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**ATTACHMENT 2
TO
OCEAN THERMAL ENERGY CONVERSION
POWER SYSTEM DEVELOPMENT-I
PHASE I — FINAL REPORT**

PRELIMINARY DESIGN REPORT — VOLUME I
LMSC-D630248 • 18 DECEMBER 1978
CONTRACT EG-77-C-03-1568

Prepared for
UNITED STATES DEPARTMENT OF ENERGY
DIVISION OF SOLAR ENERGY
by
OCEAN SYSTEMS
LOCKHEED MISSILES & SPACE COMPANY, INC.
SUNNYVALE, CALIFORNIA 94086

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**OCEAN THERMAL ENERGY CONVERSION POWER SYSTEM
DEVELOPMENT-I**

Phase I—Final Report. Preliminary Design Report—Volume 1

December 18, 1978

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

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

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

VOLUME I

18 DECEMBER 1978

CONTRACT EG-77-C-03-1568

Prepared for

**UNITED STATES DEPARTMENT OF ENERGY
DIVISION OF SOLAR ENERGY**

by

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

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FOREWORD

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

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

The corporate team members were:

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

The subcontractors were:

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

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

The complete report consists of the following volumes:

Volume I — Preliminary Design Report LMSC D-630248

Volume II — Appendixes:

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

Volume III — Appendixes:

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

Volume IV — Appendix G. Ammonia Cycle and Auxiliaries

Volume V — Appendixes:

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

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Section 1

INTRODUCTION

This report presents the results of a conceptual and preliminary design study of Ocean Thermal Energy Conversion (OTEC) closed loop ammonia power system modules performed by Lockheed Missiles & Space Company, Inc. (LMSC) for the Department of Energy Division of Solar Energy. This design study, Task 2 Preliminary Design, the second of 3 tasks in Phase I of the Power System Development-I Project was performed under contract EG-77-C-03-1568 from Feb 1978 through October 2, 1978.

1.1 OBJECTIVES/GUIDELINES

The Task 2 objectives were to develop:

- Conceptual designs for a 40- to 50-MW(e) closed cycle ammonia commercial plant size power module whose heat exchangers are immersed in seawater and whose ancillary equipments are in a shirt sleeve environment
- Preliminary designs for a modular application power system sized at 10-MW(e) whose design, construction and material selection is analogous to the 50 MW(e) module, except that titanium tubes are to be used in the heat exchangers
- Preliminary designs for heat exchanger test articles (evaporator and condenser) representative of the 50-MW(e) heat exchangers using aluminum alloy, suitable for seawater service, for testing on OTEC-1.

The reference ocean platform was specified by DOE as a surface vessel with the heat exchanger immersed in seawater to a design depth of 0-20 ft measured from the top of the heat exchanger. For the 50-MW(e) module, the OTEC 400-MW(e) Plant Ship, defined in the Platform Configuration and Integration study, was used as the reference platform.

The major guidelines under which the Task 2 study was conducted are presented in Table 1-1.

Table 1-1

TASK 2 GUIDELINES

1. Power module includes those elements and components, auxiliaries and ancillaries for power generation. Platform/ocean engineering components are not included.
2. Design temperature difference (T) between warm and cold water is 40°F (+ 4, to -11°F). The temperature range for warm water is 70-85°F and cold water 32-50°F
3. Power module design life is 30 years with 1000 start/stop cycles. This time period is interpreted to allow periodic replacement of components to provide cost-effective life.
4. Warm seawater subsystem pressure drop is 10 ft of water; cold seawater subsystem pressure drop is 15 ft. The valves may be varied if shown to be technically and economically feasible.
5. Heat exchangers are horizontal shell and tube with heat transfer enhancement.
6. Heat exchangers of the Commercial and Modular Application Power Systems are to be immersed in seawater with their ancillary equipment maintained in a shirt-sleeve environment. "Immersed" defines the depth of water at the top of the heat exchangers to be 0 to 20 feet.
7. Modular Application Power System 10-MW(e) is to be self-contained. All ammonia, electrical and support auxiliary and ancillary equipments shall be provided with the power modules in order that the operation of the power module is "self-sufficient."
8. Biofouling heat transfer resistance factor shall not exceed 0.0005 hr-ft² °F/BTU.
9. Module operability will be at least 90 percent at a 60 percent confidence level. Starting reliability will be at least 95 percent with a 60 percent confidence. For module availability use 8760 hrs/year.
10. Power systems are to be capable of normal operation at sea-state six and maintain structural integrity under sea-state nine conditions.
11. Heat exchanger test articles selection will be based on
 - a. ability to generate data for design of large heat exchangers
 - b. consideration of reducing test article cost by a size reduction
 - c. compatibility with OTEC-1 system cold and warm water and ammonia supplies.
12. Heat exchanger test articles are to have an operating life of 4 years and a total "shelf life" of 10 years.

1.2 PROJECT TEAM

The project team members and their responsibilities were:

Foster Wheeler Energy Corporation (FWEC) - responsible for verifying heat exchanger thermal performance and conducting the overall heat exchanger design.

Bechtel National, Inc. (BNI) - responsible for the ammonia cycle, ammonia auxiliaries and power control, and power distribution equipments.

Turbine Subcontractors - The choice of a turbine-generator was not finalized. The availability of axial and radial inflow turbines, and the fact that one subcontractor specializes in only one type of turbine, made it prudent to have two subcontractors:

- Rotoflow Corporation - responsible for the radial inflow turbine and turbine control design, and the selection of the matching electrical generator.
- AIRCO Cryogenics - responsible for the preliminary design of the axial flow turbine and turbine control design and the selection of the matching electrical generator.

Lockheed Electronics Company (LEC) - responsible for the design and manufacture of the Data Acquisition and Control System (DACS).

Lockheed Shipbuilding and Construction Co. (LSCC) - responsible for the establishment of facilities, equipments, schedules, and costs for the assembly of power system modules.

Lockheed Missiles & Space Co., Inc., Ocean Systems Division - responsible for project management and system engineering, heat exchanger performance, seawater system, biofouling countermeasures and corrosion control and diagnostic instrumentation.

Section 2

SUMMARY

The results of the Task 2 efforts are as follows:

1. Heat exchangers decrease in cost per unit of power with increasing size. The 50-MW(e) heat exchanger costs per kW were approximately 10 percent less than the 25-MW(e) size.
2. Aluminum heat exchangers are less expensive on a life-cycle basis than titanium or stainless steel.
3. Sea water depth to top of heat exchangers is 20 ft so minimize loads on the heat exchanger shells.
4. Spherical shell heat exchangers are cost effective and minimize system complexity through the use of the generous vapor gallery
5. Multi-level liquid ammonia distribution through an irrigation tube arrangement ensures tube wetting while preventing tube flooding.
6. Ammonia-side heat transfer enhancement is slightly cost effective; seawater-side enhancement appears not to be cost effective.
7. Variable-flow seawater pumps are cost effective at operating sites where there is a seasonal ΔT variation larger than 4°F.
8. Multiple pumps reduce the length of the seawater path and improves system availability.

9. Radial inflow turbines are slightly lower in cost and are not affected by power system vapor quality, based on experience with geothermal applications.
10. Biofouling countermeasures schemes (chemical and mechanical) of chlorination and M.A.N. brushes are effective and compatible with the heat exchangers.
11. Design and construction of test articles, modular application power system and commercial size power modules are similar for heat exchangers and ammonia cycle equipments.
12. Fully integrated power modules externally adaptable to spar-type platforms are least expensive, lend themselves most readily to scheduled maintenance, and require least capital for construction facilities.
13. Self-contained, 10-MW(e) Modular Application Power System titanium-tubed heat exchangers, are twice as costly per unit of power as 25-MW(e) aluminum-tubed heat exchanger for modules integrated into a 100-MW(e) demonstration plant.
14. Modular Application Power System module is a complete configuration. It provides all necessary equipment for power system operation and flexibility in the equipment arrangements, demonstrates technical feasibility, verifies economics, and allows gaining of operating experience.
15. Heat Exchanger Test Article Systems are representative of full-sized heat exchangers, and, with vapor augmentation and liquid distribution control, provide ability to verify thermal and fluid dynamic performance and mechanical characteristics, and to obtain operational characteristics.

2.1 TRADE STUDIES

Major design decisions for the 40-to 50-MW(e) power module, made as a result of trade studies conducted during the preliminary design, as follows:

Heat exchangers. Monolithic spherical shell and tube heat exchangers are most cost effective for the power module application. Multiple heat exchangers of this type in a single power module lead to increased complexity and cost, and offer no advantage in availability and only a slight advantage in producibility. For production purposes, monolithic heat exchangers have the system cost advantage.

The spherical shell configuration reduces system complexity and cost by permitting use of the volume within the shell for installation of the mist extractors, all liquid ammonia distribution piping, and integral sumps. Thus the design pressure, environmental conditions, and costs are lower than if these components were independent items. The larger vapor gallery within the spherical shell improves vapor flow distribution and reduces flow losses. There is no significant difference in cost between a spherical shell and a cylindrical shell of the same thermal size. With the above advantages in mind, spherical heat exchangers were selected for the 50-MW(e) commercial plant module conceptual design. This selection was carried through to the 10-MW(e) demonstration module and the test article heat exchangers.

Materials. Candidate materials considered in this study for heat exchanger tubes are aluminum, stainless steel, and titanium, in order of increasing cost and projected useful life. Aluminum is the most cost effective on a life cycle basis and was chosen for the 50-MW(e) power module. Aluminum was also specified by DOE for the heat exchanger test articles (HETA). Verification of the material properties in an ammonia/seawater environment will be part of the OTEC-1 tests. For the 10-MW(e) power module, however, the use of titanium

tubes was dictated by DOE since, for a power plant demonstration unit, the lowest risk approach is of prime importance. Titanium, although much higher in cost than aluminum, has a proven record of long life and trouble free operation in seawater, affording a lower risk.

Seawater and Ammonia Systems. Variable seawater flow has the advantage over constant flow in increasing net plant output at sites where seasonal ΔT variations exceed 4°F or for extended periods exceed 2.5°F .

Seawater pump drive options investigated were:

- Constant speed electric drive with fixed pitch impeller
- Constant speed electric drive with variable pitch impellers
- Variable speed electric drive with fixed pitch impeller
- Constant speed electric drive with fluid or eddy current coupling with controllable output speed
- Ammonia-turbine driven constant pitch impellers

For operating sites with seasonal seawater ΔT variations smaller than 4°F the constant speed electric drive with fixed impeller was the lowest cost choice. However, for seawater ΔT variations larger than 4°F , a variable speed drive or variable pitch impeller is required. A final choice was not made at this time because not enough data was received on the ammonia-turbine drive which looked good after a cursory investigation. A detailed analysis of the cost, performance and efficiency, and producibility of the ammonia turbine drive has as yet to be performed by selected vendors.

Ammonia flow will be provided by multiple, partial capacity pumps with a spare in the system. A combination of throttling and speed control appears to be the most effective means of flow control over the required range. Speed control by itself over the wide flow range required would be rather high in cost.

Module Configuration. Analyses leading to design decisions which were incorporated in the various power module configurations include the following:

- Sensitivity analysis of heat exchanger costs as a function of tube diameter and length/diameter ratio determined the heat exchanger dimensions.
- Heat transfer enhanced ammonia-side and non-enhanced (smooth) seawater side heat exchanger tubes are most cost effective.
- Heat exchanger cost was analyzed as a function of shell and internal support structural options, leading to the immersed-segmented baseline design.
- Heat exchanger tube vibration analysis leads to selection of the tube support baffle spacing.
- The immersed power module (heat exchangers side by side, 20 ft below the sea surface) has a higher cost than the submerged (evaporator above condenser, at 150 ft and 290 ft depth, respectively) module.
- The segmented power module (heat exchangers separately removable) provide greater maintenance flexibility than the unitized (heat exchangers joined structurally).
- Two stage distillation was selected over single stage for the ammonia cleanup system.
- The semi-circular tube bundle configuration with vertical splitter plate was selected for the test article with both enhanced and smooth tubes for test purposes.

2.2 CONFIGURATIONS

Studies and analyses of equipment, equipment arrangements, handling techniques, maintenance procedures and cost impact have lead to the baseline configuration to be the "immersed-segmented" power system module. The term "immersed" describes the 0 to 20 ft depth position of the heat exchanger top. The term "segmented" indicates that the power system contains several major hardware subsystem segments, which are installed separately on the ocean platform to collectively form the power system, and which can be removed separately for maintenance or replacement. Immersed heat exchangers reduce the design pressure in the ammonia side of the heat exchanger from 250 psig for an uninsulated pressure vessel on deck to 153 psig, thereby reducing the heat exchanger costs by 20 percent.

One concept for the immersed power system concept structurally connects all the power system components (including the heat exchangers) into one integral module which is designated as the "immersed-unitized" power system. This complete power module with all rotating machinery incorporated is attached to the ocean platform as a unit and is removed as a unit for maintenance or refurbishment. This difference relates to maintenance philosophies for the subsystem. The unitized power system also provides one vertical access trunk with a cross member for each machinery room. The unitized power system is disconnected from the platform and towed to shore for major maintenance and refurbishment, thereby requiring less equipment, space, and crew on the platform.

An alternative conceptual configuration is the "submerged power system" in which the heat exchangers are deeply submerged, as for installation on a spar platform. The deeply submerged heat exchangers are more nearly pressure-balanced (seawater head) thereby reducing the structural loads, weights and costs. The operating conditions for the heat exchangers are also improved

since spar platform motions and seawater circuit dynamics are reduced from those of a surface platform using immersed heat exchangers. The unitized concept has been applied to the submerged power system to avoid making and breaking many subsystem connections below the surface of the water for power system maintenance and refurbishment. This concept uses the same components as the immersed configurations, with the already noted exception of the heat exchangers.

The arrangement of the components in the power module emphasizes both compactness to minimize cost of piping, structure, and flow losses, and convenient access for maintenance purposes. The reference ship-type platform was assumed to have adjacent cold and warm seawater connections at the same depth of centerline. This interface definition forced a side-by-side arrangement of the evaporator and condenser. All ammonia lines are designed to be connected above the water surface.

Key features of the equipment and the general arrangement are:

- Control of liquid ammonia film throughout the evaporator to ensure that optimum heat transfer is accomplished by irrigating the tube bundle in sections each of which contain an optimum number of tubes per column. Adequate tube wetting can be provided at a reflux rate chosen to minimize the cost and parasitic power of ammonia distribution pumping.
- Integral circumferential fins on the exterior of the tube significantly increase ammonia heat transfer rates with no increase in parasitic power. The fins also improve the uniformity of flow of the thin film of ammonia.
- Tube size is optimized in terms of total power, systems cost, and overall performance. The selection of tube material and heat transfer configuration also influences the selection of tube size. For the 50-MW(e) module, the aluminum 5052 tube has a 1.5 in. O.D. with a nominal wall thickness of 0.1025 in.

- Aluminum tube material was selected because of its cost effectiveness and low replacement (once every 10 years or better) costs over the 30-year life of the power system.
- Locating hook and vane mist extractors inside the spherical shell eliminates the need for a separate pressure vessel for the mist extractor and allows simple gravity drainage of the liquid ammonia into the evaporator sump.
- Double tubesheets of aluminum with the inside tube sheet in an annular configuration reduces the tubesheet thickness, weight and cost of material and machining. Structural analysis has confirmed adequacy of annular inner tubesheets using the outer tube rows for stiffness.
- Biofouling in heat exchangers is controlled by a combination of chemical and mechanical systems. Chlorination limits the biofouling, and a M.A.N. brush device cleans each tube at regular intervals.
- Ammonia piping sized for optimum cost and performance results in a vapor velocity of 140 to 160 fps and a liquid velocity of about 20 fps.
- Multiple ammonia feed pumps minimize the impact of pump failure. Commonality of parts in the ammonia pumps has been achieved by using a single spool pump for ammonia transfer and a double spool pump of identical design for ammonia distribution.
- Multiple seawater pumps for each heat exchanger minimize the drop in level of power output in the event of a pump malfunction. Inlet and discharge diffuser lengths, related to pump wheel diameter, are reduced by using multiple pumps, thereby reducing seawater duct length and, consequently, cost.
- Both the radial and axial inflow turbines adapted from existing commercial hardware can meet all requirements for the turbine-generator system.

- The control system operates in an automatic control mode with manual override. The control and instrumentation system uses analog signal sensing, digital signal transmission and computation. All control, diagnostic, housekeeping and safety data will be acquired using off-the-shelf instrumentation.

2.2.1 50-MW(e) Power Modules

The 50-MW(e) Commercial plant size Power Module (CPM) is discussed in this section and illustrated in Fig. 2-1.

2.2.1.1 Immersed-Segmented Arrangement. The power module consists of two major assemblies, each incorporating one of the heat exchangers and components for various subsystems. For the radial inflow turbine configuration, the turbine and generator are included with the condenser assembly while for the alternate axial flow turbine configuration, they are part of the evaporator assembly.

The heat exchangers are immersed in seawater inside or outside the platform hull. The top of the heat exchanger is 20 ft below the waterline in order to minimize loads imposed on the heat exchanger shells and attachments to the platform due to ocean surface motions. Module attachment points to the platform are primarily at each seawater duct/pump housing and the main deck.

The evaporator has a 78.74-ft diameter truncated spherical aluminum shell with the upstream truncated end attached to the seawater pump ducting which in turn is mated to the platform. A machinery room, attached to the bottom of the evaporator, houses the ammonia distribution pumps and related plumbing in a shirt-sleeve environment. Access to the machinery room is via a 12-ft diameter

Fig. 2-1 General Arrangement 50-MW(e)
Power Module, Immersed-Segmented

multipurpose trunk. In addition to providing for personnel access and equipment replacement, the trunk also serves as a conduit for Heating, Ventilating, and Air Conditioning (HVAC) ducting, instrumentation and control cables, electrical cables, safety subsystems and drain lines. Two ammonia vapor pipes are attached to and lead to closure valves and flanges located on the top deck for connection to the turbine vapor supply pipes.

The condenser also has a 78.74 ft diameter truncated spherical aluminum shell tilted 15 deg (Fig. 2-1) to provide for improved condensate drainage. This rotation requires an upstream inlet transition piece as part of the seawater ducting. The downstream end is provided with a structural ring for attachment to the cold seawater discharge pipe. As with the evaporator, the machinery room containing the ammonia condensate pumps and related plumbing, is attached to the base of the condenser. The condensate return line is attached to the condenser, rises to the top deck, and is provided with a closure valve and flange for connection to the evaporator ammonia supply piping.

The turbine-generator sets are mounted on the top deck, and low angle diffusers connect the turbine discharges to the condenser vapor inlet ports.

Platform related structure provides the connection between the evaporator and its section of the operations and machinery deck. This structure is designed to accommodate all loads associated with deployment and platform installation. The evaporator portion of the machinery deck incorporates the ammonia cleanup subsystem, the non-condensable gas removal subsystem, the nitrogen purge and storage subsystem, the ammonia analysis and leak detection subsystems, and the chlorination subsystem. The machinery deck structure and the installation of all of its equipment is part of the platform.

The radial inflow turbine-generator arrangement provides 70-MW(e) gross using four turbine casings (8 wheels) and two 35-MW(e) generators in two units.

The generators are driven from both ends through a coupling arrangement such that if one wheel or casing fails then the remaining two wheels can continue to drive the generator. The dimensions of each unit are 41 ft long, 13 ft high, and 11 ft wide. The axial flow turbine-generator arrangement provides one wheel and casing with the turbine wheel and generator rotor mounted on the same shaft. The unit is 59 ft long and approximately 34 ft diameter on the casing.

Four single stage, high flow, low head, axial flow pumps provide the seawater to each heat exchanger. The pumps are clustered within a 55 ft diameter to enhance flow distribution to heat exchanger with a minimum transition ducting length.

Platform-related structure provides the connection between the condenser and its section of the operations and machinery deck. This deck contains the data acquisition and control subsystem, the electrical equipment, and the generator cooling subsystem. Again, the machinery deck structure and the installation of its equipment is part of the platform. All appendages to each heat exchanger, including ammonia piping and access tunnel, are rigidly supported by the heat exchanger shell or machinery space. In transit, the exposed openings of each segmented heat exchanger piping and access are sealed with blind flanges to preclude entrance of seawater.

The segmented module arrangement weighs approximately 10,800 long tons, with a net positive buoyancy of approximately 4,100 tons when immersed to the operational depth. This net buoyancy is the result of the following buoyancy elements:

- Operating spaces situated above the heat exchangers
- Watertight volumes of the heat exchangers
- Two machinery rooms located below the heat exchangers
- Machinery room access trunk

The segmented evaporator, including operations space, machinery room and access trunk, weighs 4800 tons with a net positive buoyancy of 3200 tons when immersed to the attachment depth. The related segmented condenser subassembly weighs 6000 tons with a net positive buoyancy of 850 tons when similarly immersed. The excessive net positive buoyancy for each subassembly leads to an unusual set of stability conditions when the subassembly is floated out of dock and towed from shore to site. Variable ballast requirements can be reduced by installing fixed ballast in the machinery room and access trunks.

