 1-2. Elevation: 12.5 MWe Power Module

stator vanes than with fixed vanes. These vanes vary both the orifice size and jet angle to maximize the use of energy in the turbine. A throttling valve, on the other hand, creates a pressure drop with a loss of available system energy.

For an 8.25 (gross) turbine operating 3000 hours per year off the design point, the present value of the energy saved over 30 years is approximately \$185,000. This is compared to the approximately \$50,000 initial cost of the variable stator vanes.

Ammonia Recovery

A feature of the Ammonia Storage and Supply Subsystem is the use of the warm seawater in the rectification column reboiler. The rectification column is used to recover dry ammonia from the wet heel (mixture of water and ammonia) left after vaporizing the initial load of ammonia in the evaporators. (Commercial ammonia contains too much water for efficient heat exchanger operation.) As make-up ammonia is brought on board, it too must be dried. The warm seawater, instead of electrical power, vaporizes the ammonia. This saves 165 KW compared to a fractional horsepower motor required to pump the seawater.

Alternate Hulls

Several alternate hull concepts are presently being evaluated by DOE in separate studies. These concepts may conveniently be divided into two categories:

- A. Surface interacting
- B. Surface decoupled

as shown in Figures 1-3 and 1-4. The candidate hulls belonging to Category A include the ship or barge, the disk shaped barge, and the spherical or tuned sphere. The Category B hulls include the semi-submersible, the spar buoy, and the submersible.

The compatibility of TRW's heat exchanger design with these hull concepts can easily be evaluated considering key heat exchanger characteristics versus constraints dictated by the hulls concepts. Key heat exchanger characteristics include:

1. Shell-and-tube design with vertical tubes, water on the tube-side, ammonia on the shell side.

A SURFACE, INTERACTING

SHIP OR BARGE

DISK SHAPED BARGE

SPHERICAL OR TUNED SPHERE

Figure 1-3. Surface Interacting Hulls

B SURFACE DECOUPLED

Figure 1-4. Surface Decoupled Hulls

2. Falling film ammonia heat transfer mode, heat transfer enhancement (flutes) on both ammonia side and water side.
3. Open top design (open water box). Water flows through the tubes by maintaining a head of water (free surface) relative to the sea-water level. The upper tube sheet is at or near the sea-water level.
4. On-line tube cleaning by vertical, multiple spindle, rod brushes interfacing at the top of the heat exchangers.

These characteristics are compatible with all the surface interacting hull forms; the heat exchangers may be located internal to the hull as shown in Figures 1-1 and 1-2, or external, immersed in sea water. The surface decoupled hull forms are not compatible with the open top nor the rod brush concept characteristics (items 3 and 4 above). A closed top is required to allow submergence of the heat exchangers below sea level, whether internally or externally mounted. This would then require a modification of the cleaning concept. An Amertap, MAN or similar mechanical system could be adapted without changing the basic characteristics of the heat exchanger concept (items 1 and 2 above). An advantage of submergence of the heat exchangers is the possible partial pressure balancing between the inside and outside of the tubes as well as between inside and outside of the shell (if externally mounted).

In Table 1-2 is shown a summary of the discussion above.

Table 1-2. Compatibility of Heat Exchanger Design and Alternate Hull Concepts

Hull Concept	Heat Exchanger Characteristics			
	1. Vertical Shell and Tube Design	2. Falling Film Ammonia Heat Transfer Fluted Tubes	3. Open Water Box Free Water Surface Over Top Tube Sheet	4. On Line Cleaning With Vertical Rod Brushes
<p>A. <u>Surface Interacting:</u></p> <p>1. Ship or Barge</p> <p>2. Disk Shaped Barge</p> <p>3. Spherical or Tuned Sphere</p>	<p>Compatible, Internal or Immersed</p> <p>↓</p>	<p>Compatible, Internal or Immersed</p> <p>↓</p>	<p>Compatible, Internal or Immersed</p> <p>↓</p>	<p>Compatible, Internal or Immersed</p> <p>↓</p>
<p>B. <u>Surface Decoupled:</u></p> <p>4. Semi-Submersible</p> <p>5. Spar Buoy</p> <p>6. Submersible</p>	<p>↓</p>	<p>↓</p>	<p>Redesign, Closed Water Box, "Forced" Water Flow with no Free Surface. Internal or External Mounting Feasible.</p>	<p>Different Cleaning Concept. Several Candidates. Internal or External Mounting Feasible.</p>

Performance, design, and cost characteristics of the selected baseline 12.5 MWe power module are described in the following tables with references to applicable report sections for further descriptions:

<u>Table</u>	<u>Title</u>	<u>Report Section</u>
A	Baseline Power Module	2.3.1, 2.3.4
B	Rotating Machinery	3.2
C	Turbine and Pump Conditions at Design Point Operation	3.2
D	Evaporator Design Data	2.3.3, 3.1.1
E	Condenser Design Data	2.3.3, 3.1.1
F	Evaporator Thermal Data	2.3.3, 3.1.1
G	Condenser Thermal Data	2.3.3, 3.1.1
H	Power Plant Cost Breakdown	5
I-1	Evaporator Cost Breakdown	5
I-2	Condenser Cost Breakdown	5

Table A. Baseline Power Module (12.5 MW_e Nominal)

Gross power	16.51 MW _e
Net power	12.50 MW _e
Gross power/net power	1.32
Warm water pumping power	1.52 MW _e
Cold water pumping power	2.03 MW _e
Ammonia feed pumping power	0.34 MW _e
Ammonia reflux pumping power	0.02 MW _e
Chlorination power	0.05 MW _e
Mechanical cleaning power	0.01 MW _e
Controls, miscellaneous	0.04 MW _e

Table B. Rotating Machinery (per Baseline Module of 12.5 MW_e (net))

Item	Number	Type	RPM	Horsepower per Module	Efficiency (%)	Comments
Turbine and Diffuser	2	Axial flow multi-stage	1800	22,700	85	2 turbines drive one generator
Generator	1	AC 3Ø 4.16 kV	1800		97.8	
Warm Water Pump	2	Vertical propeller	129	2000	87	
Warm Water Pump Motor	2	Vertical gear box	900		92.5	
Cold Water Pump	2	Vertical propeller	139	3000	87	
Cold Water Pump Motor	2	Vertical gear box	900		92.5	
Ammonia Feed Pump	3	Centrifugal single stage end suction	1760	500	80	One pump is shared standby
Ammonia Feed Pump Motor	3	Explosion proof	1760		91	
Ammonia Reflux Pump	3	Centrifugal single stage end suction	1170	50	80	One pump is shared standby
Ammonia Reflux Pump Motor	3	Explosion proof	1170		91	

Table C. Turbine and Pump Conditions at Design Point Operation

	Turbine (6.25 MW _e Nominal)	
Flow Rate	1,708,900 lb/hr	
<u>Inlet</u>		
Pressure	128.8 psia	
Temperature	70°F	
Velocity	78.9 ft/sec	
Quality	> 99%	
<u>Outlet</u>		
Pressure	89.2 psia	
Temperature	50°F	
Velocity	40.1 ft/sec	
Quality	>96.5%	
	<u>Ammonia Pumps*</u>	
	<u>Feed Pump</u>	<u>Reflux Pump</u>
Flow rate (lb/hr)	1,708,900	281,300
Pressure rise (psid)	54.2	14.4
	<u>Water Pumps*</u>	
	<u>Warm Water</u>	<u>Cold Water</u>
Flow rate (gpm)	358,100	381,800
Pressure rise (ft)	8.2	11.2

*One for each nominal 6.25 MW_e heat exchanger.

Table D. Evaporator Design Data

	Evaporator (6.25 MW _e Nominal)
Number per module	2
Thermal rating	907 MBtu/hr
Water flow rate	350,100 gpm
Ammonia flow rate	1,708,900 lb/hr
Design pressure	160 psia
Operating pressure	128.8 psia
Overall length	70. ft
Water enhancement factor	1.5
Ammonia enhancement factor*	1.5
Total number of tubes	5,307
Tube OD	2 in.
Tube wall, corrugated	0.028 in.
Tube pitch	2.5 in.
Tube layout	Triangular
Tube sheet diameter	17.6 ft
Effective tube length	42 ft
Actual tube length	45.5 ft
Tube sheet thickness	4 in.
Number of tube supports	4
Tube support thickness	2 in.
Tube support spacing	8 ft
Shell OD	17.6 ft
Shell wall thickness	1-3/16 in.
Void fraction	0.54
Tube material	Titanium
Tube sheet material	SA516, GR70
Shell material	SA516, GR70
Baffle material	SA516; SA283; SA285, GRB
Coating materials	Coal tar epoxy (waterboxes)

*Area ratios, fluid mechanic effect not included.

Table E. Condenser Design Data

	Condenser (6.25 MW _e Nominal)
Number per module	2
Thermal rating	879 MBtu/hr
Water flow rate	381,800 gpm
Ammonia flow rate	1,708,900 lb/hr
Design pressure	115. psia
Operating pressure	89.2 psia
Overall length	76 ft
Water enhancement factor	1.5
Ammonia enhancement factor*	1.5
Total number of tubes	7,221
Tube OD	2 in.
Tube wall, corrugated	0.028 in.
Tube pitch	2.5 in.
Tube layout	Triangular
Tube sheet diameter	20.7 ft
Effective tube length	42 ft
Actual tube length	43 ft
Tube sheet thickness	4 in.
Number of tube supports	4
Tube support thickness	2 in.
Tube support spacing	8 ft
Shell OD	20.7 ft
Shell wall thickness	1-3/16 in.
Void fraction	0.53
Tube material	Titanium
Tube sheet material	SA516, GR70
Shell material	SA516, GR70
Baffle material	SA516; SA283; SA285, GRB
Coating materials	Coal tar epoxy (waterboxes)

*Area ratio, fluid mechanic effect not included.

Table F. Evaporator Thermal Data (6.25 MW_e Nominal)

Water Flow Rate	183,200,000 lb/hr
Water Flow per tube	9.59 lb/sec
Water Velocity inside tube	7.76 ft/sec
Reynolds Number inside tube	121,400
Prandtl Number inside tube	6.28
Inlet Water Temperature	80°F
Outlet Water Temperature	74.8°F
Average Water Temperature	77.4°F
LMTD	7.02°F
Average Tube wall surface Temperature	73°F
Average Shell wall surface Temperature	71°F
hw (including enhancement) referred to Dw*	1951 Btu/hr ft ² °F
h fouling referred to Dw	10,000 Btu/hr ft ² °F
h wall referred to D _o - 0.088 in.†	8700 Btu/hr ft ² °F
h ammonia referred to D†	8052 Btu/hr ft ² °F
U _o referred to Dw	1176 Btu/hr ft ² °F
Ammonia inlet temperature	52.9°F
Ammonia outlet temperature	70°F
Outlet Vapor Quality	>99%
Outlet Maximum Vapor Velocity	≤3.0 ft/sec
Total Pressure Drop of Water**	3.32 psid
Total Pressure Drop of Ammonia***	14.4 psid
<u>For Vertical Evaporators</u>	
Feed rate per tube	375.0 lb/hr
Feed film Reynolds Number	8490
Exit Film Reynolds Number	1200
Maximum Distribution Δp	<0.01 psid

*See Appendix B for definition of these diameters.

**Tubeside pressure drop including entrance and exit losses.

***Equal to recirculation pump pressure rise.

†Ammonia coefficient includes 65 mill aluminum wall conductance, therefore wall coefficient has been adjusted upwards to give correct composite coefficient.

Table G. Condenser Thermal Data (6.25 MW_e Nominal)

Water Flow Rate	196,000,000 lb/hr
Water Flow per tube	7.54 lb/sec
Water Velocity inside tube	6.08 ft/sec
Reynolds Number inside tube	57,700
Prandtl Number inside tube	10.97
Inlet Water Temperature	40°F
Outlet Water Temperature	44.7°F
Average Water Temperature	42.35°F
LMTD	7.36°F
Average Tube wall surface Temperature	
Average Shell Wall surface Temperature	
hw (including enhancement) referred to Dw*	1216 Btu/hr ft ² F
h fouling referred to Dw	15,000 Btu/hr ft ² F
h wall referred to D _o - 0.088 in.†	8700 Btu/hr ft ² F
h ammonia referred to D†	4000 Btu/hr ft ² F
Uo referred to Dw	798 Btu/hr ft ² F
Ammonia inlet temperature	50°F
Ammonia outlet temperature	49.9°F
Inlet vapor quality	>96.5%
Inlet maximum vapor velocity	≤2.5 ft/sec
Total Pressure Drop of Water**	2.32 psid
Total Pressure Drop of Ammonia	<0.01 psid
<u>For Vertical Condensers</u>	
Feed rate per tube	236.7 lb/hr
Feed film Reynolds Number	0
Exit Film Reynolds Number	4740
Maximum Distribution Δp	<0.01 psid

*See Appendix B for definition of these diameters.

**Tubeside pressure drop including entrance and exit losses.

†Ammonia coefficient includes 65 mill aluminum wall conductance, therefore wall coefficient has been adjusted upwards to give correct composit coefficient.

Table H. Power Plant Cost Breakdown

	Prototype(a) (\$/KW _e)	Production (\$/KW _e)
Evaporator	342 (d)	213 (e)
Condenser	422 (d)	260 (e)
Turbine	80	50
Generator	56	50
Water pumps, motors, etc.	129	110
Ammonia piping and valves	10	6
Ammonia feed pumps	8	7
Ammonia Recirculation Pumps	4	4
Control Systems	12	10
Electrical equipment	26	23
Cleaning Systems	13	8
Chlorination	7	6
Ammonia Support System	38	24
Installation	109	33
Total (Direct Cost)	1256	804
Indirect Costs (Engr, Admin)	195 (b)	40 (c)
Total	1451	844

(a) Basic 12.5 MWe power system module with 2 evaporators and 2 condensers.

(b) Engineering design and field supervision 13 percent, home office and field administration 2.5 percent.

(c) Field engineer and supervision 2.5 percent, administration 2.5 percent.

(d) Includes freight (Condenser 51\$/KW and Evaporator 38\$/KW).

(e) Shipyard dedicated heat exchanger fabrication facilities - no freight.

Table I-1. Evaporator Cost Breakdown

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	143.6	37.9	143.6	13.4
Tubesheets (no cladding)*	0	29.6	5.0	3.6
Tube Support Plates	0.6	2.9	0.6	1.0
Shell	7.7	12.6	7.7	9.6
Water Boxes	1.0	19.5	1.0	6.9
Nozzles	1.2	7.8	1.2	2.8
Ammonia Distribution	1.5	5.2	1.5	1.8
Bustle	1.1	19.3	1.1	6.9
Transition Inlet	1.1	8.1	1.1	2.9
External Support	0.3	3.2	0.3	1.1
Freight		37.7		
Subtotals	163.1	179	163.1	50
TOTAL \$/KW _e	342.1		213.1	

Evaporator Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 17.6 Feet
 Tube Material - - - - - Titanium (Corrugated)
 Tube Length - - - - - 45.5 Feet, overall
 Number of Tubes - - - - - 5,307
 Tube Diameter - - - - - 2 Inches, O.D.

*Tubes rolled, not welded

Table I-2. Condenser Cost Breakdown

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	189.0	52.7	189.0	15.7
Tubesheets (no cladding)*	7.6	19.4	7.6	5.8
Tube support plates	0.9	4.5	0.9	1.3
Shell	10.6	34.4	10.6	10.3
Water boxes	1.4	24.3	1.4	7.3
Nozzles	1.3	9.7	1.3	2.9
Transition inlet	1.4	10.0	1.4	3.0
External support	0.4	4.0	0.4	1.2
Freight	---	50.2	---	---
Subtotals	212.6	209.2	212.6	47.5
Total \$/KW _e	421.8		260.1	

Condenser Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 20.7 Feet

Tube Material - - - - - Titanium (Corrugated)

Tube Length - - - - - 43 Feet, overall

Number of Tubes - - - - - 7,221

Tube Diameter - - - - - 2 inches, O.D.

*Tubes rolled, not welded

2. POWER SYSTEM CONCEPTUAL DESIGN

2.1 CONCEPTUAL DESIGN APPROACH

2.1.1 Design Requirements and Guidelines

This section lists the requirements and basic constraints utilized during the development of a conceptual engineering design for an OTEC power system module as a unit of a 100 MWe power plant.

2.1.1.1 Applicable Documents

STANDARDS:

NBS Circular 142

Tables of the Thermodynamic Properties of Ammonia

NBS Circular 564

Tables of the Thermal Properties of Gases

OSHA Standard 1910.111

Storage and Handling of Anhydrous Ammonia

SPECIFICATIONS:

USCG Booklet 115

Marine Engineering Regulations and Materials Specifications

OTHER:

ANSI/ASME BPV-VIII-1
1 July 1977

ASME Boiler and Pressure Vessel Code, An American National Standard, Section VIII, Rules for Construction of Pressure Vessels, Division 1

RFP# EG-77-R-03-1400

Ocean Thermal Energy Conversion (OTEC) Power System Development

2.1.1.2 Design Life Goals

The power module shall be designed for the following:

- a. 30-year plant life
- b. 1000 start/stop cycles
- c. 6000 hrs/yr operation (minimum).

2.1.1.3 Startup/Shutdown

The auxiliary starting power and operation auxiliary power shall include sequential logic and procedures for phased test equipment and prototype module(s) startup/shutdown. The staged application of torque on pump motors shall be a consideration in the design.

2.1.1.4 Control and Instrumentation (C&I)

Design of the OTEC power system shall include transient characteristics and component response.

2.1.1.5 Electric Power System Utilization

The power module shall initially be conditioned for use as utility power generation. The following power usage criteria shall be addressed:

- a. Part load and turndown capability
- b. Load shedding capability
- c. Startup load carrying capability
- d. Electrical energy conditioning (transformation, rectification).

The main requirements guiding the design of the electric power system are: operating flexibility, growth capability, high reliability, ease of maintenance equipment replacement, and low cost.

Because the proposed plant is an experimental installation; operating flexibility is important. Such flexibility requires the capability to control and operate power sources separately; turn loads on and off independently; increase and reduce the quantity and the magnitude of these loads; operate the system in a variety of modes and conditions; and monitor the electrical conditions of the system.

Growth capability refers primarily to the requirement that the power module can be coupled to other modules to form a 100 MWe plant.

High reliability of operation requires capability for fault sensing and isolation, redundancy, and high reliability components.

Ease of maintenance in servicing and equipment replacement is required to enable the plant to perform its test bed and utility power plant function.

Low cost must be achieved to make this new source of power more competitive with conventional plants.

2.1.1.6 Biofouling

Biofouling of the heat exchanger shall be controlled such that a fouling factor not to exceed $0.0005 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$ is maintained.

2.1.1.7 Physical Characteristics

Properties for the working fluid, ammonia (NH_3), shall be in accordance with NBS Circulars 142 and 564.

2.1.1.8 Plant Availability

Operability assurances which shall be met after an initial checkout period are:

- a. Operability (actual operational time $\div$ required operation time) shall be no less than 0.90.
- b. System starting reliability (number of successful starts $\div$ attempted starts) shall be no less than 0.95.

2.1.1.9 Maintenance

- a. Redundancy shall be provided as required to ensure continuity of operation of critical functions (e.g., controls, lubrication, cooling).

- b. Provisions shall be made for efficient maintenance during downtime of critical equipment.

2.1.1.10 Nominal Design Temperatures

The system shall produce required power output with the following design temperatures:

- a. OTEC design point nominal temperature difference (ΔT) is 40°F.
- b. Ocean surface (warm) water temperature is 80°F.
- c. Ocean subsurface (cold) water temperature is 40°F.

When exposed to alternate environments, the system shall operate with reduced power output with the following design temperatures:

- a. Ocean surface water range from 70 to 85°F.
- b. Ocean subsurface water range from 32 to 50°F.
- c. OTEC design point temperature difference (ΔT) is (+4°F, -11°F).

2.1.1.11 Containment Vessel and Sea Dynamics

Because the specific containment vessel for the Demonstration Plant has not been chosen, a surface platform ship shall be used as the reference hull during conceptual design (Figure 2-1).

Current plans call for studies pertaining to five candidate hull shapes to be completed coincident with the initiation of Phase II. As the hull studies progress, DOE will select a reference hull configuration which may differ from the basic reference platform for the conceptual design. Therefore, considerations shall be given during the Phase I design tasks to the adaptability of the contractor's design(s) to the various candidate hull shapes.

NOMINAL DIMENSIONS AND RESPONSE

LENGTH = 750 FT
 BEAM = 300 FT
 DRAFT = 100 FT

SEA KEEPING PARAMETERS (SEA STATE SIX)

H - 21.4' SU - 7.5'
 P - .4 SW - 9.5'
 R - .2 Y - .8

SEA KEEPING PARAMETERS (SEA STATE NINE)

H - 41' SU - 15'
 P - 4 SW - 22'
 R - 2 Y - 2.5

Figure 2-1. Ship Type Platform

The power system shall be capable of normal operation in a surface vessel under sea-state-six conditions, and shall survive in a secured condition in a sea-state-nine. A sea-state-six condition has a significant wave height of 10 feet; a sea state nine has a significant wave height of 30 feet and is indicative of hurricane conditions.

Based on the attendant wave form forcing functions, nominal vertical and horizontal accelerations of the vessel are:

<u>Direction</u>	<u>Sea State</u>	
	<u>Six</u>	<u>Nine</u>
Vertical	0.05g	0.15g
Horizontal	0.01g	0.07g

In the initial studies, the OTEC power shall be capable of withstanding ship motions that normally govern marine propulsion and auxiliary machinery.

Hull motions associated with frequencies below 1 Hz have a maximum acceleration of 0.5g in the vertical, fore and aft, and athwartships directions.

Maximum hull angular velocities of 0.5 radian/sec in roll, and 0.1 radians/sec in pitch and yaw shall be considered.

For components connected to the hull structure, the following limits shall be observed (see Figure 2-2):

<u>Frequency</u>	<u>Limit (Vertical and horizontal)</u>	
	<u>Hard Mounted</u>	<u>Soft Mounted</u>
1-30 Hz	1 in/sec	2 in/sec
30-100 Hz	0.5g	1.0g

SHOCK SPECTRA (X10)

Figure 2-2. Hull Motion Limits, Velocity vs. Frequency

Hard mounted refers to equipment components which are mounted directly to the hull. Soft mounted refers to equipment that is mounted on structures that would tend to amplify the motion.

2.1.1.12 Handling and Installation

Consideration shall be given to transportation, general shipyard constraints for material handling, and installation. Special equipment for handling, installation, servicing, and operation shall be identified.

2.1.1.13 Material, Processes, and Parts

- a. The hardware shall have an operational life in an ocean environment (saltwater corrosion, biofouling) consistent with the design life noted in Section 2.1.1.2. Sealing problems associated with the various subsystems shall be addressed (NH_3 , H_2O , lube, inert gas purge).
- b. Equipment which cannot be fabricated of materials resistant to an ocean environment will be designed with sealed castings, compartments, or protective coatings to ensure that design life, availability, and reliability goals are met.
- c. As materials are evaluated and selected for operation in these environments, they shall be clearly identified and the criteria for selection (design technique description) documented.

2.1.1.14 Safety

The following safety factors shall be considered

- a. Human factors (heat, light, ventilation, adequate exists, working space, equipment, accessibility) shall be consistent with recognized standards (ABS, USCG).
- b. Other factors such as fire hazard, explosion hazard, development of toxic compounds, electrical hazards, shall be consistent with applicable OSHA and marine engineering regulations.
- c. Designs shall be fail safe.

2.1.1.15 Reliability

The reliability requirements (see Section 2.1.1.9) shall be demonstrated at a 60 percent confidence level.

2.1.2 Modularity Options

An OTEC Power Plant with an output of 100 MWe (under present guidelines) can be developed in a wide variety of design configurations. The primary option is the size of the submodules (hereafter referred to as power modules) that will make up the 100 MWe power plant. This is directly related to considerations of heat exchanger size and number, as well as the options relating to water flow, pumping arrangements, and ammonia loop and turbine staging. The conceptual design studies have attempted to consider the multiplicity of such options within the practical framework of system cost to arrive at an "optimum" configuration. Design layouts were generated for all cases investigated to ensure practicality of arrangements and to provide the necessary detail for estimating the costs in Section 5.

2.1.2.1 Module Sizing

Because the OTEC heat exchangers represent the dominant cost items of the OTEC Power System, low-cost, efficient designs must be selected as the basic building blocks in configuring any of the optimized power modules. The unit costs of the heat exchangers are dependent on their size, an illustration of this relationship is presented in Figure 2-3 where heat exchanger size is indicated by shell diameter. The direct relationship between heat exchanger size and module size is also indicated, with the module size defined by the power output capability of the Rankine cycle heat engine.

There are four separate cost factors affected by heat exchanger or module size illustrated in Figure 2-3:

- (1) fabrication, shipping, and installation
- (2) hull space
- (3) manifolding and pumps
- (4) plant down time.

The first three items show a decreasing trend initially, followed by either a leveling off or increase at some intermediate size. Cost associated with plant down time, on the other hand, are expected to increase monotonically as the size of the module components get larger and, hence,

Figure 2-3. Unit Cost Trends of Heat Exchangers Versus Module/Shell Size

fewer in number for a given power plant output. This increasing cost effect is related to the more significant failure modes that can occur when larger and fewer module components are used as discussed in Section 4.

While the decreasing trend of manifolding and pump costs is a straightforward relation, hull space costs and fabrication, shipping, and installation costs are somewhat more involved. Absolute costs related to hull space will depend on the particular platform type being considered, but generally follow a decreasing trend with size initially with an increase at intermediate size as indicated in Section 5.2. The impact on module size selection will depend on both the magnitude of the cost variations and the point at which the economy of increasing hull size is invalidated by construction limitations or other practical constraints. These factors and their dependency on hull type are discussed further in Section 2.2.

Fabrication, shipping, and installation, being the dominant cost category, can be expected to dictate the trend of the total cost relation with heat exchanger or module size. This relationship will depend on the ground rules used in making the cost estimates. Two of the curves plotted in Figure 2-3 illustrate the two major underlying assumptions: current technology cost basis and future technology cost basis. The latter assumption results in lower estimated costs of the larger heat exchanger sizes and is premised on the development of special fabrication processes and facilities to efficiently produce large-scale heat exchangers. Such processes and required facilities are within the present state of the art and, given some reasonable production level for OTEC plants, could readily be implemented in the future. The cost savings could be significant and would tend to drive the optimum module to a larger size. At some point, however, increasing plant downtime costs or, possibly, hull costs would dominate and limit the heat exchanger size tradeoff.

Another significant feature of the fabrication, shipping, and installation cost curve is the step changes that are believed to occur at two points as heat exchanger size increases. These points reflect physical size constraints caused by either rail shipment limitations (approximately 18 feet) or handling equipment limitations of existing fabrication shops (approximately 35 feet). Again, special facilities and production and

shipping procedures developed specifically for OTEC would effectively eliminate the impact of these size implications as shown in Figure 2-3.

The unit cost analysis discussed above indicates that four shell diameter size ranges must be considered:

- 18-foot diameter shell
- 25-foot diameter shell
- 35-foot diameter shell
- 50-foot diameter shell

The first and third correspond to the step changes in the fabrication, shipping, and installation costs associated with the physical size constraints of current technology. Unit costs for sizes smaller than the 18-foot diameter are projected to be significantly higher due primarily to inefficient hull packaging. The 50-foot diameter shell, while not necessarily an upper limit on practical shell size, is one that remains within low unit cost bounds of feasible technology.

Corresponding to these four selected shell sizes, conceptual heat exchangers were designed to provide efficient heat transfer performance for use in configuring the detailed modularity options. The logic involved in this development is shown in Figure 2-4.