2.2.1.2 Immersed-Unitized Arrangement. A general arrangement drawing of this alternate configuration is shown in Fig. 2-2. With the unitized arrangement, the power module is constructed as a complete unit in a drydock with the major components, piping, and structural supports rigidly attached together. The unitized configuration differs from the segmented such that, in the unitized configuration:

- The control and machinery deck is a single structure which eliminates two bulkheads required for the segmented arrangement.
- A single vertical access trunk is provided with a cross tunnel connecting both machinery rooms. This also reduces the condensate piping run lengths between the heat exchangers.
- The all-welded unitized structure reduces the structural support scantling requirements.
- The turbine-generator set is located within the machinery space. This lower elevation reduces the length of the vapor lines and exhaust diffuser by approximately 25 ft.

The unitized module arrangement weighs approximately 10,600 tons with a net positive buoyancy of 3,900 tons when immersed to the operational orientation. The decrease in weight and buoyancy from the immersed-segmented arrangement is primarily due to the following:

- LEGEND**
- 1 GENERATOR
 - 2 TURBINE
 - 3 DIFFUSER
 - 4 AMMONIA VAPOR PIPE
 - 5 MIST EXTRACTOR
 - 6 MACHINERY ROOM, MIST TRUNK
 - 7 AMMONIA CLEANUP SYSTEM
 - 8 SEAWATER PUMP
 - 9 NON-CONDENSIBLE GAS REMOVAL SYSTEM
 - 10 GENERATOR COUPLING
 - 11 NITROGEN PURGE AND STORAGE SYSTEM
 - 12 CHLORINATION SYSTEM
 - 13 AMMONIA ANALYSIS SYSTEM
 - 14 DATA ACQUISITION AND CONTROL CENTER
 - 15 ELECTRICAL SWITCHGEAR
 - 16 TRANSFORMER
 - 17 LEAK DETECTION (H₂O IN SEAWATER)
 - 18 CONDENSATE PUMP
 - 19 DISTRIBUTION PUMP
 - 20 REMOTE TERMINAL UNIT (RTU)
 - 21 CONDENSER INLET TRANSITION PIECE

Fig. 2-2 General Arrangement 50 MW(e)
Power Module, Immersed-Unitized

- A weight decrease of 170 tons from the single vertical access trunk to the heat exchanger machinery rooms, with a buoyancy decrease of 210 tons
- A weight decrease of 80 tons from the shorter lengths of piping, diffuser, and structural members associated with the lowered turbine-generator set. The corresponding buoyancy decrease is 90 tons.

2.2.1.3 Submerged Arrangement. A general arrangement drawing of this alternate configuration is shown in Fig. 2-3. With the submerged arrangement, the power module is constructed as a complete unit in a drydock with the major components, piping and structural supports rigidly attached together. The evaporator is mounted directly above the condenser with a cylindrical machinery space in between. This space provides a shirtsleeve environment for the turbine-generator and all the control and auxiliary subsystems with the exception of the condensate pumps and related plumbing which is located in the pump room at the bottom of the condenser. Access to the pump room is via a 12 ft diameter multipurpose access trunk. Access to the control room and machinery space is via the module access and utility trunk.

A cluster of four seawater pumps is arranged within each of the seawater ducts. The ducting is attached to the downstream truncated end of the heat exchangers.

The condenser is tilted 15 degrees to provide for improved condensate drainage. This rotation requires an upstream inlet transition piece. The downstream end is provided with a structural ring for attachment to the cold seawater discharge pipe.

A vertical shaft oriented turbine-generator is a prime consideration for the submerged arrangement. Available space and orientation of the turbine-generator dictates the selection of the vertical shaft machine.

LEGEND

- 1. GENERATOR
- 2. TURBINE
- 3. DIFFUSER
- 4. AMMONIA VAPOR PIPE
- 5. MIST EXTRACTOR
- 6. PUMP ROOM ACCESS TRUNK
- 7. AMMONIA CLEANUP SYSTEM
- 8. NON-CONDENSIBLE GAS REMOVAL SYSTEM
- 9. GENERATOR COOLING
- 10. NITROGEN PURGE AND STORAGE SYSTEM
- 11. CHLORINATION SYSTEM
- 12. AMMONIA ANALYSIS SYSTEM
- 13. DATA ACQUISITION AND CONTROL CENTER
- 14. ELECTRICAL SWITCHGEAR
- 15. TRANSFORMER
- 16. LEAK DETECTION (NH₃ TO SEAWATER)
- 17. CONDENSATE PUMPS
- 18. DISTRIBUTION PUMPS
- 19. REMOTE TERMINAL UNIT (RTU)
- 20. PLATFORM/MODULE ACCESS AND UTILITY TRUNK
- 21. BALLAST
- 22. SEAWATER PUMP

Fig. 2-3 General Arrangement 50 MW(e)
Power Module, Submerged

The submerged module arrangement weighs approximately 10,700 long tons with a net positive buoyancy of approximately 4,000 tons when submerged to the operational depth. The unique configuration of the submerged module necessitates towing the module to the site in a horizontal attitude and upending the unit at the site to facilitate mating. Stability considerations will establish the disposition of ballast to obtain the various attitudes during the transit, mating, and demating operations.

2.2.2 10-MW(e) Power Module

The 10-MW(e) Modular Application Power System (MAPS) represents a major step towards commercialization by demonstrating the performance of the actual hardware of an OTEC system. Because the MAPS configuration is analogous in design and construction to the 50-MW(e) module, it can be used as a basis for projecting the cost effectiveness of commercial power systems. Because of the similarities between the 10- and 50-MW(e) modules, this section will identify and discuss the differences between the two modules. The 10-MW(e) module differs from the 50- in terms of design and manufacturing as follows:

- The MAPS is self-contained and requires a minimum of platform support. The module contains all ammonia and electrical auxiliaries and ancillary equipment for power system checkout and operation.
- The heat exchanger tube material is titanium by DOE directive.
- The tubesheets are single steel sheets with titanium cladding.
- The heat exchanger shells are steel. Galvanic protection is incorporated to ensure compatibility with the titanium cladding in seawater.
- The tooling for the rotating machinery is in accordance with standard shop practice.
- The facilities and equipment are readily available to fabricate, handle, deliver, and assemble all components.

Two self-contained power system module configurations are applicable to the 10-MW(e) MAPS. These configurations also are designated as "immersed-unitized." The former is adaptable to a greater variety of platform configurations and requires more platform installation activities, while the latter provides a more stable arrangement for deployment and minimizes platform installation activities. Lockheed has selected the "immersed-segmented" arrangement as the primary choice and the "immersed-unitized" arrangement as an alternative. The seawater pumps are included as part of the power module and the piping is assumed to be compatible with the platform piping. The ammonia vapor and liquid piping, downstream of the condenser, is assumed to be compatible with the heat exchanger structural support ring. These assumptions are a result of the DOE directive in that the platform is not defined.

A general arrangement drawing of the immersed-segmented module is shown in Fig. 2-4.

The design of the 10-MW(e) MAPS is a small-scale 50-MW(e) module. The descriptions of the "immersed-segmented", "immersed-unitized" given in paragraphs 2.2.1.1 and 2.2.1.2, respectively, are also applicable to the 10-MW(e) modules and are not repeated. Size, weight, and buoyancy are of course different from the 50-MW(e) module and thus are presented here for the 10-MW(e) module.

The evaporator and condenser have a 39.43-ft diameter truncated spherical steel shell mated to the seawater pump ducting at the upstream truncated end which in turn is mated to the platform. The condenser's truncated spherical shell is tilted 15 deg, requiring a seawater inlet transition ring and a discharge structural ring. Machinery rooms housing the ammonia distribution and condensate pumps and related plumbing, fulfill the requirement for those ancillary equipment in a shirt-sleeve environment. Access to these machinery rooms is via a 9-ft diameter multipurpose access trunk. In addition to

Fig. 2-4 General Arrangement 10 MW(e)
Power Module, Immersed-Segmented

providing for personnel access and equipment replacement, the trunk serves as a conduit for HVAC ducting, instrumentation and control cables, electrical cables, safety related subsystems, and drain lines.

The major ammonia piping consists of short lengths from the condenser to the condensate pumps; long lengths from the condensate pumps through the condenser shell to the evaporator shell and the evaporator sump; from the evaporator shell to the evaporator distribution orifices; and from the evaporator to the turbine. Liquid lines, where possible, are internal to the heat exchanger shell to protect the lines and reduce line structural requirements. All lines are fabricated from carbon steel. Welded construction is used except at certain points (above seawater level) where connections will be broken to remove either the evaporator or the condenser while the remainder of the module is left in place.

Ammonia flow to the condensate and distribution pumps is provided via single penetrations of the condenser and evaporator shells and supply suction headers. The condensate and distribution pump suction lines are connected to these headers. Each suction line contains a temporary startup strainer, a stop valve, and an expansion joint. The expansion joint is provided to minimize loads due to relative thermal expansion between the suction lines and the pump room walls.

Each of the condensate and distribution pump discharge lines contains a discharge stop-check valve and expansion joint. The run from the condensate pump discharge manifold passes from the pump room into the condenser, exits the condenser shell, and rises above the water line to flanges and valves which permit heat exchanger removal, descends below the water line and enters the evaporator shell, and is discharged to the sump through a large diameter sparger ring.

The pipe run from the distribution pump discharge passes from the pump room into the evaporator where it connects with the evaporator internal distribution

system. There are vents at each pump discharge port and drain provisions at the bottom of each pump barrel and pump discharge manifold. There are two vapor piping runs (from the evaporator to the turbine) each of which carries one-half of the ammonia vapor.

Each pump suction line contains a stop valve; each pump discharge line contains a stop-check valve; and each manifolded main discharge line contains a flow-modulating valve. Isolation valves in the evaporator vapor lines and in the main condensate line are installed to permit removal of a condenser or evaporator. Butterfly valves are used throughout since they provide the most compact configuration for large diameter valving. The selected valves have a special seating geometry which provides reliable and tight sealing.

The segmented module arrangement requires mechanical joints at four major ammonia piping runs, one turbine diffuser and two heat exchangers. With excessive buoyancy, the mating interface joint could be designed to facilitate self-alignment and incremental compressive loading of the joint interface as the heat exchanger assembly is mated by deballasting. The machinery room is circumferentially welded to the bottom of the heat exchanger. All appendages to the heat exchanger, including ammonia piping and access tunnel, are rigidly supported to the heat exchanger shell or machinery space. In transit, the exposed openings of each segment's heat exchanger piping and access are sealed with blind flanges to preclude entrance of seawater.

The segmented module arrangement weighs approximately 2,350 long tons with a net positive buoyancy of approximately 1,360 tons when immersed to the operational depth. This excess buoyancy is the result of the following buoyancy elements:

- Operation spaces situated above the heat exchangers
- Watertight volumes of the heat exchangers
- Two machinery rooms located below the heat exchangers
- Machinery room access trunk

The segmented evaporator, including operations space, machinery room and access trunk, weighs 980 tons with a net positive buoyancy of 925 tons when immersed to the attachment depth. The related segmented condenser subassembly weighs 1360 tons with a net positive buoyancy of 430 tons when similarly immersed.

The segmented condenser assembly requires 440 tons of ballast of which 80 tons is water ballast and 360 tons is fixed ballast in the machinery room and access tunnel. The segmented evaporator assembly requires 920 tons of ballast of which 90 tons is water ballast and 830 tons of fixed ballast in the machinery room, access tunnel, and in the structural support pockets above the heat exchanger.

All of the ballasting considerations above can be reduced to the degree that the platform can alter its own draft with the employment of the platform's trim and ballast system.

Assembly of the condenser, evaporator, and related appendages and equipment into module subassemblies will take place at the delivery point. Subassemblies for each module, shown in Fig. 2-5 for the immersed-segmented baseline configuration as well as all power system equipment to be installed on the platform, will be shipped to the delivery point in either a completely or partially assembled condition.

Final assembly will utilize the facilities and equipment available at the delivery point. These facilities and equipment should include means for heavy lift, installation, test, and checkout. As an example, the welding of the machinery room to the heat exchanger will require preinstallation of machinery foundations, piping and cabling, and preparation of surfaces for welding. Fit-up, welding, and nondestructive testing (NDT) of welds will be conducted in accordance with applicable code requirements. At this point, final installation of machinery would either take place dockside or after deployment and attachment to the platform depending on the orientation and loads encountered during launch and mating, and interface agreements.

Fig. 2-5 Heat Exchanger Assembly (Immersed-Segmented)

A general arrangement drawing of the unitized configuration is shown in Fig. 2-6. All module equipments are the same for both the unitized and the segmented arrangements. Location of the turbine-generator within the machinery space reduces the length of ammonia vapor piping by about 15 ft. The unitized module arrangement weighs approximately 2,140 tons with a net positive buoyancy of 1,050 tons when immersed to the operational orientation. The decrease in weight and buoyancy as compared to the segmented module is primarily due to the following:

- A decrease of some 40 tons by providing a single vertical access trunk to the heat exchanger machinery rooms; buoyancy decrease is about 160 tons
- A decrease of 25 tons for the shorter lengths of piping, diffuser, and structural members associated with the lowered turbine-generator set; the corresponding buoyancy decrease is 25 tons

The net positive buoyancy provides a convenient upward locking force when the module is mated to the surface platform. However, prior to being mated to the surface platform, the module must be immersed to the mating position with 1,050 tons of ballast. The 1,050 tons of ballast required can be compensated by installing 700 tons of fixed ballast in the machinery rooms and access trunks. This leaves a minimum of 350 tons of portable ballast for trimming and heeling moment compensation. Subsequent to positioning the module in the mating attitude, the ballast is partially removed to lock-in the module, and a horizontal restraining force is applied. The final assembly procedure for that module configuration will be similar to that described above for the immersed-segmented configuration.

2.2.3 Heat Exchanger Test Articles

The Heat Exchanger Test Articles (HETA) are scaled models to those designed for the 50-MW(e) Commercial Plant Size Power Modules (CPM). Refinement of heat exchanger configuration, design, and production techniques to ensure low

Fig. 2-6 General Arrangement 10 MW(e)
Power Module, Immersed-Unitized

cost and high performance requires experience with representative heat exchangers operating in the ocean environment. The test article heat exchangers provide needed cost and performance data and operating experience. The design of the test articles has been directed to verifying at minimum cost the configuration, thermal, hydraulic, mechanical and operational characteristics of horizontal shell and tube heat exchangers.

The heat exchanger test article configurations fulfill these objectives. The approach is compatible with the expected capabilities of the OTEC-1. The design is analogous to the physical and structural arrangements of the CPM and the MAPS heat exchangers. Hence, the cost data from this hardware design and procurement will be applicable to the cost models for the CPM. The features of the test article heat exchangers are described below. Figure 2-7 shows the test article evaporator.

Fig. 2-7 Configuration of Test Article Heat Exchanger
0.2 MW(e) Nominal Evaporator

The sizing of the heat exchangers resulted in nominal 0.2-MW(e) net output based on projected OTEC-1 head losses and the performance of the enhanced tube bundle. The combined capacity from the smooth and enhanced tube bundles will be 0.38-MW(e). The test article configuration incorporates many of the features of the commercial power system (1) including the use of aluminum in the spherical shell, the tubesheet and the tubes, (2) in the use of shell-side heat transfer enhancement on the horizontal tubes, (3) in the incorporation of the mist extractors and ammonia sumps in the spherical shell, and (4) in the inclination of the condenser to improve drainage of ammonia condensate.

The evaporator and condenser are contained in spherical aluminum shells with 17.4 ft OD. The 8.6-ft diam aluminum tubesheet is 2 in. thick. The tube bundle is divided into halves by a vertical divider plate.

In the evaporator, 2386 7/8 in. diameter aluminum smooth tubes are in one one-half bundle, and 2386 enhanced tubes with external fins on the shell side are in the other one-half bundle. Hook and vane mist extractors are located interior to the shell at the two evaporator vapor discharge ports. A system for generating and supplying supplementary ammonia vapor is provided with the evaporator. This vapor is injected from perforations in the central divider. Without this system, the flow conditions around the tube bundles could not be modeled. The distribution of liquid ammonia in the evaporator is provided by irrigating tubes which are located at the top of each column of seawater tubes. Valves located outside the shell allow local control of the distribution of the liquid ammonia in the bundle for optimum control of the ammonia film. Chlorination (OTEC-1 supplied) is used to control the growth of biofouling, and M.A.N. brushes are installed in each tube to clean it mechanically by reversing water flow (OTEC-1 accomplishment). The shells are mounted on legs for installation on OTEC-1. The length of the legs (or spacer pads) can be varied to allow operation at off-design angles or inclinations. The test articles are equipped with a complete set of instrumentation for measurement of thermal and hydraulic performance.

The condenser is similar to the evaporator in that the tube bundle is divided into two halves. The condenser has 2830 smooth and 2830 enhanced tubes of 7/8-in. diam. aluminum occupying the same relative position as in the evaporator allowing each half to operate independently.

The HETA configuration can provide for the simultaneous testing of several aluminum tube allows (e.g. Alclad 3003, Alclad 3004 and 5052). Selection of bundle segments and arrays provides this feature. To assist in the definition of aluminum heat exchanger corrosion and erosion four to six tubes can be extracted from each heat exchanger (evaporator and condenser) every 3 or 4 months for methodological examination.

Structural dynamic modeling information is obtained from the evaporator through the use of additional vapor provided by the external vapor supply to simulate the excitation dynamics of the tube configurations under full-size module vapor velocity conditions.

Existing commercial practices will be used to manufacture, assemble, handle, and deliver the heat exchangers and ancillary equipment to ensure that schedule and reliability requirements will be met. The accumulated actual cost data will be readily applicable to projections of the economics of the 50- and 10-MW(e) power systems.

2.3 SYSTEM CHARACTERISTICS

The following tables present the key system and equipment physical and performance characteristics of the power modules. These tables are in alphanumeric sequence using the alpha from the DOE designation and the numeric of "50" or "10" for the module net power output. Most tables summarize two configurations: the power module for the "reference baseline" is the configuration with a 10 ft head loss in the warm water system and a 15 ft head loss in the cold water system, the second column "optimized geometry" is defined as the head losses estimated for the optimized surface platform.

2.4 AVAILABILITY

Availability is that fraction of time that the power system is capable of providing its expected output. The availability of the power system is a function of the module equipment reliability (mean time between failures), repair time if a failure occurs (mean time to repair), and scheduled maintenance intervals.

The power systems incorporate redundant or multiple equipment in control subsystems to ensure continuous operation or minimum performance degradation in the event of component failure. Redundancy or multiple equipment areas are:

- Evaporator/Condenser tube contingency (7 percent excess)
- Ammonia Cycle — condensate pumps (one spare) and distribution pumps (one spare), ammonia turbine (4 radial turbines and 2 generators for the CPM)
- Control System — central processor and interfacing equipment (spare)
- Seawater System — four pumps (three provide 96% of module expected performance)

The calculated reliability for both the 50-MW(e) CPM and the 10-MW(e) MAPS is over 92 percent. The calculated availability for the self contained, titanium-tubed, 10-MW(e) MAPS heat exchanger and the platform-integrated, aluminum tubed 50-MW(e) CPM is 94.7 and 95.9 percent, respectively, based on 8760 hr/year and on refurbishment (major overhaul) of the MAPS being accomplished at sea and the CPM using a spare module for changeout at sea with refurbishment at dockside. The MAPS refurbishment requires 30 days every 4 years and the CPM changeout requires 8 days every 3 years, with a spare module.

2.5 COST ANALYSIS

The costs for the 50-MW(e) commercial size power module and the 10-MW(e) MAPS are summaries of capital and tooling costs. The costs for power of the immersed-unitized configuration are expected to be almost identical to those of the immersed-segmented, and consequently were not estimated separately. Previous work carried out by LMSC under this contract indicated that the capital costs of submerged power system configuration are 5 to 10 percent lower than those of immersed configurations. However, the platforms associated with these two configurations are markedly different and were excluded from the design effort reported upon here; consequently, costs for submerged power systems are not presented here, as comparison with the costs of other configurations would not be meaningful.