The sized heat exchangers are combined with corresponding Rankine cycle performance calculations (Figure 2-4) to provide power output estimates for the individual power modules. An iterative loop is needed to arrive at a set of modularity options that are convenient fractions of the overall power module output. The resulting options shown in Figure 2-5 correspond to this design selection process.

It should be noted that the module power output (net) associated with each of the sizes shown in Figure 2-4 is based on a nominal tube length of 25 feet. Subsequent tradeoffs have shown that the optimum tube length is about 40 feet, resulting in somewhat smaller shell diameters to generate the indicated power levels. This refinement of heat exchanger design parameters does not change the primary size and power module configuration presented here.

Figure 2-4. Logic For System Modularity Size Selection

OPTIONS	25 MWe	12.5 MWe	6.25 MWe	3.125 MWe
HEAT EXCHANGERS SIZE POWER	NOM 50' DIA	NOM 35' DIA	NOM 25' DIA	NOM 18' DIA
MODULE OPTIONS
HEAT EXCHANGERS SIZE	NOM 35' DIA	NOM 25' DIA	NOM 18' DIA	
POWER MODULE SUB-OPTIONS

EVAPORATOR
CONDENSER

TURBINE
GENERATOR

Figure 2-5. Modularity Sizing Options

Schematic drawings of the selected modularity options are summarized in Figure 2-5. In addition to the primary options corresponding to one Rankine cycle loop per module, suboptions of the power module configuration were selected for consideration as indicated. These result from the coupling of the hardware for two Rankine cycles having one-half the module power output, but with a single turbine and/or a single generator arranged as illustrated in Figure 2-6. By using one generator per module instead of two, an efficiency gain of approximately 1 percent is achieved (97.8 versus 96.8) plus a cost savings of about \$300,000.

2.1.2.2 Seawater Flow/Pump Arrangements

Important design alternatives are related to the seawater ducting arrangements which are applicable for both the suboption configurations shown in Figure 2-5 or for combinations of two or more of the single evaporator/condenser modules that make up the power module. Either a series or a parallel arrangement could be used to feed the separate evaporators or condensers from a single inlet duct. For multiple combinations of evaporators and condensers in either the 6.25 or 3.125 modularity options, a more complicated series/parallel feeding arrangement such as that illustrated in Figure 2-7 has been considered.

Although the typical arrangement is parallel feeding, the series arrangement can, perhaps, offer some advantages in terms of efficient thermal performance. This could be true if high ammonia side heat transfer coefficient required short tubes and consequently high flow rates of seawater in the conventional parallel arrangement. It may be necessary to combine the seawater series arrangement with the ammonia loop in order to achieve any benefits of the series option. The combined scheme shown in the diagram in Figure 2-8 was considered, as was the simple series arrangement in the tradeoff studies which are detailed in Section 3.

The number, type, and size of seawater pumps were also included in the conceptual design analyses. Because their selection depends on volume flow rate performance compared with unit pump costs, they are directly coupled to the module size and the ducting arrangements considered above. Their cost tradeoffs represent a secondary factor compared to the heat exchanger options, the dominant cost item.

Figure 2-6. Sub-Option Modules — Turbine/Generator Arrangements

TOTAL OUTPUT: 25KWe
 HEAT EXCHANGERS: 8 EVAPORATORS (18' DIA)
 8 CONDENSERS (18' DIA)

Figure 2-7. Series/Parallel Arrangement

(a) SERIES WATER FLOW ARRANGEMENT

(b) AMMONIA LOOP STAGING

Figure 2-8. Ammonia Loop Staging for Series Heat Exchanger Arrangement

2.1.2.3 Ammonia Loop Arrangements

In addition to the ammonia loop staging (associated with seawater series flow arrangements), there are relatively straightforward considerations of the ammonia side modularity. The basic analyses can be performed independently from the more involved heat exchanger-related tradeoffs. The options are the number and type of both the pumps and turbines associated with each of the Rankine cycle modularity options. This basically involves the consideration of multiple-stage versus single stage designs, and depends on the performance/cost tradeoffs.

2.1.3 Optimization Methodology

The overall criteria for optimizing the OTEC power module is to minimize unit cost of output energy generated over the power plant lifetime. This involves consideration of both capital costs and plant availability, with the composite effect expressed in terms of "mills per kilowatt-hour." While capital costs include a factor for the platform, only one type, a ship hull, was assumed in the initial set of power system tradeoffs based on guidelines established for the study in Section 2.1.1. The assumption was that power module size and related component selection will not be greatly affected by considerations of detailed ocean system alternatives, although the absolute cost values may change significantly. This assumption is probably not valid, however, if unit costs of module options do not vary greatly and/or if the integrated design of the power and ocean systems results in strongly coupled cost savings effects. Subsequent analysis has been performed on an alternative hull configuration in order to investigate the impact on the initial tradeoff results. A further discussion of these points is presented in the next section where costs are summarized and baseline power module selection is made.

The overall approach to the assessment of optimum power module size and component arrangement is shown in Figure 2-9. Optimization is first performed for each power module size and includes the various component and flow arrangements described in Section 2.1.2. Comparisons are then made among each optimum power module design in the selection process. Details of this procedure are described in subsequent sections of this report.

There are two major analytical tools that were developed and used in the optimization study: the OTEC II Design and Optimization Model and the Module Availability Model. The OTEC II Design and Optimization Model (described in detail in Appendix E) consists of the system cost/performance relations and an optimization search routine that finds a minimum by systematically varying design parameters (e.g., heat exchanger tube size, tube length, and seawater flow rates) for a given set of input conditions. The current version is an update of TRW's previously developed model whose major changes include improved cost algorithms and the addition of optimization variables associated with advanced shell/tube heat exchangers.

Figure 2-9. Approach to Optimum Module Selection

The plant availability or operability model incorporates formulations of component failure modes and estimates mean failure rates to give probabilistic estimates of available plant operating hours over its design lifetime. This calculation, combined with the optimized capital cost estimates, gives the unit cost per delivered energy that is used as the basis of module comparison. Further description and input data for the Module Availability Model is provided in Section 4.

2.2 BASELINE POWER SYSTEM SELECTION

A baseline OTEC Power System Conceptual Design has been selected based on the tradeoff analyses presented in the following sections. The selected system consists of the basic module size, component and ducting arrangements, and hull layouts for 100 MWe OTEC power plant. The primary selection basis was unit energy cost over the plant lifetime, as discussed in Section 2.1.3.

A summary of the bottom line costs of the optimized OTEC Power System designs is given in Figures 2-10 and 2-11 as a function of power module size. All assumptions and suboptimization studies of ducting arrangements, etc., which lead to these final results are discussed in subsequent report sections. Key design features include the use of titanium tubes in the shell/tube heat exchangers and implementation within a ship hull platform.

Unit capital costs are broken down in Figure 2-10 by platform and power plant, as well as by the number of evaporators and condensers per module. Referring back to Figure 2-7, this latter design aspect corresponds to a coupling of two Rankine cycle components into one module versus using a single loop with larger individual components. Rationale for this suboption is given in Section 6.

The most significant trends shown in Figure 2-10 are that power plant cost is not a strong function of module size, but that the ship platform cost is. The platform costs decrease initially as might seem reasonable intuitively, but then increase again beyond either the 6.25 or 12.5 MWe module size, depending on whether one or two sets of hardware components are used in the module. This result is displayed in Section 5.2 design layouts of hull and components. The total cost trend

Figure 2-10. Unit Capital Cost Variation With Module Size

Figure 2-11. Plant Availability and Delivered Energy Cost Versus Power Module Size

resulting from these composite effects is the occurrence of minimum points at the 6.25 MWe (single Rankine cycle arrangement) or the 12.5 MWe (coupled cycle) module size, with rather significant cost increases at smaller sizes and relatively small or no increases at larger sizes. The coupled Rankine cycle arrangement has somewhat lower costs overall, but only about 4 percent, at the respective minimum cost points.

Figure 2-11 shows total capital costs of a production plant versus the costs for the initial plant (a factor of 0.7 has been used) presented in the previous figure. Plant downtime fraction (equal to one minus availability as defined in Section 4.1) and busbar energy costs corresponding to these data* are also shown. The effect of increasing plant downtime with power module size will tend to decrease available busbar energy at larger power module sizes. The net effect is that busbar energy cost reaches a somewhat more sharply defined minimum, as can be seen for the coupled Rankine cycle module curve.

The 12.5 module option involving a coupled Rankine cycle arrangement was selected as the optimum design based on the above results. It represents the minimum delivered energy cost design compared with all other alternatives investigated in the study. A complete description of this baseline power system for OTEC is presented in the following section.

While not a criteria in its selection, the Baseline power module has the additional feature of having component heat exchanger shell sizes (about 20-feet diameter) and individual Rankine cycle power outputs (6.25 MWe) convenient for test article consideration. The planned test phase of subscale OTEC systems, particularly the 5 MWe experiments, could involve full scale components of the prototype 6.25 MWe power submodule cycle. This idea is developed further in Section 6.

Before proceeding, the selection of the optimum module size should be considered further in terms of its sensitivity to certain key assumptions used in the present analyses. The most potentially critical assumption

* Busbar energy cost is calculated by dividing capital costs by the availability factor times total potential plant hours over its assumed 30 year lifetime.

appears to be that of a ship hull for the platform. The results of the module size tradeoff runs indicate that the ship platform cost variations are important to the total cost trends. To investigate this result further, additional design layouts and costing analysis were performed for a circular barge hull configuration. Cost variations with module size were found to be similar, thereby increasing the confidence level of this trend of decreasing costs initially with an increase at larger sizes. The result can be generalized to include all surface coupled hulls, but the tradeoff may have somewhat different results for other hull types, e.g., submersibles. The effect, if any, would most likely be to increase optimum module size as, again, intuition says that packaging efficiency increases with size. Other compensating factors at larger module sizes such as plant downtime may drive the optimum more strongly. The conclusion is that the 12.5 module size is probably near optimal for all hull configurations. A rough estimate based on sensitivities calculated during the present study is that it will probably not operate further than about 5 percent off optimum for any other hull.

A brief discussion of the sensitivity of the power module design to alternate hull configurations is included in Section 1.1.

The other possible consideration in the optimum module selection process that is, perhaps, not adequately addressed is also related to the platform. The question is the net cost benefit of integrated design of hull, power system, cold water pipe and all other interrelated systems. It is estimated that absolute system costs can be significantly reduced by a more system-inclusive design optimization study, but that present conceptual tradeoffs of power module arrangements and relative benefits will not be changed. This is an inherent assumption of the present design process.

2.3 BASELINE POWER SYSTEM DESCRIPTION

As described in the previous section, a 12.5 MWe (net) power module was selected as the optimum cost effective module size responsive to the design requirements of RFP EG-77-R-03-1440 delineated in Section 2.1.1. The 12.5 MWe power module includes twin 6.25 MWe vertical-tube evaporators which feed ammonia vapor through two 6.25 MWe turbines, which drive a common 12.5 MWe generator, and then into twin vertical condensers (see Section 2.3.1). The liquid ammonia condensate from each condenser is transferred to its respective evaporator with an ammonia feed pump. The parallel manifolds are provided with shared redundant pump (stand-by) to improve reliability (Sections 2.3.1 and 4.1). Ammonia recirculating pumps, manifolded similar to the feed pumps, collect excess liquid from the bottom of each evaporator and return it to the evaporator liquid ammonia distribution plenum. One seawater pump is provided for each heat exchanger (Sections 2.3.1 and 3.2.2).

The baseline power module/plant design is characterized by the following features:

a. Open Trough Seawater Distribution

An open trough cold and warm water distribution system was selected to minimize the effects of potential damage mechanisms obtained by hull motions due to sea state conditions (Section 2.1.1). The open trough system (Section 2.3.2) avoids water hammer effects inherent in closed systems, decouples water side hydraulics, minimizes cavitation potential, and eliminates the requirement for expensive water side valves and piping (Section 3.3.2). The open water troughs, coupled with vertical orientation of the heat exchangers, results in a compact power plant configuration (Section 2.3.2).

b. Open Top Heat Exchanger with Free Water Surface

The heat exchangers are uniquely designed with open tops to simplify water box design and reduce cost. The open top introduces the capability for in-service cleaning of tubes with an innovative cleaning machine (Section 3.1.3). The open top with free water surface also facilitates identification and plugging

of leaking tubes without removal of waterbox heads (Section 4.3).

c. Heat Exchangers

The heat exchangers designs are based on vertical tubes with heat transfer enhancements on the ammonia and water sides. The material selected for the baseline heat exchanger tubes is titanium (Sections 3.1 and 5.2.3) with recommendations for further work in aluminum technology (Section 6.3). This concept, with a minimized fouling factor ($0.0001 \text{ F-hr-ft}^2/\text{B}$) maintained by in-service tube cleaning chlorination, and results in an overall evaporator U of $1176 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ and condenser U of $798 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ (see Tables 2-6 and 2-7).

Performance, design, and cost characteristics of the selected baseline 12.5 MWe power module are described in the following tables with references to applicable report sections for further descriptions:

<u>Table</u>	<u>Title</u>	<u>Report Section</u>
2-1	Baseline Power Module	2.3.1, 2.3.4
2-2	Rotating Machinery	3.2
2-3	Turbine and Pump Conditions at Design Point Operation	3.2
2-4	Evaporator Design Data	2.3.3, 3.1.1
2-5	Condenser Design Data	2.3.3, 3.1.1
2-6	Evaporator Thermal Data	2.3.3, 3.1.1
2-7	Condenser Thermal Data	2.3.3, 3.1.1
2-8	Power Plant Cost Breakdown	5
2-9	Evaporator Cost Breakdown	5
2-10	Condenser Cost Breakdown	5

Table 2-1. Baseline Power Module (12.5 MW_e Nominal)

Gross power	16.51 MW _e
Net power	12.50 MW _e
Gross power/net power	1.32
Warm water pumping power	1.52 MW _e
Cold water pumping power	2.03 MW _e
Ammonia feed pumping power	0.34 MW _e
Ammonia reflux pumping power	0.02 MW _e
Chlorination power	0.05 MW _e
Mechanical cleaning power	0.01 MW _e
Controls, miscellaneous	0.04 MW _e

Table 2-2. Rotating Machinery (per Baseline Module of 12.5 MW_e (net))

Item	Number	Type	RPM	Horsepower per Module	Efficiency (%)	Comments
Turbine and Diffuser	2	Axial flow multi-stage	1800	22,700	85	2 turbines drive one generator
Generator	1	AC 3Ø 4.16 kV	1800		97.8	
Warm Water Pump	2	Vertical propeller	129	2000	87	
Warm Water Pump Motor	2	Vertical gear box	900		92.5	
Cold Water Pump	2	Vertical propeller	139	3000	87	
Cold Water Pump Motor	2	Vertical gear box	900		92.5	
Ammonia Feed Pump	3	Centrifugal single stage end suction	1760	500	80	One pump is shared standby
Ammonia Feed Pump Motor	3	Explosion proof	1760		91	
Ammonia Reflux Pump	3	Centrifugal single stage end suction	1170	50	80	One pump is shared standby
Ammonia Reflux Pump Motor	3	Explosion proof	1170		91	

Table 2-3. Turbine and Pump Conditions at Design Point Operation

	Turbine (6.25 MW _e Nominal)	
Flow Rate	1,708,900 lb/hr	
<u>Inlet</u>		
Pressure	128.8 psia	
Temperature	70°F	
Velocity	78.9 ft/sec	
Quality	> 99%	
<u>Outlet</u>		
Pressure	89.2 psia	
Temperature	50°F	
Velocity	40.1 ft/sec	
Quality	>96.5%	
	<u>Ammonia Pumps*</u>	
	<u>Feed Pump</u>	<u>Reflux Pump</u>
Flow rate (lb/hr)	1,708,900	281,300
Pressure rise (psid)	54.2	14.4
	<u>Water Pumps*</u>	
	<u>Warm Water</u>	<u>Cold Water</u>
Flow rate (gpm)	358,100	381,800
Pressure rise (ft)	8.2	11.2

*One for each nominal 6.25 MW_e heat exchanger.

Table 2-4. Evaporator Design Data

	Evaporator (6.25 MW _e Nominal)
Number per module	2
Thermal rating	907 MBtu/hr
Water flow rate	350,100 gpm
Ammonia flow rate	1,708,900 lb/hr
Design pressure	160 psia
Operating pressure	128.8 psia
Overall length	70. ft
Water enhancement factor	1.5
Ammonia enhancement factor*	1.5
Total number of tubes	5,307
Tube OD	2 in.
Tube wall, corrugated	0.028 in.
Tube pitch	2.5 in.
Tube layout	Triangular
Tube sheet diameter	17.6 ft
Effective tube length	42 ft
Actual tube length	45.5 ft
Tube sheet thickness	4 in.
Number of tube supports	4
Tube support thickness	2 in.
Tube support spacing	8 ft
Shell OD	17.6 ft
Shell wall thickness	1-3/16 in.
Void fraction	0.54
Tube material	Titanium
Tube sheet material	SA516, GR70
Shell material	SA516, GR70
Baffle material	SA516; SA283; SA285, GRB
Coating materials	Coal tar epoxy (waterboxes)

*Area ratios, fluid mechanic effect not included.

Table 2-5. Condenser Design Data

	Condenser (6.25 MW _e Nominal)
Number per module	2
Thermal rating	879 MBtu/hr
Water flow rate	381,800 gpm
Ammonia flow rate	1,708,900 lb/hr
Design pressure	115. psia
Operating pressure	89.2 psia
Overall length	76 ft
Water enhancement factor	1.5
Ammonia enhancement factor*	1.5
Total number of tubes	7,221
Tube OD	2 in.
Tube wall, corrugated	0.028 in.
Tube pitch	2.5 in.
Tube layout	Triangular
Tube sheet diameter	20.7 ft
Effective tube length	42 ft
Actual tube length	43 ft
Tube sheet thickness	4 in.
Number of tube supports	4
Tube support thickness	2 in.
Tube support spacing	8 ft
Shell OD	20.7 ft
Shell wall thickness	1-3/16 in.
Void fraction	0.53
Tube material	Titanium
Tube sheet material	SA516, GR70
Shell material	SA516, GR70
Baffle material	SA516; SA283; SA285, GRB
Coating materials	Coal tar epoxy (waterboxes)

*Area ratio, fluid mechanic effect not included.

Table 2-6. Evaporator Thermal Data (6.25 MW_e Nominal)

Water Flow Rate	183,200,000 lb/hr
Water Flow per tube	9.59 lb/sec
Water Velocity inside tube	7.76 ft/sec
Reynolds Number inside tube	121,400
Prandtl Number inside tube	6.28
Inlet Water Temperature	80°F
Outlet Water Temperature	74.8°F
Average Water Temperature	77.4°F
LMTD	7.02°F
Average Tube wall surface Temperature	73°F
Average Shell wall surface Temperature	71°F
hw (including enhancement) referred to Dw*	1951 Btu/hr ft ² F
h fouling referred to Dw	10,000 Btu/hr ft ² F
h wall referred to D _o - 0.088 in. [†]	8700 Btu/hr ft ² F
h ammonia referred to D [†]	8052 Btu/hr ft ² F
U _o referred to Dw	1176 Btu/hr ft ² F
Ammonia inlet temperature	52.9°F
Ammonia outlet temperature	70°F
Outlet Vapor Quality	>99%
Outlet Maximum Vapor Velocity	<3.0 ft/sec
Total Pressure Drop of Water**	3.32 psid
Total Pressure Drop of Ammonia***	14.4 psid
<u>For Vertical Evaporators</u>	
Feed rate per tube	375.0 lb/hr
Feed film Reynolds Number	8490
Exit Film Reynolds Number	1200
Maximum Distribution Δp	<0.01 psid

*See Appendix B for definition of these diameters.

**Tubeside pressure drop including entrance and exit losses.

***Equal to recirculation pump pressure rise.

[†]Ammonia coefficient includes 65 mill aluminum wall conductance, therefore wall coefficient has been adjusted upwards to give correct composite coefficient.

Table 2-7. Condenser Thermal Data (6.25 MW_e Nominal)

Water Flow Rate	196,000,000 lb/hr
Water Flow per tube	7.54 lb/sec
Water Velocity inside tube	6.08 ft/sec
Reynolds Number inside tube	57,700
Prandtl Number inside tube	10.97
Inlet Water Temperature	40 ⁰ F
Outlet Water Temperature	44.7 ⁰ F
Average Water Temperature	42.35 ⁰ F
LMTD	7.36 ⁰ F
Average Tube wall surface Temperature	
Average Shell Wall surface Temperature	
hw (including enhancement) referred to Dw*	1216 Btu/hr ft ²⁰ F
h fouling referred to Dw	15,000 Btu/hr ft ²⁰ F
h wall referred to D _o - 0.088 in. [†]	8700 Btu/hr ft ²⁰ F
h ammonia referred to D [†]	4000 Btu/hr ft ²⁰ F
U _o referred to Dw	798 Btu/hr ft ²⁰ F
Ammonia inlet temperature	50 ⁰ F
Ammonia outlet temperature	49.9 ⁰ F
Inlet vapor quality	>96.5%
Inlet maximum vapor velocity	≤2.5 ft/sec
Total Pressure Drop of Water**	2.32 psid
Total Pressure Drop of Ammonia	<0.01 psid
<u>For Vertical Condensers</u>	
Feed rate per tube	236.7 lb/hr
Feed film Reynolds Number	0
Exit Film Reynolds Number	4740
Maximum Distribution Δp	<0.01 psid

*See Appendix B for definition of these diameters.

**Tubeside pressure drop including entrance and exit losses.

[†]Ammonia coefficient includes 65 mill aluminum wall conductance, therefore wall coefficient has been adjusted upwards to give correct composite coefficient.

Table 2-8. Power Plant Cost Breakdown

	Prototype(a) (\$/KW _e)	Production (\$/KW _e)
Evaporator	342 (d)	213 (e)
Condenser	422 (d)	260 (e)
Turbine	80	50
Generator	56	50
Water pumps, motors, etc.	129	110
Ammonia piping and valves	10	6
Ammonia feed pumps	8	7
Ammonia Recirculation Pumps	4	4
Control Systems	12	10
Electrical equipment	26	23
Cleaning Systems	13	8
Chlorination	7	6
Ammonia Support System	38	24
Installation	109	33
Total (Direct Cost)	1256	804
Indirect Costs (Engr, Admin)	195 (b)	40 (c)
Total	1451	844

- (a) Basic 12.5 MWe power system module with 2 evaporators and 2 condensers.
- (b) Engineering design and field supervision 13 percent, home office and field administration 2.5 percent.
- (c) Field engineer and supervision 2.5 percent, administration 2.5 percent.
- (d) Includes freight (Condenser 51\$/KW and Evaporator 38\$/KW).
- (e) Shipyard dedicated heat exchanger fabrication facilities - no freight.

Table 2-9. Evaporator Cost Breakdown

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	143.6	37.9	143.6	13.4
Tubesheets (no cladding)*	5.0	10.2	5.0	3.6
Tube Support Plates	0.6	2.9	0.6	1.0
Shell	7.7	27.2	7.7	9.6
Water Boxes	1.0	19.5	1.0	6.9
Nozzles	1.2	7.8	1.2	2.8
Ammonia Distribution	1.5	5.2	1.5	1.8
Bustle	1.1	19.3	1.1	6.9
Transition Inlet	1.1	8.1	1.1	2.9
External Support	0.3	3.2	0.3	1.1
Freight		37.7		
Subtotals	163.1	179	163.1	50
TOTAL \$/KW _e	342.1		213.1	

Evaporator Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 17.6 Feet
 Tube Material- - - - - Titanium (Corrugated)
 Tube Length- - - - - 45.5 Feet, overall
 Number of Tubes- - - - - 5,307
 Tube Diameter- - - - - 2 Inches, O.D.

*Tubes rolled, not welded.

Table 2-10. Condenser Cost Breakdown

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	189.0	52.7	189.0	15.7
Tubesheets (No cladding*)	7.6	19.4	7.6	5.8
Tube support plates	0.9	4.5	0.9	1.3
Shell	10.6	34.4	10.6	10.3
Water boxes	1.4	24.3	1.4	7.3
Nozzles	1.3	9.7	1.3	2.9
Transition inlet	1.4	10.0	1.4	3.0
External support	0.4	4.0	0.4	1.2
Freight	---	50.2	---	---
Subtotals	212.6	209.2	212.6	47.5
Total \$/KW _e	421.8		260.1	

Condenser Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 20.7 Feet

Tube Material - - - - - Titanium (Corrugated)

Tube Length - - - - - 43 Feet, overall

Number of Tubes - - - - - 7,221

Tube Diameter - - - - - 2 inches, O.D.

*Tubes rolled, not welded.

2.3.1 Flow Diagrams

Process flow diagrams were developed to initiate and guide the conceptual design development.

The process flow diagram describing the baseline 12.5 MWe power module is shown in Figure 2-12. The diagram displays fluid state conditions throughout the process, electrical loads, pipe line sizes, and instrumentation and control.

The ammonia storage and recovery system for each installed 50 MWe of plant power is displayed in Figure 2-13.

The liquid nitrogen and gasifying purge system for each installed 50 MWe of plant power is displayed in Figure 2-14 (see Section 3.2.4).

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

Figure 2-12. 12.5 MWe Power Module Process Flow Diagram Selected System

Figure 2-13. Ammonia Storage & Recovery Subsystem

0	INITIAL ISSUE	1-12-78	ROP
LTR	DESCRIPTION	DATE	APPROVED
REVISIONS			
OTEC POWER SYSTEM DEVELOPMENT		TRW SYSTEMS GROUP ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
LIQUID NITROGEN STORAGE			
DATE	JAN 5 1978	CHG. NO.	6850.38-51
DRAWN	R J COREY		
CHG.			
APPROVED	R O PEARSON	APPROVED - PROGRAM OFFICE	

Figure 2-14. Liquid Nitrogen Storage

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

2.3.2 General Arrangement Drawings

The general arrangement of a 100 MWe power plant utilizing the selected baseline 12.5 MWe power modules is shown in Figure 2-15 as configured in a surface platform/ship hull, in accordance with DOE RFP EG-77-R-03-1400. A compact layout was accomplished with our selected baseline design that is significantly smaller than the DOE reference hull, as noted below and in Section 2.3.5.

ITEM	DOE REFERENCE HULL	TRW BASELINE HULL
Length (feet)	750	353
Beam (feet)	300	286
Draft (feet)	100	75

By examination the selected baseline power module design is directly applicable to the circular barge-type platform and, with modification, to the spar and submersible platforms.

The general arrangement of the power module was developed in sufficient detail to establish high confidence definition of space requirements, as well as credible cost estimates. The general arrangements are depicted in Figures 2-16, -17, -18, and -19.

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

OTEC POWER SYSTEM DEVELOPMENT		TRW SYSTEMS GROUP	
100 MWe POWER PLANT SURFACE BARGE PLATFORM GENERAL ARRANGEMENT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE 1-19-78		DWG. NO.	6850.38-56
DRAWN R. COREY			
CHECKED			
APPROVED R.O. PEARSON	APPROVED - PROGRAM OFFICE		

Figure 2-15. 100 MWe Power Plant
Surface Barge Platform General
Arrangement

Figure 2-16. 12.5 MWe Power Module
General Arrangement Elevation

2-53/54

POWER STATION PLAN

Figure 2-17. 12.5 MWe Power Module
General Arrangement Power Station
Plan

1/8" = 1'-0"
SCALE 5 0 5 10 15 20 FEET

O INITIAL ISSUE		1-12-76	ROP
LTR	DESCRIPTION	DATE	APPROVED
REVISIONS			
OTEC POWER SYSTEM DEVELOPMENT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
12.5 MWe POWER MODULE GENERAL ARRANGEMENT POWER STATION PLAN			
DATE JAN 6 1976		DWG. NO.	
DRAWN R. COREY		6850.38-46	
CHKD.			
APPROVED R. O. PEARSON		APPROVED - PROGRAM OFFICE

PUMP STATION PLAN

1/8" = 1'-0"
SCALE
5 10 15 20
FEET

0	INITIAL ISSUE	1-12-78	RoP
LTR	DESCRIPTION	DATE	APPROVED
REVISIONS			
OTEC POWER SYSTEM DEVELOPMENT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
12.5 MWe POWER MODULE GENERAL ARRANGEMENT PUMP STATION PLAN		DATE JAN 6 1978 DRAWN R. LOREY CHIEF APPROVED R.D. PEARSON APPROVED - PROGRAM OFFICE	
		DWS. NO. 6850.38-47	

Figure 2-18. 12.5 MWe Power Module
General Arrangement Pump Station Plan

(A) FRAMING PLAN AT TURBINES

(B) FRAMING PLAN AT SUMPS

ELEVATION

0	INITIAL ISSUE	1-12-78	ROP
LTR	DESCRIPTION	DATE	APPROVED
REVISIONS			
OTEC POWER SYSTEM DEVELOPMENT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
12.5 Mw POWER MODULE GENERAL ARRANGEMENT STEEL FRAMING & ELEVATION			
DATE	1-6-78	CHG. NO.	6850.38-49
DRAWN	R. COREY		
CHECKED			
APPROVED	R.O. PEARSON	APPROVED - PROGRAM OFFICE	

1/8" = 1'-0"
SCALE
0 5 10 15 20
FEET

Figure 2-19. 12.5 MWe Power Module General Arrangement Steel Framing and Elevation

2-57/58

2.3.3 System Performance Specifications and Characteristics

This section summarizes performance data for the conceptual design which uses titanium heat exchanger tubes. The following paragraphs give the results for the major subsystems with evaporator performance (CMU data), condenser performance (Oak Ridge data) and reduced fouling factors 0.0001 (evaporator)/0.000067 (condenser) resulting from the combination of chlorination and cleaning.

2.3.3.1 Cold Water Supply Pipe - Per 100 MWe

Pipe Length, ft	3,400
Pipe Inner Diameter, ft	49.8
Water Velocity, fps	7
Volume Flow Rate, cfs	13,610
Mass Flow Rate, lb/sec	871,100

2.3.3.2 Evaporator (Including Preheater) - One Per 6.25 MWe: 2 Evaporators Per 12.5 MWe Module

Tube Length-Effective, ft	42
Tube Length-Total, ft	45.5
Tube Inner Diameter, in	1.884
Tube Outer Diameter, in	2.0
Enhancement Ratio	1.5
Shell Diameter	17.6
Number of Tubes Per Exchanger	5,307
Tubeside Water Velocity, fps	7.76
Tubeside Pressure Drop, psi (1)	3.32 ⁽¹⁾
Heat Transfer Area, ft ² (2)	109,970 ⁽²⁾
Thermal Duty, Btu/hr x 10 ⁻⁸	9.07

Overall U, Btu/hr - Ft² - °F 1176⁽³⁾

Temperatures

Water In, °F 80

Water Out, °F 74.8

Ammonia Evaporation, °F 70

LMTD, °F 7.02

Shellside Operating Pressure, psia 128.8

Fouling Factor, hr-ft² - °F/Btu 0.0001

NOTES:

- (1) Includes tubes entrance and exit losses
- (2) Referred to inside area of smooth tube having same flow area as enhanced tube
- (3) Referred to inside smooth tube area defined in 2 above

2.3.3.3 Warm Water Return Pipe, One Per Exchanger

Pipe Length, ft 100

Pipe Inner Diameter, ft 12.0

Water Velocity, fps 7

Volume Flow Rate, cfs 798

Mass Flow Rate, lb/sec 50,880

2.3.3.4 Condenser - One Per 6.25 MWe: 2 Condensers Per Module, Each Exchanger

Tube Length-Effective, ft 42

Tube Length-Total, ft 43

Tube Inner Diameter, in 1.884

Tube Outer Diameter, in 2.0

Enhancement Ratio 1.5

Shell Diameter 20.7

Number of Tubes Per Exchanger 7,221

Tubeside Water Velocity, fps	6.08
Tubeside Pressure Drop, psi	2.32 ⁽¹⁾
Heat Transfer Area, ft ²	149,600 ⁽²⁾
Thermal Duty, Btu/hr x 10 ⁻⁸	8.79
Overall U, Btu/hr - ft ² - °F	798 ⁽³⁾
Temperatures, °F	
Water In, °F	40
Water Out, °F	44.7
Ammonia Condenser, °F	50
LMTD, °F	7.36 ⁽⁴⁾
Shellside Operating Pressure, psia	89.2
Fouling Factor, hr-ft ² - °F/Btu	0.000067

NOTES:

- (1) Includes tube entrance and exit losses
- (2) Referred to inside area of smooth tube having same flow area as enhanced tube
- (3) Referred to inside smooth tube area defined in 2 above
- (4) Corrected for shellside pressure drop

2.3.3.5 Cold Water Return Pipe, One Per Exchanger

Pipe Length, ft	100
Pipe Inner Diameter, ft	12.4
Water Velocity, fps	7
Volume Flow Rate, cfs	851
Mass Flow Rate, lb/sec	54,440

2.3.3.6 Ammonia Cycle Data Per 6-1/4 MWe

Enthalpy Drop Across Turbine, Btu/lb	16.81
Turbine Exit Moisture, %	2.5 + inlet moisture (%)
Ammonia Mass Flow Rate, lb/sec	474.7
Volumetric Flow Rates of Ammonia:	
Evaporator Exit, acfs	1,098
Turbine Exit, acfs	1,526
Condenser Exit, acfs	12.17

2.3.3.7 Total Water Side Pressure Drop*

Warm Water Loop psi (ft)	3.96 (8.2)
Cold Water Loop, psi (ft)	4.96 (11.2)

*Figure 2-20 indicates head losses due to entrances, exits and screens assumed in the optimization study.

1) GRAVITY HEAD DUE TO DENSITY DIFFERENCE TAKEN AS 3 FT = 1.33 PSI

2) ALL RESISTANCE COEFFICIENTS ARE REFERRED TO NOMINAL DUCT VELOCITY OF 7 FPS

3) HEAD LOSS:

$$h = K \frac{V^2}{2g}$$

4) ROUGHNESS FACTORS (FT)
 COLD WATER DUCTS $\epsilon = 0.01$
 WARM WATER DUCTS $\epsilon = 0.02$
 HEAT EXCHANGER TUBES, $\epsilon/D = 0.0002$

Figure 2 - 20 Head Losses Due To Entrances, Exits, And Screens

2.3.4 Equipment Itemization (size, weight, and envelope of major components and subsystems)

Equipment requirements for a baseline 12.5 MW_e power module are shown in Table 2-11. Information includes weight, number required, cost, dimensions, and design descriptor.

Table 2-11. Equipment List 12.5 MW_{Net} Power Module

Item	Component	Weight (K lbs)	No. Req'd	Unit Cost (K\$)	Dimensions (ft)			Design Information					
					Bowl Dia	Disch Dia	Shaft L	Flow KG/PM	TDH FT	Power HP	Speed RPM	η %	
1	W W Pump	100	2	313	8	12	55	358.1*	8.2	874	129	87	Vertical Propeller Wet Pit Pump
2	C W Pump	100	2	336	8	12	55	381.8	11.2	1272	139	87	Vertical Propeller Wet Pit Pump
					H	W	L						
3	NH ₃ Feed Pump	5	3	25	4.6	4.8	8.0	5.5	200	250	1760	72	Centrifugal Single Stage, End Suction
4	NH ₃ Rec Pump	2	3	15	4.4	4.1	7.0	2.2	55	25	1170	72	Centrifugal Single Stage, End Suction
	<u>Motors</u>												
5	W W Motor	35	2	75	12	6	6	--	--	1000	900	92.5	900 RPM Motor and Gear Box in Vert. Stack
6	C W Motor	37	2	75	12	6	6	--	--	1500	900	92.5	900 RMP Motor and Gear Box in Vert. Stack
7	NH ₃ Feed Motor	2.2	3	10.5	2.3	1.9	1.4	--	--	250	1760		Class 1, Div. 1, Group D.
8	NH ₃ Rec Motor	0.3	3	9.4	3.9	2.5	1.8	--	--	25	1170		Class 1, Div. 1, Group D.

*This flow rate has been adjusted upwards by 2.3% to reflect corrected heat transfer data (since Conceptual Design Review)

Table 2-11. Equipment List 12.5 MW_{Net} Power Module (Continued)

Item	Component	Weight (K lbs)	No. Req'd	Unit Cost (K\$)	Dimensions (ft)			Design Information
	<u>H.E.</u>				<u>Dia</u>	<u>L</u>		
9	Evap	500	2	2,138	17.6	45.5		2.0" Dia TI tubes (5307)
10	Cond	500	2	2,638	20.7	43		2.0" Dia TI tubes (7221)
11	Turbines/Generator	220	1	1,700	<u>H</u> 11	<u>W</u> 8	<u>L</u> 44	2 Turbines drive one generator RPM/1800
12	Lube and seal oil console	40 (wet)	1	175	10 16	14 14	32 22	Dimensions depend on arrangement
	<u>NH₃ valves</u>				<u>Dia (Inches)</u>			
13	V ₁	10.3	4	8.9	51			Butterfly valve
14	V ₂	2.4	8	1.0	12			Butterfly valve
15	V ₃	2.4	2	4.2	12			Check valve
16	V ₄	3.6	2	6.3	18			Control Valve
17	V ₅	3.6	2	6.3	18			Check valve

Table 2-11. Equipment List 12.5 MW_{Net} Power Module (Continued)

Item	Component	Weight (K lbs)	No. Req'd	Unit Cost (K\$)	Dimensions (ft)		Design Information	
	<u>NH₃ valves (cont)</u>				<u>Dia (Inches)</u>			
18	V ₆	3.6	8	1.6	18		Butterfly valve	
19	V ₇	4.4	2	7.7	22		Control valve	
20								
					<u>Dia</u>	<u>L</u>		
21	Evap. Sumps	6.9	2	10.4	9	12	0.5 inch thick steel	
22	Con Sumps	6.9	2	10.4	9	12	0.5 inch thick steel	
23	15 Mil Screen filters	4	2	6	5	6	at \$2/dia. in. ft	
	<u>NH₃ pipes</u>				<u>Length</u>		<u>Lb ft</u>	<u>Wall thickness (in)</u>
24	84 inches dia	6.4	1 lot	2.4	14.4		445.9 carbon steel	0.5
25	51 inches dia	12.1	1 lot	4.6	45		269.7 carbon steel	0.5
26	22 inches dia	9.7	1 lot	4.9	112		86.44	0.375 ----

Table 2-11. Equipment List 12.5 MW_{Net} Power Module (Continued)

Item	Component	Weight (K lbs)	No. Req'd	Unit Cost (\$)	Dimensions (ft)	at \$2/dia. in. ft Design Information			
	<u>NH₃ pipes (Cont)</u>				<u>Length</u>	<u>Lb ft</u>	<u>Material</u>	<u>Wall Thickness (in.)</u>	<u>Schedule</u>
27	24 inches dia	7.9		4.1	169	45.56	Carbon steel	0.375	20-Std
28	18 inches	11.1		5.7	157	70.59	Carbon steel	0.375	Std
29									
30	Chloropack	0.8	1/2	180	Negligible				
31	Tube cleaners	0.6	1/2	330	2 x 2 x 45				
32	N ₂ purge system	266	1/4	422	48 x 56 x 34				
33	NH ₃ support system	1840	1/4	725	40 x 56 x 128	Dimensions for NH ₃ support and hardware			
34	NH ₃ system hardware	80	1/4	63		Connecting valves, pipes, regulators, etc.			
35	Instrumentation and control								
36	Turbine control								

Table 2-11. Equipment List 12.5 MW_{Net} Power Module (Continued)

Item	Component	Weight (K lbs)	No. Req'd	Unit Cost (K\$)	Dimensions (ft)			Design Information
					H	W	D	
37	Station service transformer No. 1	55	1/2	89	14.2	9.6	9.6	13.8 KV to 4360/2520 V, 12000 KVA, type OA
38	Station service transformer No. 2	12	1/2	13	7.9	5.7	6.25	13.8 KV to 480 V, 1500 KVA, type OA
39	Switch gear box	195	1/2	98	17	4.0	3.5	Includes exiter cabinet
40	Emergency bearing oil pumps	10	1/2	20	Space - Nil			
41	Motor control centers	4	1/2	18	7.5	10	2	
42	Motor control centers	3.5	1/2	15	7.5	10	2	
43	Generator main bus lead	10	1/2	41	3.5	2.5	100 +	
44	Low voltage power control ctr	10	1/2	30	7.5	11.67	4	
45	Hook-up cable and cable ladder	20	1/2	40				
46	Ammonia Inventory	230		19				Operational inventory of power module only.

2.3.5 Critical Interfaces

The critical interfaces of a 100 MWe power plant, including 8 power modules of 12.5 MWe each, two ammonia support systems, and two power centers, are shown in Figure 2-21. Interfaces shown include containment, hull penetration, and water trough dimensions. In addition, the dimensional configuration of the cold water supply pipe and warm and cold water discharge pipes are shown establishing pressure drop calculations utilized in developing the baseline system (see Section 2.3.3).

The estimated weight of a 100 MWe power plant, excluding ship systems, is as follows:

<u>Item</u>	<u>Weight (LT)</u>	<u>No. Required Per 100 MWe</u>	<u>Total Weight Per 100 MWe (LT)</u>
Power Module (12.5 MWe)	1,373	8	10,984
Ammonia Support	1,021	2	2,043
Electrical Power Center	286	2	572
Biofouling	3	4	<u>12</u>
		TOTAL	13,608

Figure 2-21. 100 MWe Power Plant Critical Interfaces

OTEC POWER SYSTEM DEVELOPMENT		TRW SYSTEM GROUP ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY
100 MWe POWER PLANT CRITICAL INTERFACES		
DATE 1-13-78		DESIG. NO. 6850.38-57
DRAWN R. COREY		
CHECKED		
APPROVED R. G. PEARSON	APPROVED - PROGRAM OFFICE	

3. TRADE STUDIES AND ANALYSES

3.1 HEAT EXCHANGER

3.1.1 Design/Productibility Summary

Although the conceptual forms of the OTEC evaporator and condenser, Figures 3-1 and 3-2 respectively, are of similar design, each has its unique features. Both designs are vertical units using minimum hull area and incorporating a fixed-tubes sheet, non-removable tube bundle with enhanced titanium tubes sealed to the carbon steel tube sheet. The remainder of the units are of all-carbon steel construction; protective, non-metallic coatings will be applied to the water boxes and tube sheets to inhibit corrosion. The inlet water box is an open (rapid access) channel-type header to permit in situ mechanical defouling. The water enters via a flared, venturi-type inlet nozzle to reduce inlet pressure drop and to enhance water side distribution. The exit water box is a straight flow (no bends), truncated cone that converges the flow to the exit pipe, thus minimizing water exit pressure drop. The tube bundle in both the evaporator and condenser does not fill the entire shell. A vapor belt surrounds each tube bundle to permit uninhibited flow of the vapor from (evaporator) or to (condenser) all parts of the tube bundle and the entering or exiting vapor nozzle. A rodded bundle may be inserted within this annular space at the vapor entrance in the condenser to break and disperse the entering fluid and to prevent impingement damage to the tubes. A similar rod bundle and a splash baffle can be inserted at the vapor exiting nozzle in the evaporator to ensure dry exiting ammonia vapor if, indeed, a demister is required.

The performances of both the condenser and the evaporator are independent of the degree of modularization (i.e., size of heat exchanger). The current size-to-cost evaluations which were limited to a single fabricator (by schedule) result in an optimum heat exchanger diameter of eighteen feet. It is anticipated that larger diameter heat exchangers will contribute to power plant cost reductions.

The evaporator contains an ammonia distributing chamber on the shell side to distribute the liquid ammonia to the tubes individually for optimum evaporator performance. The distribution chamber in evaporators larger

Figure 3-1. Evaporator

Figure 3-2. Condenser

than eighteen feet in diameter will contain liquid distributing lanes to assure proper liquid ammonia penetration of the tube bundle. The condenser will contain purging nozzles to remove non-condensables from dead areas and from positions at the end of the vapor train. With a triangular pitch and tube pitch-to-diameter ratio of 1.25, the pressure drop of the vapor penetrating the tube bundle in both the evaporator and the condenser is small, on the order of 0.045 psi or less for a 50-foot diameter bundle; thus, larger tube pitch-to-diameter ratios, square pitches, or other special provisions such as the inclusion of vapor lanes to reduce pressure drop and enhance shell side distribution, are not required.

The tubes for both units have enhancement on both the ammonia and water sides. This enhancement consists of flutes fabricated by upsetting-rolling. Corrosion data indicate titanium has a greater than thirty year life in the environment of the OTEC system, whereas aluminum has a life uncertainty. The high life expectancy of titanium, coupled with its lower tube wall thickness, makes for a conservative, viable design choice.

Extended studies were conducted to determine to what extent enhanced heat transfer devices would strengthen the performance of the OTEC heat exchangers and which enhancing devices were most suitable. Preliminary studies indicated the OTEC heat exchangers are predominately rate limited, as opposed to temperature difference limited and, as such, their performance is greatly improved by introducing enhancing devices. Many types of enhancing devices were considered, including finned, spiral, and fluted tubes. These devices were all compared as individual tubes, to a common basis, with the performance of smooth tubes, and also as a grouping of tubes in a tube bundle. The comparisons considered heat exchanger costs, increased heat transfer, and parasitic power requirements within the OTEC system.

After the initial study, only those tubes which furnished increased heat transfer without increased parasitic power requirements over that of a smooth tube were considered. Cost ratios (the cost per foot of an enhanced tube to the cost per foot of a smooth tube) were used for the comparisons. Within the scope of the enhancements investigated, enhanced tubes were more economical than smooth tubes for all degrees of enhancement when the cost ratio is 1.6 or less. The results of these investigations proved that fluted titanium tubes are superior. Present day technology attributes

a 1.54 cost ratio for upsetting-rolling titanium tubes into the fluted shape. Furthermore, there is reason to believe that in the quantities required by OTEC, and with the introduction of improved and diverse (and not yet evaluated) technologies, this cost ratio will decrease.

Both heat exchangers were investigated to determine the influence of mal-distribution on the water side and both heat exchangers were found to be insensitive to this influence. As already stated, vapor bundle penetration and any associated mal-distribution on the shell side resulting from poor bundle penetration is not a problem in either the evaporator or condenser.

The thermal equations used for both evaporation and condensation and the thermal and pressure drop equations used on the seawater side for fluted tubes were based on data developed by experimentation in the heat transfer laboratories of Carnegie-Mellon University. More recent experimental condensing data on fluted tubes has been developed by Oak Ridge National Laboratory. The Oak Ridge data show much higher heat transfer coefficients than the Carnegie-Mellon data. Much of the conceptual design was completed using the Carnegie-Mellon data; however, verification of the Oak Ridge data has led to its use for the baseline design.

The TRW OTEC design uses a thin film evaporation process in which vapor forms on the surface of the liquid ammonia. This is in contrast to a boiling process where bubbles of vapor form on the heat transfer surface and then rise and escape from the liquid surface. Because the thin film evaporation process does not expel liquid droplets into the surrounding vapor, the rising vapor contains no moisture. Velocities of the vapor at the tube bundle periphery, where it is greatest, are less than five feet/second at the construction point between tubes. This velocity is too low to abrade or remove liquid particles from the tubes. The lack of moisture in the exiting vapor has been verified by observations at the Carnegie-Mellon heat transfer laboratory and during tests at Argonne National Laboratory. Moisture removal devices, such as demisters, are therefore unnecessary in the TRW-OTEC system.

Inquiry specifications were prepared at the onset of this conceptual investigation to determine: 1) preliminary cost data, 2) relative costs of the different components of the heat exchangers (to identify high-yield

cost reduction design targets), and 3) potential difficulties and problem areas. The results of this investigation indicated that non-standard design approaches to tube sheet and tube-to-tube sheet joining will provide the greatest opportunity for further cost reduction. These approaches may require consideration of unorthodox design procedures as set forth in prevailing design codes, such as the A.S.M.E. Unfired Pressure Vessel Code. Tube materials and forms, as a basis for high-yield cost reductions, were also considered and, as stated above, upsetting-rolling forming of fluted titanium tubes was chosen.

The present design is based on a single steel tube sheet with a cladding (Figure 3-3).^{*} Among the more prominent alternative tube sheet designs currently under consideration are sandwich combined metallic, non-metallic, with punched (thus reducing high drilling costs) tube holes or double tube sheets, possibly also punched, which have a high moment of inertia (using the same principle as used in structural "I" beams).

The design of the tube-to-tube sheet joint is based on light rolling and seal welding. This approach is highly labor intensive in nondedicated facilities. Alternative approaches being considered are special rolling/welding tools, rolled-only joints, and explosive bonded joints. Whatever approach proves reliable for the 30-year life requirement will contribute significant cost reductions to production procurements.

The cost investigations also indicate that the most economical heat exchanger is the largest heat exchanger which can be fabricated on a river or sea port in a dedicated facility. The added costs required for tooling necessary to build heat exchangers larger than currently possible in vendor's shops (about 35 feet) can only be justified for quantity production. Additional tooling will also be required for in situ construction or assembly of large heat exchangers on the floating hull. Quantity production of OTEC plants will probably justify the creation of an entire facility to build the hull and the heat exchanger at a single location.

The design analysis and producibility evaluations discussed above have involved a conceptual heat exchanger design that fits into present fabrication limitations and utilizes existing tools and fixtures. Although

^{*}Cost projections assume technology developments to eliminate requirement for cladding of tube sheets and welding of tubes.

Figure 3-3. Titanium Tubed Heat Exchanger Tube/Tube Sheet Detail

these constraints result in a higher cost per unit of heat transfer than can ultimately be realized through design and fabrication improvements, they provide a firm base for evaluating the upper limits of OTEC power costs and identification of potentials for technology development.

Tables 2-9 and 2-10 display a cost breakdown for the evaporator and condenser. Appendices A, B, and D provide supporting rationale and detailed discussions of the design and producibility activities as prepared by TRW, CMU, and C. F. Braun.

3.1.2 Materials and Processes

During the conceptual design phase, materials and processes evaluations focused on selection of the heat exchangers, materials and fabrication processes. Because of the proportionately high cost of the heat exchanger tubes and tube-to-tube sheet attachment, these evaluations also concentrated on selection of optimum cost tube materials. The performance of the vertical tube falling film concepts is highly dependent upon ammonia side flutes, therefore the tube materials evaluation included in-depth investigations into production methods and costs for the required tube profiles. The recommended design materials are listed in Table 3-1.

Initial tube material assessments concentrated on aluminum because of its quality of extrudability. It had been demonstrated that flutes of the desired configuration could be formed on both the external and internal surfaces of aluminum tube made of 6063 alloy. Further, early studies favor aluminum because of its relatively low initial cost. A problem with the corrosion processes of aluminum in sea water and the requirement for a 30-year plant life precipitated extensive investigations into aluminum corrosion protection, as well as alternative materials.

Of the alternative materials, several candidates were chosen, including aluminum 5052 and SJM 620 lead because of their extrusion forming potential. Initial concern with the extrudability of aluminum 5052 was removed by Alcoa's recent development work in this area. Most other materials were considered unsuitable, except as smooth inner liners of bimetallic tubes, due to the absence of known technology to enhance the tube surfaces with flutes of the desired profile (see Section 3.1.1). A process of up-setting/rolling tubes into fluted profiles was uncovered at Grob Manufacturing. This process is similar to that for producing gear teeth and is adaptable

Table 3-1. Recommended Design Materials

Tubes	Titanium Tubes, Fluted By Upsetting/Rolling, 0.028 Inch Wall x 2.00 Inch Diameter*
Tube Sheet**	SA 516, GR 70 Steel (Clad With Titanium Sheet Using Explosive Bonding if welding proves required).
Shell**	SA 516, GR 70
Water Boxes*	SA 516, GR 70
Internal Structure*	SA 516; AS 283; SA 285, GR B
Ammonia System Components	Steel Or CRES (Specific Composition Dependent On Availability)
Water System Components	Steel or CRES (Specific Composition Dependent On Availability), Aluminum Bronze, Glass Fiber Reinforced Plastic (Candidates).

*Final thickness and diameter will be determined during preliminary design.

**No anticipated compatibility problems with ammonia. Surfaces exposed to seawater will be coated.

to almost any material in the desired profiles. With this process capability, materials such as Titanium 50A, Allegheny 6X corrosion resistant steel, and 706 copper-nickel were included in the evaluation process.

In addition to exploring other materials, methods of protecting aluminum from corrosion were investigated. These processes include aluminum cladding, zinc, and a proprietary process of Marston Excelsior. The compatibility of these coatings with the cleaning (bio-fouling removal) mechanism, and their ability to be applied to the desired internal tube profile, were major considerations in the evaluation of the coatings, as was their ability to protect aluminum in the seawater environment.

As indicated by Table 3-1, the recommended material for tubes is titanium formed by upsetting/rolling, with an 0.028-inch wall. This selection was based on the relative cost of producing titanium rolled heat exchangers of an expected life of 30 years, versus producing and replacing the tubes (once) of aluminum tubed heat exchangers of an expected life of 15 years. (See Section 5.8 for discussion of the present-value analysis of alternative heat exchangers.) Other materials and processes have also been

evaluated in terms of their compatibility with the economic goals of OTEC and consistent with the design requirements. In-depth discussions of these evaluations and the supporting data are contained in Appendix H.

3.1.3 Biofouling Control

There is a specific requirement of the project to maintain the thermal resistance due to fouling at a value of $0.0005 \text{ hr-ft}^2 \text{ }^\circ\text{F/Btu}$ or less. Furthermore this fouling resistance affects the thermal performance of the heat exchangers (Figure 3-4) necessitating increased heat transfer surface area for a given thermal duty (i.e., the higher the fouling factor, the greater the amount of surface area required to effect a specific heat transfer rate). It is therefore critical to the economic evaluations of heat exchangers design to assess the costs of maintaining a low fouling resistance (i.e., power required for cleaning mechanisms) versus the cost for added heat exchanger surface. These tradeoff studies, along with determinations of the process of fouling buildup and removal, have formed the essence of the biofouling control conceptual design activities.

The conclusions of the studies have led to an open head heat exchanger which allows continuous access to the waterside of the tubes to effect mechanical cleaning operations. A design has been developed using the concept of multiple spindle drilling units to periodically clean each tube of the exchanger (for analysis, the cleaning frequency is anticipated to be once a week). This design is a modification of existing tube cleaning technology used by industry and will evolve to a numerically controlled multiple-head device moved by bridge crane from section to section and between exchangers (Figure 3-5). In addition to the mechanical cleaning process, a low dose rate of chlorination (about 0.05 PPM chlorine) will be continuously injected into the water supply. This added prevention is to decrease the buildup rate of biomass in the tubes and to provide some protection (and less frequent maintenance times) for portions of the system not cleaned (e.g., tube sheet, ducts, pumps). The combination of these two processes is anticipated to yield a nominal fouling resistance of $0.0001 \text{ hr-ft}^2 \text{ }^\circ\text{F/Btu}$ for the heat exchangers (based upon DOE tests to date). A detailed discussion of the tradeoff studies is given in Appendix C.