2.5.1 50-MW(e) Power Module (CPM)

Table 2-1 summarizes the capital costs of the various components of 50-MW(e) power systems (CPM). These CPMs are considered to be integrated into the 400-MW(e) plant where ammonia and electrical power auxiliaries and the control system are shared between power modules. The total cost of production power systems (including heat exchanger replacement after 15 years) declines from \$1915/kWe for the first unit to \$2150/kWe for the eighth. Prototype

Table 2-1
50-MW(e) POWER SYSTEM COST SUMMARY [\$/kW(e)]

	Prototype Unit	First Production Unit	Eighth Production Unit
Turbine	52	52	43
Generator	35	35	32
Warm Seawater Pumps	96	96	80
Cold Seawater Pumps	96	96	80
Ammonia Condensate Pumps	10	10	10
Ammonia Distribution Pumps	12	12	12
Ammonia System Piping	17	17	17
Ammonia System Valving	19	19	19
Control Systems	6	5	5
Chlorination System	23	23	23
Mechanical Cleaning System	13	13	13
Ammonia Support Systems	34	34	34
Electrical Equipment	112	112	112
Auxiliary Systems	13	13	13
Evaporator*	402	398	378
Condenser*	423	418	397
Mist Extractor	4	4	4
Structures	62	62	54
Component Installation	55	55	55
Subtotal	1,484	1,474	1,381
Contingency	297	147	138
Home Office and Engineering	223	147	138
Fees	148	147	138
TOTAL	2,152	1,915	1,795

*Present worth including replacements. Assumptions: 15-year life;
6% escalation rate; 19% cost of money

unit costs are estimated at approximately \$2150/kWe. Should heat exchanger life be only 10 years, the present cost of production power systems would increase by just under \$400/kWe.

The largest cost elements in the power system are the heat exchangers; their combined present costs, assuming replacement after 15 years, account for over half the total power system costs. Heat exchanger costs are discussed in more detail later in this section. Seawater pumps are the second costliest item, accounting for roughly an eighth of total cost. The cost estimates shown in Table 2-1 reflect the results of a parametric study carried out by the Worthington Pump Corporation. This study indicates that, contrary to expectations, seawater pumps are subject to diseconomies of scale; it also suggests that seawater pump costs are roughly 30 percent higher than originally expected. Other quotes, received from M.A.N. Corporation, are approximately 10 percent lower, and suggest that economies of scale are indeed present in seawater pump manufacture. Clearly, there still appears to be considerable uncertainty concerning the costs of pumps large enough for OTEC applications. The only other cost element accounting for over 5 percent of total cost is electrical equipment; roughly two thirds of this cost is associated with electrical cabling. Although in some ways they are more related to the platform than to the power system, the costs of the envisaged pump machinery rooms and access trunks, and of equipment installation, are included in Table 2-1 and, when combined, are slightly in excess of \$100/kW(e). The estimated costs given for controls and ammonia support (mainly startup and shutdown) systems are also platform-dependent, in that they are assumed to be shared among eight modules in a 400-MW(e) power plant.

The allowances for contingency, home office and engineering, and systems contractor and integrator fees and the exact distribution of these allowances for each component varies depending on the nature of the work involved, the practices of contractor(s), and the type of contract under consideration. For the purposes of estimating production unit costs, an allowance of 10 percent for each element was selected as giving an approximately correct

total cost for the power system as a whole. In practice, the importance of these three cost categories is indeed likely to be roughly equal; their exact distribution is of course of only secondary interest for estimating purposes. Contingency allowances were doubled for prototype units and home office and engineering allowances increased by 50 percent, to reflect the added uncertainty and difficulty associated with prototype construction. These might also lead to an increase in the fees charged by systems integrators and contractors and to lower proportional increases in other allowances. However, the combined allowances chosen are believed to yield a reasonable estimate of prototype costs, regardless of the actual distribution of cost among the three types of charges for which the allowances are made.

Heat Exchanger. Table 2-2 summarizes heat exchanger costs. The estimate reflects direct shop labor rates of \$19/hr for the prototype and first units and \$16/hr for the eighth set. This reduction in labor rates is due to the expected effect of spreading overhead over a larger labor pool. In addition, a 6 percent reduction in labor hours is expected to occur in going from the prototype to the first production unit, after which labor hours are expected to decline following a learning curve with slope 0.97. Material costs are estimated on a weight basis at a unit cost of \$1.50/lb, and tubing costs are based on a unit cost of \$1.40/ft for finned 1.5 in. diam tube. As was to be expected, the cost of aluminum tubing represents the largest component of total costs, accounting for roughly 55 percent of it. Roughly two thirds of this cost corresponds to the cost of the tubing proper, and the remainder to the cost of the enhancement process. The second largest cost element, and the largest labor item in the total, is the cost of tube bundle assembly, accounting for about half of labor costs and roughly an eighth of total costs. Shell material is the only other cost component accounting for more than 10 percent of total costs. Overall, material costs dominate, representing roughly three quarters of total costs. As a result, no drastic decreases in cost are likely to take place as a result of multiple-unit production, and the eighth production units are expected to be only some 10 percent less costly than the prototypes.

Table 2-2
50-MW(e) ALUMINUM HEAT EXCHANGER
COST BREAKDOWN [\$/kW(e)]

ITEM	PROTOTYPE			FIRST PRODUCTION			EIGHTH PRODUCTION		
	SUBCONTRACT & MATERIAL	LABOR	TOTAL	SUBCONTRACT & MATERIAL	LABOR	TOTAL	SUBCONTRACT & MATERIAL	LABOR	TOTAL
TUBES	292*	75	367	292*	71	363	290*	54	344
TUBESHEETS AND RINGS	8	9	17	8	8	16	7	6	13
TUBE SUPPORTS	20	40	60	20	37	57	19	29	48
SHELL	62	6	68	62	6	68	58	5	63
WATERBOXES	3	11	14	3	11	14	3	8	11
NOZZLES	2	-	2	2	-	2	2	-	2
AMMONIA DISTB. & NONCOND. REMOVAL SYSTEM	2	4	6	2	4	6	2	3	5
FINAL ASSEMBLY	1	7	8	1	6	7	1	5	6
TOTAL	390	152	542	390	143	533	382	110	492

*TUBES AT \$1.00/FT BASIC + \$0.40/FT ENHANCEMENT

Nonrecurring Costs. The possible desirability of building OTEC power system components in dedicated facilities has been investigated briefly. Turbines, seawater pumps and heat exchangers immediately suggest themselves as obvious candidate components for manufacture in dedicated facilities.

In the course of discussions with turbine manufacturers, it was found that currently existing industrial facilities are adequate for OTEC ammonia turbine manufacture and, consequently, that dedicated facilities are unnecessary. While the size of OTEC turbines is quite large in relation to their output, in absolute terms it is not beyond the current state of practice. The situation concerning seawater pump manufacture is somewhat similar. While the size of contemplated OTEC pumps is beyond the experience of some manufacturers, several others appear capable of producing OTEC pumps in existing facilities. Initial tooling pattern costs for pumps in the 1000 cfs range are estimated by the Worthington Pump Corporation at between \$100,000 and \$200,000.

Foster Wheeler has investigated the question of heat exchanger producibility in some depth, and concluded that heat exchangers sized for 50 MW(e) can be built in its existing facilities. The required tooling cost, however, is quite considerable, amounting to approximately \$4,000,000 for the prototype heat exchanger, \$2,000,000 for the first production units, and roughly \$800,000 apiece for succeeding heat exchanger sets.

It should be stressed that manufacture of OTEC power system componentry being economically competitive without need of dedicated facilities does not necessarily imply that such facilities are not necessary or desirable for the production of complete OTEC plants (including the platform system).

2.5.2 10-MW(e) Power Module (MAPS)

This section is analogous in spirit and structure to the foregoing section which presented the capital cost summary for the 50-MW(e) power system. Table 2-3, similar to Table 2-1 in that section, summarizes the system's capital costs. The cost per kilowatt of 10-MW(e) power systems is some 50 percent higher than that of the 50-MW(e) units, as a result of using Ti-tubed heat exchangers, the self contained module requirement, and of failing to capture some available economies of scale. As discussed in the preceding section, data received from Worthington Pump were used in estimating seawater pump costs for Table 2-3; in this instance, the pump costs indicated by M.A.N. Corporation are roughly 10 percent higher. Except for the seawater pumps, all power system components are costlier (per kW) for the 10-MW(e) modules than for the 50-MW(e) units. Total production unit costs are estimated to range from about \$3000/kW for the first to \$2800/kW for the eighth; the prototype unit cost is estimated at roughly \$3350/kW.

Heat exchanger costs, again dominant, are discussed in more detail in following paragraphs; of course, Ti-tubed heat exchangers are not expected to require replacement in the course of the 30-year power system life. It will be noticed that the cost of controls and ammonia support systems per kW(e) is much higher than in the previous case, because the 10-MW(e) power module is entirely self-contained, and is thus required to incorporate some dedicated equipment that was assumed shared among several modules for the 50-MW(e) power systems. The platform-dependent costs of access trunks and pump machinery rooms and of component installation have again been included in the total, and jointly amount to roughly \$120/kW(e).

A cost breakdown for the 10-MW(e) heat exchangers is shown in Table 2-4. The prototype costs are based on the Phase II proposal with the numbers for current 1978 costs before escalation; otherwise, the cost estimation procedure is entirely analogous to that outlined for the 50-MW(e) heat exchangers. The shop labor shows a reduction in hours between first and eighth units

Table 2-3
10-MW(e) POWER SYSTEM COST SUMMARY
[\$/kW(e)]

	Prototype Unit	First Production Unit	Production Unit
Turbine	71	71	58
Generator	47	47	44
Warm Seawater Pumps	85	85	70
Cold Seawater Pumps	85	85	70
Ammonia Condensate Pumps	15	15	15
Ammonia Distribution Pumps	17	17	17
Ammonia System Piping	23	23	23
Ammonia System Valving	26	26	26
Control Systems	30	30	30
Chlorination System	23	23	23
Mechanical Cleaning System	28	28	28
Ammonia Support Systems	265	265	265
Electrical Equipment	140	140	140
Auxiliary Systems	34	34	34
Evaporator	592	584	551
Condenser	689	680	645
Mist Extractor	7	7	7
Structures	71	71	60
Component Installation	65	65	65
Subtotal	2,313	2,296	2,171
Contingency	462	230	217
Home Office and Engineering	347	230	217
Fees	231	230	217
TOTAL	<u>3,353</u>	<u>2,986</u>	<u>2,822</u>

due to a learning curve with slope 0.97, and also a reduction in hourly rate from \$19 to \$16/hr due to spreading the overhead over a larger number of manhours. The material costs show little reduction, as the quantities for one unit are substantial and little quantity discount can be expected.

It should be mentioned that, for a production run of 8 heat exchanger sets, the incorporation of a tube welding and finning line at the heat exchanger manufacturer's facility, could give a potential \$900,000 additional cost saving for the eighth production set which is not included in the cost estimate of Table 2-4.

Of course, Titanium tubing is by far the dominating cost component, accounting for roughly three quarters of total cost; some 25 percent of tubing costs represents the cost of heat transfer enhancement. The only other cost element of some individual importance is tube bundle assembly, accounting for some 10 percent of total cost. As would be expected under these circumstances, the effect of series production on costs is quite weak, and the eighth production set of heat exchangers is only some 7 percent less costly than the prototype units.

Table 2-4

HEAT EXCHANGER COST BREAKDOWN, 10-MW(e) UNITS [\$/kW(e)]

	Production			First Production			Eighth Production		
	Subcontract and Material	Labor	Total	Subcontract and Material	Labor	Total	Subcontract and Material	Labor	Total
Tubes	907*	131	1038	907*	123	1030	900*	94	994
Tubesheets and Rings	58	20	78	58	18	76	54	14	68
Tube Support Plates and Supports	12	45	57	12	42	54	11	33	44
Shell	48	22	70	48	20	68	45	16	61
Water Boxes	3	14	17	3	13	10	3	10	13
Nozzles	3	-	3	3	-	3	3	-	3
Ammonia Distribution and Noncondensable Removal System	2	5	7	2	4	6	2	3	5
Final Assembly	1	10	11	1	10	11	1	7	8
TOTAL	1034	247	1281	1034	230	1264	1019	177	1196

*Tubes at \$2.25/ft basic plus \$0.80/ft enhancement

*Back-up
check*

Blank

2-40

Section 3

POWER SYSTEM DESIGN RATIONALE

The rationale used in the derivation of power module configurations, for PSD-I, Phase I, Task 2, are in the form of the DOE groundrules and guidelines, system trade studies, and system performance modeling. The DOE guidelines are presented in Section 2, as they related to the conceptual design activities for the Commercial size Power Module (CPM), and the preliminary design activities for the Modular Application Power System (MAPS) and Heat Exchanger Test Articles (HETA).

The trade studies, conducted for the CPM and MAPS, are described. A description of the power system analytical model and the related computer programs used to generate performance and functional characteristics are provided along with pertinent analytical results.

The design and functional requirements used to define the power system and Heat Exchangers Test Article (HETA) configurations are presented in Section 2. The basic requirements for the 40- to 50-MW(e) CPM did not change from Task I requirements except to use a 400-MW(e) plant rather than the 100-MW(e) plant as a design parameter.

3.1 SYSTEM TRADE STUDIES

Major system trade studies were conducted on

- o Materials and related life cycle costs for 50-MW(e) heat exchangers
- o Cost effectiveness of heat transfer enhancement techniques
- o Modularity of heat exchangers for a 50-MW(e) power module
- o Constant versus variable seawater flow
- o Effects of turbine efficiency
- o Heat exchanger tube length to diameter ratio

Some of the work was performed during the conceptual design phase, with subsequent studies in preliminary design confirming and refining previous conclusions. The results are presented here with summaries of the more recent work.

3.1.1 Heat Exchanger Materials

Life cycle costs were compared for heat exchangers with titanium, stainless steel, and aluminum tubes, each with compatible tube sheet and shell materials, and a range of tube sizes and shell geometries. (See Ocean Thermal Energy Conversion (OTEC) Power System Development Conceptual Design, Contract No. EG-77-C-03-1568, LMSC-D566744, 30 January 1978). Aluminum heat exchangers are preferable if their useful life exceeds eight years; the estimated life for aluminum heat exchangers is in excess of twelve years.

Recently available data on Alclad 3003 indicates that the life of this alloy is superior to that of 5052 alloy in terms of pitting resistance in an OTEC-type seawater environment. Discussions with Kaiser Aluminum and Gifford-Hill representatives (Oct 1978) indicated that Alclad 3004 and Alclad 3003 have good seawater pitting resistance. The cost (\$/ft) of Alclad 3003 or Alclad 3004 is about 30 percent less than 5052. Alclad 3004 has been tentatively selected for the CPM heat exchanger tube material. This alloy allows a high production rate of tube fabrication (roll and weld), and by selection of cladding thickness should give the 12 year life expectation with a predictable failure rate. The final choice of tube material will be made based on the heat exchanger test article test results. The configuration and costs presented in this report reflect the use of 5052 aluminum alloy for the tube material. Selected physical properties for the various aluminum tubing alloys are given in Table 3-1.

3.1.2 Heat Transfer Enhancement

Independent Research testing in LMSC's Palo Alto Research Laboratories, in progress since completion of the conceptual design, shows that ammonia side

Table 3-1
HEAT EXCHANGER TUBE ALUMINUM ALLOYS

	Composition				Density lb/in. ³	K BTU/ hr/ft-°F	Temper	Tensile 10 ³ psi	Shear 10 ³ psi	Fatigue 10 ³ psi	Modulus 10 ⁶ psi
	Mn	Mg	Cr	Zn							
Alclad 3003	1.2	-	-	-	0.099	98	0	6	11	7	10
							HB	27	16	10	
Alclad 3004	1.2	1.0	-	-	0.098	91	0	10	16	14	10
							H3B	36	21	18	
5052	-	2.5	0.25	-	0.097	77	0	13	13	16	10.2
							H3B	37	24	20	
Cladding	-	-	-	1.0	0.098	124	-	-	-	-	-

Cladding thickness estimated at 10 percent thickness of base material.

enhancement is effective. Currently, fifteen to twenty (15 to 20) circumferential fins/in. appear to be optimum. Further testing presently underway will provide design refinements. This design improvement has not been included in the present heat exchanger design. The heat exchanger test articles have been designed to test more than one type of tube simultaneously in its separate halves. Results from these OTEC-1 tests will supply useful data on this subject.

Cost effective techniques for seawater-side enhancement have not yet been identified. A study enclosed in Appendix A states that water side enhancement could reduce material cost about 40 percent and increase pumping power about 25 percent. Enhancement can save cost by increasing the heat transfer rate, thereby reducing required tube area and shell size for a given power requirement. It has been stated, however, that the added cost of the more elaborate tube seawater side enhancement and the higher seawater pumping losses more than cancel the savings (Ref: W. L. Ownes, "Comments on Seawater Side Enhancement for OTEC Heat Exchangers," Proceedings, 4th Annual Conference on OTEC, New Orleans, LA, March 1977).

3.1.3 Modular Heat Exchangers

Having one evaporator and one condenser results in the lowest cost for a power module of any size up to the 50-MW(e) limit of this investigation. The single heat exchanger in itself is less costly by avoiding the cost and complexity of added piping pressure vessels and valves. Compared to a 50-MW(e) system with a single heat exchanger, two 25-MW(e) modules are 10 to 15 percent more costly, while ten 5-MW(e) modules are 55 to 77 percent higher in cost.

It may be argued that modular heat exchangers improve operational availability by allowing repair without totally shutting down the plant. However, with the proper design and maintainability approach, this is not the case. An excess (7 percent) of heat exchanger tubes are provided (by design) so that tube leaks may be corrected by application of inserts or by plugging the tubes. This and other in-service maintenance can be accomplished during scheduled

maintenance periods with no impact on availability for the single heat exchanger configuration.

There are producibility limits which preclude the single heat exchanger configuration on the basis of equipment size. The heat exchanger assembly process was analyzed with regard to material size, handling of equipment, and fabrication procedures.

It was concluded that monolithic 50-MW(e) heat exchangers are technically feasible and have the lowest cost per kW(e) of plant output. Module assembly procedures for the 50-MW(e) heat exchangers are within the present capability of a number of shipyards.

A cursory review was made of several assembly methods using subassemblies. No module assembly concept was found to have a cost advantage over the assembly technique selected. In summary, the single monolithic heat exchanger configuration is producible and has the lowest cost (\$/kW).

3.1.4 Variability of Seawater Flow

If the OTEC plant is to operate at a seawater ΔT other than its baseline design value, the ability to vary seawater flow can improve net plant output as compared to constant flow systems. For any given ΔT , there is an optimum flow which balances heat transfer (evaporation rate) against pumping losses. At seawater ΔT s higher than the design baseline (40°F), higher evaporation rates and pressures (if the warm water temperature increased) provide higher turbine output. With variable seawater flow capability, an optimum in net power output may be obtained by balancing heat transfer against turbine gross power output and pumping power required. This results in higher seawater flow rates and higher net power output than the baseline. Conversely, if the seawater ΔT should decrease, the decreasing available turbine gross output can be conserved by decreasing parasitic losses. As a result, the plant with variable seawater flow produces more net output than the constant-flow plant at any off-design seawater ΔT .

These trends have been quantified using computerized performance calculations, with results displayed on curves of the ratio of net plant output over design output as a function of seawater ΔT , with both constant and optimized seawater flow (Fig. 3-1). The constant and optimum flow are quite close for ΔT variations of $\pm 2^\circ\text{F}$ around the design point, but diverge markedly for larger ΔT variations. It is concluded that variable seawater flow is a desirable feature of an OTEC plant that will operate at a site where ΔT variations exceed 4°F or where operation is expected for extended periods where the ΔT variation is greater than 2.5°F . Further, as shown in Fig. 3-2, the absolute temperature has very little effect over the water temperature range of 70 to 85°F . The ΔT dominates the power output.

3.1.5 Seawater Pump Drives

The two basic assumptions for variable seawater flow are a variable pump rpm or pump with variable pitch impellers. No investigation was made into throttling or bleed-off arrangements. The methods investigated were

- o Constant speed electric motor with fixed pitch impeller blades
- o Constant speed electric motor with variable pitch impeller blades (two alternatives first, blades bolted in position to hub second, link to remote control actuator)
- o Constant speed electric motor with variable speed coupling and fixed pitch impeller blades (fluid coupling or eddy current coupling)
- o Variable speed electric motor with fixed pitch impeller blades
 - 2 speed/multipole wound rotor with slip controller, synchronous motor with frequency control DC motor
- o Ammonia turbine drive with fixed pitch impeller blades

The seawater pump drive options are shown in Table 3-2 with associated advantages and disadvantages. The final selection for the 50-MW(e) power module has not yet been made because the distinctions in cost performance and reliability require further design detail and vendor participation to reach a resolution.

Fig. 3-1 Off-Design Performance of Power System With Optimized and Constant Flow Rate

Fig. 3-2 Off-Design Performance of Power System With the Effect of Absolute Temperature.