Figure 3-4. Effects of Fouling on Heat Transfer

Figure 3-5. Tube Cleaning Machine

3.2 MACHINERY AND SUBSYSTEMS

3.2.1 Turbine-Generator

The turbine-generator requirements for a 12.5 MWe deliverable power output module are summarized in Table 3-2. Various options were analyzed to achieve the goal of optimum efficiency of the turbine-generator set. The design selected has the following characteristics:

- Axial flow
- Multiple stage
- Proven technology from existing expanders and ammonia compressors
- Low-pressure drop variable stator vanes for speed control

The efficiency of axial turbines can be maximized by the proper selection of rotation speed and the number of stages. (In the size range being considered, this is essentially independent of size.) This will yield a specific speed (N_s) where maximum efficiency can be achieved as shown in Figure 3-6. Examples of various parameter combinations yielding high efficiency for a given power output (e.g., 25 MWe) are shown. In addition to the thermodynamic efficiency shown in the figure, turbine-generator set efficiency must also account for the following losses: in the bearings, turbine wheel windage, control vanes, and generator windage (which can be minimized by hydrogen cooling for a net gain of approximately 0.1 to 0.2%). These degradations from the ideal isentropic efficiency result in a projected efficiency of >85%. Axial turbines offer other advantages, including an excellent reliability record and a large background of experience because no other type of turbine is being built in the desired size. Table 3-3 shows the design characteristics of a typical 6.25 MWe building block turbine. Tables 3-4 and 3-5 list a few of the expanders and compressors that approach the sizes required and that incorporate technology applicable to an ammonia turbine suitable for an OTEC power system. For the conceptual turbine configuration selected, i.e., a single flow, multi-stage, axial between-the-bearings design with variable stator vanes for control, the following cost effects were considered:

- Overhung turbines cost slightly less than between-the-bearings units.

WHERE:

N = SPEED, RPM

Q' = ACFM AT WHEEL EXIT

H = ISENTROPIC HEAD, FT-LBS/LB

Figure 3-6. Turbine Performance

- Side nozzles (which would be required with the concept adopted) cost more with overhung turbines.
- While additional stages slightly increase the cost, this is more than balanced by the ability to increase efficiency with a multi-stage design.
- Single flow turbines are less expensive than double flow of the same capacity (See Figure 3-7).
- Although the total cost increases as the turbine size decreases, overall power system cost was optimized with the 6.25 MWe turbine.
- The variable stator vanes increase the efficiency 2 to 3% over the use of a throttling valve when operating at off-design-point conditions (when seawater temperature difference is greater or less than 40 F). Over the 30 years life, this translates into a \$135,000 present value savings over the cost of the vanes for each turbine.

Table 3-2. Turbine-Generator Requirements

- Output = 16.51 MWe (gross) at 60 Hz
- Input = 3,417,800 lb/hr ammonia vapor at 70°F and >99% quality inlet; 50°F outlet at >96.5 quality
- Maximize overall efficiency considering modularity options
- Provision for control during input variation
- Use existing technology
- 30 year life
- Base loaded duty cycle

Turbine components using available technology are being considered. Figures 3-8, 3-9, and 3-10 illustrate commercially used stator vanes for control; multi-stage, between-the-bearings turbine, similar to turbines contemplated for OTEC; and bearings and seals suitable for use with ammonia.

Because maximum use of the ocean's available heat energy will minimize the power system cost, ways to accomplish this with the turbine have been considered. Specifically, utilization of the variation in ammonia

Table 3-3. A Typical 6.25 MWe Turbine's Parameters

- Gross Power Output - 8.25 MWe (generator)
- Multi-Stage, 3600 rpm
- Specific Speed - 108
- Efficiency - ~85%
- Fluid Conditions (NH_3)
 - Inlet 70°F (128.8 psia) at >99% quality
 - Outlet 50°F (89.2 psia) at >96.5% quality
 - Flow 1,708,900 lb/hr
- Materials
 - Bearings - steel backed babbitt
 - Shaft - SAE 4140 or equivalent
 - Wheels and Blades - low alloy steel (e.g., A-286)
 - Casing and Bearing Housing - cast or fabricated carbon steel

Table 3-4. Hot Gas Expanders

User	Location	Date	Supplier	Inlet Temp (°F)	Horsepower	Speed (rpm)
Shell	Canada	1964	Elliott	1200	3,500	7,000
Shell	Norco, Louisiana	1965	I-R	1100	9,100	3,500
Gulf	Alliarce, Louisiana	1971	Elliott	1250	22,300	3,000
Sun	Toledo, Ohio	1973	I-R	1250	21,300	3,000
Mobil	Beaumont, Texas	1977	I-R	1285	20,600	3,000
Diamond Shamrock	Amarillo, Texas	1978	I-R	1300	10,900	6,070
Phillips	Sweeny, Texas	1979	Elliott	1300	35,000	3,000
Sun	Marcus Hook, PA	1979	I-R	1315	21,200	3,000
Shell	Martinez, California	1968	I-R	1200	16,800	3,000
Gulf	Edmonton, Alberta	1970	I-R	1270	8,200	4,350
Mobile	Beaumont, Texas	1968	Elliott	1250	9,600	10,040
Esso Chemie	Holland	1968	I-R	1130	14,520	6,550

Table 3-5. Centrifugal Ammonia Compressors

User	Location	Date	Supplier	Model	Horsepower	Speed (rpm)
Federation Chemicals	Trinidad	1966	Clark	2M10-9	8040	10,700
Collier	Brea, California	1967	Clark	2M8-6	4670	10,840
Mobil Chemical	Beaumont, Texas	1967	Clark	2M10-8	3270	10,800
Esso Chemie	Holland	1968	GHH/Elliott	8RM-9K31/29MC-7	590/5350	8720/ 11,900
First Miss (was Sinclair)	Clinton, Iowa	1969	Clark	2M7-8, 2MC-7	8850	10,240
Asahi	Japan	1972	Clark	2M7, 2MC-6	4900	10,200
NSM	Holland	1972	Nuobo Pignone	MCL457, 2MCL456	9000	4,900
Collier	Kenai, Alaska	1977	Elliott	38M6-5, 38M9-8	8720	8,000

Figure 3-7. Turbine-Generator Cost

Figure 3-8. Variable Stator Actuating Mechanism

Figure 3-9. Multi-Stage Between the Bearings Turbine

Turbine Tilting Shoe Bearing

Turbine Iso-Carbon Shaft Seal
Figure 3-10.

temperature and pressure, and mass flow rate due to water temperature variations were addressed.*

There are various ways to optimize turbine efficiencies for variable inlet conditions (multiple staging, reaction staging, variable stator vanes, etc.). During the preliminary design phase of this program, a performance analysis will be conducted to establish the most cost effective approach.

Conceptually, the use of variable stator vanes can provide two advantageous effects, variable nozzle angle and variable nozzle area. First, for an axial, impulse, full admission turbine, the ideal maximum efficiency for a stage occurs at a velocity ratio (turbine pitch line velocity, U , $\div$ nozzle exit velocity, C_0) of 0.5. Considering realistic blade turning angles, 0.52 is a better value. Since the turbine is operating at constant speed and the turbine rotor blading geometry is fixed, any increase or decrease in the nozzle exit velocity will result in a reduction in turbine efficiency. By changing the nozzle vane angle, and the resultant velocity vector, this decrease in turbine performance can be minimized.

The nozzle exit velocity is primarily a function of the square root of the enthalpy drop across the turbine. For $\pm 30^\circ\text{F}$ temperature change, the enthalpy change is ± 0.5 Btu/lb, which changes C_0 only ± 1.25 percent. Hence, the estimated impact on turbine static efficiency of a change in velocity ratio of this magnitude is comparatively small (see Figure 3-11).

However, the more significant impact on turbine performance results from the change in ammonia mass flow rate. An example of the effect of varying mass flow is shown in Figure 3-12. Variable vanes can provide variable nozzle areas which can accommodate the increased flow without the efficiency penalty. As can be noted, a turbine efficiency loss of 1.5 percent would result without using a variable-area nozzle ring with pivoted vanes. While this application is for a radial turbine, the effect is valid for axial turbines also.

*For the OTEC ammonia-driven turbine, variations in the sea water surface temperature will result in variations in ammonia inlet conditions to the turbine; typically $70^\circ\text{F} \pm 30^\circ\text{F}$ temperature, 128.8 psia ± 6.7 psi pressure, and ± 15 percent mass flow rate.

Figure 3-11. Estimated Effect of Velocity Ratio Change on Turbine Static Efficiency

Figure 3-12. Estimated Effect of Ammonia Flow Rate Change on Turbine Static Efficiency

Although variable stator vanes are not employed in conventional steam turbo-generator plants (most operate with constant turbine inlet conditions), the technology associated with their design and operation is within the current state-of-art. Figure 3-13 shows a schematic of currently used radial flow turbine variable area inlet vanes. Figure 3-14 is a photograph of the installation of variable stator vanes on an axial compressor.

All that is required to ensure satisfactory turbine performance is that the turbine blading be designed to assure that no destabilizing aerodynamic conditions are encountered (i.e., excessive blading angle-of-attack) over the range of variations in turbine inlet conditions for the possible vane positions. Mechanical design solutions for applying them to an OTEC turbine are considered by manufacturers to be straightforward.

As a result of system optimization analyses, two 6.25 MWe turbines connected to a 12.5 MWe generator was selected as the 12.5 MWe powerplant module. Figure 3-15 shows some of the physical characteristics used in tradeoffs performed prior to the selection of the preferred module size. Two 6.25 MWe turbines with generator have a greater volume and are heavier than a single 12.5 MWe unit, but have such compensating features as:

- Higher efficiency at part load
- Easier maintenance
- Less data extrapolation because the power module will be the same size as the test article.
- Less power output loss if one unit is not operating.

Figure 3-16 illustrates the effect of a unit failure on power module availability based on a 12.5 MWe system. For example, if a single turbine is used, its failure would mean a loss of 12.5 MWe (or 100%) of the system; with a 6.25 turbine, only 50% of the power is lost.

A hydrogen cooled AC generator is planned to maximize efficiency. This is advisable even if DC power transmission is considered; if necessary, higher efficiency is obtained in the voltage step-up process, with AC power and rectification than with DC power generation at a low voltage with a step-up stage which must include inverters. Except for special applications requiring limited amounts of power, the large majority of power generation in the U.S. is 60 Hz AC.

Figure 3-13. Variable Area Nozzle Arrangements With Pivoted Inlet Vanes for Radial Inward Flow Turbines

Figure 3-14. Axial Compressor With Variable Stator Vanes

MODULE SIZE, MWe	A	B	C	D	E	F	L	WEIGHT LB.
25	10	10	20	13	13	9	50	310,000
12.5	10	10	18	11	11	8	44	220,000
6.25	8	9	14	10	10	8	38	170,000

(SIZES IN FEET)

- 3600 RPM UNITS WITH SINGLE FLOW, SINGLE STAGE TURBINES
- 1800 RPM UNITS ~ 2 FT SHORTER, 1-2 FT LARGER IN HEIGHT AND WIDTH

Figure 3-15. Modularity Size Effect

Figure 3-16. Modularity Performance Curve

The advantages and disadvantages of DC generation compared to AC are summarized below:

- Advantages:

- Makes possible direct utilization by such users as electrochemical industries and vehicular propulsion (although conversion or rectification from an AC supply is more common).

- Disadvantages:

- Higher acquisition cost for generating equipment and for most utilization equipment
- Higher losses
- Higher maintenance cost
- Lower reliability
- High transmission cost, except for long distance and at high voltage
- DC generators are not usually practical in single units in sizes above 5000 KW and at voltages above 3000.

3.2.2 Seawater Pumps and Drives

3.2.2.1 Seawater Pump Requirements

OTEC seawater pump requirements are as follows:

- System Design. For a 12.5 MWe module, the requirements are 763,600 gpm cold seawater flow with a system pressure drop of 11.2 feet (nearly all static head), and 716,200 gpm warm seawater flow with system pressure drop of 8.2 feet (nearly all static head).
- High Efficiency. The seawater pumps constitute approximately 75% of total parasitic load. Therefore, the major effort was to maximize the overall efficiency of the pumping system consistent with the OTEC system requirements. Pump efficiency, including losses from pump suction bell to discharge elbow, should be $\geq 87\%$.
- Compatibility with Ocean Environment. The pumps will be operating continuously in an ocean environment for 30 years with up to 1000 starts and stops. At times they will operate in a sea-state-six condition, which would result in a total dynamic head variation of up to 30% oscillating every 15 seconds.

3.2.2.2 Seawater Pump Type and Speed Selection

The seawater pump selection rationale was based on the analysis and evaluation of performance, physical arrangement, and cost.

The number of pumps required depends on the number of heat exchangers, the physical arrangement of pumps and pipes, maintainability-reliability-manufacturability considerations, and start-up power considerations.

The desirable number of pumps would equal the number of heat exchangers; this eliminates complicated and expensive manifolding and valving arrangements. Although the use of two pumps per heat exchanger could be achieved without extensive manifolding and valving (by dividing the water-box above the tubesheet into two compartments), the use of one pump for more than one heat exchanger is not desirable because it would require diversion of large quantities of water when one heat exchanger is shut down for maintenance.

Because 1, 2, 4, or 8 evaporators and condensers were considered for a 25 MWe power module, the same number of pumps were evaluated with warm water flows ranging from 170,050 gpm to 1,400,400 gpm at 8.2 feet of total dynamic head (TDH); and 190,900 to 1,527,200 gpm cold seawater flows at 11.2 ft TDH.

Our survey of manufacturers indicated that for the range of flow up to 765,000 gpm, the type of pump, and the associated TDH given above, there are no economies of scale. The estimated cost of pumps is approximately one dollar per gpm for the first 10 units; the price decreases to approximately 85 cents per gpm in larger quantities. These data were derived from four sources.

In evaluating the pump efficiency, the Typical Pump Specific Speed (N_s) versus Efficiency (η) curve in Figure 3-17 illustrates two important pump characteristics: the maximum allowable N_s for a minimum desirable pump η , and the relationship of the type of pump and impeller shape for a given specific speed.

Figure 3-17 Typical Pump Specific Speed Versus Efficiency

The specific speed is a useful parameter because it has a definite value for a pump of a given design operating at maximum efficiency. The specific speed is defined as

$$N_s = \frac{[N(\text{pump rpm})][(Q(\text{gpm}))^{0.5}]}{[H(\text{head in ft})]^{0.75}}$$

The pump system head is determined by the system configuration (heat exchanger/hull relationship); therefore, the variables that affect N_s are the pump flow rate, Q , and pump speed, N .

With the module concept of two condensers and two evaporators per 12.5 MWe module, the flow rate required per condenser is 381,800 gpm, with a system pressure drop of 11.2 feet. The flow rate required per evaporator is 358,100 gpm, with a system pressure drop of 8.2 feet. Because it is desirable to use one pump per heat exchanger, N_s of the pump would essentially depend on the speed of the pump.

The higher the speed of the pump, the smaller and less expensive it is, and the higher the N_s . However, there is a direct relationship between pump N_s and pump η : the pump η is maximum at approximately $N_s = 3000$, which results in a cold seawater pump speed of 31 rpm. Pumps operating at such low speeds are very large and expensive. Furthermore, when the Reynold's number in certain sections of the pump drops below a particular critical range, viscous effects begin to play a dominant role and alter the flow pattern, which increases the friction factor and reduces efficiency.

Both good engineering practice and economic considerations therefore require the selection of a pump with a high N_s and a pump efficiency of 85 to 90 percent. For this reason, a maximum specific speed for a seawater pump of 14,000 was selected.

As Figure 3-17 indicates, the type of pump to achieve an $N_s = 14,000$ is an axial (propeller type) impeller, with a projected efficiency of approximately 87%. A survey of manufacturers' data of vertical wet pit propeller pumps indicated a maximum efficiency point of 87% for large pumps at this specific speed to be entirely reasonable.

For the noted flows and pressure drops, combined with $N_s = 14,000$, the warm seawater pump speed will be 129 rpm and the cold seawater pump speed will be 139 rpm.

Seawater Pump Configurations

As discussed above, the flow regime of seawater pumps leads directly to the selection of axial flow-type pumps.

Three axial flow type pump configurations (Figure 3-18) were considered:

- Vertical wet pit propeller pump
- Horizontal, external drive
- Self-contained bulb

- VERTICAL PIT TYPE (PREFERRED CONFIGURATION)
 - COMPACT PHYSICAL ARRANGEMENT
 - CLOSED SHAFT
 - FILTERED SEAWATER LUBRICATED BEARINGS

- HORIZONTAL EXTERNAL DRIVE TYPE
 - REQUIRES GREATER SUBMERGENCE FOR FLOATING BARGE VESSEL (INCONSISTENT WITH CONCEPTS STUDIED)

- SELF-CONTAINED BULB TYPE
 - REQUIRES GREATER SUBMERGENCE
 - REQUIRES LARGER SEAWATER CONDUITS
 - UNSUITABLE PHYSICAL ARRANGEMENT FOR CONCEPTS STUDIED

Figure 3-18. Seawater Pump Configurations

The large, self-contained bulb-type axial pumps are unsuitable for the compact physical arrangement required by the baseline ship platform module since they require large seawater conduits and a greater submergence than the vertical wet pit pumps. The external drive type pumps also require a greater submergence than the wet pit type. Therefore, the seawater pumps picked for the conceptual design of the OTEC 12.5 MWe power module are the vertical wet pit propeller type.

The wet pit propeller type pumps can have either of two shaft arrangements: open shaft lubricated with seawater or a closed shaft with oil, freshwater, or filtered seawater used for the lubrication of the shaft bearings. A closed shaft configuration is preferred because it affords prolonged bearing life. Using lubricating oil was rejected because continuous loss of lubricating liquid into the seawater system would be environmentally unsound, and because oil depositing on the heat exchanger tubes would cause degradation of the heat transfer coefficient. The freshwater option was rejected because it requires the storage of large amounts of fresh water for continuous replenishment. Seawater which is filtered before use and then returned to the pump outlet is preferred for lubricating the shaft bearings.

Other characteristics that influenced the selection of the OTEC seawater pumps were high reliability (an off-the-shelf design); compatibility with the ocean environment (pump motors and drive system components isolated from seawater contact, and a closed shaft arrangement to protect shaft bearings); ease of maintainability/accessibility to minimize repair time; and availability (several manufacturers).

3.2.2.3 Seawater Pump Performance

The seawater pump performance graphs of 3-19 and 3-20 indicate variations of pump flow resulting from variations in the wet pit (seawater trough) water level due to the heave, pitch, and roll of the ship platform.

The pumps will be operating in up to sea-state-six conditions which result in ± 10.5 ft sea level oscillations every 16 seconds due to heave. With proper intake control gates, this variation will be damped to a maximum of ± 1 ft of wet pit surface variation.

Figure 3-19. Cold Seawater Pump Performance

Figure 3-20. Warm Seawater Pump Performance

The graphs indicate the desired typical head/flow characteristics of the pumps, the effect of sump level variation on flow, and the effect on efficiency with typical efficiency characteristics. These data were taken from actual performance curves supplied by manufacturers for typical pumps.

The propeller blade geometry influences the slope of the head/flow curve. A steep curve is preferred because it will result in smaller flow variations with changes in head.

It is also desirable to have a flatter efficiency curve to maintain as high a level of efficiency as possible, when the sump level drops, to minimize power surges of the pump motor. Oversizing the pump often results in a flatter efficiency curve.

Seawater Flow Control

Current system design requirements indicate that the optimum utilization of equipment is achieved with constant seawater flow through the heat exchangers. Therefore, the only control of seawater flow would be by turning the pumps on and off. Because the size of each heat exchanger is 6.25 MWe nominal, power output control of 50 percent increments is possible for a 12.5 MWe nominal power module.

Should it be desirable to control the seawater flow through the heat exchanger to adjust power output due to seasonal surface water temperature variations, three options are available. Each has different motor/drive efficiency, pump system efficiency at full and part load, reliability, and incremental cost (see Table 3-6).

The variable pump speed option could be implemented by DC motors or variable fluid drives. The DC motor provides very high motor/drive efficiency and pump system efficiency at full and part load, a relatively high system reliability, but an incremental cost of \$150,000 per pump. The variable fluid drive arrangement decreases the drive efficiency by 5 percent and is the least reliable option evaluated.

The variable pitch propeller pump option provides very high drive and system efficiencies and high reliability, but is the most costly option.

Table 3-6. Seawater Flow Control

Options	Full Load Drive Efficiency (%)	Pump Efficiency at Normal Flow (%)	Relative Part Load System Efficiency	System Reliability Ranking	Incremental Unit Cost (\$)
Variable Pump Speed					
• DC motor	92	86	Very high	2	150 K
• Variable fluid drive	87	85	Moderate	5	50 K
Variable Pitch Propeller	92	86	High	4	500 K
Bypass Valves and Conduits	92.5	87	Low	3	150 K
Baseline (constant speed induction motor with gear reduction)	92.5	87	-	1	-

Note: For variable flow ($\pm 25\%$ of normal flow), pump efficiencies are optimized for normal flow requirements.

Our survey of manufacturers indicates that the cost of a variable pitch propeller pump could be as high as twice the cost of fixed pitch propeller pump.

The bypass conduit and valve option is as expensive as the DC motor arrangement; it is also very inefficient and less reliable than the DC motor.

Therefore, if a variable seawater flow control is required, the DC motor arrangement is the preferred option to maintain the present high system reliability even though it costs \$100,000 more than the variable fluid drive arrangement.

3.2.2.4 Seawater Pump Drive Requirements

The following requirements were used to select and size electric motors for seawater pump drive.

- Seawater pumps operate at constant speeds; 139 and 129 rpm for the cold and warm water pumps, respectively. Full load in terms of shaft power at pump input is 1272 hp for cold water pumps, 854 hp for warm water pumps. This power is not constant; it varies with dynamic head variations due to wave action. The variations result in calculated load pulsations of up to 30 percent (establishment of these variations is discussed elsewhere in this report). The period of these pulsations is expected to be approximately 15 seconds, and motor rating must be selected to withstand the pulsations without excessive heating. Because the thermal constant of motors is much larger than the period of the pulsations, the RMS value of the required power was used rather than the absolute peak value. The resulting selection of standard NEMA motor sizes is 1500 and 1000 hp for the cold and warm water pumps, respectively.
- The starting torque requirement was selected by similarity with centrifugal pump installations. Ten to twenty percent of full load appears to be typical (however, low-speed vertical pumps with massive Kingsbury type thrust bearing may require forty percent starting torque).
- The standard NEMA synchronous speed selected for the AC motors is 900 rpm.

These requirements are summarized in Table 3-7.

Table 3-7. Seawater Pump Drives Requirements

	COLD WATER	WARM WATER
FULL LOAD – PUMP DRIVE SHAFT HP	1.272	854
LOAD PULSATIONS (DYNAMIC HEAD VARIATIONS), %	30% max	30% max
FULL LOAD SPEED (RPM)	145	135
STARTING TORQUE (% FL)	~20	~20
SELECTION		
STANDARD NEMA SIZE AC MOTOR (HP)	1,500	1,000
STANDARD NEMA SIZE SYNCHRONOUS SPEED (RPM)	900	900
SPEED REDUCING GEAR BOX	6.5	7.0

Seawater Pump Drive Comparison

Three basic types of electric motors can be used to drive the seawater pumps: synchronous, induction, and DC motors. Motors can be the low-speed type for direct drive of the pumps through rigid coupling to the pump shafts, or the high-speed type with a speed-reducing gear box. It is estimated that a gear box will reduce the overall efficiency by approximately 2 percent.

Because of their inherent variable speed capability, DC motors would only be used with direct coupling and would not require a gear box. The main advantage of DC motors is their ability to operate over a wide speed range. Seawater pumps are basically constant-speed devices and are planned to be operated that way so this advantage is of no particular value for this application. The main disadvantage of DC motors is higher cost. High initial cost is due to the fabrication and assembly of commutators and rotor windings, and to the need for a converter to rectify AC power into DC. Maintenance cost is higher because of this increased complexity and wearing elements. The overall efficiency of a DC motor drive is estimated at 92 percent at full load, including the static converter losses. Its cost is estimated at approximately \$250,000. Losses and cost are evenly divided between the motor and the converter.

Induction motors, especially those of the squirrel-cage type, offer the advantages of simple and rugged construction, low maintenance, simple controls, and low overall cost. Based on manufacturer's data, the cost of a standard design motor with gear box for speed reduction is estimated at approximately \$75,000. The cost of a special design for low speed operation is estimated at \$175,000. Induction motors have the disadvantage of high inrush KVA with full voltage start (typically 600% of full load KVA), but this can be alleviated by using reduced voltage start. In the power range of interest, a low speed direct drive full load efficiency is estimated at 92.5 percent, which is also the approximate total efficiency of a high speed motor with gearbox.

Synchronous motors have the advantages of high efficiency, high-power factor, and low KVA inrush capability for low torque applications. The

full load efficiency (including gearbox) is estimated at 93.5 percent. Disadvantages include the need for an exciting source and field control means, which could cause complications even when the motor is of the brushless type, and slightly higher maintenance costs. The cost of a special design motor for low-speed operation is estimated to be approximately the same as for an induction motor, i.e., \$175,000 with an efficiency of 94.5 percent. The cost of a high-speed synchronous motor is estimated to be approximately 30 percent above that of a high-speed induction motor.

The above comparison is summarized in Table 3-8. The drive selected for the seawater pumps consists of an induction motor with a speed reducing gear box. This selection was made primarily on the basis of cost. Additional synchronous motor efficiency does not save enough electricity to balance the capital cost of the electricity generation compared to the increased drive cost.

Use of an Ammonia Turbine as a Seawater Pump Drive

Ammonia turbines were evaluated as potential seawater pump drives. This was done because of the potential increase in overall system efficiency by eliminating the inefficiencies associated with converting power from shaft (the power turbines) to electric and back to shaft (the seawater pumps). Although up to 2% efficiency increase might be achieved, they were determined unattractive compared to electric motor drives and were rejected from further consideration. Factors which contributed to its elimination were:

- The need for a fossil fueled auxiliary evaporator and other support elements necessary to generate the high pressure ammonia vapor to drive the seawater pump turbines during a bootstrap-type startup sequence.
- The need to design the ammonia loop and heat exchangers for higher operating pressures which occur during system startup. Because cold water is not available for condensing at startup, and the "warm" water is used to condense the ammonia, elevating the initial operating pressure range results.
- The added system complexity and leakage potential and attendant degradation in reliability arising from the auxiliary high pressure ammonia vapor generation system. Also required for startup are the controls and valving required to switch over to power module supplied ammonia for steady state operation.

Table 3-8. Seawater Pump Drive Comparison

MOTOR TYPE	MOTOR SPEED	OVERALL EFFICIENCY (%)	COST (\$K)	REMARKS
SYNCHRONOUS	LOW	94.5	175	HIGH POWER FACTOR COMPLEX STARTER REQUIRED FOR LOW KVA INRUSH
	HIGH	93.5	100	
INDUCTION	LOW	92.5	175	RUGGED CONSTRUCTION REDUCED MAINTENANCE SIMPLE CONTROLS
	HIGH	92.5	75	
DC	LOW	92	250	VARIABLE SPEED REQUIRES STATIC CONVERTER HIGH MAINTENANCE

- HIGH-SPEED MOTORS REQUIRE GEAR BOX WHICH IS INCLUDED IN OVERALL EFFICIENCY AND COST
- DC MOTOR COST INCLUDES STATIC CONVERTER

3.2.3 Ammonia Pump Requirements

The power system module requires both ammonia feed and recirculation pumps. The pumps circulate anhydrous ammonia between the condenser and the evaporator. The rate of feed per evaporator is 5470 gpm, and the rate of recirculation per evaporator is 1680 gpm. Each evaporator and condenser is piped to an individual ammonia turbine, constituting an independent ammonia circuit. A single feed and a single recirculation pump is used per evaporator. Their requirements are:

- Feed pump: 5470 gpm at 54.2 psi system pressure, equivalent to 200 feet total dynamic head (TDH).