Table 3-2
SEAWATER PUMP DRIVE OPTIONS

Option	Advantages	Disadvantages	Comments
Constant-speed electric drive with fixed pitch impeller blades	Simplest motor and power distribution requirements; lowest cost	Does not provide variable flow	Cost trade-off based on site ΔT variation.
Constant-speed electric drive with variable-pitch pump (baseline)	Simplifies motor and power distribution requirements. High efficiency	More costly and mechanically complex pump assembly	Operating experience with this type of pump proved adequate reliability
Constant-speed electric drive with variable speed coupling	Simplifies motor and power distribution. No load motor startup Infinite adjustability	Low efficiency in-coupling. Heat dissipation controls/auxiliaries	Size/operating range available.
Variable-speed electric drive with fixed-pitch pump	Simplifies pump mechanism	Costly control and power conversion system. Lowest efficiency	Controllers have not been built at required power levels and voltages
Direct ammonia turbine drive of seawater pumps.	Eliminates power conversion losses. May be most efficient	Special turbine design. Complex vapor manifolding and control	Requires detailed efficiency tradeoff versus baseline constant-speed electric drive.

3-8

3.1.6 Materials for 10-MW(e) and Test Article Heat Exchangers

According to DOE directive, the 10-MW(e) heat exchanger tube material is to be titanium. This is in response to programmatic requirements which place highest confidence in reliability as the foremost design criterion, particularly for the first 10-MW(e) power system.

Aluminum is the recommended material for the test article. The choice of aluminum was directed by DOE to provide at-sea OTEC operating experience and verification of life-cycle cost-effectiveness for the 50-MW(e) design.

3.1.7 Effect of Turbine Efficiency

The power system cost sensitivity to the turbine cost for turbine costs in the range of \$40 to \$100/kW and efficiencies of 83 to 86 percent was established.

The approach taken to determine the sensitivity was to define heat exchanger configurations, turbine-generator sizes and seawater pump flow rates as functions of turbine efficiency with the system resized to provide 50 MW(e). The optimization program was used to resize the heat exchangers. Estimated cost data for these heat exchangers are plotted in Fig. 3-3 for labor costs of \$15 and \$20/hr. The turbine generator and seawater pump costs were added and the results are as shown by the solid curves of Fig. 3-4. Discussions with the turbine subcontractors indicated that the cost gradient indicated for efficiency improvement was considerably higher than they would have predicted. The sensitivity of the results to turbine cost gradient was evaluated by keeping the midpoint at the same value and halving the gradient, reducing it from \$20 to \$10 per percent efficiency. The results shown in the dashed curves of Fig. 3-4 indicate that the objective should be to attain the highest efficiency with a reasonable cost gradient.

3.1.8 Effect of Tube Length/Diameter Ratio

Studies conducted for a previous project indicated that the optimum tube length/diameter ratio (L/D) was between 350 and 400 for tube diameters of

PSD 244

Fig. 3-3 Effect of Turbine Efficiency on Heat Exchanger Costs

PSD 245

Fig. 3-4 Effect of Turbine Efficiency on Module Component Costs

2.0 and 3.0 in. A similar study was conducted for the 1.5-in. diameter aluminum tube proposed for the commercial size module. A family of configurations was developed by varying the number of tubes in the heat exchangers. The resultant heat exchangers were costed and as shown in Fig. 3-5, the minimum cost evaporator is at an L/D of approximately 350 while the condenser minimum is at an L/D of approximately 400.

The effect of tube L/D on system costs must include the effect on turbine-generator and seawater system sizes and costs. The lower the value of L/D, the greater the number of tubes required to provide the heat transfer area and consequently, a higher seawater flow rate required to maintain a reasonable overall heat transfer coefficient. The results of using a cost of \$50/kW for the turbine and \$600/CFS for the seawater pumps are shown in Fig. 3-6. From these curves it is seen that the optimum tube L/D value lies between 450 and 500.

3.2 SYSTEM MODELING AND RESULTS

LMSC has developed three computer programs for analyzing the OTEC power module, one for design optimization under prescribed design conditions, and two for calculating performance at off-design conditions. The computer programs for off-design performance calculations differ, in that one program determines performance with fixed seawater and ammonia flow rates while the other maximizes the net power by varying the flow rates. All three programs use the basic analytical model depicted in Fig. 3-7.

3.2.1 Optimization Program

The primary program is a "quasi-optimization" program which determines the minimum area heat exchanger for a given set of thermal resource conditions and physical parameters. The input parameters include the physical geometry of the external seawater system, such as the cold water pipe length and diameter, cleaning system losses plus additional pressure drops required to meet imposed pressure drop values, and line lengths for the ammonia system. The heat

Fig. 3-5 Effect of Tube L/D on Heat Exchanger Costs

Fig. 3-6 Effect of Tube L/D on Module Component Costs

Fig. 3-7 Power Cycle Analytical Model

exchanger parameters specified include type of material, number of tubes, tube diameter and wall thickness, heat transfer enhancement factor, and a design value for the biofouling resistance.

The nomenclature associated with the mathematical model is given in Table 3-3. The computational procedure is multi-iterative in that it starts with initial values for the thermodynamic parameters and converges to the final values. Based on the results of developmental computations, the thermal resource T split was approximately 25 percent across each heat exchanger and 50 percent across the turbine. The computations therefore begin by assigning values to T_2 and T_3 at 25 percent and 75 percent of the interval between T_{ei} and T_{ci} . The objective of the first operation is to determine the ammonia cycle characteristics to arrive at the turbine work equivalent to an initial value of gross power.

Table 3-3

COMPUTATION NOMENCLATURE

Variables		Subscripts	
A	Heat exchanger area, ft^2	a	Ammonia
C	Specific heat, $\text{Btu/lb } ^\circ\text{F}$	c	Condenser
H	Enthalpy, Btu/lb	cwp	Cold Water Pipe
LMTD	Log mean temperature difference $^\circ\text{F}$	e	Evaporator
		g	Generator
P	Pressure, lb/ft^2	i	Seawater inlet
Q	Total heat flux, Btu/sec	is	Isentropic expansion
T	Temperature, $^\circ\text{F}$	l	Liquid ammonia
U	Overall heat transfer coefficient, $\text{Btu/sec-ft}^2 \text{ } ^\circ\text{F}$	o	Seawater outlet
		p	Pump
W	Fluid mass flow rate, lb/sec	t	Turbine
X	Vapor quality, percent	w	Seawater
		Δ	Difference
		n	Efficiency
		2	Turbine inlet
		3	Turbine outlet

Based on the value of T_2 , an initial value for W_a and an assigned turbine inlet quality of 90 percent, the turbine efficiency, and ΔH_t are determined. The ammonia is assumed to be 100 percent liquid leaving the condenser, and the condensate pump adds enthalpy based on the required pumping head and pump efficiency. The evaporator, in turn, is assumed to provide sufficient heat flux for the required pumping head and pump efficiency. The evaporator, in turn, is assumed to provide sufficient heat flux for the required ammonia flow.

When the ammonia cycle has been established to provide the gross power, subroutines are called which size the heat exchangers. The starting point for these subroutines is an input value for seawater velocity in the tubes. This determines the seawater-side heat transfer coefficient, the seawater flow

rates, and related pressure drops. The gross generator power is then reduced by the various pump power requirements and auxiliary power requirements to arrive at the power system net power output. This value is compared with the desired net power and, if the error is greater than allowable, the initial conditions are adjusted as required. When a solution has evolved, perturbations are made in individual parameters to determine whether the objective of a minimum tube area has been achieved. The final results are then presented in the format shown in Appendix A.

Real life optimization would require development of a family of "optimized" power systems. The desired optimization function, such as power module cost or total plant cost, could then be applied to evaluate the resulting configurations.

3.2.2 Off-Design Performance Programs

The fixed flow and optimized flow off-design programs use the same initial assumption with regard to the split of the available seawater temperature difference. The ammonia cycle is established for these initial conditions and the required heat fluxes are determined. The heat exchangers are then balanced with the available seawater-side heat fluxes. If the differential between available seawater-side and required ammonia-side heat flux is greater than the allowable error, new values are assigned to T_2 and T_3 , and the process is repeated until the solution converges. The resultant gross and net powers are then calculated. The fixed flow program stops at this point and prints out the desired data.

The flow optimization program continues by storing this net power as a reference value, then adjusting seawater and ammonia flow to arrive at a new net power output. A simple trial and error search strategy is used to continue this process until no further net power increase can be realized. At that point the program stops and prints out the corresponding data as the optimum operating point.

3.2.3 Analytical Results

The configurations resulting from the optimization program are described in appropriate sections. The sensitivity of each design to the imposed design requirements in lieu of an optimized geometric configuration is also discussed. The analytical results presented in this section are related to the off-design performance of the CPM and MAPS. Performance of the HETA is closely allied to the OTEC-1 installation. Off-design performance will be determined on the basis of available seawater thermal resources and will be in the form of thermal flux rather than net power output.

The relative off-design performance characteristics, defined as net power available divided by the design value, for the 50-MW(e) design configuration was found to be similar to the off-design performance characteristics of the 25-MW(e) design configuration determined as part of the Task 1 effort. The differences are within the accuracies of the program error functions. The off-design performance of the 10-MW(e) Modular Application Power System has similar characteristics. The relative performance of the power system, designed for the $40^{\circ}\text{F } \Delta T$, is shown in Fig. 3-1. This curve can be used for determining the off-design power available for both the CPM and MAPS.

The sensitivity of performance to absolute seawater temperatures was also investigated for the 50-MW(e) module. The data presented in Fig. 3-2 are based on a constant 40°F cold seawater temperature with the warm seawater temperature varied as required to obtain the indicated ΔT . Additional computations were made with the seawater temperatures varied as shown in Table 3-4. The results, shown in Fig. 3-2, indicate that the temperature differential is the prime factor and that absolute temperatures have a negligible effect on performance.

Table 3-4
TEMPERATURE CONDITIONS (°F)

<u>Warm</u>	<u>Cold</u>	<u>ΔT</u>
85	41	44
85	50	35
80	50	30
80	36	44
70	32	38
70	41	29

*Back-up
Blank*

Blank

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Section 4

POWER SYSTEM MODULE

The physical and functional characteristics of both the 50-MW(e) Commercial Size Power Module (CPM) and the 10-MW(e) Modular Application Power System (MAPS) are presented in this section. It comprises the formal and analytic basis for the conclusions and recommendation of Section 2. The major subdivisions included in this section are

- System Description (4.1) - discussion and design of three power module configurations, immersed-segmented, immersed-unitized and submerged. (Appendix C)
- System Performance (4.2) - discussion of computer programs used for heat exchanger analysis, design parameters and their effect on cost and performance (Appendix A)
- Heat Exchanger (4.3) - discussion of cost trades, tube heat transfer enhancement, stress analysis, configuration and heat exchanger design parameters (Appendix E)
- Rotating Machinery (4.4) - discussion of seawater pumps characteristics and design options; turbine/generator options and analysis and design parameters (Appendix F)
- Auxiliary Systems (4.5) - discussion of ammonia systems, design parameters and operations procedures (Appendix G), power distribution system (Appendix H) and ballast control
- Control and Instrumentation (4.6) - discussion of the power module control requirements with selected instrumentation (Appendix I)
- Material Selection (4.7) - discussion of heat exchanger, tubes, turbine and piping material selection and ammonia purity requirements
- Operability and Maintainability (4.8) - discussion of biofouling, corrosion and their prevention techniques and overall maintenance methods
- Producibility (4.0) - discussion of fabrication and installation methods (Appendix J)

- Cost Analysis (4.10) - discussion of projected power module costs (Appendix K)
- Energy Resource Analysis (4.11) - discussion of a net energy balance to produce a 10-MW(e) power module

4.1 POWER MODULE DESCRIPTIONS

Based on the results of PSD-I Conceptual and Preliminary Design activities and the OTEC Plant Configuration and Integration Study, three configurations for the 50-MW(e) CPM and two self contained configurations for the ammonia power MAPS have evolved. As discussed in the introductory paragraphs of Section 2 these configurations are designated as "immersed-segmented"; "immersed-unitized"; and for the CPM "submerged-unitized". The "immersed-segmented" is adaptable to a greater variety of platform configurations and requires more platform installation activities while the "immersed-unitized" and "submerged-unitized" provide a more stable arrangement for deployment and minimize platform installation activities. The concept followed in this study is the "immersed-segmented" arrangement as the baseline design. The "immersed-unitized" and the "submerged" arrangements are presented as alternates.

The seawater pumps are included as part of the power module in the configuration discussed herein. The ducting is assumed to be compatible with the platform ducting. The efflux ducting, downstream of the condenser, is assumed to be compatible with the heat exchanger structural support ring. Details of these interfaces, as well as any other mechanical, electrical, or functional interfaces, are not defined at this time due to the absence of platform definition.

4.1.1 Immersed-Segmented Arrangement

The general arrangement drawings of the 50-MW(e) CPM (Fig. 4-1) and the 10-MW(e) MAPS (Fig. 4-2) show the power module to be operationally self-sufficient in that all control and auxiliary subsystems, with the exception

- LEGEND
1. GENERATOR (2)
 2. TURBINE (4)
 3. DIFFUSER
 4. AMMONIA VAPOR PIPE
 5. AMMONIA CONDENSATE PIPE
 6. MIST EXTRACTOR
 7. MACHINERY ROOM ACCESS TRUNK
 8. AMMONIA CLEANUP SYSTEM
 9. NON-CONDENSIBLE GAS REMOVAL SYSTEM
 10. GENERATOR COOLING
 11. NITROGEN PURGE AND STORAGE SYSTEM
 12. CHLORINATION SYSTEM
 13. AMMONIA ANALYSIS SYSTEM
 14. DATA ACQUISITION AND CONTROL CENTER
 15. ELECTRICAL SWITCHGEAR
 16. TRANSFORMER
 17. LEAK DETECTION (NH₃ IN/IN SEAWATER)
 18. CONDENSATE PUMP
 19. DISTRIBUTION PUMP
 20. REMOTE TERMINAL UNIT (RTU)
 21. CONDENSER INLET TRANSITION PIECE
 22. SEAWATER PUMP (CLUSTER OF 4 PUMPS)
 23. TRUNK ACCESS COMPARTMENT
 24. INTERFACE STRUCTURE

Fig. 4-1 General Arrangement, 50-MW(e) Power Module, Immersed-Segmented

4-4

Fig. 4-2 General Arrangement, 10-MW(e) Fower Module, Immersed-Segmented

of startup, standby, and emergency power, are provided. The CPM module external dimensions are, approximately, 190 ft wide, 170 ft high (with 140 ft extending below the water line) and 150 ft thick including the pump ducts. The corresponding MAPS module dimensions are 110 ft, 110 ft (80 ft) and 75 ft.

These modules consist of two major assemblies, each incorporating one of the heat exchangers and components for various subsystems. For the radial inflow turbine configuration, the turbine and generator are included with the condenser assembly. For the axial flow turbine configuration, turbine and generator are part of the evaporator assembly. Consideration was given to provide separate transit and installation for the turbine and generator. However, incorporation of the turbine and generator into one of the assemblies reduces the amount of ballast required to overcome the buoyancy.

The evaporators have truncated spherical shells, aluminum for CPM and steel for MAPS. (Table 4-1), with the upstream truncated end attached to the seawater pump ducting which in turn is mated to the platform. A machinery room is attached to the bottom of the evaporator and houses the ammonia distribution pumps and related plumbing in a shirt-sleeve environment as per the design requirements (Section 2). Access to the machinery room, for routine maintenance or equipment replacement, is via a multipurpose access trunk. In addition to providing for personnel access and equipment replacement, the trunk serves as a conduit for HVAC ducting, instrumentation and control cables, electrical cables, safety subsystems and drain lines.

Two ammonia turbine supply vapor pipes come from the evaporator with vertical lengths leading to closure valves and flanges located on the top deck. The condensate return pipe with closure valve and flange, is also attached to the evaporator.

Platform related structure provides the connection between the evaporator and its section of the operations and machinery deck. This structure is designed to accommodate all loads associated with deployment and platform installation. The evaporator portion of the machinery deck incorporates the ammonia

cleanup subsystem, the nitrogen purge and storage subsystem, the ammonia analysis and leak detection subsystems, and the chlorination subsystem. The machinery deck structure and the installation of all of its equipment is part of the platform.

The condensers also have truncated spherical shells, aluminum for CPM modules and steel for MAPS modules (Table 4-1) and are tilted 15 deg to provide adequate condensate drainage. This rotation requires an upstream inlet transition piece as part of the seawater piping. The downstream end is provided with a structural ring for attachment to the cold seawater discharge pipe.

Table 4-1
EVAPORATOR AND CONDENSER MODULE DIMENSIONS

<u>Component</u>	<u>50-MW(e)</u>	<u>10-MW(e)</u>
Evaporator Diam	78.74 ft	30.43 ft
Condenser Diam	78.74 ft	39.43 ft
Access Trunk Diam	12 ft	9 ft

As with the evaporator, a machinery room containing the ammonia condensate pumps and related plumbing is attached to the base of the condenser. The condensate return line passes through the condenser, rises to the top deck, and is provided with a closure valve and flange. The turbine-generator set is mounted on the top deck and a low angle diffuser connects the turbine discharge with the condenser vapor inlet.

Platform-related structure provides the connection between the condenser and its section of the operations and machinery deck. This deck, the operating space, contains the data acquisition and control subsystem, the electrical equipment, and the generator cooling subsystem. Again, the machinery deck structure and the installation of its equipment is part of the platform. Estimated weights and buoyancies of the various components which comprise the

immersed-segmented module are shown in Table 4-2 for 50-MW(e) and in Table 4-3 for 10-MW(e). The operations and machinery deck and module/platform interface structure may vary as the result of platform design.

All appendages to each heat exchanger, including ammonia piping and access tunnel, are rigidly supported by the heat exchanger shell or machinery space. In transit, the exposed openings of each segmented heat exchanger, piping and access are sealed with blind flanges to preclude entrance of seawater.

The segmented module arrangement approximate weights and net positive buoyancies when immersed to the operational depth are also shown in Tables 4-2 and 4-3 and this net buoyancy is the result of the following:

- Operation spaces situated above the heat exchangers
- Watertight volumes of the heat exchangers
- Immersion of the heat exchangers to 20 ft below the waterline
- Machinery rooms located below the heat exchangers
- Machinery room access hatch and trunk

The large net positive buoyancy for each subassembly necessitates a unique set of stability calculations to establish the amount and distribution of ballast when the subassembly is floated out of dock and towed from shore to site. Another set of stability calculations is required to submerge and mate the module to the platform. Variable ballast requirements can be reduced by installing fixed ballast in the machinery room and access trunks. After the module is mated to the platform, part of the variable ballast is removed.

All of the ballasting considerations may be reduced to the degree that the surface platform can alter its own draft with the employment of the platform's trim and ballast system.

Table 4-2

ESTIMATED WEIGHT AND BUOYANCY
50-MW(e) SEGMENTED POWER MODULE
(With Operating Space)

<u>Condenser Module</u>		
<u>Item</u>	<u>Weight (lb)</u>	<u>Buoyancy (lb)</u>
Condenser	5,604,000	8,067,400
Pump Room	580,600	1,973,000
Pump Room Access Trunk	473,900	1,366,100
Condensate Pumps (3)	96,600	—
Condenser Inlet Transition	47,800	6,300
Condenser CW Discharge Pipe	48,000	—
Condenser Plumbing	21,500	—
Turbine Exhaust Diffuser	1,355,000	402,100
Turbine and Generator Foundation	10,000	—
Structure Support	65,500	8,600
Operation Space (Platform)	1,354,300	3,030,900
Operation Equipment	2,818,600	—
Seawater Pumps	1,400,000	365,700
Seawater Pump Ducting	<u>757,100</u>	<u>99,100</u>
Total	13,413,400	15,319,200
<u>Evaporator Module</u>		
Evaporator	5,082,000	8,894,100
Pump Room	660,700	3,526,700
Pump Room Access Trunk	473,900	1,366,100
Distribution Pumps (4)	108,000	—
Ammonia Vapor Plumbing	436,100	325,700
Structure Support	150,700	19,700
Operation Space (Platform)	1,489,700	3,333,900
Operation Equipment	189,800	—
Seawater Pumps	1,400,000	365,700
Seawater Pump Ducting	<u>757,100</u>	<u>99,100</u>
Total	10,748,000	17,931,000
Grand Total	<u>24,161,400</u>	<u>33,250,200</u>
		9,088,880

Table 4-3

ESTIMATED WEIGHT AND BUOYANCY
10-MW(e) SEGMENTED POWER MODULE
(With Operating Space)

<u>Condenser Module</u>		
<u>Item</u>	<u>Weight (lb)</u>	<u>Buoyancy (lb)</u>
Condenser	1,004,000	1,500,000
Pump Room	100,000	613,000
Pump Room Access Trunk	93,000	335,000
Condensate Pumps (3)	42,000	-
Condenser Inlet Transition	21,000	3,000
Condenser CW Discharge Pipe	24,000	-
Condenser Plumbing	12,000	-
Turbine Exhaust Diffuser	33,000	40,000
Turbine and Generator Foundation	5,000	-
Structure Support	32,000	4,000
Operation Space (Platform)	365,000	1,521,000
Operation Equipment	<u>1,319,000</u>	<u>-</u>
Total	3,050,000	4,016,000
<u>Evaporator Module</u>		
Evaporator	940,000	1,610,000
Pump Room	100,000	613,000
Pump Room Access Trunk	93,000	335,000
Distribution Pumps (4)	48,000	-
Ammonia Vapor Plumbing	96,000	54,000
Structure Support	32,000	4,000
Operation Space (Platform)	383,000	1,659,000
Operation Equipment	<u>510,000</u>	<u>-</u>
Total	2,202,000	4,275,000
Grand Total	<u>5,252,000</u>	<u>8,291,000</u>
		3,039,000

4.1.2 Immersed-Unitized Arrangement

A general arrangement drawing of this configuration is shown in Fig. 4-3 and Fig. 4-4 for the 50-MW(e) and 10-MW(e) modules, respectively. With the unitized arrangement, the power module is constructed as a complete unit in a drydock with the major components, piping, and structural supports rigidly attached together. The major differences between the unitized and the segmented configurations are as follows:

- The control and machinery deck for the unitized arrangement is a single entity which eliminates the two bulkheads required for the segmented arrangement
- A single vertical access trunk is provided with a cross tunnel connecting both machinery rooms, thus reducing the condensate piping run lengths between the heat exchangers
- The all-welded unitized structure reduces the structural support scantling requirements
- The turbine-generator set is relocated within the machinery space; this lower elevation reduces the length of each vapor line and the exhaust diffuser by approximately 25 ft for the 50-MW(e) module and 15 ft for the 10-MW(e) module
- For the CPM the overall width is reduced 10 ft and the height by 20 ft
- For the MAPS the overall width is reduced 15 ft and the height 10 ft

The unitized module arrangement approximate weights and buoyancy decrease, as compared to the segmented module, are tabulated in Table 4-4 for the CPMs and the MAPs. This decrease in weight and buoyancy is primarily due to the use of a single vertical access trunk to the heat exchanger machinery rooms, and the shorter lengths of piping, diffuser, and structural members associated with the lowered turbine-generator set.