- Recirculation pump: 1680 gpm at 14.4 psi system pressure, equivalent to 55 feet TDH.
- Pump compliance with the latest revision of American Petroleum Institute (API) 610 specifications.
- Pump material in accordance with API Code S-1.
- Pump efficiencies $\geq 80\%$.
- Pump net positive suction head (NPSH) ≤ 20 feet.

3.2.3.1 Ammonia Pump Configuration and Selection Rationale

For an ammonia feed pump with 5470 gpm flow and 200 feet head, the specific speed $N_s = 2448$ at 1760 rpm pump speed. Similarly, for the recirculation pump with 1680 gpm flow and 55 feet head, the specific speed $N_s = 2374$ at 1170 rpm pump speed. Figure 3-21 shows that for specific speeds $N_s = 2000$ to 3000, pump efficiencies of 80% or better are attainable with centrifugal type impellers.

The American Petroleum Institute recommends a radially split centrifugal pump for pumping liquids with specific gravity of 0.7 or less to minimize potential leaks. Therefore, both the ammonia feed and the ammonia recirculation pumps selected are horizontal, single-stage centrifugal pumps, with radially split case and closed impeller.

Can type vertical pumps with mixed flow impellers and horizontal propeller type pumps were evaluated during the design evolution when system requirements indicated the need for high flow/low head recirculation pumps. For the system concept flow rates and system pressure requirements, however, centrifugal pumps provide the highest efficiencies and the most compact physical arrangement.

Critical issues in the selection of the ammonia pumps are ammonia leaks and pump cavitation. The use of radially split centrifugal pumps will minimize the potential of ammonia leaks. Pump cavitation could be minimized by the selection of pumps requiring low NPSH. The NPSH requirements for the horizontal, single-stage centrifugal pumps are approximately 20 feet. The present physical arrangement of pumps allows the pump to be

Figure 3-21. Typical Pump Specific Speed vs Impeller Shape and Efficiency

located 25 feet below the liquid level of the ammonia in the condenser sump thus avoiding pump cavitation. If during preliminary design, changes require pumps with lower NPSH, they could be provided by selecting larger centrifugal pumps and reducing the pump speed.

3.2.3.2 Ammonia Pump Modularity and Performance

Each evaporator and condenser set is connected to its respective ammonia turbine with separate ammonia loops, thus requiring an individual set of feed and recirculation pumps. One 100% capacity or two 50% capacity feed and recirculation pumps were considered for each set of evaporator and condenser. To improve reliability and minimize cost and space requirements, single 100% capacity feed and recirculation pumps were selected for each evaporator/condenser set with single (redundant) 100% capacity standby feed and recirculation pumps shared between two evaporator/condenser sets.

To control the feed and recirculation rates of ammonia to the evaporator, variable speed pumps or pump bypass piping and valving arrangements are required. Variable speed pumps require either a variable speed fluid coupling between the pump and the motor or direct coupled DC motors, which operate at higher efficiencies at part load. However, the additional cost of each variable speed approach (whether with fluid drive coupling or DC motor) will be more than \$10,000, compared to the bypass piping and valving arrangement. Because OTEC is a base-loaded plant with continuous full-load power generation anticipated, the savings in power consumption of the relatively small motors (225 HP for the feed and 25 HP for the recirculation pump) do not justify the additional \$10,000 per pump capital investment. Therefore, bypass conduits and control valves will be provided.

3.2.4 Ammonia Systems

The ammonia support systems consist of the subsystems and components required to:

- Store ammonia and nitrogen
- Charge the power system with ammonia
- Purge the system with nitrogen after removing and prior to filling with ammonia

- Remove moisture from the ammonia
- Recover ammonia during shutdown
- Prevent and recover ammonia leakage.

The TRW design approach for the ammonia support systems emphasizes low-risk implementation. The techniques and components selected are based on proven technologies and, whenever possible, commercially available equipment with a maximum application of experience and expertise on similar existing systems.

The ammonia storage system was originally sized to provide a capacity of one million pounds, based on a preliminary calculation of the system capacity. It was intended to provide the ammonia required for charging an entire 25 MWe power system module. The module was subsequently broken down in smaller building blocks with independent ammonia loops. The total NH_3 charge of the building blocks required to produce 25 MWe is probably in excess of 1,000,000 lb. The volume will be recalculated during the preliminary design task.* In the meantime, however, the capacity of the storage is useful because only a single module or building block will be charged at a given time; charging the next module can be postponed until the storage system has been refilled. Salient features of the NH_3 storage system are:

- Capability to withstand a pressure of 250 psig (115°F)
- No parasitic power required
- No external insulation on tanks
- Water spray safety for emergency temperature control to prevent overheating of storage tanks.

The nitrogen will be stored in liquid form at 250 psig at approximately -255°F, in 9000 gallon tanks. This will provide 1,024,000 SCF, sufficient for 3 cycles of 5 purge volumes each. Autorefrigeration provides for compact, cryogenic storage, which is less costly than high-pressure storage.

*A recent estimate indicates that approximately 230,000 lb is required to charge each 12.5 MWe power module as indicated in Table 2-11 (p. 2-69).

Figure 3-22 is a schematic of the ammonia charging system. Because commercial ammonia contains up to 0.5 percent water, the system includes a rectifying stage. The evaporator is used as a batch drying stage. The ammonia from the charge storage tank is fed to a sump in batches of about 27,000 lbs each. The commercial ammonia is pumped to the evaporator, the dry vapor goes directly to the condenser via a turbine bypass duct, and the remaining ammonia with about 2 percent water is drained back to the sump. With the transfer pump, the wet ammonia is pumped to the wet heel storage tank. This process continues until the full module capacity has been evaporated, transferred to the condenser, liquified, and held.

The wet heel ammonia is processed through a rectification column using warm seawater as a heat source in the reboiler. Figure 3-23 is a schematic of the rectification system, which is capable of processing ammonia at 1000 lb/hr. The bottoms of the process are ammonia with 25 percent water, which is disposed of as waste. Ninety-four percent of the heel ammonia is recovered with 1.5 percent of the initial load lost (the cost of which is about \$1500). Additional ammonia recovery is uneconomical, e.g., 166 KW would be required to recover an additional 4 percent of the heel.

The water entrained in the system either by slow seepage or introduced by the make-up ammonia is periodically collected from the sump and transferred to the wet heel tank. This stored wet ammonia is also periodically processed through the rectifying column and sent to the charge storage tank. A diagram of the full ammonia storage and recovery system is shown in Figure 3-24.

The nitrogen purging system provides dilution of ammonia vapor prior to opening the ammonia power loop for repair or during shutdown of the power system. Nitrogen dilution of air is also required prior to charging the system with ammonia. A full purging cycle is experienced when the system is opened and then closed. With the system full of ammonia at atmospheric pressure, nitrogen is injected in small increments and the mixture vented until the ammonia is diluted to a concentration below the flammability range when mixed with air. The system may then be flushed with air and opened. After closing the system and prior to the injection

Figure 3-22. Ammonia Systems: Charging and Purifying

Figure 3-23. Ammonia Systems: Rectification

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

Figure 3-24. Ammonia Storage and Recovery Subsystem

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

of ammonia, the air is diluted with nitrogen in the same manner as above until air concentration is below the flammability range when the ammonia is introduced. Prior to the introduction of liquid ammonia, ammonia vapor from the top of the storage tanks is injected in increments, again as discussed above. This minimizes the amount of non-condensable nitrogen in the system.

Figure 3-25 illustrates the improvement in purge effectiveness when smaller increments are used with a given total volume of purge gas. To obtain the correct results, TRW recommends the use of 3 volumes of nitrogen to dilute the ammonia, and 2 additional volumes of nitrogen prior to recharging the system with ammonia after it is closed. About 4 volumes of ammonia vapor are planned.

Figure 3-26 shows a diagram of the nitrogen storage system that includes two 9000 gallon cryogenic storage tanks and the heater required to vaporize and increase the nitrogen temperature as it is fed into the system. Care must be taken to prevent frost formation in the nitrogen feed lines from the cooling effect of expansion from storage pressure (250 psig) to atmospheric pressure.

Although ammonia leaks may occur in any part of the system, they are most likely to occur in the seals around rotating shafts. Figure 3-27 illustrates the types of seals most likely to be used on turbine and pump shafts. The oil seals that will be used around the turbine shaft have been extensively used on ammonia compressors, and usually run 3 to 5 years without maintenance. Double mechanical seals will be used on pump shafts. The space between the seals will be flooded with methanol from a small surge pot located slightly above the pump. The methanol serves as anti-freeze and lubricant. A vent line is connected to the stuffing box between the seal faces and is routed to a safe, elevated point. In the event of an inboard seal failure, the outboard seal ensures that essentially all the ammonia will go out the vent.

In addition, leaked ammonia is collected in a vapor collection drum connected through a manifold to the affected seals in the system, as shown in the leakage collection system of the figure.

Final selection of components and proper sizing of tanks and other major components will be accomplished during the preliminary design task.

Figure 3-25. Ammonia Systems: Purging

Figure 3-26. Liquid Nitrogen Storage

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

TURBINE SHAFT SEALING

PUMP SHAFT SEALING

LEAKAGE COLLECTION SYSTEM

Figure 3-27. Ammonia Systems: Leakage Control

3.3 INSTRUMENTATION AND CONTROL

3.3.1 Power System

Assumptions

- 1) The plant is intended to operate continuously at constant design maximum gross electrical output; however, it can be operated at reduced gross output if required.
- 2) Both the cold water pump and the warm water pump operate continuously at full speed. No control either of pump or of water flow is provided.
- 3) The turbine and generator are sized to accept the largest gross output expected for the site (i.e., the output generated under the most favorable warm water to cold water temperature difference expected for the site). The turbine operating point for maximum efficiency is presumed to be close to the nominal gross power level (i.e., corresponding to the nominal or average temperature difference at the site). The latter point is not essential; sizing to accept the expected maximum power level is desirable.
- 4) The expected maximum net electrical output of the installation is small (a few percent) compared to the total minimum grid load. If the minimum grid load is small, it may be necessary to equip the turbine with a governor that operates on the turbine stator vane positioning system.

Approach to System Control

- A) Control the liquid ammonia feed flow into the evaporator by automatic adjustment of a flow (pressure) regulating throttle valve in response to a flow rate signal in the evaporator feed line or differential pressure signal across the film forming orifice plate.
- B) Control the ammonia liquid level in the recirculation sump by automatic adjustment of feed valve in the feed pump line in response to high and low liquid level signals in the recirculation sump.
- C) Control the ammonia liquid level in the condenser hot well by automatic on-off operation of the makeup pump and associated valving and of the letdown valving in response to high and low liquid level signals in the condenser hot well.
- D) If required, control turbine speed and generator frequency by automatic adjustment of variable stator vanes in response to a turbine speed signal.

A discussion of each of the control functions follows. Refer to Figure 3-28 and the Equipment Identification Key.

Equipment Identification Key for Figure 3-28

- 1) Warm water pump
- 2) Warm water inlet header
- 3) Evaporator fluted tube (typical)
- 4) Warm water discharge header
- 5-7) Deleted
- 8) Ammonia feed header
- 9) Ammonia feed control valve and controller
- 10) Main evaporator ammonia vapor volume
- 11) Ammonia recirculation sump
- 12) Ammonia recirculation pump
- 13) Ammonia bypass valve
- 14) Ammonia feed flow regulator
- 15-16) Deleted
- 17) Condenser ammonia vapor volume
- 18) Condenser sump
- 19) Cold water pump
- 20) Cold water inlet header
- 21) Condenser fluted tube (typical)
- 22) Cold water discharge header
- 23) Liquid ammonia feed pump
- 24) Ammonia let-down/make-up controller

Figure 3-28. Power Module Schematic

Control Functions

A) Liquid Ammonia Feed Flow

Two options that accomplish regulation of the flow of ammonia through the film forming orifices at the top of the evaporator are possible:

1. Use a liquid ammonia flow meter in the ammonia feed line to the feed header to monitor flow rate into the closed header, and use a flow-regulating valve with an automatic valve position controller to maintain flow at the required preset value.
2. Use a differential pressure sensor to monitor the pressure difference between a fixed point in the closed feed header and an arbitrary point in the ammonia vapor region of the evaporator, and use a pressure regulating valve with an automatic valve position controller to maintain a preset head across the film forming orifices.

B) Recirculation Sump Level Control

Again, two options, each with a possible disadvantage, can be considered.

1. This is the option shown in Figure 3-28. The sump level signal actuates a control valve in the ammonia feed line. A low level signal opens the control valve and a high level signal closes the control valve.
2. A second option (not shown in Figure 3-28) is to connect the outlet of the ammonia feed pump/flow control to the recirculation sump and size the recirculation pump to handle the total flow (including recirculated fluid). The sump level would be controlled as in option 1 by adjusting the control valve downstream of the feed pump.

Of the two options, the first may have a slight edge from a cost point of view. It requires a somewhat smaller sump and the recirculation pump is smaller since it handles only the reflux. On the other hand, interaction between the sump level control and the evaporator feed control may require some care to avoid the control actions "fighting" each other. The second option, with the two pumps essentially in series is simpler from a control point of view and has the added advantage that the recirculation pump now handles sub-cooled ammonia such that the pump setting depth can be reduced significantly. The choice between the two approaches will be made during Preliminary Design.

C) Condenser Sump Level Control

Flow of liquid ammonia from the condenser sump to the feed pump suction header is controlled by the evaporator feed control and the evaporator sump level control. During normal operation the condenser sump level will remain essentially constant provided there is no loss of ammonia due to leaks. During transient operation or as a result of leaks, makeup or letdown of ammonia may be required to maintain level in the condenser sump. Figure 3-28 indicates a simple way to accomplish this by control valves in the lines connecting the power module to the ammonia storage subsystem. High and low liquid signals function as on-off signals for valve control.

D) Steady-State Turbine Control

The turbogenerator speed control is provided by the power grid to which it is connected. When the generator moves from synchronous speed, the grid supplies power to return the generator to synchronism. As the turbine's ammonia vapor inlet conditions vary (due to seasonal water temperature change) a computer will adjust the variable stator vanes in the turbine to maximize its shaft torque, thereby causing the turbine to operate at its most efficient point. Analytic simulations are planned to calculate the cost benefit produced by this control loop to determine whether vane control is worthwhile. Pending the results of simulation, no explicit turbine speed control is planned during synchronized operation in order to avoid the pressure drops caused by a turbine inlet control valve or bypass-control valve. The large external grid is expected to provide adequate frequency control. While the major torque-control loop maximizes turbine output torque, the generator exciter control loop maintains constant line voltage and reactive-kva flow. The exact relation between the exciter loop and torque control loop will be defined during preliminary design.

E) Startup and Shutdown Turbine Control

During startup, when there is no load on the turbine, the operator (or an automatic startup computer) will control the bypass valve and the variable stator vanes to bring the unloaded turbine-generator up to near synchronous speed and to hold it at that speed. The operator (or the start-up computer) will synchronize the generator and then fully close the bypass valve, thus eliminating steady-state losses caused by flow through the bypass valve.

During shutdown, the bypass will be opened and the turbine inlet valve tripped, with no speed control on the turbine as it coasts down.

F) Protection

An over-speed trip is provided that opens the bypass valve and trips the main ammonia flow when the turbine's acceleration or speed exceeds preset limits. Thus, the only steady-state losses caused by the protection system are those from the main ammonia trip valve. That valve is essential and cannot be eliminated. Simulations during the preliminary design phase will determine the loss in steady-state power caused by several representative valve pressure drops.

3.3.1.1 Electrical Power System Single Line Diagram

Figure 3-29 illustrates the recommended bus and circuit breaker schemes. The bus scheme includes a double bus arrangement consisting of a main bus and a transfer bus. It provides dual feed to output loads as well as to station service loads. Its purpose is to prevent complete station shutdowns in case of bus fault or equipment failure. This scheme reduces the likelihood of extended outages of any circuit due to circuit breaker trouble and permits breaker maintenance without a feeder outage.

A ring bus scheme is used for bus sectionalization. The purpose of sectionalization between modules is several-fold. It limits the short circuit duty imposed upon the circuit breakers. It reduces the mechanical stresses on buses and equipment. It localizes shutdowns due to bus or equipment failures and prevents complete station shutdowns. It provides backup protection against feeder-breaker failures. The ring bus sectionalization scheme maintains connections between the bus sections even in the case of loss of one section and facilitates transfer of generator capacity between sections.

The combination ring-bus and transfer bus arrangement maintains continuity of the ring bus operation in the event of trouble in any section.

Step-down transformers are included for the various loads. Large loads such as the seawater pump motors are assumed to be operating at 4.16KV. Small loads are assumed to operate at 480V.

Figure 3-29. Nominal Module One Line Electrical Diagram

3.3.2 Water Supply And Distribution System Dynamics

A study was performed on the OTEC cold and warm water systems flow dynamics to determine flow transients and oscillations caused by operational conditions and by hull motion due to sea state conditions. The operational conditions include startup, shutdown, load rejection, and equipment failure. Hull motion includes vertical motion (heave), horizontal motion (surge and sway), and rotation about horizontal axes (roll and pitch). This study was performed in order to:

- Determine the nature and approximate magnitude of transients and oscillations in pressures, velocities, and water surfaces in the distribution system.
- Interpret the significance of the transients and oscillations in the system design.
- Recommend design features to reduce or modify the transients and oscillations in instances where they are unacceptable.

Because of the preliminary nature of this study, the distribution system was broken down into simple elements for analysis and, in some cases, relationships were linearized to achieve rapid, efficient, and timely approximations to support the design effort.

Three types of water distribution systems were evaluated: a completely closed conduit distribution system, a decoupled conduit distribution system, and an open trough distribution system.

3.3.2.1 Completely Closed Conduit System

In the completely closed conduit water system, the total path from intake to discharge consists of a closed conduit with no openings to the atmosphere. While this alternative was examined for both the cold and warm water systems, most attention was given to the cold water system because the long, vertical cold water supply conduit makes it more crucial. The dynamics of flow in the completely closed cold water system were examined for: 1) pump start up and 2) vertical hull motion in sea state six.

System Startup

The major problem in pump startup in this system is the maximum drop in suction head at the starting pump before sufficient water arrives from the vertical supply conduit. If the suction head falls too low, it will cause cavitation of the pump. The parameter which can be adjusted to avoid the problem is the rate of increase of pump speed. The slower the increase, the less the drop in suction head. For a typical configuration, cavitation can occur if the starting pump reaches full speed in less than 15 seconds. Starting additional pumps decreases the suction head of pumps already running, as well as of the pump being started. Cavitation is again avoided by limiting the rate of increase of pump speed and flow rate.

The decrease in flow at pumps already running also means a decrease in flow at the heat exchangers already operating. Rapid starting of pumps in the completely closed system can therefore interfere with power generation in units already running.

Hull Motion

In order to study vertical hull motion, the system was approximated by an inverted U-tube with both legs down in the water and a pump in the upper end of one leg. One leg represented the vertical cold water supply column; the other the hydraulic equivalent of the heat exchangers and vertical discharge pipes in parallel. The horizontal section between the two legs represented the hydraulic equivalent of the pipes from the supply conduit to the heat exchangers in parallel. The heave and vertical accelerations for sea state six were incorporated into a sinusoidal variation of hull elevation with time which was applied to the inverted U-tube model utilizing linearized equations for the fluid motion.

Sea-state-six pressures and velocities were estimated at discrete points in the hull motion. Maximum pressures in conduits were of the order of 60 psi; pressures of this magnitude occurred in the top of the condenser on the cold water side. For a mean water velocity of 7 fps, at steady state, the mean velocity at sea state six is reduced by about 4% to 6.7 fps. The variation in velocity at this sea state is from 6.2 to 7.3 or + 8%.

3.3.2.2 Decoupled Closed Conduit System

In an early design, the closed conduit system for cold water was decoupled by disconnecting the conduits at the pumps. Instead of a vertical conduit connected to the pump intake, the vertical conduit was larger than the pump and extended up around it to form an open standpipe. The pump was located inside the standpipe, which served as the forebay or suction well for the pump. The dynamics of flow in this system were examined for 1) pump startup and 2) vertical hull motion in sea state six.

System Startup

The dynamics of startup in the decoupled system are the same as in the completely-closed system, if water-surface elevation in the standpipe is substituted for suction head at the pump. The fall in the standpipe water level is equivalent to the decrease in suction head. The occurrence of cavitation and the variation of flow in the heat exchangers are similar for both systems, and are avoided in both systems by limiting the rate of increase in pump speed during pump starting. However, the limit is not quite the same for both: the standpipe in the decoupled system holds a larger volume of water than the pump conduit in the completely closed system. Because of the storage, the water level drop is less than the decrease in suction head for the same rate of increase in pump speed. A pump starting the decoupled system can be brought up to full speed in about 10 seconds.

Hull Motion

In the decoupled system, the vertical hull motion does not produce the high pressures produced in the completely closed system. However, the variation in flow through the heat exchanger is much larger. The rise and fall of water surface in the standpipe around the pump is of the same order as the heave of the hull; that is, ± 8 to 10 feet from the mean elevation at sea state six. The variation caused a similar variation in head on the pumps, and this amount of change in head will change the flow rate pumped by $\pm 33\%$. This is a major disadvantage to the decoupled system.

3.3.2.3 Open Trough System

The design shown in Section 2.3.2 was used for evaluation of the dynamics of flow in the open trough system. The dynamics were examined

for pump startup, vertical hull motion (heave), horizontal hull motion (surge), and rotation about horizontal axes (pitch and roll).

System Startup

In the open trough, the water surface does not drop in the same manner as water level in the standpipe or suction head in the pump conduit. As soon as the water falls at one location in the trough, water from other locations rushes to that first location and forms a surface wave. Because the depth and the area of the vertical cross-section of the cold water trough are so large, the waves formed by starting one pump are small. If the largest pump anticipated in the design were brought up to full speed instantaneously, the resulting surface wave in the trough would be less than 1 foot high. Consequently, there is no limit on how fast the pumps are brought up to speed.

If several pumps were started at the same time, however, and each one brought up to full speed instantaneously, difficulties could occur. Depending on the location in the pumps in the trough, the surface wave formed could be significant. The inertia of the vertical supply conduit limits the rate at which the flow in that conduit can increase. Instantaneous startup of several pumps simultaneously would also remove a significant amount of water from the trough and lower the water level by 10 feet or more before the inflow from the supply conduit could catch up. Therefore, pumps should be started sequentially with 30 or 60 seconds between each start, but each pump can be started instantaneously. Steady state drawdown is approximately 4 feet with all pumps operating.

The startup of the warm water pumps presents no difficulties. The warm water troughs are connected to the open sea through openings in the hull. If these openings are large enough, water rushes into the trough quickly enough to prevent significant water level drop in the trough. The pumps in each trough should be started sequentially.

Vertical Hull Motion

The heave of the hull and vertical movement of the open trough system produces vertical motion of the water in the trough relative to the hull. It also produces vertical motion of the water in the top water box of the heat exchanger relative to the top tube sheet.

If there were no friction losses in the vertical supply conduit and the horizontal area of the trough were the same as the cross section area of the conduit, the water column in the conduit would not move in space due to its inertia. The motion of the water relative to the conduit and trough would be exactly equal and approximate the vertical hull motion. The same would be true in the heat exchanger if the horizontal area of the top water box equalled the cross section area of the discharge conduit and there were no friction losses. The vertical motion of the water in the trough or water box would have an amplitude of ± 10.5 feet in sea state six.

The area ratios and friction losses that actually occur reduce this amplitude significantly. To estimate the reduced amplitude, the equations of motion were set up for vertical flow in the supply conduit and trough and in the heat exchanger and discharge conduit. A constant rate of pumping was assumed and other simplifying assumptions were made. The equations were solved numerically for a limited number of time increments, and the results are shown in Figure 3-30. From these calculations it was estimated that the amplitude of vertical motion of the water surface in the trough relative to the trough in sea state six is 6 feet. That is, the water surface stays within ± 3 feet of the steady level in calm sea. In the heat exchanger, the motion of the water surface relative to the tube sheet stays within ± 1 foot of the steady level in calm sea.

If it is desired to reduce the vertical motion in the trough even further, the adjustment of gates in the walls across the trough can be used. More detailed computation or hydraulic model studies would be necessary to determine how much additional reduction is possible.

In the warm water trough there is no vertical motion of the water level, from heave, relative to the sea surface. The sea water flows directly into the trough. With respect to the hull, the water level rises as the hull moves down and falls as the hull moves up. Similar motion is induced by roll. As one side of the hull goes up or down, the depth of water in the trough decreases or increases accordingly. The change in water level due to these effects is ± 3 ft.

Figure 3-30. Transient Response of Trough and Heat Exchanger Liquid Levels

Pump head depends upon the relative water level in the water trough and heat exchanger.

The combination of the vertical motion in the trough and heat exchanger can change the head on the cold water pumps by ± 4 feet in either the cold or warm water system. This change of head causes a change in flow rate of ± 10 to 12%.

Since a constant rate of flow was assumed for the computation of vertical motion, the results are conservative. When the pump head increases, the flow rate decreases, the water level on the heat exchanger falls, and the pump head begins to decrease again. The opposite happens when the pump rate decreases. The variation in pump head will therefore tend to decrease the vertical motion that causes the variation.

The variation in pump flow rate and the vertical motion of water in the heat exchanger and discharge conduct produce a total variation in flow rate through the heat exchanger. The conservative assumption is that the maximum change due to pump flow rate occurs simultaneously with the maximum due to vertical motion. This would cause the flow rate to vary by ± 20 to 30%. The heat exchanger water boxes can be designed to minimize pump head variation and to reduce the flow rate variation. Overflow channels can be included on the top of the heat exchangers to return seawater to cold and warm water troughs as a further means of optimizing the design of the seawater systems.

Horizontal and Rotational Hull Motion

The surge and pitch of the hull produce standing shallow water waves in the trough (Figure 3-31). At certain frequencies, the waves are in resonance and the amplitude increases without limit until the waves break. For the length of trough, the fundamental resonant frequency is about 14 seconds.

Without hydraulic model tests or detailed numerical computations, it is difficult to predict the maximum amplitude of the waves in resonance before they break. Based on values of wave steepness, the amplitude of the standing waves could be as great as 7 feet. These amplitudes can occur in relatively calm seas when the frequency of the surge and pitch are

Figure 3-31. Shallow Water Standing Waves

suitable for resonance. Wave motion is superimposed upon vertical motion causing increased variations in pressure head.

The amplitude of the standing waves in resonance can be reduced and controlled by gated walls or bulkheads across the trough. The gates in the walls would be adjustable and controlled as part of the operation. Whether two or four walls are required would have to be determined by hydraulic model studies and other aspects of operation. By opening various gates in each wall, the trough can be detuned and the resonant waves broken up. It is estimated that gate walls can reduce the maximum amplitude of the standing waves to 2 feet.

3.3.2.4 Summary

The open trough system has been chosen as the primary design concept for OTEC. The system has some problems which must be considered in the system design:

- Standing waves in the open trough
- Vertical motion of water surfaces in the trough and on top of the heat exchangers
- Variation of pump head and rate of flow through heat exchanger

These problems are less severe or more easily avoided than those associated with the two other alternative systems. The high internal pressures of the closed system and the large changes in pump flow rate of the decoupled closed system are avoided. The open trough system also minimizes the effects of potential damage to mechanisms. Cavitation is much less likely in this system and the possibility of water hammer effects, inherent in closed systems, is also avoided. The open trough system is also the preferred alternative from the standpoint of system dynamic considerations. This, combined with other performance and operational considerations, resulted in its selection as the primary design concept.