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Fig. 4-3 General Arrangement, 50-MW(e) Power Module, Immersed-Unitized

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Fig. 4-4 General Arrangement, 10-MW(e) Power Module, Immersed-Unitized

Table 4-4

IMMERSED-UNITIZED MODULE WEIGHTS AND
DIFFERENTIAL WEIGHTS AS COMPARED TO
THE IMMERSED-SEGMENTED MODULE

<u>Item or Component</u>	<u>Module</u>	
	<u>50-MW(e)</u>	<u>10-MW(e)</u>
Single vertical access trunk	-380,800	-94,080
Shorter Piping, diffuser, structural members	-170,240	-53,760
Total Module Weight	23,833,600 10,640 Long Tons	5,104,160 2,280 Long Tons
Total Net Buoyancy	8,749,080	2,765,720

4.1.3 Submerged-Unitized Arrangement

An alternate configuration for the immersed (segmented and unitized) module is a submerged-unitized arrangement as shown in Fig. 4-5. This configuration was considered only for the 50-MW(e) module. For the MAPS, this approach did not satisfy the operational requirements provided by DOE.

The submerged-unitized power module overall dimensions are approximately 96 ft wide, 275 ft high and 135 ft thick. The submerged power module arrangement, is constructed as a complete unit in a drydock with the major components, piping, and structural supports rigidly fastened together. The evaporator is mounted directly above the condenser with a cylindrical machinery space in between. This space provides a shirt-sleeve environment both for the turbine-generator and for all the control and auxiliary subsystems. Condensate pumps and related plumbing are located in the pump room attached to the bottom of the condenser. Access to the pump room for routine maintenance or equipment replacement used via a multipurpose excess trunk. Access to the control room and machinery space is via the module access and utility trunk. A cluster of four seawater pumps is arranged within each of the seawater pipes. The pipes are attached to the downstream truncated end of the heat

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LEGEND

1. GENERATOR
2. TURBINE
3. DIFFUSER
4. AMMONIA VAPOR PIPE
5. MIST EXTRACTOR
6. PUMP ROOM ACCESS TRUNK
7. AMMONIA CLEANUP SYSTEM
8. NON-CONDENSIBLE GAS REMOVAL SYSTEM
9. GENERATOR COOLING
10. NITROGEN PURGE AND STORAGE SYSTEM
11. CHLORINATION SYSTEM
12. AMMONIA ANALYSIS SYSTEM
13. DATA ACQUISITION AND CONTROL CENTER
14. ELECTRICAL SWITCHGEAR
15. TRANSFORMER
16. LEAK DETECTION (NH₃ TO SEAWATER)
17. CONDENSATE PUMPS
18. DISTRIBUTION PUMPS
19. REMOTE TERMINAL UNIT (RTU)
20. PLATFORM/MODULE ACCESS AND UTILITY TRUNK
21. BALLAST
22. SEAWATER PUMP

Fig. 4-5 General Arrangement, 50-MW(e) Power Module, Submerged

exchangers. An option to this arrangement is to place the seawater pumps upstream of the heat exchangers. Also the seawater pumps can be rotated 45 deg about the cluster axis such that the two top pumps are at the same depth.

A vertical shaft turbine-generator was a prime consideration for the submerged arrangement. Such a machine in the required power range is, however, unavailable at this time.

The submerged module arrangement weights approximately 10,730 long tons with a net positive buoyancy of approximately 4,010 tons when submerged to operational depth. The configuration of the submerged module necessitates towing the module to the site in a horizontal attitude and upending the unit at the site to facilitate mating. Extensive stability consideration will be required to establish effective disposition of ballast to obtain the various attitudes during transporting, mating, and demating operations. Weights and buoyancies of the various components which comprise the submerged module have been estimated and are shown in Table 4-5.

The net positive buoyance for all arrangements provides a convenient upward locking force when the module is mated to the surface platform. However, prior to being mated to the surface platform, the module must be immersed to the mating position by ballasting. Large displacement volumes of the operations space, heat exchangers, machinery rooms, and the machinery room accesses located well below the water surface will necessitate detailed weight, buoyancy, and moment computations to establish effective disposition of ballast in order to attain adequate transverse and longitudinal stability. The required ballasting can be compensated for by installing fixed ballast in the machinery rooms and access trunks. This facilitates a minimum of portable and variable ballast for trimming and heeling moment compensation. Subsequent to positioning the module in the mating attitude, the ballast is partially removed to lock in the module, and a horizontal restraining force is applied.

Table 4-5

ESTIMATED WEIGHT AND BUOYANCE
50-MW(e) SUBMERGED POWER MODULE

<u>Item</u>	<u>Weight (lb)</u>	<u>Buoyancy (lb)</u>
Condenser	5,604,000	8,067,400
Evaporator	5,082,000	8,894,100
Condenser Pump Room	669,500	2,796,600
Condenser S.W. Pump Housing	757,100	99,100
Evaporator S.W. Pump Housing	757,100	99,100
Operation Space	3,144,400	11,184,100
Condenser Pump Room Access	217,700	703,700
Condensate Return Pipe	19,300	-
NH ₃ Vapor Pipe	369,200	579,100
Diffuser	10,200	-
Condensate Pumps	96,600	-
Distribution Pumps	108,000	-
Seawater Pumps	2,800,000	365,700
Seawater Pump Housing Support	1,395,000	232,200
Control Room Equipment	2,998,400	-
Grand Total	<u>24,028,500</u>	<u>33,021,200</u>
		8,992,700

4.1.4 Module Equipment

The major components/equipment for the 50-MW(e) power system module are summarized in Table 4-6. Table 4-7 presents the components for the 10-MW(e) MAPS. Several of these components are dependent on the selection of a final platform configuration and the degree of power module integration with the platform. Further details of various components are contained in Appendix D.

Table 4-6
50-MW(e) POWER SYSTEM MAJOR EQUIPMENT

Component	Description	No.	Dimensions	Weight (lb)	Comments
Evaporator	Aluminum Spherical Shell W/1.50 in. O.D. Aluminum Tubing	1	78.74 ft O.D. eff. tube length 52.83 ft	5,082,000	89,000 tubes, 0.1025-in. tube-wall thickness, 1700 tons positive buoyancy
Condenser	Aluminum Spherical Shell W/1.50 in. O.D. Aluminum Tubing	1	78.74 ft O.D. eff. tube length 52.83 ft	5,604,000	106,000 tubes, 0.1025-in. tube-wall thickness, 1100 tons positive buoyancy
Turbine	Rotoflow Radial Inflow Turbine	4	8 ft x 11 ft x 10 ft	200,000	Steel
	or Airco Axial Flow	1		270,000	Steel
Generator	Rotoflow, 4160 Volts, 60 Hz, 1800/3600 RPM	2	31 ft x 12 ft x 13 ft	392,000	2-35-MW(e) generator, self-contained, cooling and lubricating system
	or Airco, 4160 Volts 60 Hz, 1800/3600 RPM	1	30 ft x 20 ft x 20 ft	312,000	70-MW(e) generator, self-contained, cooling and lubricating system
Seawater	Cluster of 4 pumps within each seawater duct to the heat exchanger	8	TBD	350,000	

Table 4-6 (Cont.)

Component	Description	No.	Dimensions	Weight (lb)	Comments
Condensate Pumps	Two-state Centrifugal	3	5.5 ft x 5.5 ft x 27 ft with motor	32,200 each	
Distribution Pumps	Single-State Centrifugal	4	5.5 ft x 5.5 ft x 21 ft with motor	27,000 each	
Ammonia Cleanup System	Vertical Pressure Vessel with Pump	1	Column 84-in. diam. 27 ft height	13,800	
Leak Detection	Pump With Analyzer	1	TBD	3,200 (est)	Seawater into NH ₃
Ammonia Storage System	Storage Tank with Pump and Compressor				Platform Mounted
Overpressure Vent	Safety Relief Valves	2	TBD		
Noncondensable Gas Removal System	Secondary Condenser and Compressor with Vent	1	TBD	14,900	
Standby Power System	Diesel-Generators 1-MW(e) Power Total	2	TBD		Platform Mounted
Electrical Power Distribution System	Startup, Normal, Standby, and Emergency	-	---	2,339,000	Cables, transformers, switchgear, and other equipment

Table 4-6. (Cont.)

Component	Description	No.	Dimensions	Weight (lb)	Comments
Equipment Cooling	Seawater-Fresh Water Cooling System	1	TBD	24,000	
Nitrogen Purge	Liquid Nitrogen Supply Tank or Cylinders, Piping, Pressure in Reducing Valve	1	TBD	10,800	
Fouling Counter-measures	Mechanical Tube Cleaning and Chlorination System	2	TBD	3,000	
			TBD	6,500	
Data Acquisition and Control Systems (DACS)	Remote Terminal Units, Data Processing, Computer Programmer, and Control and Display Consoles	3	TBD	2,000	
		1	TBD	8,000	

Table 4-7

10-MW(e) POWER SYSTEM MAJOR EQUIPMENT

Component	Description	No.	Dimensions	Weight (lb)	Comments
Evaporator	Steel Spherical Shell W/1.00 in. O.D. Titanium Tubing	1	39.43 ft O.D. eff. tube length 26.20 ft	940,000	51,400 tubes, 0.025-in. tube-wall thickness. 895 tons positive buoy- ancy
Condenser	Steel Spherical Shell W/1.00 in. O.D. Titanium Tubing	1	39.43 ft O.D. eff. tube length 26.20 ft	1,004,000	64,000 tubes, 0.025-in. tube-wall thickness. 751 tons positive buoy- ancy
Turbine	Rotoflow Radial Inflow Turbine	1	9 ft x 7 ft x 7 ft	40,000	Steel
	or Airco Axial Flow	1	28 ft x 9 ft x 9 ft	30,000	Steel
Generator	Rotoflow, 4160 Volts, 60 Hz, 1800/3600 RPM	1	15 ft x 10 ft x 7 ft	118,000	14-MW(e) generator, self-contained, cooling and lubricating system
	or Airco, 4160 Volts 60 Hz, 1800/3600 RPM	1	11 ft x 8 ft x 8 ft	118,000	14-MW(e) generator, self-contained, cooling and lubricating system
Condensate Pumps	Two-Stage Centrifugal	3	3 ft x 3 ft x 11 ft with motor	13,800 each	1,368,072 lb/hr at 64.2 psi, 182 hp motor
Distribution Pumps	Single-Stage Centrifugal	4	3 ft x 3 ft x 10 ft with motor	12,000 each	1,368,072 lb/hr at 43.0 psi, 122 hp motor
Ammonia Cleanup System	Vertical Pressure Vessel with Pump	1	Column 48-in. diam. 20 ft height	6,900	1.46 MW(e) intermittent power required
Leak Detection	Pump With Analyzer	1	—	1,200 (est)	Seawater into NH ₃
Ammonia Storage System	Storage Tank With Pump and Compressor	4	Storage tank 6200 ft. ³ miscellaneous equipment	210,000	150% operating capacity plus 16% ullage
				111,000	
Overpressure Vent	Safety Relief Valves	2	5-in. pipe size	—	
Noncondensable Gas Removal System	Secondary Condenser and Compressor with Vent	1	—	9,300	
Standby Power System	Diesel-Generators 1-MW(e) Power Total	2	—	102,000	Platform mounted
Electrical Power Distribution System	Startup, Normal, Standby, and Emergency	—	—	1,033,200	Cables, transformers, switchgear, and other equipment
Equipment Cooling	Seawater-Fresh Water Cooling System	1	—	12,000	
Nitrogen Purge	Liquid Nitrogen Supply Tank or Cylinders, Piping, Pressure in Reducing Valve	1	—	4,800	
Fouling Counter- Measures	Mechanical Tube Clean- ing and Chlorination System	2	—	2,000	
		1	—	3,500	
Data Acquisition and Control Systems (DACS)	Remote Terminal Units, Data Processing, Computer Programmer, and Control and Display Consoles	3	—	1,500	
		1	—	5,000	

4.1.5 Assembly and Installation

The assembly and installation approach for the CPM and the MAPS power modules is in principle the same. However, because the 50-MW(e) module is much larger than the 10-MW(e) module, the bulkier components present different handling problems.

The assembly of the components for the module subassemblies is shown in Fig. 4-6. This assembly procedure, i.e., assembly of the evaporator, condenser, and their related appendages and equipment into module subassemblies, will take place at the delivery point. Subassemblies for each heat exchanger section, as well as all power system equipment to be installed on the platform, will be shipped to the delivery and final assembly point either completely or partially assembled. The handling facilities and equipment available at the point of delivery are more than adequate for handling the 10-MW(e) segmented module subassemblies. However, for a unitized module and for the 50-MW(e) module subassemblies, lifting capacity is not available although other means of final assembly were investigated. The most promising are those that use the inherent flotation capability of the subassemblies. These methods are further discussed in Appendix J.

For the 10-MW(e) segmented module, flotation or barge loading and towing to the plant site are possible. For the unitized and the 50-MW(e) module assembly, flotation and towing are probably the most cost effective.

With the unitized arrangement, the power module is constructed as a complete unit in a drydock with the major components, piping, and structural supports rigidly attached together. The major differences between the unitized and the segmented configuration are as follows:

- The control and machinery deck is a single entity which eliminates two bulkheads required for the segmented arrangement
- A single vertical access trunk is provided with a cross tunnel connecting both machinery rooms. This also reduces the condensate piping run lengths between the heat exchangers

Fig. 4-6 Heat Exchanger Assembly

- The all-welded unitized structure reduces the structural support scantling requirements.
- The turbine-generator set is relocated within the machinery space. This lower elevation reduces the length of each vapor line and the exhaust diffuser by approximately 15 ft.

The final assembly procedure for that module configuration will be similar to that described above for the immersed-segmented configuration, except that the heat exchangers will be attached by connecting structure and they have a common access trunk.

At the platform, the final installation of the module units takes place by attitude control of the unit and ballasting to the required depth for mating. Details of the possible mating procedures are shown in Appendix J. These procedures are not complicated, but should probably take place at low ocean wave activity. After mating the module to the platform, ballast is removed and the final pipe, electrical, and access trunk connections above the water (immersed modules) are made. After a final inspection, the module is ready for startup operations.

4.1.6 Major Module/Platform Interfaces

For the module types derived during conceptual and preliminary design, families of interfaces between the power module/heat exchangers and the platform were initially identified. These are shown in Fig. 4-7 for immersed segmented MAPS module.

Typically, in overview, the 10-MW(e) segmented module/platform interfaces require the platform to provide

- Structural attachment and support
- Buoyancy control
- Seawater piping input (warm and cold)
- Turbine-generator sync. signal

Fig. 4-7 Interfaces, 10-MW(e) Power Module, Immersed - Segmented

- Distribution of net power output
- Seawater outflow piping (if required)
- Auxiliary power
- Personnel access

In addition, the platform provides the housing and structural foundations and support within that housing for all elements of the modular application power system except the two heat exchangers, the machinery rooms attached to the lower end of the heat exchangers, and the pumps and piping located within the machinery room and heat exchangers.

Three major interface requirements need early resolution since they impact the basic configuration design. These are all associated with the design of the platform seawater systems.

First, it is assumed that the structural arrangement of the heat exchanger attachment points will be compatible with the structural ring for water pipe interface and the saddle for load support (buoyancy). Second, the depth of submergence of the heat exchanger depends on the depth of the plenum chambers onboard the platform. Finally, the seawater pumps must be capable of providing at least 50 percent of the design flow in the reverse direction. The first item determines the method used for deployment and attachment of the power module. The second determines the length of vapor and condensate lines, machinery room access trunks, and any other piping and cable length as well as ballast and mating procedures. The final item is associated with the selection of the M.A.N. brushes as part of the biofouling countermeasures subsystem.

Structural Interfaces: A conservative approach to designing the structural interface between the heat exchangers and the module support structure is to assume that the sea-state motions of the coupled module and platform are the same as the motions of the uncoupled module during transit. This would allow proceeding with module joint design until the actual platform/module motions are defined. The immersed configurations would be attached to the platform at

the inlet ends of the seawater pump ducts, and where the module support structure mates with the surface platform. The structural mating interfaces of the submerged configuration would be at the inlets to the heat exchangers and the access trunk. The segmented module would require additional mechanical joints at four major ammonia piping runs, one to four turbine diffusers depending on whether or not the turbine is the radial flow or axial flow, and a joint between each section of the operations and machinery deck. The mating interface between the modules and the surface platform should be designed to facilitate selfalignment and utilize their inherent buoyancy to maintain the joint-locking force.

Module Design Loads: Several loading conditions are identified with each step in the assembly, deployment, and operations sequence. These steps are as follows:

1. Assembly in shipyard
2. Towing to site
3. Attachment to platform
4. Operation

The loads to be considered are primary loads only, acting on the overall structure itself, and include pressure, static and dynamic bending moments, shear, and torsion. In addition, rigid-body accelerations and other responses in a seaway, such as slamming, complete the extent of loading to be examined.

A second category of interfaces concerns interfaces within the power system module. These interfaces are shown in Fig. 4-8.

An interface data sheet is prepared to record first the types of data required to define the interface and finally, as the detailed design progresses, to record specific data affecting the interface. A number of data sheets were started in preliminary form during the Phase I effort. A typical data sheet is shown in Fig. 4-9. The data sheets and interface drawings which will be developed as part of the detailed design and analysis effort will become the Interface Control Documents (ICDs) for the power system. These ICDs will be

HEAT EXCHANGER - Evaporator
- Condenser

SEAWATER SUBSYSTEM - Warm
- Cold

TURBINE-GENERATOR

AMMONIA - Vapor & Condens. Piping, Valves

- Storage and Fill

- Drain (Unloading)

- Vapor Recovery

- Purge

- Noncondensable Gas Removal

- Sampling

- Cleanup

- Vent

- Leak Detection

CONTROL SYSTEM

INSTRUMENTATION

ELECTRICAL (primary)

AUX. SYSTEMS - Aux/Standby Pwr.

- Startup

- Power Distrib.

- Lube & Cooling
(except T-G)

- Fouling Countermeasures

- Corrosion Control

- Safety & Emergency

STRUCTURE (Module)

PERSONNEL SUBSYSTEM

SUPPORT EQUIPMENT

▽

▽

X ▽

X ▽

▽

X X X ▽

X X X ▽

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Note:
For Module vs. Platform
Interfaces, see separate
tabulation

Fig. 4-8 Interfaces Within Modular Application Power System 10-MW(e)
Immersed-Segmented

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Interfaces between Module Power Distribution Subsystem and Platform

A. PHYSICAL

- (1) Cable from turbine-generator to platform power distrib. bus.*
- (2) Cable from platform bus to module 4160 Vac bus.*
- (3) Cable from platform standby/emergency bus to module 480 Vac bus.*
- (4) Note. Also power instrumentation cabling.