3.3.3 Power Module Startup

For the OTEC power module to produce power, power must be supplied to the auxiliary equipment required for its operation. Calculations show that for a module producing 12.5 MWe net power, the auxiliary equipment will require 4.0 MWe: most for the seawater pumps. During normal operation, the OTEC power module can supply power to this auxiliary equipment. During startup, however, an auxiliary power source is required.

Startup sequence and power buildup are illustrated in Figure 3-32 (25 MWe output) equivalent to two 12.5 MWe modules. The seawater pumps are started sequentially. After the first set of pumps is started (one for cold water to the condenser and one for the warm water to the evaporator) the module begins delivering power as indicated by the dotted line curve in the figure. The gross power output is expected to be slightly less than 25% of the full output because of the lower efficiency of the generator at 50% full load, which was estimated to be approximately 8 MWe.

With a second set of pumps running, two turbines and one generator would operate at full load. The gross power output would be 16.5 MWe.

Figure 3-32 also illustrates the need for an auxiliary source of power during the startup phase. That power source must be capable of starting and carrying the first set of pumps and other parasitic loads. The steady state load value is estimated at 2 MWe. For starting subsequent pump sets, the output of the OTEC module is available to be paralleled with the auxiliary power source. The effectiveness of this parallel operation will depend on the transient characteristics of the turbine-generator sets.

Auxiliary Power Source Size

The curves shown in Figure 3-33 summarize the calculations of the auxiliary power source (APS) required to start a 12.5 MWe power module. The main criterion for determining the size of the APS was the limitation of voltage drop during the startup of the seawater pumps. Two values of maximum voltage drop, 10% and 15%, were considered. These values are commonly used in systems of comparable capacity. Voltage drops in electrical power systems are not only annoying in terms of light flicker or dimming, but are also serious from the power utilization standpoint. It

Figure 3-32 Power Module Startup

Figure 3-33. Auxiliary Power Source Size (10 and 15 percent Maximum Voltage Drop)

causes contactors to drop out; motors to slow down, stall, fail to start or accelerate; or interference with the proper operation of voltage-sensitive devices.

It was assumed that seawater pump motors were started sequentially one at a time. It was also assumed that with full voltage start the starting KVA of each motor was six times its full load rating and, with reduced voltage start, the starting KVA was only 2.3 times its full load rating. Finally, it was assumed that the transient reactance of the generator was 20%, based on its similarity with most current designs.

For a two-turbine/four-pump module, a seven MVA auxiliary power source is required to limit the voltage drop to 10% during startup with reduced voltage start. With full voltage start a twenty MVA auxiliary power source would be required.

Power Output Versus Water Flow Characteristic

The bootstrap approach for module startup is based on the capability of the module to supply partial output with partial waterflow.

In Figure 3-34 the upper left hand curve shows the NH_3 vaporization rate in the heat exchanger as a function of waterflow. Heat exchanger efficiency is higher at low water flow rates.

The upper right hand curve shows output electrical power of the turbine-generator as a function of NH_3 vapor flow. Efficiency is higher at high flow rates.

The combination of the upper two curves results in the output electrical power of the turbine generator as a function of waterflow rate in heat exchanger. The curve is close to linear.

Figure 3-34. Power Module: Power Output Versus Waterflow Characteristic

4. OPERABILITY/MAINTAINABILITY

4.1 AVAILABILITY

4.1.1 Requirements

4.1.1.1 Operability

The RFP requirement, "Operability (i.e., actual operational time divided by required operational time) shall be 90% at a 60% confidence level," can be interpreted several ways depending on the definitions of "actual operational time" and "required operational time." The problem with "actual operational time" is that in the proposed modular configuration, partial (less than 100 MWe) power output is anticipated. The "required operational time" is interpreted to be 6000 hours per year.

To preclude misinterpretation, the decision was made to measure availability (i.e., operability) in energy units. With this approach, power generated during partial operation could be accounted for. Furthermore, with the inherent capability of the modular configuration, the 6000 hours per year requirement is restrictive and unnecessary; i.e., the power system should be capable of operating for longer periods of time. Availability is therefore defined as:

$$\text{Availability} = \frac{\text{net generation per year (MWe - Hours)}}{\text{Hours in a year times installed capacity (MWe)}}$$

which is similar to the Federal Power Commission's definition of plant factor. It should be noted that the interpretation of the operability requirement does not include a confidence level. Confidence level within a requirement is not significant, but would be appropriate when associated with the operational demonstration of the requirement.

4.1.1.2 System Starting Reliability

The RFP requirement, "System starting reliability (i.e., number of successful starts divided by attempted starts) shall be 95% at a 60% confidence level," is interpreted to apply to the 100 MWe power plant, rather than a module. As is the case with operability, including a confidence level in the requirement is not significant but would be appropriate when associated with the operational demonstration of the requirement.

Because component start-up failure rate data for OTEC hardware were not available, it was necessary to use a previously developed prediction method (described in paragraph 4.1.4.2) based on operating failure rates.

4.1.2 Operational Considerations

4.1.2.1 Failure Modes and Effects

Power System Machinery. Turbine failure modes are most likely to be shaft seal leakage, bearing wear, and stator vane control linkage and bushing wear resulting in degraded performance.

Generator failure modes are most likely to be bearing wear and cooling system failure/degradation resulting in degraded performance or module shutdown.

Seawater pump/motor failure modes are most likely to be shaft seal leakage and bearing wear resulting in degraded performance. Ammonia feed and recirculation pumps/motors failure modes are most likely to be shaft seal leakage and bearing wear resulting in degraded performance.

Heat Exchanger Tubes. Heat exchanger tube failure modes include corrosion, erosion, stress corrosion cracking, corrosion fatigue and vibration failure. The degree of degradation or failure depends upon material selection. For aluminum, the most likely failure modes are pitting corrosion and corrosion fatigue. For titanium, the most likely failure mode is crevice corrosion. The effect would range from degraded performance (due to leakage) to module shutdown (due to rupture).

4.1.2.2 Maintenance

Repairs will be accomplished on-site to the maximum extent possible. This will require detailed planning for spares provisioning and depot support requirements during the preliminary design phase.

Power System Machinery and Support Systems. Periodic inspections of power system machinery and support systems will be performed to facilitate detection and prevention of incipient failures. The inspections will be accomplished while the power plant module is shut down for other reasons, or when there is an indication of malfunction. The extent and level of inspection will be determined, with vendor participation, during the preliminary design phase.

Heat Exchanger Tube Leak Detection and Isolation. The power plant module will be shut down immediately and tube(s) will be inspected and isolated if and when there is an indication of tube rupture. Degradation due to tube leakage within an acceptable (both performance and environmental) level does not require immediate action. Several candidate methods for on-line leak detection have been investigated. An attractive approach is to use leak detecting sensors on the cleaning machine brushes; pH sensors and other candidate sensors will be evaluated during preliminary design. Tube sealing after a leak has been identified is discussed in section 4.3.4.

4.1.3 Availability Prediction

Plant availability estimates were determined for all options using estimates of Mean-Time-Between-Failures (MTBF) and Mean-Time-To-Repair (MTTR) of critical components. For these analyses, availability was defined as:

$$\text{Availability} = \frac{\text{net generation per year (MWe - Hours)}}{\text{Hours in a year times installed capacity (MWe)}}$$

In order to provide availability estimates for use in the tradeoff studies, it was necessary to formulate a set of assumptions, construct availability models for all options, and establish MTBF's and MTTR's for critical components.

4.1.3.1 Assumptions

The initial model assumptions are as follows:

- a) Repair team comprised of specialists capable of repairing individual critical items, i.e., a team exists for each critical item type.
- b) Failure rates and repair rates of machinery (e.g., pumps, generators, turbines) are independent of module size.
- c) Failure rates and repair rates of heat exchangers (not tubes) are affected by module size.
- d) Failure rates and repair rates of heat exchanger tubes are independent of application (i.e., condenser versus evaporator).

- e) Replacement items are available upon demand (i.e., no pipeline constraints at this time).
- f) Tube failures can be readily isolated to a particular heat exchanger.
- g) Critical items can be isolated and purged of NH_3 for maintenance.
- h) Redundancy is provided to ensure continuity of instrumentation and control.
- i) Partial operation of a module is permissible down to 25% as follows:

(1) 25 MWe (two-evaporator two-condenser)

This system option consists of two power loops (or branches) with a common generator and support subsystems. Each loop includes one evaporator, one condenser, two warm seawater pumps, two cold seawater pumps, one turbine, one NH_3 feed pump, one NH_3 recirculation pump, and the necessary piping, valves, etc.

- Twenty-five percent power output is possible with one loop inoperative, common components operable, and the remaining loop producing one-half installed capacity. Operational components within the operative loop are:
 - 1 of 2 warm seawater pumps
 - 1 of 2 cold seawater pumps
 - 1 of 1 or standby NH_3 feed pump
 - 1 of 1 or standby NH_3 recirculation pump
 - turbine
 - evaporator
 - condenser
 - piping, valves, etc.
- Fifty percent power output is possible with one loop inoperative and the remaining loop operative, or both loops producing one-half installed capacity (see twenty-five percent output).
- Seventy-five percent power output is possible with one loop totally operative and the remaining loop producing one-half installed capacity (see twenty-five percent output).

(2) 12.5 MWe (two-evaporator two-condenser)

This system option consists of two power loops with a common generator and support subsystems. Each loop includes one evaporator, one condenser, one warm seawater pump, one cold seawater pump, one NH_3 feed pump, one NH_3 recirculation pump, and necessary piping, valves, etc.

- Fifty percent power output is possible with one loop inoperative, common components operable, and the remaining loop producing installed capacity. Operational components within the operative loop are:

- 1 of 1 warm seawater pump
- 1 of 1 cold seawater pump
- 1 of 1 or standby NH_3 feed pump
- 1 of 1 or standby NH_3 recirculation pump
- turbine
- evaporator
- condenser
- piping, valves, etc.

(3) 6.25 MWe (two-evaporator two-condenser)

Same as (2), above

(4) 25 MWe (one-evaporator one-condenser)

This system option includes one evaporator, one condenser, two warm seawater pumps, two cold seawater pumps, two NH_3 feed pumps, one standby NH_3 feed pump, two NH_3 recirculation pumps, one standby NH_3 recirculation pump, turbine, generator, support subsystems, and necessary piping, valves, etc.

- Fifty percent power output is possible provided that the following components are operable:

- 1 or more warm seawater pumps
- 1 or more cold seawater pumps
- 1 of 2 or standby NH_3 feed pumps
- 1 of 2 or standby NH_3 recirculation pump

- turbine
- generator
- evaporator
- condenser
- support subsystems
- piping, valves, etc.

(5) 12.5 MWe (one-evaporator one-condenser)

Same as (4), above

(6) 6.25 MWe (one-evaporator one condenser)

This system option includes one evaporator, one condenser, one warm seawater pump, one cold seawater pump, two NH_3 feed pumps, one standby NH_3 feed pump, two NH_3 recirculation pumps, one standby NH_3 recirculation pump, turbine, generator, support subsystems, and necessary piping, valves, etc.

- Fifty percent power output is possible provided that the following components are operable:

- 1 of 1 warm seawater pumps
- 1 of 1 cold seawater pumps
- 1 of 2 or standby NH_3 feed pumps
- 1 of 2 or standby NH_3 recirculation pumps
- turbine
- generator
- evaporator
- condenser
- support subsystems
- piping, valves, etc.

(7) 3.125 MWe (one-evaporator one-condenser)

Same as (6), above

4.1.3.2 Availability Models

The availability models constructed include components critical to continued operation as well as components (e.g., valves) that could impact startup or delay isolation for required maintenance actions. Additionally, in order to project a more complete analysis, some ships equipments (e.g., the cold seawater duct) were included. The power module availability model (Figure 4-1) is typical of all the availability models.

4.1.3.3 Failure Data

Power System Machinery. Because numerical reliability data were not directly obtainable from users of equipments similar to proposed OTEC hardware, estimates (Table 4-1) were made by interpreting information provided by land-based power plant representatives, vendors, C.F. Braun engineers and, where possible, generic failure rate data contained in the Rome Air Development Center (RADC) Non-electronic Reliability Notebook and failure rate data contained in the Bureau of Naval Weapons Failure Rate Data (FARADA) Handbook.

Heat Exchange Tubes. Titanium tube failures (Table 4-2) were based on data extracted from the Electric Power Research Institute "Steam Plant Surface Condenser Leakage Study."⁽¹⁾ Titanium was found to have a 90 percent probability of lasting 40 years with less than 10 percent tube failures. It was further concluded that "this probability would be nearly 100 percent if it were not for guard plugs placed in tube sets that were experiencing vibration problems." The expected tube failures (by heat exchanger size and application) were calculated using the more conservative 10 percent per forty-year failure rate. Aluminum tube failures (Table 4-2) were based on the conservative assumption that the aluminum failure rate will not exceed a four to five times greater failure rate than that of aluminum/brass in clean seawater as discussed in the A.D. Little, Inc., "Survey of Condenser Tube Life in Salt Water Service."⁽²⁾ The expected tube failures (by heat exchanger size) were calculated using a 0.95%/10,000 hours failure rate.

⁽¹⁾Prepared by Bechtel for EPRI, March 1977, Report No. EPRI-NP-481.
⁽²⁾February 1967, Report No. PB178 311.

Figure 4-1. Baseline Availability Model

Table 4-1: Mean-Time-Between-Failure (MTBF) and Mean-Time-To-Repair (MTTR)

ITEM	MTBF (YR)	MTTR (HR)
CENTRAL COLD SEAWATER DUCT**	10	112
COLDWATER TROUGH AND SLUICE GATE**	10	24
COLD SEAWATER PUMP AND MOTOR	1	40
COLD SEAWATER DISCHARGE DUCT**	10	24
WARM SEAWATER TROUGH AND SLUICE GATE**	10	24
WARM SEAWATER PUMP AND MOTOR	1	40
WARM SEAWATER DISCHARGE DUCT**	10	24
NH ₃ PURGE/REPLENISHMENT	10	24
CONDENSER TUBES	*	*
CONDENSER SHELL AND CONNECTIONS	5	*
NH ₃ FEED PUMP AND MOTOR	1	8
NH ₃ FEED PUMP SUCTION VALVE	30	8
NH ₃ FEED PUMP DISCHARGE VALVE	30	8
NH ₃ RECIRCULATION PUMP AND MOTOR	1	8
NH ₃ RECIRCULATION PUMP SUCTION VALVE	30	8
NH ₃ RECIRCULATION PUMP DISCHARGE VALVE	30	8
EVAPORATOR TUBES	*	*
EVAPORATOR SHELL AND CONNECTIONS	5	*
EVAPORATOR OUTLET VALVE	30	8
TURBINE VANE CONTROL	2	16
NH ₃ PIPING	5	56
TURBINE	2	40
GENERATOR	10	40
SWITCHGEAR	5	8

*SEE SEPARATE SHEET.

**SHIP SYSTEMS.

Table 4-2. Heat Exchanger Tube Failures

Heat Exchanger Size (in Feet)	Estimated Tube Failures (Failures per Year)			
	Titanium		Aluminum	
	Evaporator	Condenser	Evaporator	Condenser
18	15	20	33	42
25	32	44	67	23
35	64	88	142	103
50	126	175	275	350

Determining the heat exchanger failure rate (caused by tube failures) presented some difficulty. The basic problem consists of the inability to predict frequency of tube rupture and difficulty in modeling an acceptable tube leakage rate. For availability prediction, a constant failure rate was assumed. It was also assumed that leaks would be detectable and repaired when approximately 0.2% of the tubes were leaking. Heat exchanger failure rates, caused by tube failure, for all options were based upon the 0.2% figure.

4.1.3.4 Repair Data

Power System Machinery. Numerical Mean-Time-To-Repair (MTTR) data were not directly obtainable from users of equipments similar to proposed OTEC hardware; therefore, estimates (Table 4-1) were made by interpreting information provided by land-based power plant representatives, vendors, and C. F. Braun engineers.

Heat Exchangers. Because numerical Mean-Time-To-Repair (MTTR) data were not directly obtainable from users of equipments similar to proposed OTEC hardware, estimates were made by interpreting information provided by land-based power plant representatives, vendors, and C. F. Braun engineers. The MTTR's (Tables 4-3, 4-4, and 4-5) reflect TRW's best estimates for each required action.

4.1.3.5 Analysis

Computer Program. Computer programs (based on a logic structure and consistent with availability models) were developed for all power plant options. Structuring in this manner enhanced ease of program verification and permitted rapid incorporation of changes as the power plant design evolved.

The application of the logic technique requires the grouping (Table 4-6) of subsystems/components according to failure impact. System power output varies with the combination of component success/failure; the impact ranges from total system failure (zero power output) to total system success (100% power output). The power module computer model block diagram (Figure 4-2) is typical of all the block diagrams developed for the various options. To illustrate the logical equations (Figure 4-2), the various groups were redesignated as alphabetical components (e.g., Group I became Component A).

Table 4-3. Repair Action Times (Hours): Titanium Heat Exchanger Tubes

ACTION	HEAT EXCHANGER SIZE (FT)			
	18	25	35	50
SHUTDOWN	1	1	1	1
N ₂ PURGE	8	12	14	16
EQUIPMENT REMOVAL (E.G., ANTI-FOULING)	2	2	2	2
TUBE REPAIR*				
IDENTIFICATION	2	2.5	3	3.5
PLUGGING	5	10	22	42
EQUIPMENT REPLACEMENT (E.G., ANTI-FOULING)	2	2	2	2
NH ₃ CHARGE	8	12	14	16
RESTART	4	4	4	4
TOTAL	32**	45.5**	62**	86.5**

*BASED ON 10%/40 YEAR FAILURE RATE.

**HOURS/SHUTDOWN (9 MONTH INTERVALS).

Table 4-4. Repair Action Times (Hours): Aluminum Heat Exchanger Tubes

ACTION	HEAT EXCHANGER SIZE (FT)			
	18	25	35	50
SHUTDOWN	1	1	1	1
N ₂ PURGE	8	12	14	16
EQUIPMENT REMOVAL (E.G., ANTI-FOULING)	2	2	2	2
TUBE REPAIR*				
IDENTIFICATION	2	2.5	3	3.5
PLUGGING	3.5	7	15	29
EQUIPMENT REPLACEMENT (E.G., ANTI-FOULING)	2	2	2	2
NH ₃ CHARGE	8	12	14	16
RESTART	4	4	4	4
TOTAL	31.5**	42.5**	55**	73.5**

*BASED ON 0.95%/10K HOUR FAILURE RATE.

**HOURS/SHUTDOWN (3-MONTH INTERVALS).

Table 4-5. Repair Action Times (Hours) of Heat Exchanger Connections (Nozzles)

ACTION	HEAT EXCHANGER SIZE			
	18 FEET	25 FEET	35 FEET	50 FEET
SHUTDOWN	1	1	1	1
N ₂ PURGE	8	12	14	16
REPAIR	4	6	7	8
NH ₃ CHARGE	8	12	14	16
STARTUP	<u>4</u>	<u>4</u>	<u>4</u>	<u>4</u>
TOTAL	25	35	40	45

Table 4-6. Logic Structure
(Grouped According to Failure Consequence)

GROUP	ITEM
<p>I: ENTIRE MODULE FAILURE/DEGRADATION</p> <p>II: SINGLE BRANCH FAILURE/DEGRADATION</p> <p>III: SINGLE FEED PUMP SUBSYSTEM FAILURE</p> <p>IV: SINGLE RECIRCULATING PUMP SUBSYSTEM FAILURE</p>	<p>CENTRAL CW DUCT CW THROUGH AND SLUICE GATE WW THROUGH AND SLUICE GATE SWITCH GEAR NH₃ PURGE/REPLENISHMENT GENERATOR</p> <p>CW PUMP AND MOTOR CW DISCHARGE DUCTS WW PUMP AND MOTOR WW DISCHARGE DUCTS CONDENSER TUBING EVAPORATOR TUBING EVAPORATOR OUTLET VALVE TURBINE VANE CONTROL VALVE NH₃ PIPING CONDENSER SHELL AND CONNECTIONS EVAPORATOR SHELL AND CONNECTIONS TURBINE</p> <p>NH₃ FEED PUMP AND MOTOR NH₃ FEED PUMP DISCHARGE VALVE NH₃ FEED PUMP SUCTION VALVE</p> <p>NH₃ RECIRCULATION PUMP AND MOTOR NH₃ RECIRCULATION PUMP DISCHARGE VALVE NH₃ RECIRCULATION PUMP SUCTION VALVE</p>

AT LEAST 50% POWER

$$A \times [B \times (D + E) \times (G + H) + C \times (E + F) \times (H + J)]$$

LOGICAL EQUATION

100% POWER

$$A \times B \times C \times (D \times E + D \times F + E \times F) \times (G \times H + G \times J + H \times J)$$

LOGICAL EQUATION

Figure 4-2. Availability Model Logic Diagram

The probability of each combination was calculated and summed for the appropriate levels of system output. When all possible success/failure combinations had been considered, the resultant probabilities of each level of operation were used to compute availability.

Availability Estimates. Availability estimates were calculated for all options using titanium and aluminum tube failure and repair rates. The results (Table 4-7) demonstrate that the 0.90 requirement would be achieved by all options. Figures 4-3 and 4-4 depict availability estimates in graphical form. Additional computer runs were made to assess the sensitivity (on availability) to changing tube failure rates. The results are shown in Figures 4-5 and 4-6.

4.1.4 System Starting Reliability Prediction

4.1.4.1 Prediction Model

Numerical startup reliability data were not directly obtainable from available data sources; therefore, a model previously developed and used on missile programs by The Boeing Company was used. This model equates the equipment reliability to one hour of operation, to the probability of a successful startup as follows:

$$\text{System starting reliability} = e^{-\lambda t}$$

where λ = plant failure rate (in failures per hour)

t = one hour

4.1.4.2 System Starting Reliability Estimates

Power plant starting reliability was calculated for the two-evaporator two-condenser 12.5 MWe configuration option utilizing titanium tubes. The predicted 0.987 exceeds the 0.95 requirement. Table 4-8 contains the failure data used in the calculation.

Table 4-7. Availability Estimates - All Options

TWO EVAPORATOR — TWO CONDENSER				ONE EVAPORATOR — ONE CONDENSER			
NOMINAL DIAMETER (FEET)	MODULE OUTPUT (MWe)	AVAILABILITY		NOMINAL DIAMETER (FEET)	MODULE OUTPUT (MWe)	AVAILABILITY	
		ALUMINUM	TITANIUM			ALUMINUM	TITANIUM
18	6.25	0.966	0.979	18	3.125	0.957	0.970
25*	12.5	0.957	0.974	25	6.25	0.949	0.966
35	25.0	0.939	0.961	35	12.5	0.939	0.961
				50	25.0	0.925	0.954

*BASELINE POWER MODULE

*BASELINE POWER MODULE

Figure 4-3. Availability - Two Evaporator - Two Condenser Configuration

NOMINAL H.E. DIA.	MODULE OUTPUT	ALUMINUM	TITANIUM
25	6.25 MWe	0.949	0.966
35	12.5 MWe	0.934	0.961
50	25.0 MWe	0.925	0.954

Figure 4-4. Availability - One Evaporator - One Condenser Configuration

Figure 4-5. Availability - Two Evaporator - Two Condenser Configuration

Figure 4-6. Availability - One Evaporator - One Condenser Configuration

Table 4-8. System Starting Failure Rate

ITEM	FAILURE RATE (FAILURES PER YEAR)	NUMBER PER PLANT	FAILURE RATE 1 PLANT (FAILURES 1 YEAR)
CENTRAL COLD SEAWATER DUCT	0.1	1	0.1
COLD SEAWATER TROUGH AND SLUICE GATE	0.1	8	0.8
COLD SEAWATER PUMP AND MOTOR	1.0	16	16.0
COLD SEAWATER DISCHARGE DUCT	0.1	16	1.6
WARM SEAWATER TROUGH AND SLUICE GATE	0.1	8	0.8
WARM SEAWATER PUMP AND MOTOR	1.0	16	16.0
WARM SEAWATER DISCHARGE DUCT	0.1	16	1.6
NH ₃ PURGE/REPLENISHMENT	0.1	8	0.8
CONDENSER TUBES	1.33	16	21.28
CONDENSER SHELL AND CONNECTIONS	0.2	16	3.2
EVAPORATOR TUBES	1.33	16	21.28
EVAPORATOR SHELL AND CONNECTIONS	0.2	16	3.2
EVAPORATOR OUTLET VALVE	0.03	16	0.48
TURBINE VANE CONTROL	0.5	16	8.0
NH ₃ PIPING	0.2	16	3.2
TURBINE	0.5	16	8.0
GENERATOR	0.1	8	0.8
SWITCH GEAR	0.2	8	1.6
NH ₃ FEED PUMP AND MOTOR	1.0	*	0.8
NH ₃ RECIRCULATION PUMP AND MOTOR	1.0	*	0.8
SYSTEM STARTING FAILURE RATE			108.9
SYSTEM STARTING RELIABILITY =			0.987

* REQUIRES TWO OUT OF THREE PER MODULE

4.2 SAFETY ISSUES

The key safety issues for the OTEC plant associated with the PSD include:

- Fire
- Electrical hazards
- Explosion hazards
- Development of toxic compounds.

A fire protection system will be provided for anhydrous ammonia fires, oil fires, and electrical fires.

Ammonia is flammable and will burn in air at concentrations between 16 and 25 percent. The ignition temperature is high (1204°F) and, under many conditions, ammonia will not continue to burn if the source of ignition is removed. Extinguishing methods require detection systems stopping the flow of gas, with provisions for water spray to cool hot surfaces and protect workers, and dilution ventilation systems to be provided by ship systems. Adequate ventilation systems should be provided for all apparatus and equipment below decks. A complete air change of between once per minute and once every three minutes is required to allow operating and maintenance personnel to work around the equipment. The air change will also prevent the formation of explosive mixtures of ammonia vapor with air.

Potential sources for oil fires are turbine lubrication oil, turbine control system oil, and high-pressure hydraulic control system oil. The flash point of these oils is about 400°F and the ignition temperature is 700°F. Extinguishing methods should include dry chemical foam and aqueous film forming foam (AFFF) systems.

To avoid electrical fires and explosion hazards, the electrical system will be designed in accordance with Class I, Group D environment, and in accordance with National Electrical Code Article 501. In addition, hand CO₂ systems and halon systems will be provided in critical areas where the potential for electrical fires exists.

Ammonia reacts with mercury to form an explosive compound and is therefore a hazard. Instruments containing mercury must not be used in ammonia services.

Each pipe flange is a potential source of ammonia leakage and should be limited to equipment connections only. All other connections, including valves, should be welded.

The use of toxic compounds, is a major issue which requires the OTEC plant be designed to chemical safety data sheet Sd-8 of the Manufacturing Chemists' Association Inc., and the Compressed Gas Association (CGA) pamphlet G-2, anhydrous NH_3 .

A training program for all employees handling ammonia is essential. Operators and maintenance men must wear coverall goggles whenever operating or working on ammonia valves and ammonia-containing equipment, such as pumps, turbines, or loading storage tanks. All employees and supervisors must be thoroughly familiar with the location of all turret nozzles, fire hydrants, safety showers, eyewash fountains, ammonia canister masks, fresh air packs, and emergency oxygen supplies. The plant should also have a minimum of two protective suits complete with masks which have their own air or oxygen supply. The suits should be located in the control room, accessible to all employees at all times. Supervisors should be trained in the use of suits, and a man wearing protective equipment should be backed up by a safety man who is also dressed in protective equipment.

For detection of ammonia leaks, small size sulfur dioxide cylinders will be provided. In the presence of SO_2 , ammonia forms a white cloud of ammonium sulfide.

Preliminary design planning will provide procedures for preparing equipment for maintenance, with particular emphasis on safety procedures for welding and other heat-generating work.