*Define connections:

- Size
- Type
- Number
- Connectors, etc.
- Location of interface on platform/module

B. FUNCTIONAL

- (1) Primary power to startup from turbine-generator
 - (a) Voltage: 13,800
 - (b) Frequency (Hz)
 - (c) Max. Power (KVA)
 - (d) Continuous duty
- (2) Standby/Emergency power from platform
 - (a) Voltage
 - (b) Frequency
 - (c) Max. Power
 - (d) Max. period/duty cycle

C. OPERATIONAL

- (1) Turbine-generator is synchronized with platform and suppliers net power output to platform
- (2) Platform supplies power to module power distrib. bus for:
 - (a) Startup: Control subsyst., instrumentation, safety, hotel load (lighting, etc.), ammonia pumps and valves, filling
 - (b) Shutdown: Controls, instrum., safety, hotel load, pumps and valves, purging
 - (c) Maintenance: Tools, hotel, instrum., machinery (cranes, etc.)
 - (d) Preparation and operational readiness at platform
 - (e) Disassembly and/or removal from platform
- (3) Switching requirements
- (4) Safety requirements

Fig. 4-9 Interface Data Sheet (preliminary)

modified, reviewed, formalized, and controlled as a consequence of the PSD Interface Control Working Group (ICWG) activities anticipated during the detailed design and fabrication period.

4.2 SYSTEM PERFORMANCE

Three computer programs have been used for the analytical modeling of the OTEC power system. All three programs use the same basic thermodynamic equations for the determination of heat exchanger total heat flux and turbine work and the same fluid dynamic equations for determination of parasitic power losses. The first program (Power Cycle Optimization) defines an optimum size for the heat exchangers by varying seawater and ammonia flow rates to arrive at the optimum configuration for a given net output. The primary option minimizes the heat transfer area and an alternative option minimizes heat exchanger cost based on available material and fabrication cost algorithms. Typical outputs from this program for the 50- and the 10-MW(e) modules are presented in Appendix A. The second program (Broan) determines the off-design performance of the optimized configuration, with fixed values of seawater and ammonia flow rates. The third program (POCYOP) also determines the off-design performance of the selected configuration; however, this program varies seawater and ammonia flow rate to arrive at the maximum net power output.

4.2.1 Configuration Analysis

The optimization program was used to determine the heat exchanger sizes as well as ammonia and seawater flow rates required for the 50-MW(e) net CPM and the 10-MW(e) MAPS. The design condition is for a 40°F ΔT with 80°F warm seawater and 40° cold seawater and pressure losses in the warm and cold seawater systems 10 and 15 ft of water respectively. The results of these computations are presented in Appendix A. Computations were also made for a power system based on an optimized configuration with calculated seawater system pressure losses.

The results are summarized in Tables 4-8 (A50) and 4-9 (A10). A reduction in required pumping power of approximately 1 percent is noted for CPM and 5 percent for MAPS. The Reference Baseline column presents values based in the designated 10 and 15 ft of water pressure loss, and the Optimized Geometry column is based on the surface platform from the Platform Configuration and Integration Study.

Table 4-8 (A50)

50-MW(e) COMMERCIAL POWER SYSTEM
MODULE-POWER ALLOCATIONS

Item	Power (MW) (except as noted)	
	Reference Baseline ΔP (a)	Optimized Geometry (b)
Turbine Power	68.3	67.8
Generator Power	67.3	66.8
Warm Water Pumping Power	6.7	6.7
Cold Water Pumping Power	8.7	8.2
Ammonia Pumping Power	1.4	1.4
Controls, Hotel, Internal Losses, etc.	0.4	0.4
Chlorination Power ^(c)	0.08 (to 0.64)	0.08 (to 0.64)
Mechanical Cleaning Power ^(d)	—	—
Net Power Out	50.0	50.0
Turbine Gross/Net Power (ratio)	1.37	1.36

- (a) Reference baseline includes 10 ft warm (evaporator) and 15 ft cold (condenser) seawater head losses
- (b) Optimized geometry represents surface platform estimated seawater head losses
- (c) 0.2 mg/liter to (1.6) mg/liter dosage
- (d) System sized with added heat exchanger pressure loss; cleaning system operated by seawater flow reversal

Table 4-9 (A10)

10-MW(e) MODULAR APPLICATION POWER SYSTEM
POWER ALLOCATIONS

Item	Power (MW)	
	Reference Baseline P (a)	Optimized Geometry (b)
Turbine Power	14.8	14.0
Generator Power	14.6	13.8
Warm Water Pumping Power	1.7	1.3
Cold Water Pumping Power	2.2	1.8
Ammonia Pumping Power	0.3	0.3
Controls, Hotel, Internal Costs, etc.	0.4	0.4
Chlorination Power ^(c)	0.02 (to 0.16)	0.02 (to 0.16)
Mechanical Cleaning Power ^(d)	—	—
Net Power Out	10.0	10.0
Turbine Gross/Net Power (ratio)	1.48	1.40

- (a) Reference baseline includes 10 ft (evaporator) and 15 ft (condenser) of seawater head losses
- (b) Optimized geometry represents surface platform estimated seawater head losses
- (c) 0.2 mg/liter to 1.6 mg/liter dosage
- (d) System sized with added heat exchanger pressure loss. Cleaning system operated by seawater flow reversal

4.2.2 Off-Design Performance

The off-design performance of power system modules designed to provide 50- and 10-MW(e) net at the design temperature differential of 40°F , has been computed using both off-design programs. The results, presented in Fig. 4-10 and 4-11 are based on the off-design performance characteristics of the radial in-flow turbine (Appendix F).

In order to evaluate the effect of absolute temperature on performance, off-design performance computations were made with two approaches. In the first approach, the cold seawater temperature was maintained at 40°F and warm seawater temperature varied to provide the temperature differential. In the second approach, both warm and cold seawater temperatures were varied as shown in Table 4-10. The net power versus ΔT for both approaches for the 10-MW(e) case is shown in Fig. 4-12 and it is apparent that absolute temperature has a negligible effect on net power available.

Fig. 4-10 Off-Design Performance of 50-MW(e) (Net) Module - Optimized and Constant Flow

Fig. 4-11 Off-Design Performance of 10-MW(e) (Net) Module - Optimized and Constant Flow

Table 4-10

TEMPERATURE CONDITIONS (°F)

<u>Warm</u>	<u>Cold</u>	<u>ΔT</u>
85	41	44
85	50	35
80	50	30
80	36	44
70	32	38
70	41	29

Fig. 4-12 Performance Characteristics of the 10-MW(e) Power System Module at Part Load

4.2.3 Performance Characteristics of the 10-MW(e) Power System Module at Part Load

The performance characteristics of the 10-MW(e) power system module at part load are dependent upon, available thermal resource (see Fig. 4-12) and percent load requirement.

First, with the available thermal resource reduced below the design ΔT of 40°F, the control system central processing unit using the sensed seawater temperatures will command the ammonia condensate and distribution pump flow rates to maximize the net power output. These commands will incorporate the

reduced efficiency of the equipment operating at off-design conditions. The turbine, operating at constant speed throughout its load range, will provide less torque based on lower ammonia enthalpy and flow rate, hence there will be less generator output. Second, with a reduction in load requirement (not thermal resource limited), the turbine rpm will start to increase with the reduced torque on the turbine and the inlet vanes will close reducing the turbine rpm to its set point. The ammonia pressure and temperature will increase in the evaporator and decrease in the condenser with the reduced ammonia flow rate to a new stabilized condition and as the level in the evaporator sump starts to rise due to reduced vaporization rate, the distribution and condensate pump flow rates will be reduced to maintain ammonia flow rates and appropriate sump levels. The seawater flow rate will be reduced also. The central processing unit (CPU) receives the signals for part load and issues the commands to provide the load output while minimizing the flows of ammonia and seawater to produce the load.

4.2.4 Effect of Turbine Efficiency

To establish the sensitivity of power system cost to turbine cost (for turbine costs in the range of \$40 to \$100/kW corresponding to efficiencies of 83 to 86 percent) with the power system sized to provide 50-MW(e), the following variables were defined as a function of turbine efficiency:

- Heat exchanger configurations
- Turbine generator sizes
- Seawater pump flow rates

The Optimization Program was used to resize the heat exchangers. Estimated costs for these heat exchangers are plotted in Fig. 4-13 for labor costs of \$15/hr and \$20/hr. The turbine generator and seawater pump costs were added, and the results are as shown in Fig. 4-14. The turbine subcontractors pointed out that the cost gradient indicated for efficiency improvement was considerably higher than they would have predicted. The sensitivity of the results to turbine cost gradient was evaluated by keeping the midpoint at the same value and reducing the gradient from \$20 to \$10/percent efficiency. The results

Fig. 4-13 Effect of Turbine Efficiency on Heat Exchanger Costs

Fig. 4-14 Effect of Turbine Efficiency on Module Component Costs

are shown in Fig. 4-15, they indicate that the objective should be to attain the highest efficiency with a reasonable cost gradient.

4.2.5 Effect of Heat Exchanger Tube Length/Diameter Ratio

Studies conducted for a previous project indicated that the optimum tube length/diameter ratio (L/D) was between 350 and 400 for tube diameters of 2.0 and 3.0 in. A similar study was conducted for the 1.5-in. diameter aluminum tube proposed for the commercial size module. A family of configurations was developed by varying the number of tubes in the heat exchangers. The resultant heat exchanger costs are shown in Fig. 4-16. The minimum cost evaporator is at an L/D of approximately 350 while the condenser minimum is at an L/D of approximately 400.

The effect of tube L/D on system costs must include the effect on turbine-generator and seawater system sizes and costs. The lower the value of L/D for a decreasing L, the greater the number of tubes required to provide the heat transfer area and consequently, a higher seawater flow rate required to maintain a reasonable overall heat transfer coefficient. Using a cost of \$50/kW for the turbine and \$600/CFS for the seawater pumps, the results are as shown in Fig. 4-17. From these curves it is seen that the optimum tube L/D has a value between 450 and 500.

4.2.6 Effect of Biofouling.

The maximum allowable biofouling thermal resistance by DOE directive is $0.0005 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$. Test data presented in subsection 2.3.1.5 indicate that the minimum possible value, following a cleaning cycle, is less than 0.0001. Based on this range, a conservative value of 0.0003 was selected for sizing the heat exchangers. The predicted effect of biofouling on the performance of the power system is obtained by varying the biofouling resistance in the off-design performance program. The overall heat transfer coefficient, U, is determined from

Fig. 4-15 Effect of Turbine Efficiency on Module Component Costs

Fig. 4-16 Effect of Tube L/D on Heat Exchanger Costs

Fig. 4-17 Effect of Tube L/D on Module Component LCosts

$$U = \frac{1}{\frac{1}{U_1} + \frac{1}{U_2} + \frac{1}{U_3} + FR}$$

where

U_1 = seawater heat transfer coefficient

U_2 = ammonia heat transfer coefficient

U_3 = tube wall conductivity

FR = fouling resistance factor

The results of varying FR from zero to 0.0005 are shown in Fig. 4-18 for the 10-MW(e) power system operating with the design thermal resource temperature differential of 40°F and considering that seawater and ammonia flow rates are optimized as required. From this figure it is seen that an increase in the fouling resistance factor of 0.0001 (from 0.0002 to 0.0003 as an example) reduces the net power output by 0.6-MW(e). At a cost of power of 4c/kWh, this equates to a revenue loss of \$576/day or approximately \$210,000/year. The

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

importance of controlling the biofouling layer is demonstrated by the relative effect of biofouling on heat exchanger performance. For example, a decrease in the fouling resistance from the nominal value of 0.0003 to the minimum anticipated value of 0.0001 for the baseline case (ammonia heat transfer coefficient of 3000) causes a 14 percent increase in overall heat transfer coefficient of the titanium evaporator. For the condenser, this increase is 10.5 percent.

4.2.7 Platform Influences on System Performance

The configuration and performance data presented are based on ammonia evaporation heat transfer data obtained from thermodynamic testing of horizontally oriented tubes. Recent tests of the LMSC Palo Alto Research Laboratories on an enhanced tube, inclined at approximately 1 deg, indicated an improvement in heat transfer characteristics at low flow rates. This phenomenon is attributed to improved tube wetting. Additional testing will be accomplished

at higher angles representative of anticipated platform/heat exchanger pitch angles during platform/operation.

References cited in the LMSC Conceptual Design Report (LMSC-D566744), indicate an improvement in condensation rate with tube inclinations up to 15 deg. An analysis by A.G. Sheynkman and V.N. Linetskiy ("Hydrodynamics and Heat Transfer by Film Condensation of Stationary Steam on an Inclined Tube," Heat Transfer-Soviet Research, Vol. 1, No. 3, May 1969) indicates an improvement of condensation on inclined plain tubes within a bundle at the higher flow rates experienced by those tubes. Their results indicate an improvement on the order of 12 percent for a Reynolds number of 300 at an angle of inclination to the horizontal of approximately 15 deg. Tube bundle inclination is also expected to reduce the tube to tube flow by diverting a portion of the liquid drainage to the tube support plates. This is expected to be available for condensation by virtue of the reduction in tube to tube flow rate typical of a horizontal tube orientation. The influence of condenser tube bundle inclination will be determined with the test article which is designed to allow a range of orientations to the horizontal.

The condenser tubes are inclined at 15 deg to the horizontal. Platform pitch angularity limits, defined in the Phase I RFP, should have no detrimental effect on cycle performance.

The effect of heat exchanger roll angle and vertical acceleration on performance will be determined as part of the continuing LMSC Independent Research and Development program. The heat exchangers incorporate 7 percent additional tubes to provide for tube manufacturing tolerance and tube losses due to plugging. The additional tube area will also aid in maintaining required tube wetted area at extreme roll angles.

The MAPS platform, because of its smaller size relative to the CPM platform, is expected to have greater seaway-induced motions. Consequently, platform effects on MAPS performance could be of greater magnitude than for the CPM.

4.3 HEAT EXCHANGERS

During the Conceptual Design task, cost trade studies were made for three different heat exchanger material combinations over a size range from 5- to 50-MW(e). The results indicated that the heat exchanger costs per unit of power decreased with size, the aluminum heat exchangers were the least expensive and lightest, a spherical shell was about the same cost as a cylindrical shell, and a 1.5 to 2.0 in. tube diameter was optimum. Based on these results and DOE Task 2 directives, the reference heat exchanger design was selected as a single, spherical, 50-MW(e) module which would be completely fabricated of aluminum. The heat exchangers are approximately 79 ft diameter spheres weighing about 1,900 tons each.

The 10-MW(e) heat exchangers are similar in design to the 50-MW(e) heat exchangers except for the reduction in size and the selection of material. The tubes are titanium (per DOE Guidelines) the tubesheets are titanium clad carbon steel, and the balance of the unit is carbon steel for the 10-MW(e) units. The heat exchangers are 40 ft diameter spheres and weigh about 500 tons. Heat exchanger supporting data is included in Appendix E.

The shell diameter, tube length, and required heat transfer surface were selected using a computer program to optimize the entire power module system. Both the evaporator and condenser will be designed and fabricated in accordance with ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. They are designed for immersion in seawater at 20 ft of water above the top of the heat exchangers. The design pressure inside the shell (ammonia side) is 168 psig maximum and 0 psia minimum.

The heat exchangers have been designed for fabrication of plate, tube, bar, and pipe. There are no forgings or intricate machining operations. Machining, such as shell segments or tubesheet segments, is done on existing machines before the parts are welded into large pieces. Final "machining," such as cutting the shell opening for the tubesheet/transition ring assembly, can be done with flame cutting equipment and will not require special

equipment. The stress analysis takes into account shop tolerances for mismatch in the axial direction between the shell and the transition ring.

4.3.1 Evaporator Design

The evaporator has two vapor discharge ports for the ammonia going to the turbine. A hood and vane separator (or mist contractor) inside the shell at each discharge ensures that liquid ammonia is not carried through to the turbine. The mist extractors gravity drain into the sump in the evaporator.

The three liquid ammonia ports are for the ammonia coming from the condenser, for the discharge from the evaporator sump, and for the return from the recirculation pumps. The recirculation pumps supply 15 percent of the overall system ammonia flowrate to the ammonia feed distribution manifolds so that the tubes are fully covered with an ammonia film to prevent ineffective tube surface due to dryout. The excess ammonia returns to the sump in the bottom of the evaporator and is recirculated.

The liquid ammonia enters the evaporator through the ammonia feed inlet and is distributed to rectangular, horizontal manifolds behind the tubesheet(s). The manifolds fit in the space created by deleting tubes at the horizontal tubesheet welds. The bottoms of the manifolds are perforated to allow the ammonia to irrigate the ends of the water tubes. The manifolds supply feed tubes are on the same 30 deg. triangular pitch as the water tubes. There is one horizontal feed tube for each column of water tubes. The feed tubes extend to the inner face of the second tubesheet, the end is plugged and is supported by a drilled plated which fastens to the tubesheet. The ammonia leaves the feed tubes through a row of holes along the top of each tube, hits a deflector plate made of a half tube, runs around the outside of the feed tube, and irrigates the column of water tubes below the feed tube. There are 30 to 70 water tubes in each column. The excess ammonia from the last water tube in the column passes over the next lower level feed/deflector tube and is added to the ammonia which is entering at that level. Some of the excess ammonia from

Fig. 4-19 Evaporator Configuration, 50-MW(e)

Fig. 4-19 Evaporator Configuration, 50-MW(e)

The 89,000 tubes are 1.5 in. OD with a 0.1025 in. minimum wall on a 1.875 in. triangular pitch. The tubes are circumferentially finned on the outside surface. All tubes are welded to the outer tubesheet with either a fillet weld (baseline) or an end weld (backing). About 460 of the tubes are also welded to the inner tubesheet annulus to provide stiffening for the outer rim of the outer tubesheet.

The tubesheets are 1.5 in. aluminum plate. The maximum available plate width is 15 ft, so that several plates would be welded together to form a tubesheet. If tubes are omitted to allow for tubesheet weld seams, the bundle diameter will become slightly larger. The inner tubesheet is annular with a 47.7 ft opening. The two tubesheets at each end are enclosed on the outer edge by a rolled transition ring which mates to the shell.

The aluminum tube support plates are also made up in sections with seams in line with the tubesheet seams. There are a total of ten support plates rather than the eight shown on the drawing. The tube support plates are enclosed by rings which are supported from the shell with pipe spacers. About 120 tie rods with tubular spacers are attached to one tubesheet to maintain the spacing between the support plates.

The shells are partial spheres which are made of 72 cold formed segments. The shell vapor and liquid ammonia ports are reinforced with formed segments and use heavy wall cylinders to strengthen the openings.

The evaporator tubes are arranged in five banks of tubes with an ammonia feed system for each bank. The present arrangement drawing shows the ammonia from the condenser entering through the ammonia inlet pipe at the bottom of the evaporator and feeding the lower three banks of tubes. Horizontal headers inside both outer tubesheets distribute the ammonia across the tube bundle. The area between the double tubesheets forms the upper header. The headers feed perforated ammonia feed tubes which are the same diameter as the water tubes and in the same pitch pattern as the water tubes. There is one ammonia

feed tube for each column of water tubes. The water tube pattern is modified to "square-off" the ends of the tube bank so that a minimum of 35 tubes and a maximum of 68 tubes are in any single column. Ammonia is fed at 150 percent of the predicted evaporation rate so that the tubes will not dry out at the lower end of each tube bank. All tube banks are fed from both sides of each header and from both ends of each feed tube to reduce the size of these components. Preliminary design refinement is required to determine sizes of orifices and pumps required to maintain a uniform ammonia flow in all tube banks despite the large differences in elevation.

Design data and thermal data for the evaporator are summarized in Tables 4-11 (D50) and 4-12 (F50).

Table 4-11 (D50)

HEAT EXCHANGER DESIGN DATA (EVAPORATOR) 50-MW(e) MODULE

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Number per Module	1	1
Thermal Rating (Mbtu/hr)	7280	7200
Water Flow Rate (gpm)	2,666,500	2,726,600
Ammonia Flow Rate (lb/hr)	20,746,400	20,500,000
Design Pressure (psi)(c)	168	168
Operating Pressure (psia)	130	130
Overall Length (ft)(d)	82.8	79.2
Water Enhancement Factor	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0
Total Number of Tubes	89,000	89,000
Tube OD (in.)	1.5	1.5
Tube Wall (in.)(e)	0.1025/0.090	0.1025/0.090
Tube Pitch (in.)	1.875	1.875
Tube Layout (angle from horizontal)	30°	30°
Tubesheet Diameter (ft)	55.3	55.3
Effective Tube Length (ft)	54.0	51.6
Actual Tube Length (ft)	55.2	52.8
Tubesheet Thickness (in.)	1.5	1.5
Second Tubesheet Thickness (in.) (if applicable) (f)	1.5	1.5
Number of Tube Supports	10	10
Tube Support Thickness (in.)	1.0	1.0
Tube Support Spacing (ft)	5.0	4.8
Shell OD (ft)	78.7	77.0
Shell Wall Thickness (in.)	6.5	6.5
Void Fraction	0.60	0.60
Tube Material	A1 5052	A1 5052
Tubesheet Material (g)	A1 5083	A1 5083
Shell Material	A1 5083	A1 5083
Baffle Material	A1 5083	A1 5083
Cladding Material	NA	NA
Coating Materials	(h)	(h)
Demister Size (ft x ft x ft)	27.5 x 27.5 x 0.75	27.5 x 27.5 x 0.75
Demister Design Pressure (psia)	(i)	(i)
Design Inlet Quality (%)	95	95
Design Outlet Quality (%)	99	99

(a) 10 ft warm and 15 ft cold
seawater head losses

(b) Estimated seawater
head losses

(c) Burst

(d) With inlet waterbox

(e) Mean/Minimum

(f) Annual Type

(g) Design for costing only - current
investigation indicates alclad 3004
is superior

(h) Not defined at this time

(i) Mist extractor inside evaporator shell

Table 4-12 (F50)
HEAT EXCHANGER THERMAL DATA
EVAPORAOR 50-MW(e) MODULE

Item	Reference Baseline $\Delta P(a)$	Optimized Geometry (b)
Water Flow Rate (lb/hr)	1365 x 10 ⁶	1396 x 10 ⁶
Water Flow Per Tube (lb/sec)	4.26	4.36
Water Velocity Inside Tube (ft/sec)	7.30	7.46
Reynolds Number Inside Tube (-)	70,136	71,719
Prandtl Number Inside Tube (-)	6.16	6.16
Inlet Water Temperature (°F)	80	80
Outlet Water Temperature (°F)	74.20	74.47
Average Water Temperature (°F)	77.14	77.24
LMTD (°F)	6.312	6.698
Average Tube Wall Surface Temp (°F)	71.3	71.0
Average Shell Wall Surface Temp (°F)	71.5	71.2
hw (Enhanced) (Btu/hr ft ² °F)	1,329	1,355
h Fouling (Btu/hr ft ² °F)	3,333	3,333
h Wall (Btu/hr ft ² °F)	9,709	9,709
h Ammonia (Btu/hr ft ² °F)(c)	3,475	3,475
Uo (Btu/hr ft ² °F)(c)	693	700
Ammonia Inlet Temperature (°F)	49.61	49.31
Ammonia Outlet Temperature (°F)	70.31	70.01
Vapor Quality (Outlet, %)	99	99
Maximum Vapor Velocity (Outlet, ft/sec)	19.31	19.1
Maximum Vapor Velocity (Between Tubes, ft/sec)	8.2	8.9
Total Pressure Drop of Water (psid)	4.46	4.39
Total Pressure Drop of Ammonia (psid)(d)	0.56	0.64
Distribution Tubes/Heat Transfer Tubes	0.016	0.016
Maximum Feed Rate (lb/ft sec)	0.10	0.10
Minimum Feed Rate (lb/ft sec)	0.05	0.05
Design Feed Rate (lb/ft sec)	0.05	0.08
Recirculation Ratio (-)	0.5	0.5
Maximum Distribution System P (psid)	15	15

- (a) 10 ft warm and 15 ft cold seawater system head losses
 (b) Estimated seawater head losses
 (c) Based on tube ID
 (d) Across tube bundle radius using Eqs. 6.13 and 6.13 (b) of
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4.3.1.2 10-MW(e) Evaporator. The general arrangement of the MAPS evaporator is shown in Fig. 4-20 (drawing 58-3157-6-0210). The shell is approximately 40 ft in diameter, 26 ft long and weighs about 500 tons. See Appendix C for drawings and Appendix E for support data.

The liquid ammonia enters the evaporator through the ammonia feed inlet and is distributed to rectangular, horizontal manifolds at four elevations behind one tubesheet. The manifolds extend 18 in. from the inside face of the tubesheet and are 3.5 in. high. The manifolds fit in the space created by deleting tubes at the horizontal tubesheet welds. The bottoms of the manifolds are perforated to allow the ammonia to irrigate the last 18 in. of the water tubes. The manifolds supply 1 in. OD carbon steel feed tubes which are on the same 30 degree triangular pitch as the water tubes. There are 30 to 63 water tubes in each column.

The evaporator has 51,400 tubes which are welded titanium tubes of 1 in. OD with a smooth inside surface and a circumferential fin on the outer surface. The tube wall is 0.028 in. minimum at the root of the fins, and the fins are 0.025 in. high with 15 to 20 fins per in. The tubes will have smooth lands at the ends and at the tube support plates which are also 1 in. OD with about a 0.050 in. wall. The tubes are on a 1.25 in. pitch. The tubes will be welded to the tubesheets with an end weld.

The tubesheets are 1 in. thick carbon steel with a 3/16 in. thick titanium Detacclad cladding. The clad material is available in 92 in. widths, so that 8 sheets must be welded together to form a tubesheet. The weld seams have been located to match the tube lanes in the evaporator due to the ammonia feed system manifolds. The titanium is removed at each weld seam in order to gain access to both faces of the base (carbon steel) material for welding. The tubes which must be omitted in the horizontal direction due to the missing titanium allow room for the horizontal ammonia feed headers. Three columns of tubes must also be deleted to accommodate the single vertical weld seam. Design calculations are based on the carbon steel thickness only, and the titanium straps will not be welded between tubesheet sections. It would be

Fig. 4-20 Evaporator Configuration, MAPS

difficult to seal the ends of the straps. An epoxy or similar protective material can be applied to the weld seams to protect them from corrosion on the water side. The function of the titanium clad is to provide a compatible material for welding the tubes. The tubesheets are enclosed on the periphery by a carbon steel transition ring. This ring allows for a fabrication mismatch between the tubesheet and shell. It will also provide the attachment point for the water boxes. The ring will be modified in the detail design when the interface with the water boxes is defined and interaction loads are specified.

The tube support plates are also made up in sections having seams in line with the tubesheet seams. The shell is a sphere which is made of 16 "orange peel" segments. Design data and thermal data for the evaporator are shown in Tables 4-13 (D10) and 4-14 (F10).

Table 4-13 (D10)
HEAT EXCHANGER DESIGN DATA (EVAPORATOR) MAPS

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Number per Module	1	1
Thermal Rating (Mbtu/hr)	1,562	1,440
Water Flow Rate (gpm)	677,500	712,750
Ammonia Flow Rate (lb/hr)	4,455,300	4,104,000
Design Pressure (psi) (c)	168	168
Operating Pressure (psia)	130	130
Overall Length (ft) (d)	40.8	36.8
Water Enhancement Factor	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0
Total Number of Tubes	51,400	51,400
Tube OD (in.)	1.0	1.0
Tube Wall (in.) (e)	0.028	0.028
Tube Pitch (in.)	1.25	1.25
Tube Layout (angle from horizontal)	30 deg	30 deg
Tubesheet Diameter (ft)	29.3	29.3
Effective Tube Length (ft)	25.5	24.1
Actual Tube Length (ft)	26.2	24.8
Tubesheet Thickness (in.)	1.19	1.19
Second Tubesheet Thickness (in.) (if applicable) (f)	-	-
Number of Tube Supports	7	6
Tube Support Thickness (in.)	1.0	1.0
Tube Support Spacing (ft)	3.9	3.7
Shell OD (ft)	39.4	38.4
Shell Wall Thickness (in.)	2.0	2.0
Void Fraction	0.60	0.60
Tube Material	Ti	Ti
Tubesheet Material	SA516 GR70	SA516 GR70
Shell Material	SA516 GR70	SA516 GR70
Baffle Material	A36	A36
Cladding Material	Ti	Ti
Coating Materials	(g)	(g)
Demister Size (ft x ft x ft)	4.25 x 6 x 0.75	4.25 x 6 x 0.75
Demister Design Pressure (psia)	(h)	(h)
Design Inlet Quality (%)	95	95
Design Outlet Quality (%)	99	99
(a) 10 ft warm and 15 ft cold seawater head losses	(e) Minimum	
(b) Estimated seawater head losses	(f) Annular type	
(c) Burst	(g) Not defined at this time	
(d) With inlet water box	(h) Mist Extractor inside evaporator shell	

Table 4-14 (F10)
HEAT EXCHANGER THERMAL DATA (EVAPORATOR) MAPS

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Water Flow Rate (lb/hr)	347 x 10 ⁶	365 x 10 ⁶
Water Flow Per Tube (lb/sec)	1.875	1.972
Water Velocity Inside Tube (ft/sec)	6.47	6.807
Reynolds Number Inside Tube (-)	45,327	47,686
Prandtl Number Inside Tube (-)	6.16	6.16
Inlet Water Temperature (°F)	80	80
Outlet Water Temperature (°F)	75.17	75.77
Average Water Temperature (°F)	77.585	77.885
LMTD (° F)	7.004	7.455
Average Tube Wall Surface Temp.(°F)	71	71.1
Average Shell Wall Surface Temp.(°F)	71.5	71.4
hw (including Enhancement) (Btu/hr ft ² °F)	1,272	1328
h Fouling (Btu/hr ft ² °F)	3,333	3,333
h Wall (Btu/hr ft ² °F)	4,587	4,587
h Ammonia (Btu/hr ft ² °F)	3,180	3,180
Uo (Btu/hr ft ² °F) (c)	618	631
Ammonia Inlet Temperature (°F)	49.93	49.76
Ammonia Outlet Temperature (°F)	70.24	70.16
Vapor Quality (Outlet, %)	99	99
Maximum Vapor Velocity (Outlet, ft/sec)	24.4	22.5
Maximum Vapor Velocity (Between tubes ft/sec)	6.6	6.1
Total Pressure Drop of Water (psid)	4.43	2.81
Total Pressure Drop of Ammonia (psid) (d)	0.284	0.241
Distribution Tubes/Heat Transfer Tubes	0.016	0.026
Maximum Feed Rate (lb/ft sec)	0.053	0.053
Minimum Feed Rate (lb/ft sec)	0.028	0.028
Design Feed Rate (lb/ft sec)	0.044	0.044
Recirculation Ratio (-)	0.5	0.5
Maximum Distribution System P (psid)	15	15

(a) 10 ft warm and 15 ft cold seawater head losses

(b) Estimated seawater head losses

(c) Based on tube ID

(d) Across tube bundle radius using Eqs. 6.13 and 6.13 (b) of
HEAT TRANSMISSION by McAdams

4.3.2 Condenser Design

Depending on the turbine type (single or dual discharge), the condensers can have either one or two vapor inlet ports. An impingement plate at each inlet prohibits direct vapor inlet blast on the tubes and provides for better vapor distribution through the tube bundle.

The condenser is pitched with its tubes 12 to 15 deg off horizontal to encourage the condensing ammonia to run along the tubes and down the tube support plates. The single liquid discharge from the bottom of the condenser goes to the condensate pump. The adapter ring, which is welded to the bottom of the condenser shell, mates with a pod which contains the pump. The pump liquid discharge line reenters the bottom of the condenser and exits through the shell at an elevation which matches the liquid inlet to the evaporator.

The condenser has a shroud over 3 percent of the tubes in the center of the bundle to aid in removal of noncondensable gases. The gases which collect inside the shroud are removed from the tube bundle by three vertical tubes which are perpendicular to the water tubes and in line with the tubesheet seams. The final choice of the number of such tubes inside the shroud will be according to the specification for noncondensable gas removal rate. The shroud in the center portion of the tube bundle is a standard industrial method for removing gases from the bundle at its coolest point.

The shell support plates, internal structure and tube-sheet-shell transition ring are the same as for the evaporator. This minimizes development and tooling costs.

4.3.2.1 50-MW(e) CPM Condenser. The general arrangement of the 50-MW(e) condenser is shown in Fig. 4-21 (drawing 8-58-3157-0100). The condenser is about 79 ft in diameter, 55 ft long, and weighs 2800 tons. The 106,000 water tubes are identical to those used in the evaporator condenser tubesheets, tube support plates and shell design and manufacturing concepts are also similar to those discussed in the preceeding section.

Fig. 4-21 Condenser Configuration, 50-MW(e)

Two 20.ft vapor inlet ports are connected to the ammonia outlet pipes from the turbine. An impingement plate at each inlet directs the ammonia vapor around the tube bundle for more uniform vapor distribution.

Design data and thermal data for the condenser are shown in Tables 4-15 (E50) and 4-16 (G50)

Table 4-15 (E50)
HEAT EXCHANGER DESIGN DATA (CONDENSER) 50-MW(e) MODULE

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Number per Module	1	1
Thermal Rating (Mbtu/hr)	7,053	6,972
Water Flow Rate (gpm)	2,305,000	2,598,000
Ammonia Flow Rate (lb/hr)	13,831,000	13,666,300
Design Pressure (psi) (c)	168	168
Operating Pressure (psia)	90	90
Overall Length (ft) (d)	82.8	79.2
Water Enhancement Factor	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0
Total Number of Tubes	103,000	103,000
Tube OD (in.)	1.5	1.5
Tube Wall (in.) (e)	0.1025/0.090	0.1025/0.090
Tube Pitch (in.)	1.875	1.875
Tube Layout (Angle from horizontal)	30°	30°
Tubesheet Diameter (ft)	55.3	55.3
Effective Tube Length (ft)	54.0	51.6
Actual Tube Length (ft)	55.2	52.8
Tubesheet Thickness (in.)	1.5	1.5
Second Tubesheet Thickness (in.) (if applicable) (f)	1.5	1.5
Number of Tube Supports	10	10
Tube Support Thickness (in.)	1.0	1.0
Tube Support Spacing (ft)	5.0	4.8
Shell OD (ft)	78.7	77.0
Shell Wall Thickness (in.)	6.5	6.5
Void Fraction	0.60	0.60
Tube Material	A1 5052	A1 5052
Tubesheet Material	A1 5083	A1 5083
Shell Material	A1 5083	A1 5083
Baffle Material	A1 5083	A1 5083
Cladding Material	NA	NA
Coating Materials	(g)	(g)

- (a) 10 ft warm water and 15 feet cold water head losses
(b) Estimated seawater head losses
(c) Burst/Collapse
(d) Including inlet water box
(e) Mean/Minimum
(f) Annular Type
(g) Not defined at this time

Table 4-16 (G50)
HEAT EXCHANGER THERMAL DATA (CONDENSER) 50-MW(e) MODULE

Item	Reference Baseline (a)	Optimized Geometry (b)
Water Flow Rate (lb/hr)	1186 x 10 ⁶	1337 x 10 ⁶
Water Flow Per Tube (lb/sec)	3.20	3.61
Water Velocity Inside Tube (ft/sec)	5.45	6.16
Reynolds Number Inside Tube (-)	40,289	45,421
Prandtl Number Inside Tube (-)	10.6	10.6
Inlet Water Temperature (°F)	40.0	40.0
Outlet Water Temperature (°F)	46.4	45.6
Average Water Temperature (°F)	43.2	42.8
LMTD (°F)	6.168	6.463
Average Tube Wall Surface Temp (°F)	49.8	49.4
Average Shell Wall Surface Temp (°F)	54	53
h _w Enhanced (Btu/hr ft ² °F)	878	968
h Foulings (Btu/hr ft ² °F)	3,333	3,333
h _{Wall} (Btu/hr ft ² °F)	9,709	9,709
h _{Ammonia} (Btu/hr ft ² °F)	3,475	3,475
U _o (Btu/hr ft ² °F) (c)	547	580
Ammonia Inlet Temperature (°F)	50.2	50.0
Ammonia Outlet Temperature (°F)	49.6	49.3
Vapor Quality (Inlet %)	96.6	96.6
Maximum Vapor Velocity (Inlet, ft/sec)	19.4	19.2
Max. Vapor Velocity (Between Tubes, ft/sec)	9.8	11.0
Total Pressure Drop of Water (psid)	2.69	3.08
Total Pressure Drop of Ammonia (psid) (d)	1.09	1.33

- (a) 10 ft warm and 15 ft cold seawater system head losses
 (b) Estimated Seawater head losses
 (c) Based on tube ID
 (d) Across tube bundle radius using Eqs. 6.13 and 6.13 (b) of
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4.3.2.2 10-MW(e) MAPS Condenser. The condenser has 64,200 titanium seam welded tubes which are installed in the same manner as those in the evaporator.

Design data and thermal data for the condenser are shown in Tables 4-17(E10) and 4-18 (G10).

4.3.3 Heat Exchanger Studies

During Task 2 additional analyses and trade studies were conducted to ensure the results of the conceptual design studies were valid.

4.3.3.1 Cost Trade Studies. Additional cost trade studies were made for the aluminum 50-MW(e) heat exchangers using the same estimating methods as were used in the Conceptual Design task. These studies determined the cost sensitivity to tube length and to turbine efficiency. The results are summarized in Table E-1 in Appendix E. These results were used to optimize the cost of the overall system.

An alternate material combination was also sized and costed. The tubes were still 5052 aluminum, but the tubesheets were 5083 aluminum clad on a carbon steel base and the balance of the heat exchanger was carbon steel. The steel shell wall was reduced to 3.5 in. thick as compared to the 6.5 in. thick aluminum. The tubesheet thickness remained at 1.5 in. thick, but the inner partial tubesheet was eliminated. The cost of the aluminum/carbon steel heat exchanger was estimated to reach the same as that of the all-aluminum units, while weighing about 850 tons more than the all-aluminum unit. Therefore, the all-aluminum design was selected as the reference.

A trade study for four cases was made to verify that the results from the Conceptual Design using smooth tubes was applicable to enhanced tubes. Evaporators were sized for 1.5 or 2 in. OD tubes in either the immersed or submerged design conditions. The key dimensions and estimated costs are shown in Table E-2 of Appendix E. The submerged design is still less expensive than the immersed due to the lower design pressure. The 1.5 in. tube design is

Table 4-17 (E10)
HEAT EXCHANGER DESIGN DATA (CONDENSER) MAPS

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Number per module	1	1
Thermal Rating (Mbtu/hr)	1,514	1,395
Water Flow Rate (gpm)	589,000	641,500
Ammonia Flow Rate (lb/hr)	2,970,000	2,736,000
Design Pressure (psi)(c)	168	168
Operating Pressure (psia)	90	90
Overall Length (ft)(d)	40.0	38.8
Water Enhancement Factor	1.0	1.0
Ammonia Enhancement Factor (over smooth tube)	3.0	3.0
Total Number of Tubes	64,200	64,200
Tube OD (in.)	1.0	1.0
Tube Wall (in.) (e)	0.028	0.028
Tube Pitch (in.)	1.25	1.25
Tube Layout (angle from horizontal)	30°	30°
Tubesheet Diameter (ft)	29.3	29.3
Effective Tube Length (ft)	25.5	24.1
Actual Tube Length (ft)	26.2	24.8
Tubesheet Thickness (in.)	1.19	1.19
Second Tubesheet Thickness (in.) (if applicable)(f)	-	-
Number of Tube Supports	7	6
Tube Support Thickness (in.)	1.0	1.0
Tube Support Spacing (ft)	3.9	3.7
Shell OD (ft)	39.4	38.4
Shell Wall Thickness (in.)	2.0	2.0
Void Fraction	0.60	0.60
Tube Material	Ti	Ti
Tubesheet Material	SA516 GR70	SA516 CR70
Shell Material	SA516 GR70	SA516 GR70
Baffle Material	A36	A36
Cladding Material	Ti	Ti
Coating Materials	(g)	(g)

(a) 10 ft warm and 15 ft cold seawater head losses

(b) Estimated seawater head losses

(c) Burst

(d) Including inlet water box

(e) Minimum

(f) Annular Type

(g) Not defined at this time

Table 4-18 (G10)
HEAT EXCHANGER THERMAL DATA (CONDENSER) MAPS

Item	Reference Baseline ΔP (a)	Optimized Geometry (b)
Water Flow Rate (lb/ft)	303 x 10 ⁶	330 x 10 ⁶
Water Flow Per Tube (lb/sec)	1.311	1.428
Water Velocity Inside Tube (ft/sec)	4.50	4.901
Reynolds Number Inside Tube (-)	24,247	26,408
Prandtl Number Inside Tube (-)	10.6	10.6
Inlet Water Temperature (°F)	40.0	40.0
Outlet Water Temperature (°F)	45.34	44.52
Average Water Temperature (°F)	52.67	42.26
LMTD (°F)	7.079	7.459
Average Tube Wall Surface Temperature (°F)	49	49
Average Shell Wall Surface Temperature (°F)	53	53
hw (Including Enhancement (Btu/hr ft ² °F)	788	843
h Fouling (Btu/hr ft ² °F)	3333	3333
h Wall (Btu/hr ft ² °F)	4587	4587
h Ammonia (Btu/hr ft ² °F) (c)	3180	3180
Uo (Btu/hr ft ² °F) (c)	476	495
Ammonia Inlet Temperature (°F)	50.27	50.16
Ammonia Outlet Temperature (°F)	49.0	49.73
Vapor Quality (Inlet, %)	96.6	96.6
Maximum Vapor Velocity (Inlet, ft/sec)	21.73	20.1
Maximum Vapor Velocity (Between Tubes ft/sec)	8.2	3.26
Total Pressure Drop of Water (psid)	2.91	3.26
Total Pressure Drop of Ammonia (psid) (d)	0.639	0.744

- (a) 10 ft warm and 15 fr cold seawater head losses
 (b) Estimated head losses
 (c) Based on tube ID
 (d) Across tube bundle radius using Eqs 6.13 and 6.13 (b) of
 HEAT TRANSMISSION by McAdams

slightly less expensive than the 2 in. design due to the better tube packing which produces a smaller heat exchanger.