All vessels or equipment in ammonia service will be provided with relief valves in accordance with the ASME code. The relief valves are set at the design pressures of the equipment. These settings are determined by the normal criteria for sizing, and include those for fire, cooling water failure, or misoperation.

An atmospheric vent system is required for such things as ammonia safety valve discharges, emergency manual vents, and high and low point ammonia system vents and drains. The atmospheric vent system will collect all such vents and discharge to the atmosphere at a safe high point of the platform structure. A separate atmospheric vent system is required for vents that should not be subjected to high back pressures. An example of this is the vents from the methanol surge pots of tandem pump seals. A high back pressure on these would trip all the seal pressure alarms.

4.3 LEAK DETECTION AND TUBE ISOLATION

The power plant module will be shut down immediately and tube(s) will be inspected and isolated (identified and sealed) if there is an indication of tube rupture. Performance or environmental degradation within acceptable limits will not require immediate action. The heat exchanger tubes will be inspected and isolated during the next required shutdown caused by other failures/malfunctions. Leak rate criteria for shutdown and/or maintenance will be developed during the preliminary design phase.

4.3.1 Ammonia Leakage Detection

Two basic ammonia leakage detection techniques will be utilized in this system.

One technique will involve the determination of excessive ammonia liquid level change rates in the condenser sump. The present scheme of operation calls for automatic liquid ammonia level maintenance in the evaporator and condenser sumps. This scheme also incorporates an ammonia storage tank and supply system to maintain the condenser sump liquid level within limits. If the condenser sump liquid level is monitored, the rate at which the level recedes (as long as the sump is not being replenished or diminished by the storage and supply system) could provide the necessary volumetric rate of change data from which leakage rates can be calculated. Key indicators of excessive overall leakage will be abnormally long and frequent periods of supply pump operation. These observations could be converted to overall leakage rates. If one storage tank and supply pump replenishes more than one module condenser sump, cumulative volumetric flow and time monitoring may be required in each module condenser sump supply line. When large overall leakage rates are observed, these rates

will be due largely to leakage into the seawater caused by flaws in, or damage to, the seawater/ammonia barrier.

Another means that will be utilized to determine ammonia leakage into the heat exchanger seawater side is the sensing of ammonia concentrations in the seawater by use of a specific sensor connected to a direct readout and strip chart recorder. Sensing units currently available are capable of detecting concentrations in the range of 0.1 ppm to 10,000 ppm by volume. The sensing unit in this type system would be installed in the seawater effluent stream and would sense and relay signals back to the main unit. This detector will provide immediate alarm indication of a tube rupture.

The ammonia leak rate criteria will be developed during the preliminary design phase; the leak detection equipment to be used will also be defined during this phase.

4.3.2 Identification of Leaking Tubes

Because of the unique open top water box configuration of both heat exchangers, tube leak detection can be accomplished on-line (without module shut-down). As discussed earlier, our preferred method is to incorporate this capability with the cleaning machine; pH detectors attached to the cleaning brushes will sense and locate tube leaks on-line. The details will be developed during preliminary design. After identification, defective tubes will be tagged and sealed as described below.

4.3.3 Tube Sealing

Heat exchanger tubes which are found to be defective will be sealed from the seawater circuit. Based on discussions with oil and gas well completion equipment manufacturers, including Oil States Rubber Company and Baker Oil Tools, it is intended that the tubes will be sealed at the top and bottom by tubing packer cups or plugs. These cups or plugs will be mounted on a shaft and inserted into the defective tube from the open top waterbox. This shaft will then be rotated to mechanically set both the top and bottom tubing cups or plugs. Figure 4-7 shows the shaft and sealing cups in place sealing a defective tube at top and bottom simultaneously.

TUBES WITH ISOLATING PACKER CUPS INSTALLED

Figure 4-7. Heat Exchanger Leaking Tube Isolation

4.3.4 Tube Maintenance Time Line

Table 4-3 presents the estimated time requirements to perform shutdown, purge and pressurization, equipment removal, tube isolation, equipment replacement, ammonia charging, and restart operations. Specific details relative to these time lines are given in Section 4.1.

5. COST ANALYSES AND PROJECTIONS

The overall approach to the comparative assessment of different design options is described in Section 2.1.3. As indicated, minimum energy cost at the busbar is used as a figure-of-merit for comparison of design alternatives because OTEC is a capital-intensive system. Details of the supporting cost analyses and projections (Section 2.2) are presented in this section.

5.1 BASE OF COST ESTIMATES

General arrangement drawings and equipment lists were developed for each of the seven module options described in Section 2.1.2. Capital cost estimates were based on established equipment requirements determined from the design drawings (Section 2.3.2) and equipment lists (Section 2.3.4) for each option.

Verbal quotations were obtained for major equipment, such as heat exchangers, turbine-generator, water pumps and ammonia pumps; the cost of other equipment, such as piping, valves, instrumentation, and electrical equipment, was estimated from current TRW sources. Minor bulk materials not indicated on the drawings (i.e., small piping and wiring) were evaluated as a function of major subsystem bulk material and labor costs.

Estimates of busbar energy costs are based on the relationship of capital costs and availability as follows:

$$C_e = \frac{C_c (R + C_o)}{A \times H}$$

where

C_e = Busbar energy cost (mills/KWH)

C_c = Capital cost (\$)

R = Annual fixed charge rate of 15% (includes cost of money, depreciation sinking fund annuity, insurance and general expenses)

C_o = Operating and maintenance costs of 1.5%

A = Availability (see Section 4.1 for option-related quantity)

H = Plant operating time per year of 8,760 hours

Indirect costs include engineering services, home office costs, and fee. Engineering includes preliminary engineering, optimization studies, specifications detail engineering, vendor drawing review, and resident engineering services during construction. Home office costs include procurement, estimating and scheduling services, quality assurance and project management. The fee is included as a function of the total project cost. The sum of these three categories falls into historically consistent percentages in the range of 10 to 20 percent, depending on the complexity of the project. For this study, a figure of 15.5 percent has been used as typical for a prototype plant that, while new in concept, has had the benefit of several iterations of design optimization. The apportionment of percentages by type is as follows:

<u>Plant Type</u>	<u>Engineering</u>	<u>Home Office</u>	<u>Total</u>
Prototype	13%	2.5%	15.5%
Production	2.5%	2.5%	5%

5.2 COST COMPARISON OF OPTIONS

5.2.1 Modularity Options

Comparative cost estimates of power plant modularity options with titanium tubes were developed based on the methodology described in Section 5.1. Hull cost estimates were developed for each of the module sizes based on a surface platform ship configuration as shown for the baseline in Figure 2-15, and for other module sizes in Figure 5-1. Optimized ship volumes are as follows:

MODULE SIZE (MWe)	NUMBER EVAPORATOR AND CONDENSERS	SHIP VOLUME (ft ³ x 100,000)
25	1 each	14.6
25	2 each	14.7
12.5	1 each	14.7
12.5	2 each	12.9
6.25	1 each	12.9
6.25	2 each	22.1
3.125	1 each	22.1

The resultant comparative busbar energy costs* indicate that the lowest cost is obtained with a 12.5 MWe power module with two evaporators and condensers. Table 5-1 shows the cost trends not including ship volume effects.

5.2.2 Tube Material Options

Cost breakdowns for aluminum tubes heat exchangers are presented in Tables 5-2, -3, and -4 for comparison with titanium tubed heat exchangers as presented in Section 2, Tables 2-7, -8 and -9.

Based on productions first costs, the aluminum tubed heat exchanger is approximately \$94/KWe less expensive than titanium. Aluminum tubes are more susceptible to corrosion (Section 3.1.2), however, and will require expensive retubing earlier than titanium tubes. The tube material selection therefore requires a present-value life cycle cost analysis as presented in Section 5.2.3 to support a cost effective tube material selection.

* This is our figure of merit used for selection of optimum design. This cost included power plant and associated hull. Per DOE direction, ship volume will not be included in the figure of merit during preliminary design and has been removed from Table 5-1. (DOE is considering immersing the exchangers in sea water).

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

REVISED 25 MWe POWER MODULE	1-2-78	ROP
INITIAL ISSUE	11/28/77	ROP
DESCRIPTION	DATE	APPROVED
REVISIONS		
OTEC POWER SYSTEM DEVELOPMENT	TRW ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY	
DATE 11/22/77	DRW. NO.	6350.38-40
DRW. I. ENDA, BLUEP		
CHKD. PEG. BOGAIN		
APPROV. ROP	APPROV. - PROGRAM OFFICE	

Figure 5-1. 100 MWe Power Plants General Arrangement Options

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

**Table 5-1. Comparative Option Cost Estimates
(Titanium Tubes)**

Module Size (MWe)		25		12.5		6.25		3.125
Evaporators	Number	1	2	1	2	1	2	1
	Diameter (Ft)	35.2	24.9	24.9	17.6	17.6	12.4	12.4
Condensers	Number	1	2	1	2	1	2	1
	Diameter (Ft)	41.4	29.3	29.3	20.7	20.7	14.6	14.6
Power Plant Costs (\$/KW)								
Evaporator (Less Freight)		307	303	303	304	304	318	318
Condenser (Less Freight)		383	377	377	371	371	379	379
Turbine/Generator		104	124	136	142	160	160	172
NH ₃ Pumps		13	10	18	12	24	16	29
NH ₃ Piping/Valves		9	10	10	10	10	11	11
Cold Water Pumps		66	66	66	66	66	66	66
Warm Water Pumps		63	63	63	63	63	63	63
Misc. (Support, I&C, Elec)		93	93	98	101	103	113	118
Installation and Checkout		101	102	104	104	107	110	112
Subtotal (Direct)		1139	1148	1175	1173	1208	1236	1268
Indirect (Engr, Admin)		57	58	59	59	61	62	64
Power Plant Capital Cost (\$/KW)		1196	1206	1234	1232	1269	1298	1332
Experience Factor		0.7	0.7	0.7	0.7	0.7	0.7	0.7
Production Plant Cost		837	844	864	862	888	909	932
Availability		0.954	0.961	0.961	0.974	0.966	0.979	0.970
Relative Figure of Merit (Cost/Availability)		877	878	899	885	919	928	961

NOTE: Freight is included in exchanger costs in this table.
 Freight is not included in exchanger costs in Table H.

Table 5-2. Cost Breakdown (Aluminum Option)

	Prototype (a) (\$/KW _e)	Production (\$/KW _e)
Evaporator	305 (d)	184 (e)
Condenser	355 (d)	211 (e)
Turbine	80	50
Generator	62	56
Water pumps, motors, etc.	124	105
Ammonia piping and valves	48	30
Ammonia feed pumps	8	7
Ammonia recirculation pumps	4	4
Control systems	12	10
Electrical equipment	26	23
Cleaning systems	18	11
Chlorination	7	6
Ammonia support system	38	24
Installation	<u>97</u>	<u>30</u>
Total (Direct Cost)	1184	751
Indirect Costs (Engr, Admin)	<u>184 (b)</u>	<u>38 (c)</u>
Total	1368	789

(a) Basic 12.5 power system module with 2 evaporators and 2 condensers.

(b) Engineering design and field supervision 13 percent, home office and field administration 2.5 percent.

(c) Field engineer and supervision 2.5 percent, administration 2.5 percent.

(d) Includes freight (Cond 51\$/KW and Evap 38\$/KW).

(e) Shipyard dedicated heat exchanger fabrication facilities - no freight.

Table 5-3. Evaporator Cost Breakdown (Aluminum Option)

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	64.8	30.4	64.8	15.2
Tubesheets	23.6	21.6	23.6	10.9
Tube support plates	0.8	3.3	0.8	1.6
Shell	8.4	29.0	8.4	14.1
Water boxes	1.2	22.5	1.2	11.2
Nozzles	1.3	9.1	1.3	4.5
Transition inlet	1.3	9.4	1.3	4.7
NH ₃ Distribution	2.0	6.1	2.0	3.0
External Support	0.4	3.8	0.4	1.9
Bustle	1.5	22.5	1.5	11.2
Freight	---	41.6	---	---
Subtotals	105.3	199.3	105.3	78.8
Total \$/KW _e	304.6		184.1	

Evaporator Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 19.6 Feet

Tube Material - - - - - Aluminum

Tube Length - - - - - 38.5 Feet

Number of Tubes - - - - - 4,145

Tube Diameter - - - - - 2.5 inch (outer)

Table 5-4. Condenser Cost Breakdown (Aluminum Option)

Item	Prototype		Production	
	Material \$/KW _e	Labor \$/KW _e	Material \$/KW _e	Labor \$/KW _e
Tubes	83.5	44.1	83.5	20.7
Tubesheets	30.3	35.0	30.3	16.5
Tube support plates	1.0	4.0	1.0	2.1
Shell	11.1	35.8	11.1	16.9
Water boxes	1.5	26.1	1.5	12.3
Nozzles	1.4	10.5	1.4	4.9
Transition inlet	1.5	10.7	1.5	5.0
External support	0.4	4.3	0.4	2.0
Freight	---	53.2	---	---
Subtotals	130.7	224.1	130.7	80.4
Total \$/KW _e	354.8		211.1	

Condenser Configuration (6.25 MW_e Nominal)

Shell Diameter - - - - - 21.9 Feet
 Tube Material - - - - - Aluminum
 Tube Length - - - - - 40 Feet
 Number of Tubes - - - - - 5,157
 Tube Diameter - - - - - 2.5 inches (Outer)

5.2.3 Comparison of Titanium and Aluminum Heat Exchanger Costs on a Present-Worth Basis

This comparison realistically compares the costs, on a present-worst basis, of the titanium tubed evaporator and condenser to the aluminum tubed versions. All the cost comparisons in this section will be on the basis of costs for "production" heat exchangers, rather than for individual prototype units.

This comparison is made because the estimated life expectancy of aluminum tubes in seawater is 15 years, versus 30 years or more for titanium tubes. The assumption is that after 15 years the aluminum tubes will have to be replaced to permit operation for another 15 years to satisfy the design requirements of 30-year life for the evaporator and the condenser.

In performing this analysis, allowances were made for the scrap value of aluminum tubes at the 15th and 30th years, and for titanium tubes at the 30th year.

Table 5-5 shows some of the pertinent characteristics of the aluminum and titanium heat exchangers.

Table 5-5. Aluminum and Titanium Heat Exchanger Characteristics

Characteristic	EVAPORATOR		CONDENSER	
	Titanium	Aluminum	Titanium	Aluminum
Tube Length, ft.	45.5	38.5	43.0	40.0
Tube Diameter	Inside	2.058	1.884	2.058
	Outside	2.500	2.000	2.500
Number of Tubes	51.88	4145	7221	5157
Initial Total Tube Weight, lbs	134,799	226,607	177,430	292,918
Scrap Total Tube Weight, lbs	134,799	133,698	177,430	172,821
Shell Diameter, ft.	17.6	19.6	20.7	21.9

5.2.3.1 Calculation of Present Worth of Aluminum Tubed Evaporators with Retubing at 15 Years

● Total initial cost	\$1,150,625
● Retube in 15 years	
Tube material present cost \$405,036	
Tube material in 15 years, assuming 3 percent inflation per year	\$ 631,033
Tube installation present cost \$221,202	
Tube installation in 15 years, assuming 5 percent inflation per year	\$ 459,863
Retube cost in 15 years	\$1,090,896
● Bundle R&R and removal of old tubes	
In 15 years	\$ 831,600
Cost retube in 1993	\$1,922,496
Present value retube (at 10% int)	\$ 460,230
● Present value Aluminum tubed Evaporator	
Initial capital cost	\$1,150,625
Present value retube	\$ 460,230
	\$1,610,855
Less present value aluminum scrap*	- 27,385
Present value aluminum tubed evaporator	\$1,583,470

*Present Value of Evaporator Aluminum Tube Salvage

It is estimated that approximately 41 percent of the original weight of aluminum tubing will be lost due to seawater erosion and corrosion at the end of 15 years. Therefore, the scrap weight of the aluminum tubing in the evaporator is

$$0.59 \times 226,607 = 133,698 \text{ lbs.}$$

Using a 1978 scrap value of \$0.40 per pound, the 1978 scrap value is $(0.40)(133,698) = \$53,479$. Escalating at 3 percent per year to 1993 gives

\$ scrap (1993) = \$83,321. The present value of this 1993 scrap is \$19,947. The scrap value of the second (retubed) aluminum bundle is \$53,479. Escalated 3 percent per year to the year 2008, this scrap value is \$129,808. The present value of the scrapped second bundle is \$7,438.

Therefore, the present value of evaporator aluminum scrap is:

Initial bundle	\$19,947
Retubed bundle	\$ 7,438
Total P.V. aluminum scrap	\$27,385

5.2.3.2 Calculation of Present Worth of Aluminum Tubed Condenser with Retubing at 15 Years

● Condenser initial cost	\$1,319,375	
● Retube in 15 years		
Tube material present cost \$521,717 escalated at 3 percent per year for 15 years		= \$ 812,835
Tube installation present cost \$275,208 escalated at 5 percent per year for 15 years		= \$ 572,157
		<u>\$1,384,992</u>
● Bundle R&R and removal of old tubes in 15 years		\$ 623,700
Cost Retube in 15 years		\$2,008,692
Present Value Retube (at 10% int)		\$ 480,881
● Present Value Aluminum Tubed Condenser		
Initial capital cost		\$1,319,375
Present value retube		\$ 480,881
		<u>\$1,800,256</u>
Less Present value aluminum salvage*		\$ - 35,397
		<u>\$1,764,859</u>

*Present Value of Condenser Aluminum Tube Salvage

The basis for this calculation is the same as for the computation of the present value of aluminum tube salvage for the evaporator. Again, it

is estimated that approximately 41 percent of the original weight of aluminum tubing will be lost at the end of 15 years due to seawater erosion and corrosion. Therefore, the scrap weight of aluminum tubing in the condenser is:

$$0.59 \times 292,918 = 172,821 \text{ lbs.}$$

The salvage value, at \$0.40/lb, is therefore:

$$172,821 (\$0.40) = \$69,128$$

escalating at 3 percent per year gives a 1993 scrap value of \$107,701. The present value of this is \$25,784. The salvage value of the second, retubed, aluminum bundle is \$167,774 in the year 2008. The present value of this sum is \$9,613. The total present value of the condenser aluminum tube salvage (two tube bundles) is:

Initial bundle	\$25,784
Retubed bundle	\$ 9,613
Total	\$35,397

5.2.3.3 Calculation of Present Worth of a Titanium Tubed Evaporator and Condenser at the End of 30 Years

For this computation it is assumed that there is no loss of titanium during the 30-year life of the titanium condenser and evaporator. Therefore, the salvage value of titanium in 2008, using a 1978 scrap value of \$0.80/lb is:

<u>EVAPORATOR</u>	\$134,799 lbs x \$0.80/lb = \$107,839, 3 percent escalation to 2008 gives \$261,725. The present value of this sum is <u>\$14,997</u> .
<u>CONDENSER</u>	\$177,430 lbs x \$0.80/lb = \$141,944, 3 percent escalation to 2008 gives \$344,498. The present value of this sum is <u>\$19,740</u> .

The present value of the titanium tubed evaporator and condenser is:

	EVAPORATOR	CONDENSER
Initial capital cost	\$1,331,875	\$1,625,625
Less present value salvage	<u>- 14,997</u>	<u>- 19,740</u>
Present value	\$1,316,878	\$1,605,885

Table 5-6 is a summary of the life cycle analysis showing all the cost factors used to complete the present worth of the aluminum and titanium tubed heat exchangers. As previously mentioned, all cost comparisons are based on costs for "production" heat exchangers, rather than for one-of a kind prototype units.

Table 5-6. Total Evaporator and Condenser Present Value Costs for 30-Year Life

	ALUMINUM		TITANIUM	
	EVAPORATOR	CONDENSER	EVAPORATOR	CONDENSER
Initial Cost	\$1,150,625	\$1,319,375	\$1,331,875	\$1,625,625
Present Value Retube	\$ 460,230	\$ 480,881		
Present Value Salvage				
15 year	\$ 19,947	\$ 25,784		
30 year	\$ 7,438	\$ 9,613	\$ 14,997	\$ 19,740
TOTAL	\$ 27,385	\$ 35,397	\$ 14,997	\$ 19,740
Present Value Including Salvage	\$1,583,470	\$1,764,859	\$1,316,878	\$1,605,885
Present Value EVAP and COND	\$3,348,329		\$2,922,763	
Misc. Incremental Operating Costs	\$ 104,383			
Total costs for 30-year life	\$3,452,712		\$2,922,760	

Miscellaneous Incremental Operating Costs. Three costs are grouped under this item. All are a further cost penalty to the aluminum heat exchangers or a credit to the titanium heat exchangers.

These three factors are:

- 1.) Fractional loss in annual electric power production due to the fact that on an annual basis the availability of the heat exchangers is less than 1.000.

The difference in availability for the aluminum and titanium heat exchangers results in a present worth penalty for the aluminum heat exchangers in the amount of \$43,873.

- 2.) Another cost penalty for the heat exchangers is due to the fact that during continued operation, increasing numbers of tubes are expected to fail. These tubes would be plugged, and would result in reduced heat exchanger performance.

The calculated difference between titanium and aluminum tubed heat exchangers results in a present-worth cost penalty of \$56,739.

- 3.) The third factor to be determined is the lost electric power generation caused by downtime during retubing at the end of 15 years. This factor, also a penalty for the aluminum heat exchangers, has a present worth of \$3771.

It should be mentioned that all these factors were computed using a value for lost revenue of only 5 mills/Kw-hr. This is approximately equivalent to a 10% profit at present day electric power rates. Should a larger value for lost generation be used (e.g., the actual value to the nation), the miscellaneous incremental cost factor would greatly increase. This increase would severely penalize the aluminum heat exchangers. As this is a high leverage item, it appears that further economic analyses should be performed to determine lost revenue costs and their affect on aluminum versus titanium heat exchanger costs.

Calculation of Miscellaneous Incremental Operating Costs

- 1.) Incremental loss due to difference in availability between titanium and aluminum tubed heat exchangers.

Assumptions:

A_t = Annual titanium heat exchanger availability = 0.974

A_a = Annual aluminum heat exchanger availability = 0.957

KW = Electric power production per heat exchanger = 6250 KW

CLP = Value of lost revenue = \$0.005/KW-hr

i = Interest rate = 10%

Annual Incremental Cost Penalty for Aluminum

$$\begin{aligned} &= (A_t - A_a)(\text{hrs/yr.})(\text{KW})(\text{CLP}) \\ &= (0.974 - 0.957)(8760 \text{ hrs/yr.})(6250 \text{ Kw})(\$0.005/\text{Kw-hr}) \\ &= \$4654/\text{yr.} \end{aligned}$$

The present worth of the annual cost for 30 years is

$$C \left[\frac{(1+i)^n - 1}{i(1+i)^n} \right] = 4654 \left[\frac{(1+0.1)^{30} - 1}{(0.1)(1.1)^{30}} \right] = \$43,873$$

2.) Incremental Cost due to increasing numbers of failed tubes.

Assumptions:

	Loss in Capacity at N. years	Fractional Capacity loss, Increase per Year
Aluminum	12½% in 15 yeqrs	0.00833
Titanium	7.5% in 30 years	0.00250

For aluminum, the lost capacity at the end of one year is 0.833%, the second year 1.666%, etc. For titanium, the lost capacity at the end of the first year is 0.25%, and at the end of the second year is 0.50%, etc. The present worth of these geometric series of lost-capacity is 0.1927 for titanium and 0.4127 for aluminum. The present worth cost for the geometric series lost capacity for the titanium heat exchanger is

$$= (0.1927)(0.974)(8760)(6250)(0.005) = \$51,380$$

and for the aluminum heat exchanger is

$$(0.4127)(0.957)(8760)(6250)(005) = \$108,119$$

The difference in these present worth costs is

$$\$108,119 - \$51,380 = \$56,739$$

3.) Incremental cost due to lost electric power generation when retubing aluminum heat exchangers.

Assumptions: 3 weeks (504 hrs.) downtime for aluminum heat exchanger retubing consisting of the following operations:

- Shut down power module
- Removal of tube bundle
- Removal of old tubes from tube bundle
- Installation of new aluminum tubes
- Replace tube bundle in heat exchanger
- Start up power module, resume power generation.

Downtime Cost (1993) = (504 hrs)(6250 Kw)(\$0.005/Kw-hr) = \$15,750

Present Worth = \$ 3771

The total for the three miscellaneous incremental costs (penalty for aluminum heat exchangers) are

\$ 43,873

\$ 56,739

\$ 3,771

\$104,383

5.3 COST ANALYSES AND ALGORITHMS

5.3.1 Heat Exchanger Cost Model

To facilitate parametric studies of the size and cost of the various OTEC power module configurations, it became apparent that a detailed and accurate cost model would have to be developed for the OTEC heat exchangers. The heat exchangers are the most costly components of the OTEC power module. For parametric trade studies and for final decision making, it is imperative to develop and use cost models that accurately reflect the costs of both prototype and production versions of the OTEC heat exchangers. Development of these cost models should be predicated on realistic heat exchanger fabrication cost data. To obtain these cost data, C. F. Braun was charged with developing specifications for three sizes of evaporators and condensers. Heat exchanger sizes chosen for this purpose were 18-, 35-, and 50-feet in diameter using 1.5-inch diameter, 25-feet long tubes.

The specifications prepared by C. F. Braun for the heat exchangers included drawings, dimensions, materials, mechanical details, pressure, temperature and fluid requirements, and applicable design and fabrication codes. These specifications are given in Appendix A.

Because of the tight schedule, only one fabricator, Wyatt Industries, was contacted for pricing information. Wyatt was selected because of its excellent background in both shop fabrication and field erection of large pressure vessels and heat exchangers. To facilitate obtaining accurate costs, C. F. Braun assisted Wyatt in the preparation of their estimates.

The Wyatt cost estimates are exclusive of tube costs; tube costs were obtained by TRW directly and were then added to the cost model generated from the Wyatt data.

Wyatt's cost estimates for the 35- and 50-foot diameter heat exchangers include field fabrication costs and a dockside facility cost charge. Wyatt envisions a special facility, built at dock-side, to be used to fabricate the larger heat exchangers. These field fabrication facilities would have special equipment not normally available in Wyatt's shop. The smaller, 18-foot diameter heat exchanger would be completely shop-fabricated and would therefore not require extra tooling or facilities.

Although Wyatt's shop fabrication is limited to a maximum heat exchanger diameter of approximately 18 to 20 feet, other heat exchanger fabricators apparently are able to shop-fabricate larger sizes.

Other vendor contacts were made to determine interest and capability for manufacturing OTEC heat exchangers: five suppliers confirmed that they could shop fabricate the 35-foot diameter exchangers in one piece and ship them to their destination. These five suppliers include two American nuclear fabricators, located on navigable rivers, with a lifting capacity of 1000 tons. The three Japanese suppliers are located on the water and also have the necessary lifting and handling capability.

Because of the apparent capability of some vendors to shop-fabricate heat exchangers up to 35-foot diameter, the cost model developed from Wyatt's shop fabrication costs for the 18-foot heat exchangers can be used for costing heat exchangers up to 35-foot diameter.

The detailed analysis to develop the cost algorithms, based upon the specifications prepared earlier in the program by C. F. Braun, are given in Appendix A.

Table 5-7 shows the most recent reformulation of the various heat exchanger cost algorithms.

These algorithms reflect the current configuration and components of the baseline design. The tabulation is also responsive to the requests from the D.O.E. to show the various heat exchanger component parts as listed.

Continual fine tuning of the original cost model (Appendix I) has resulted in the algorithms shown in Table 5-7.

These cost algorithms represent heat exchanger fabrication costs that may be expected in the future when large heat exchanger fabrication facilities will be dedicated solely to the fabrication of production item OTEC heat exchangers at shop-fabricated costs.

Table 5-7. Cost Equations

ITEM	MATERIAL	LABOR
Tubes	$N(C_1 L + C_2) \frac{D}{1.5}$	$28.1 D^{0.7} N$
Welding	0	$12.18 N^{1.03}, N \leq 36,000$ $0.727 N^{1.3}, N > 36,000$
Tubesheets Clad	$1.24 D^{2.30} N^{1.15}$	$0.457 N^{1.34} D^{0.68} (D-0.5) + 329 (DN^{1/2} - 148)^{1.5}$ if $DN^{1/2} > 148$ else 0
Unclad	$0.105 D^{2.5} N^{1.27}$	$0.307 N^{1.33} D^{0.66} (D-0.5) + 221 (DN^{1/2} - 148)^{1.5}$
Tube Support Plates	$9.22 (10^{-4}) D^{3.046} N^{1.523}$	$0.124 N^{1.295} D^{0.591} (D-0.5)$
Shell	$0.121 D^{2.29} N^{1.145} L$	$3.16 D^{1.654} N^{0.827} L^{0.7}$
Water Boxes	$0.175 D^{2.1} N^{1.05}$	$172.5 D^{1.32} N^{0.66}$
Nozzles	$79.6 D^{0.908} N^{0.454}$	$62.3 D^{1.34} N^{0.67}$
Ammonia Distribution (Evaporator only)	$0.024 D^{2.592} N^{1.296}$	$0.0962 N^{1.38} D^{0.756} (D-0.5)$
Bustle (Evaporator only)	$0.0104 D^{2.69} N^{1.345}$	$147.4 D^{1.35} N^{0.675}$
Water Inlets	$12.0 D^{1.28} N^{0.64}$	$92.3 D^{1.27} N^{0.635}$
Supports	$0.053 D^{2.090} N^{1.045}$	$27.1 D^{1.33} N^{0.665}$

	C1	C2
Titanium	2.83	1.00
Aluminum	1.38	5.50

5.3.2 Seawater Pump Costs

The algorithm for the cost of seawater pumps was based on data obtained through telephone conversations with pump manufacturers.

Manufacturers contacted included Johnston Pump Co.; Byron Jackson, Worthington Pump Co.; and Process Equipment Co. (distributors of Ingersoll Rand and Johnston pumps).

The pumps ranged in size from 155,000 gpm through 750,000 gpm with total dynamic heads of 8 through 12 feet. These pumps were vertical, wet-pit, propeller-type pumps with cast iron or cast steel columns, approximately 40 feet long, with protective epoxy coatings, stainless steel shaft, and bronze impeller.

Four pumps were priced within this range and, in all cases, the price of the pumps was very close to \$1/gpm.

Other materials, such as all stainless steel or aluminum bronze material, if required, will more than double the price of each pump.

Economies of scale become evident with large, bulb-type, axial pumps (larger than 1,000,000 gpm). The price of a 1,790,000 gpm pump at 7.4 feet TDH was approximately \$1,300,000, or about \$0.75/gpm. However, the present configuration of the OTEC power cycle on a floating barge does not lend itself to a suitable arrangement to use large, bulb-type propeller pumps (Figure 5-2).

5.3.3 Turbine/Generators

The algorithm cost for turbines and generators is obtained from data provided by manufacturers through CF Braun to TRW.

Figures 5-3 and 5-4 give the cost as a function gross power output.

Figure 5-2. Cost of Warm/Cold Water Pumps Versus Flow Rate

Figure 5-3. Turbine Cost Versus Gross Generator Power Output

Figure 5-4. Generator Cost Versus Gross Power Output

5.4 SENSITIVITIES OF DESIGN TO KEY DESIGN REQUIREMENTS

The cost/performance of the OTEC Power System is particularly sensitive to the following key parameters:

- heat exchanger tube diameter and length
- fouling factor
- available seawater ΔT

These parameters are discussed briefly below as to their impact on system design.

Tube Diameter and Length. The variation of evaporator cost with tube diameter and length is presented in Figures 5-5 and 5-6 for aluminum and titanium tubes, respectively. Curves are drawn corresponding to constant power output and constant pumping power.

The evaporator cost using aluminum (Figure 5-5) is shown to decrease significantly as both tube diameter and length increases for all power output levels. Note that length and diameter increase simultaneously along the constant power output curves while shell diameter decreases.

The decrease in cost with increase in tube diameter and tube length is associated with the decrease in shell diameter as well as the decrease in required number of tubes (and companion tubesheet drilling). Approximately the following relationships hold true for tube length (L), number of tubes (N) and tube diameter (D) for fixed duty and pumping power:

$$N \sim D^{-2}$$

$$L \sim D^{1.2}$$

$$A \sim N \times L \times D \sim D^{0.2}$$

The required heat transfer area (A) increases slightly with D. The more drastic decrease in cost with increasing tube diameter for aluminum tubes, Figure 5-5, compared to titanium tubes, Figure 5-6, is primarily due to the large wall thickness required for aluminum. This results in larger shell diameters with aluminum tubes and correspondingly larger sensitivity to tube diameter.

The selection of optimum heat exchanger design features is constrained by efficient hull packaging and/or production and shipping considerations,

NOTE: These curves are for constant 109 mils wall thickness.
This assumption will be reexamined during preliminary design.

Figure 5-5. Cost of Heat Exchangers Versus Tube Size for Range of Module Sizes

NOTE: These curves are for constant 28 mils wall thickness. This assumption will be reexamined during Preliminary Design.

Figure 5-6. Cost of Heat Exchangers Versus Tube Size for Range of Module Sizes

as well as by the cost trends discussed above. Total direct costs for the system are presented in Figure 5-7 and, again, illustrate the decreasing relation with tube diameter and length for both the evaporator and condenser. There are two limiting factors that have been included in the design optimization. These are:

1. Considerations of packaging and impact on hull draft with increasing tube diameter and tube length, and
2. Increasing ammonia liquid loading on the tubes of both the evaporator and the condenser with increasing tube diameter. The total tube periphery is proportional to $N \times D \sim D^1$ such that liquid loading or dry-out become a constraint on tube length. Preliminary analysis has shown that the effective tube length should not exceed about 45 feet (evaporator most critical).

Note that results presented here are based on constant aluminum tube thickness of 0.109 inches.

Fouling Factor. The decrease in evaporator heat transfer performance is reflected in the increasing system cost shown as a function of fouling factor in Figure 5-8. Performance and cost is quite sensitive to this factor, which underlines the importance of its understanding and assessment to the OTEC Program in general. The design conditions assumed in the present study are indicated.

Seawater ΔT . Optimization studies show that available seawater ΔT is the most important factor dictating system cost and performance because this factor represents the critical available resource that drives the heat engine. An example of the cost relationship with the ΔT design condition is shown in Figure 5-9.

Obviously variations in system performance will occur between different ocean regions and with different times of year because of the ambient thermal variations. The magnitude of the seasonal variability and its effect on system performance is estimated in Figure 5-10. The difference in plant output between summer and winter variation may be ± 25 KW output for a nominal 100 KW plant design. Comprehensive resource surveys now being conducted at the potential OTEC sites will define these temperature variations so that cost-efficient systems can be developed.

Figure 5-7. Effect of Tube Diameter on Direct Cost

Figure 5-8. Direct Cost Versus Fouling Factor for Smooth and Enhanced Tubes

Figure 5-9. Direct Cost Variation with Design ΔT

Figure 5-10. Plant Output Variation with Available ΔT

5.5 NON-RECURRING COSTS

Non-recurring costs of two types have been identified:

- Engineering. Non-recurring prototype or first article engineering costs have been identified as 15.5% of prototype field construction costs. Subsequent production engineering costs are estimated as 5% of production field construction costs, as described in Section 5.1.
- Fabrication Facilities. New dedicated heat exchanger fabrication facilities, described in Section 2.1.2 and Appendix A, were used as the basis for developing cost estimates of heat exchangers in this study. The non-recurring cost for these facilities is estimated to be approximately 10 million dollars.

5.6 PRODUCTION COST PROJECTIONS

In manufacturing processes it is well-documented that the more often an operation is repeated, the more efficient that operation becomes. This concept is applicable to the OTEC plants: if a series of OTEC plants is built, subsequent units should show cost savings over the first unit. The cost reductions can be estimated using the algebra of the "Experience Curve."

The Experience Curve states that costs show a consistent reduction between doubled quantities. For example, a representative industrial Experience Factor for labor is 80 percent. This means that the average per unit cost of producing two units is 80 percent of the cost of the first unit, the average cost of producing four units is 80 percent of the average cost of producing two units, etc. This applies to uninterrupted production of similar items. However, the concept is also applied to engineering and planning. Note that cost reduction is more dramatic for the low-serial-numbered items. If production is already significantly advanced "down the experience curve," costs will not be substantially lower for the next few units. Note also that the cost reduction through Experience Curve processes does not take place automatically: management planning and monitoring are necessary.

For projecting costs based on limited information, certain rules of thumb are used for the choice of curve factor. TRW manufacturing uses 92 percent when projecting costs based on dollar value of the first unit. For an OTEC plant, as for any complex assembly, the net factor is a composite of the component factors. It is known that a portion of the costs is not sensitive to Experience Curve considerations, e.g., materials or off-the-shelf items. The Experience Factor will be applied to that portion of the subsystems attributed to labor. Since this portion would be assigned a dollar value rather than a man-hour value, a ninety percent factor will be used unless experience indicates another value is appropriate.

An algebraic expression for the Experience Curve aspect is as follows: if the serial number of any unit is represented as 2^m , and the Experience Factor is 100 y percent, then the labor scaling factor for 2^m units is y^m .

For ease in computation only integer m values are used here and inflation is not taken into account.

The following paragraphs give derivations of the cost ratios utilized in the conceptual design evaluations.

Heat Exchangers. Evaporators and condensers are grouped together for this calculation. There are 32 units per 100 MWe plant. The eighth plant would contain the 256th unit. Assume this is the 256th production unit: $256 = 2^8$. If H_1 is the cost of the first unit, and $0.6H_1$ is the portion attributed to labor, then H_{256} will be approximately:

$$H_{256} = H_1 (0.4 + 0.6 \times 0.9^8) = 0.66H \approx 0.7H_1$$

The value 0.7 would be the Experience Curve Cumulative Average scale factor for 256 heat exchangers.

Water Pumps. With 32 units per 100 MW plant, the eighth plant would contain the 256th unit. Assume this is the 512th production unit: $512 = 2^9$. If W_1 is the average cost of 256 production units and $0.5W_1$ is the portion attributed to labor, then W_{256} (the 512 unit average) for an 80 percent Experience Factor (80 percent is chosen because this is a regularly manufactured item)s:

$$W_{256} = W_1 (0.5 + 0.5 \times 0.8^9 / 0.8^8) = 0.9W_1^*$$

Ammonia Systems. With 48 units per 100 MW plant, the eighth plant would contain the 384th assembly. Industrial experience shows fixed costs to be one-half the total, with 30 percent savings by the 100th assembly. (Though savings may be more from quantity savings than from Experience Factors.) Assume the assembly is the 384th production unit.

* Note that $0.85 W_1$ was used in our final projections based on latest available manufacturers recommendations.

A 30 percent saving by the 100th unit is approximately obtained with an 85 percent Experience Factor for 128 units. If A_1 is the cost of the first assembly and $0.5A_1$ is the portion attributed to labor, then

$$A_{256} = A_1 (0.5 + 0.5 \times (0.85)^8) \approx 0.63A_1$$

$$A_{512} = A_1 (0.5 + 0.5 \times (0.85)^9) \approx 0.61A_1$$

The average cost of 384 assemblies would be $\approx 0.62A_1$.

Turbines. With 16 units per 100 MW plant, the eighth plant would contain the 128th unit. Assume this is the 128th production unit. If T_1 is the cost of the first unit and $0.7T_1$ is the portion attributed to labor, then T_{128} for a 90 percent experience curve (chosen because it is a new design and manufacturers are not set up for such large production runs) will be:

$$T_{128} = T_1 (0.3 + 0.7 \times 0.9^7) \approx 0.63T_1$$

Ship Systems. TRW is not directly involved in the ship systems aspect of this design. An approximation to the Experience Curve factor is taken from a MITRE Corporation review of designs by the Applied Physics Laboratory and by Lockheed. Although the two designs involve different applications and operating conditions, a rough approximation can be obtained.

APL estimates that the sixth unit hull structure cost will be about 65 percent that of the initial unit; for Lockheed, the cost is about 80 percent the cost of the first unit. A cost ratio of 0.7 is chosen, based on the above numbers. Anything more than one significant digit is not justified in this case.

Miscellaneous Auxiliaries. The construction of an OTEC plant is a custom operation. Many of the smaller pieces will be special order items, and are treated as new production. Since the thrust of the entire design effort is to use available technology and components, some of the auxiliary

items will be off-the-shelf. Consider that half the items are off-the-shelf and the other half are new production. From plant to plant there will be some design learning so that the special order items would almost always be new production from the standpoint of the Experience Curve. As an approximation to the cost of new-production items, assume serial number $4 = 2^2$ has a representative average value. Then, if M_1 is the total cost of miscellaneous auxiliaries for the first plant, the average cost of these auxiliaries for the eighth plant, M_8 , using a 90 percent Experience Factor, would be $M_8 = M_1 (0.5 + 0.5 \times 0.9^2) \approx 0.9 M_1$. Thus, the cost ratio would be 0.9.

6. RECOMMENDATIONS

6.1 PROOF-OF-CONCEPT SCALED PILOT PLANT

Recommendations for a scaled intermediate size power system (5 MWe nominal) corresponding to the full scale conceptual design have been developed based on an analysis of subscale proof-of-concept testing requirements. The objectives of and the rationale for the selected pilot plant testing are presented below.* Reduced scale heat exchangers (1 MWe nominal) will first be developed for use in early test and design verification studies on the OTEC-1 platform prior to implementation of the 5 MWe Pilot Plant. These heat exchangers are described in Section 6.2.

6.1.1 Test Objectives

The overall test objectives for the proof-of-concept power system can be summarized as follows:

- Demonstrate net power generation
- Develop system performance data required for full scale design
 - Thermal-hydraulic
 - Power cycle efficiency (gross and parasitic power)
 - Materials performance
- Analyze system dynamics (including effect of ship motions)
 - Thermal-hydraulic transients
 - Electrical transients
 - Mechanical transients (stress and strain)
- Validate biofouling and corrosion control approach
- Demonstrate instrumentation and control approach
- Develop operational procedures, including startup, shutdown, and load shedding
- Develop maintenance and repair procedures
- Develop safety procedures
- Collect reliability, availability, and life data
- Develop cost data to project full scale power system economics.

These objectives have been used as the basis for developing the scaled pilot plant configuration and recommended test procedure presented below.

* See also Appendix B-9: Rationale for module and test article sizes.

6.1.2 Analysis of Scaled Testing Requirements

Selection of a scale for the test articles will necessarily be a compromise between the cost of large scale testing on the one hand, and the risk associated with scaling up to prototype (full scale) from smaller test articles on the other hand. Because overall OTEC Program viability would be seriously jeopardized by failure of the pilot plant to perform, however, the tendency in test article selection was to move cost/risk tradeoffs in the direction of lower risk (similarly with time/risk tradeoffs). It was assumed that the cost of this intermediate size power system would not be as critical as the development of a capability to project economics to full scale operation.

The scaling problem must consider the following facts:

- Different components (heat exchangers, turbines, etc.) scale differently because scaling depends on such physical phenomena as heat transfer, fluid flow, etc.
- Certain physical phenomena are more amenable to analysis than others.

In the following paragraphs the need for scaled testing of major power system components is addressed, followed by a discussion of system considerations.

6.1.2.1 Heat Exchangers

The state of the art of two-phase heat transfer is one where empirical methods dominate. Correlations are generally valid only over a limited range of variables. For our proposed heat exchanger concept, all tubes behave (to a good approximation) the same way. For example, by design, vapor shear effects will be small. On the NH_3 side, a major uncertainty will be the effect of ship motions; both the flow of liquid down the tubes and the distribution of liquid onto the tubes (evaporator) will be affected by motion (heave, roll, etc.). On the waterside, the spatial distribution will be strongly impacted by motion, and the tube velocity will also vary with time (resulting primarily from heave).

The shell side steady state and dynamic performance can be evaluated using a relatively small bundle size (of the order of 1000 tubes). The distribution effects (water distribution to each tube and ammonia distribution in the evaporator) are highly dependent on the size of the tubesheet and the motions the exchanger experiences. While analysis and model studies on a laboratory scale will provide useful data, large scale heat exchangers must be tested in the correct dynamic environment for proof-of-concept.

6.1.2.2 Turbo-Generators

High efficiency of turbines operating with ammonia at low pressure ratios has been assumed for the OTEC power system (85 percent overall efficiency based on input from turbine manufacturers). This has not been proven on the scale required for OTEC; it depends on optimizing the fluid dynamic design. Because the thermal environment is benign, close tolerances necessary for such high efficiency can be maintained. The performance of turbines can be projected from scale tests with high confidence. The use of Moody type formulas to relate efficiency of the scaled model to the prototype is well established. The actual designs will vary with size: the wheel diameter of a single stage turbine producing about 16 MWe gross (~12.5 MWe net) will typically be 6 to 8 feet, while a 1.3 MWe gross (~1 MWe net) turbine will have a wheel diameter of about 2 feet.

6.1.2.3 Pumps

Both seawater and ammonia pumps in the sizes being considered for the prototype are commercially available and are not a major consideration with respect to performance scaling of the power plant system.

6.1.2.4 System Considerations

A full scale prototype power plant module of TRW's design will probably consist of several (one to four) evaporator/condenser loops coupled electrically by the busbar and hydraulically by the cold water pipe and the common water supply troughs. Analysis of transients caused by motion of the platform, as well as startup and shutdown transients, will be complicated by this coupling of individual evaporator-condenser

loops. Hydraulic transients have been studied in some detail and it is believed that these transients can be understood based on analysis and model basin studies. The resulting electrical transients may lead to significant problems of synchronizing the output power from each generator. This problem and its impact on scaling will have to be studied in more detail.

The motions of the platform will impact the performance of the power plant in ways which may be categorized according to whether or not they are strongly module size-dependent.

The following phenomena are clearly very dependent on physical scale:

- Spatial distribution of water to the heat exchanger tubes
- Spatial distribution of ammonia to the heat exchanger tubes
- Fluid transients in the water distribution troughs (these will depend on physical dimensions, particularly channel length and depth)
- Structural loads, both steady state and transient (equipment loads, driveshaft loads, etc.)
- Electrical power transients (magnitude of power surges and associated synchronization problems).

To realistically evaluate these phenomena experimentally, scales approaching prototype scales will be required.

Other physical phenomena are, at least to a first approximation, not strongly dependent on the physical size of the power plant.

Examples are:

- Temporal variations in water flow in the water supply and discharge pipes caused by platform motions which are functions of pipe length rather than pipe diameter (i.e., U tube-type oscillations)
- Transient response characteristics of the power loop, i.e., time constants, time delays, and frequencies of oscillations. These will typically depend on flow velocities, pipe and heat exchanger tubes, lengths, but not number of tubes.

Transients which do not depend on actual power system output should first be studied at subscale.

6.1.3 Rationale and Recommendations for Scaled System Selection

The above discussion has led us to postulate the following set of scaling guidelines. The relative importance of each item has not been assessed at this time.

- The proof-of-concept heat exchangers must be similar to the full scale prototype exchangers with respect to thermal fluid behavior. This leads to the following requirements:
 - Same tube geometry (tube diameter, tube length, and pitch)
 - Same operating conditions (temperatures, fluid velocities, and thermal duty per tube)
 - Comparable shell size to prototype to simulate fluid mechanic effects of distribution and platform motion. (This may suggest shell diameters not less than about 1/2 prototype, or possibly special shapes for the heat exchanger shells, e.g., pie shape.)
- The heat exchangers will set the size of machinery components, not the reverse.
- The proof-of-concept power module must be tested onboard a platform which will (or can be made to) respond to wind and waves in essentially the same manner as the full scale platform.
- The water supply, distribution, and return subsystems must be scaled to produce full scale transient response characteristics. This can be done by applying the laws of similitude to ensure that the hydraulic transient will be representative of the full scale plant. The lengths of the supply and return pipes, however, must be the same as for the full scale plant in order to investigate power loop transients.

- Problems associated with electrically coupling, via the busbar, of several Rankine cycle loops will require that at least two such loops (i.e., two evaporator-turbine/generator-condenser loops) be tested in parallel. This would also provide verification of the hydraulic performance of our trough concept for delivering water to multiple units.

Assuming that TRW's full scale concept will employ heat exchangers of about 20 feet shell diameter with two or more evaporators/condensers required per module, the following two stage test program is recommended for proof-of-concept:

- (1) Test a single power loop of about 5 MWe net output (6.25 MWe per full scale TRW module design) employing heat exchangers of approximately 20-feet shell diameter on an oversized platform. Provide a cold water pipe of double capacity (approximately 14-feet in diameter). This test will demonstrate individual power cycle loop performance for sizes comparable to full scale.
- (2) If the first stage of the proof-of-concept program is successful, add one more power cycle loop of the same capacity (approximately 5 MWe net). The verification of dynamics of electrically and hydraulically coupled modules, including synchronization and conditioning of the output power, will be the major objectives of this second stage of the proof-of-concept test program.

Layout drawings of a proof-of-concept power system test loop have been generated consistent with this recommended test program. Test loop hardware will be scaled to provide 6.25 MWe output, which is identical to the recommended full scale module design and is also reasonably close to the 5 MWe test guideline. These drawings, presented in Figures 6-1 through 6-4, include process flow diagram and hardware arrangement within the hull platform.

Figure 6-1. 6.25 MWe Power Module Process Flow Diagram Scaled Pilot Plant

Figure 6-2. 6.25 MWe Power Module General Arrangement Elevation

POWER STATION PLAN

PUMP STATION PLAN

1/8" = 1'-0"
SCALE
5 10 15 20
FEET

INITIAL ISSUE	1-12-78	ROP
LTR	DESCRIPTION	DATE APPROVED
REVISIONS		
OTEC POWER SYSTEM DEVELOPMENT		ENVIRONMENTAL, POWER AND FACILITIES DEPARTMENT SYSTEM INTEGRATION LABORATORY
6.25 MWe POWER MODULE GENERAL ARRANGEMENT POWER & PUMP STATIONS.		
DATE JAN 12, 1978	DRL NO.	
DRAWN R. COREY	6850.38-52	
CHECKED	APPROVED - PROGRAM OFFICE	
APPROVED R. O. PEARSON		

Figure 6-3. 6.25 MWe Power Module
General Arrangement Power
and Pump Stations

Figure 6-4. 6.25 MWe Power Module
General Arrangement
Steel Framing and
Elevation

6.2 PROOF-OF-CONCEPT SCALED HEAT EXCHANGER

Initial ocean testing of scaled power system components will take place aboard the Early Ocean Test Platform (EOTP-1, also referred to as OTEC-1). An approach to the design and test of scaled vertical tube, falling film, heat exchangers (1 MWe, nominally) has been developed for this test phase as part of the current study. The guideline in this development is that the test articles be as representative as possible of the full scale power system module.

In developing a design philosophy for the test articles, it was recognized that the functional risk that the heat exchanger concept will not work is small. Because the heat exchangers are a major portion of total cost, however, performance optimization and cost reductions should be the major objective of the test program. It was therefore decided to reduce test article scale to the extent that is physically reasonable, and to develop alternate designs (enhancement, liquid distribution, materials) for parallel testing. This approach will enable the evaluation of the optimum design within the tight schedule constraints of the overall OTEC Program.

The recommended heat exchanger test articles would be made up of 100 to 1000 tubes of full scale diameter in the vertical tube, falling film configuration presented in Section 3.1. Tube length would be maintained at near full scale dimension, about 40 feet, to enable direct scaling of heat transfer performance of individual tubes. The small tube bundles are not critical to the performance evaluation since tubes act essentially independently, a major advantage in testing the falling film concept. The coupling between tubes due to the common supply plenum (evaporator) and vapor flow effects can be explored at this scale by using a rectangular "slice" that is narrow in one direction and close to full size in the other.

The rough order of magnitude cost to fabricate one 100 tube heat exchanger is \$42,000, based on assumption of \$20 per square foot of tube area.

The small scale heat exchangers described above can be developed at a relatively low cost in a number of thermal designs and materials to perform a series of parallel tests. This will permit experimental verification of:

- Effect of tilt on tube performance
- Effect of heave on tube performance
- Vapor in-flow/out-flow interactions

for candidate exchanger designs.

In addition, resolution of other performance questions can be accomplished, for example:

- Tube cleaning techniques and efficiencies (biofouling)
- Service life (corrosion, erosion)
- Single tube heat transfer characteristics (tube interaction is small).

6.3 TECHNOLOGY DEVELOPMENT RECOMMENDATIONS

During conceptual design, performance, fabricability, and cost assumptions were required. Although the present concept and approach are not contingent upon verification of these assumptions (because a conservative posture has been adopted and the project can progress satisfactorily through preliminary design without further development), design optimizations and cost reductions resulting from technology development will be incorporated where appropriate.

This section summarizes those areas where technology development will benefit the Power System Development Program. Additional recommendations formulated during the preliminary design phase will be highlighted at that time.

6.3.1 Materials and Processes

Heat Exchanger Tube Surface Enhancement

TRW is pursuing the development of the upsetting/rolling tubeforming process to the physical dimensions required of this design. Additional development work on the process would increase the production rates thereby reducing costs. (The Grob process is discussed in detail in Appendix H.) In addition, the feasibility of other forming processes, such as brake forming or pre-forming sheets prior to tube formation, should be investigated. Such development will assure an economic alternative to forming the flutes by processes other than extrusion.

Aluminum Life Projections

Aluminum would be preferred as a tube material if the expected life in the seawater environment could be projected beyond the present design life of 15 years to approximately 30 years. The current data available on the corrosion characteristics of aluminum are the major deterrent to such projection because such data are inconclusive. As discussed in Appendix H, since some data indicates long life, others indicates short life, design assumptions/assessments have been based on averages. Firm data on aluminum would enable definitive design conclusions.

Temperature Dependence of Aluminum Corrosion Data

A limited laboratory study indicated a significant temperature dependence of the general corrosion and pitting corrosion rates for 5052 aluminum in seawater (Reference 50, Appendix H). Further testing should be conducted to substantiate these data.

Corrosion-Fatigue of Aluminum Alloys

The available corrosion-fatigue data for aluminum in seawater indicates drastic reductions in fatigue strength over the values obtained in air (Appendix H, Section 4.) The impact of these data on aluminum heat exchanger design should be established and, if necessary, further data gathered on specific OTEC environments. The effect of ammonia or seawater-ammonia mixtures on fatigue strength should also be determined.

Alclad Compatibility with Mechanical Antifouling Systems

The use of aluminum may possibly be exploited by Alclad. An area of concern, however, is in the compatibility of Alclad aluminum tubes with mechanical cleaning systems. The opinions of experts tend to doubt such compatibility due to the irregular nature of the cladding corrosion (Appendix H, Section 6). It is possible that the abrasive action of a mechanical cleaning system will tend to minimize the localized character of the surface attack, however, and the demonstrated increase in corrosion resistance of aluminum could then be exploited. Determination of Alclad performance in seawater with mechanical brushes and the Amertap antifouling system would be valuable.

Tube-to-Tube Sheet Joining

Considerable labor is presently required in the process of providing a reliable seal at the tube-to-tube sheet joint. Alternatives include mechanical expansions, explosive bonding, adhesives, 'O' rings, magna forming and welding. Although all these are alternatives feasible, specific reliability data in an OTEC environment should be developed.