4.3.3.2 Structural Analysis. The structural analysis for the preliminary design was made with a simplified ANSYS finite element analysis. The computer model used the axisymmetric conical shell element (STIF 61) for the spherical shell, tubesheet and transition ring, and a two-dimensional spar element (STIF 1) for the tubes. The model had about 200 elements, of which half were of the STIF 61 type.

This model can accurately evaluate the stresses in the shell, tubesheet and tubes but not in the junction regions of the transition ring between the shell and the tubesheet. A more detailed analysis of this region using a finite element continuum model will be required during the detail design phase. The structural analysis considered only the effects of an internal pressure of 168 psig. Thermal effects were judged to be insignificant and were neglected. Load inputs from the platform and from the water boxes were not specified and were not considered.

50-MW(e) Power Module. Eight analysis cases were run for the 50-MW(e) size heat exchangers. The model geometry for each case is described by Fig. E-1 and Table E-3 in Appendix E, and the results are tabulated in Table E-4. Case 1 through 6 studied the effect of increasing the opening in the inner tubesheet as well as changes in the junction ring geometry. Case 7 reduced the radial distance between the outer tube and the transition ring which had little effect. Increasing the tubesheet thickness in Case 8 caused unacceptable stress in the tube. The selected reference design is based on Case 5. Results can be summarized as follows:

- a. When the complete heat exchanger is modeled, the shell does not act as a true sphere. The shell wall thickness must be 6.5 in. to maintain acceptable shell pressure stresses. This thickness is about midway between that required for a true sphere and a true cylinder with the same radius.

- b. The relatively thick shell wall furnishes a high degree of stiffening and reduces the stresses at the tubesheet junction. The maximum tube compression in the outer rows is 500 psi.
- c. The inner and outer tubesheet must be 1.5 in. thick. Only the outer region of the tubesheet is required to stiffen the outer tubesheet. The radial dimension of the inner tubesheet is reduced to a 22.5 in. annulus.
- d. The water tubes can be used as stiffeners for the tubesheets. They can also be used as stays between the inner and outer tubesheets. The stay tubes are on 6 in. by 12 in. centers, and only the stay tubes are welded to the inner tubesheet. Stay tubes have a maximum compression of 1,100 psi and maximum tension of 7,700 psi depending on location. No stay tubes are required in the unperforated rim of the tubesheet.

Additional hand calculations were made for the tie rods, reinforcements for the ammonia piping connection, support plate brackets, and other areas.

Preliminary stress calculations were also made to compare the double tubesheet design described above with a design using a flexible member between the tubesheet and the shell. A spherical shell with truncated ends was loaded with internal pressure using a simplified ANSYS model. The radial deflection was about 0.33 in., and the axial deflection was about 0.06 in. It should be possible to design a flexible member to take these movements. However, it was found that the truncated shell acts like a cylinder which would require an additional two inches of shell wall thickness. This alternative was not pursued further after the reference design with the welded tubesheet-to-shell junction was shown to be feasible.

10-MW(e) MAPS. The structural analysis was made with a simplified ANSYS finite element model which was similar to the 50-MW(e) heat exchanger model previously described.

Ten analysis cases were run for the 10-MW(e) size heat exchangers. The model geometry for each case is described by Fig. E-1 and Table E-5 in Appendix D

and the results are tabulated in Table E-6. Cases 1-2 studied the effects of shell thickness and inner tubesheet inside diameter. Based on these results, the inner tubesheet was eliminated, and Case 4 proved this design to be acceptable. Cases 5 and 6 reduced the shell thickness, but both designs were unacceptable due to high stresses in the tubesheet and tubes. The thicker shell reduces the shell stresses and deflections which greatly reduce the discontinuity stresses at the junction ring. Cases 7-9 determined the effects of manufacturing tolerances on the fabrication of the shell to transition ring junction. Case 10 is for an evacuated heat exchanger which is subject to external pressure of the 20 ft column of water. Results are summarized as follows:

- o The shell and tubesheets are A516, Grade 70, carbon steel with a 3/16 in. thick titanium clad on the tubesheets. No credit was taken for the clad thickness in the stress calculations. A 2 in. thick shell and 1 in. thick steel base tubesheet were selected. A thinner shell produced excessive stresses in the outer perforated region of the tubesheet.
- o The inner tubesheet is not required, so the annular inner tubesheet and the stays were eliminated.
- o A sensitivity analysis of the connection between the shell and the transition ring showed the tubesheet stresses are unacceptable when the shell meets the ring inboard of the tubesheet. All stresses are acceptable when the shell is outboard of the tubesheet in at least the range of 0 to 2 in. axial (shell wall midpoint relative to tubesheet thickness midpoint). This allows at least a 1 in. tolerance for shop fit-up of shell-to-transition ring.
- o All stresses are acceptable for 43 psi external pressure (corresponding to 63 ft of external water pressure) and a vacuum in the ammonia space.
- o The above conclusions are valid for all regions except the shell-to-ring and ring-to-tubesheet junctions. These areas require more detailed finite element analyses which would be done in the detail design phase. Also, the final results will be affected by load inputs from the platform and the water boxes which are not defined for the preliminary design.

Additional calculations were made for the inlet and discharge piping reinforcement pads, piping thickness, pump pod, and structure for supporting the tube support plates. All calculations were based on static pressure and weight loads and did not account for platform movements which were not specified.

4.3.3.3 Dynamic Analysis. A tube vibration analysis was made to check for vortex shedding (Chen), turbulent buffeting (Owen), and fluid-elastic whirling (Conners) after the designs of the 50-MW heat exchangers were completed. Vapor velocity was chosen at 125 percent of the velocity between the tubes at the outer edge of the tube bundle assuming an average circular pitch. Damping of 1 percent was used in determining the magnification factor. It was also conservatively assumed that a tube could be scratched, and a stress concentration factor of 4 was used to calculate the tube stress. The calculations assumed a smooth tube of 0.1025 in. wall and did not take account of internal or external fins on the tubing.

50-MW(e) Power Module. The tube natural frequency is 22.6 cps, and the range of vortex shedding frequency is 18 to 55 cps at full load ammonia flowrate. Therefore, the 50-MW(e) heat exchangers will require 10 tube support plates rather than the 8 plates shown on the drawings in this report. Further analysis with the enhanced tube should not significantly affect the results unless the tube wall is appreciably thinner.

10-MW(e) MAPS. The tube natural frequency is 30 cps, and the range of vortex shedding frequency is 21 to 64 cps at full load ammonia flowrate. For a vapor velocity of 6.5 ft/sec, the allowable tube span is greater than the span used in this design. The vibratory stress at resonance is 85 percent of the endurance limit for an infinite number of cycles. Maximum tube amplitude is predicted to be 0.05 in.

4.3.3.4 Tubing Enhancement 50-MW(e) Power Module. The Conceptual Design task concluded that the aluminum circumferentially grooved tube fabricated by an embossing process or an integral rolled fin with a smooth internal bore is the most cost-effective tube. Seawater-side enhancement technique appeared to be not cost-effective due to the increased pumping power which results from the increase in friction factor. During the Preliminary Design task, inquiries were made to 31 tube manufacturers for large quantity of tubes which could be made by either the embossing or finning process. The external pattern was specified as being 25 to 30 fins per inch, 0.025 in. high, and applied to a 1.5 in. diameter 5052 aluminum tube with a 0.090 in. wall under the fin. An optional internal longitudinal fin was 0.018 in. high.

For the embossed tube, a flat strip of metal would be embossed with cross fins on one side and longitudinal fins on the other side in either one or two passes through a set of rollers. The embossed strip would be rolled and welded to form the tubing with circumferential fins on the outside and longitudinal fins on the other side in either one or two passes through a set of rollers. The embossed strip would be rolled and welded to form the tubing with circumferential fins on the outside and longitudinal fins on the inside. This fabrication method was explored with several vendors. Embossing is normally done by a coining process using a smooth bottom roll and a patterned top roll or by deep texturing using a male and female pattern on the top and bottom rolls. With the deep texturing, the metal is pushed through the thickness of the material. For enhancement running at right angles to each other, the metal would have to be pushed sideways. With the coining operation, the maximum attainable groove depth is 0.003 to 0.004 in. The cost would be about \$0.15/lb for aluminum in 18 in. widths. If new equipment could be developed to produce the deeper grooves, the embossing might be damaged during the tube rolling operations and the tube seam weld would have to contend with the varying wall thickness due to enhancement. This method does not appear to be feasible at this time, and no tube fabricator would quote on a tube.

At least two companies currently produce tubing with an internal longitudinal fin and a circumferential external fin. The tubing is extruded with the internal fins, and then the outside surface is upset to form the external fins. Wieland has made internally finned tube for M.A.N. to use with their brush cleaning system. They have made similar tubing in copper and believe it could be produced in 5052 aluminum for \$1.35/ft, but they are limited to 30.5 ft tube length. Wolverine presently makes 1.5 in. OD tube in 3003 aluminum with a higher internal and external fin and up to 11 fins/in. They estimate that our configuration could be made for about \$1.00/ft with a maximum length of 45 ft. They would require a development program to work out the production details.

Several companies indicated interest in making a tube with external finning only. The 1 1/2 in. diameter is larger than their current capabilities but should present no fabrication problems if larger manufacturing equipment is provided. No cost estimates were received for the external finning.

10-MW(e) Power Module inquiries were made for embossing or finning the titanium tube with the same configuration as the aluminum tubes. The external circumferential fins would be 0.025 in. high with 25 to 30 fins per in., and the internal axial fin would be 0.018 in. high. The minimum tube wall with the titanium would be 0.028 in.

High Performance Tube fabricates a low profile, circumferentially finned tube as a standard product. They can easily adapt to our design configuration. The base tube would be 0.049 in. wall, and smooth lands would be provided at the tubesheets and tube support plates. If the tubes were finned internally by some other process prior to external finning, the internal fins could be damaged by the mandrel while forming the external fins.

Wolverine can produce a titanium tube with 6 circumferential flutes/in. internally and a fine external fin. They have been producing circumferentially finned titanium tubes since early 1978. Other companies felt they could develop the capability with a minimum effort.

Although seamless titanium tubing is available, the cost is about twice the cost for welded tube, so welded tube has been selected. The finning process adds about 50 percent to the cost of the bare tube based on the thicker wall sections in the unfinned portion of the tube. The current price is about \$3/ft in the quantity required for the 10-MW(e) heat exchangers.

Subsequent to the above inquiries test results on the external circumferential fins at the LMSC Palo Alto laboratories indicates that the best fin configuration is 15 to 20 fins per in. with a height of 0.025 to 0.35 in.

4.3.4 Tube-to-Tubesheet Joining

The required parameters for end welding aluminum and titanium tubes were developed as part of the Task 1 work. Tensile tests were made with the weld only and with the weld combined with varying degrees of rolling.

The one inch OD x 0.065 in. wall SB-210 Grade 5052 aluminum tubes were welded in 0.75 in. thick type 5083 aluminum tubesheets. There were three 14-hole mockups which were welded with the tubes in the horizontal position. Two of these mockups were welded using a GTAW-ATW torch to deposit fusion and welds with no filler material. Testing of these welds was found to be acceptable to Section III of the ASME Code on X-ray, dye-penetrant and cross-section examination. The third 14-tube mockup was then welded for the tensile testing.

Four 14-hole titanium mockups were prepared with 1/8 in. thick titanium mechanically laminated to a one inch thick carbon steel plate. Two mockups were welded using one inch OD x 0.049 in. and 0.065 in. wall SB-338 Grade 2 titanium tubes welded to the SB-265 Grade 2 titanium tubesheets. X-ray examination of these welds revealed random microporosity (0.002 to 0.005 in.) at the root of some welds. Dye-penetrant examination of all welds was found acceptable with no indications and 20X magnification showed all leak paths in excess of 2/3T. The other two mockups were used for rolling and tensile testing.

From these tests it was concluded that either of these materials could be successfully welded using the end weld technique.

Additional development work will be done before the weld procedure is released to the shop for production welding. The welding development to date was done with a straight polarity dc which gives the maximum penetration. However, particularly for the aluminum, an ac arc will give a combination of penetration and cleaning action which would be beneficial. The problem with the ac is that the high frequency which is required to maintain the arc can also affect the system controls. An alternate method is a balanced wave dc which is used by Magnatech for example. FWEC visited Magnatech to review the equipment which they developed for Alcoa. We will continue to evaluate alternative methods.

Another potential tube-to-tubesheet weld is the internal bore weld which fuses the tube to the center of the tubesheet. This type of weld would permit the tube to extend beyond the front face of the tubesheet so that the cleaning brush cage could be fastened to the outside of the tube. With little extra cost, a second weld could be made in series for a redundant weld in the aluminum tubesheet. The thinner titanium clad would permit only a single weld. The disadvantages of this weld type are the crevice on the seawater side and the bump inside the tube from the weld. Foster Wheeler spent a few days to show that this type of weld is technically feasible, but we have not developed it to the same degree as the end weld.

The current position can be summarized as follows. If Lockheed determines that there is sufficient incentive to mount the brush cages on the external surface of the tube, the internal bore weld will be developed. The ball back position is to use the end weld and mount the brush cage internally. This may require an extra operation to provide an internal bead or the cages can be bonded in place. There is also a possibility that the tube material may change to another aluminum alloy.

Whichever weld procedure is eventually selected, it will be fully qualified prior to release for fabrication. The heat exchangers will be fabricated to Section VIII of the ASME Code, but the tube-to-tubesheet welds will meet all requirements of Sections III and IX. These include 2/3T minimum leak path, porosity level, push out tests, demonstration of weld reproducibility, etc. Also, a helium leak test will be used to validate the procedure.

Foster Wheeler's experience in fabricating tube-to-tubesheet joints in aluminum heat exchangers has been limited to the two vertical heat exchangers which were built for the Argonne OTEC test loop. These units were built to ASME Section VIII requirements and were stamped. Each heat exchanger has 240 tubes which are welded at both ends to tubesheets. The tube-to-tubesheet joint design is an end weld which was also chosen as the reference design on the Task 2 PSD-1 drawings. The Argonne units have 6063 tubes welded to a 5083 tubesheet. The development work in the PSD-1, Task 1 was done with the same equipment and techniques using a 5052 tube in a 5083 tubesheet. The limited PSD-1 development work produced an acceptable weld.

The development of the weld/procedures will follow the same successful methodology used to develop weld procedures for Inconel, Monel, Copper-Nickel and 304 Stainless Steel.

4.4 ROTATING MACHINERY

This section discusses the rotating machinery required for power module operation. A compilation of the related data is presented at the end of the section.

4.4.1 Seawater Pumps

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

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

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

The seawater pump performance requirements depend on the type of platform selected. The optimized installation requires a lower gross output power generator due to reduction in ammonia and warm seawater pumping power requirements. Pump heat loss requirements of 10 ft for the warm seawater subsystem and 15 ft for the cold seawater subsystem were a DOE design goal. For the power system modules, developed in this conceptual study, the total flow rates and heat requirements for the 50-MW(e) power module are as shown in Table 4-19. For the 10-MW(e) module, they are shown in Table 4-20. Selection of the number of pumps for each seawater subsystem requires evaluation of the economic aspects and installation effects.

Data provided by the Worthington Pump Corp., for a family of flow rates and head requirements, are shown in Fig. 4-22. The parametric data were developed

Table 4-19
50-MW(e) - PUMP FLOW RATE AND HEAD REQUIREMENTS

Item	Baseline Reference ΔP	Optimized Geometry
Warm Water Pumps		
Flow (CFS)	5,940	6,074.0
Head (FT)	10	9.8
Cold Water Pumps		
Flow (CFS)	5,134	5,788.0
Head (FT)	15	12.6

Table 4-20
10-MW(e) - PUMP FLOW RATE AND HEAD REQUIREMENTS

Item	Baseline Reference ΔP	Optimized Geometry
Warm Water Pumps		
Flow (CFS)	1,510	1,588.0
Head (FT)	10	6.4
Cold Water Pumps		
Flow (CFS)	1,312	1,430.0
Head (FT)	15	10.2

determining the sizes, weights, and power requirements for the complete pump and reduction gears. Cost data were developed using the following guidelines:

- The basic pump (inlet, impeller, case, and exit) costs \$7 per pound for material and fabrication.
- Reduction gears (VOITH data) cost \$11 per pound if greater than 2000 hp and \$19 per pound if less than or equal to 2000 hp.
- Submersible motors (Reliance Electric data) cost \$22.50 per rated hp.

Fig. 4-22 Parametric Cost Data

The increase in cost associated with the reduced head is the fact that pump speed is reduced and, therefore, a larger diameter pump is required to provide the flow rate. Four cost estimates were provided by the M.A.N. Corporation. Additional factors favoring the use of multiple pumps are the effect of pump failure on performance and the effect of pump size on the physical installation. Worthington and M.A.N. pump details are presented in Appendix F.

The off-design performance program was run at reduced seawater flow rates corresponding to the loss of a pump in one such system and in both the warm and cold water subsystems. The results, shown in Fig. 4-23, indicate that the loss of one pump in either the warm or cold subsystem has the same effect on performance. With four pumps installed, a pump loss in one subsystem results in a performance degradation of approximately 4 percent, and a pump loss in both subsystems results in a performance loss of approximately 9 percent. With three pumps installed, these performance losses are increased to 9 percent and 18 percent, respectively. In addition, the use of variable flow pumps can reduce these losses.

For the 50-MW(e) module, the physical installation of a single pump would require an impeller 22 to 24 ft in diameter with a related overall pump assembly length of 60 ft. A two-pump installation would reduce this length to 42.5 ft, but this side-by-side pump arrangement would require a long transition section to the heat exchanger face. The maximum diameter of the pump sized for the four-pump installation allows the pumps to be installed with the diameter of the heat exchanger front face. The overall diameter of the three-pump installation is greater than the heat exchanger front face diameter. In addition, a longer transition section is required to provide good flow distribution at the heat exchanger. The four-pump installation is, therefore, recommended for the 50-MW(e) power system module.

For the 10-MW(e) module, the same rationale as for the 50-MW(e) module is applied. For the MAPS, four pumps are incorporated into a pump module located at the exit plane upstream from the heat exchanger tubesheet at a distance equal to one-half of the tubesheet diameter. The pumps, which are sized for "optimum geometry," have 78 in. diameter impellers and are 209 in. overall length.

CONSTANT FLOW PUMPS

Fig. 4-23 Operational Capability With Seawater Pump Loss
Constant Flow Pumps

Variable Seawater Flow. The results of off-design performance studies, completed under Tank 1, indicated that, in addition to flow reversal required for the M.A.N. cleaning brush operation, the capability of regulating the seawater flow is desirable under several circumstances. In particular, if the seasonal ΔT variation is greater than $\pm 4^\circ \text{F}$ from the design ΔT or if it is required to maintain power module self-sufficiency at ΔT 's below 26°F , then optimizing the seawater flow rates is required.

A feasibility study was initiated on various methods of attaining flow variation. These included:

- Constant speed motor/constant pitch impeller with throttling valve
- Constant speed motor/variable pitch impeller
- Constant speed motor/variable speed couplings
- Variable speed motor/constant pitch impeller
- Ammonia turbine drive/constant pitch impeller

The first method was discarded because it requires constant input power and, in fact, may increase the power required depending on the pump characteristics. The second, third, and fourth approaches have been used in industry and are, therefore, considered feasible. The feasibility of the fifth method, the ammonia turbine-driven pumps, was determined using the following approach: from the results of the off-design, optimized flow computer program, the variation in seawater flow rate is as shown in Fig. 4-24 and the related seawater pump power variation is as shown in Fig. 4-25. The available power is a function of the work available per unit mass flow and the mass flow rate. The former depends on the isentropic enthalpy available across the turbine as shown in Fig. 4-26. The latter is a function of the pressure and temperature at the turbine inlet. The pressure drop from the evaporator outlet to the condenser inlet is small and, for this evaluation, the available pressures are assumed to be as shown in Fig. 4-27. The inlet temperature is based on the 25-50-25 percent split across the heat exchangers and turbine.

Fig. 4-24 Seawater Flow Variation

Fig. 4-25 Seawater Pump Power Variation

The effect of a change in warm seawater temperature which reduced the T from 40°F to 30°F is evaluated as follows:

$$\text{Case 1 } T_{\text{warm}} = 80^{\circ}\text{F} \quad T_{\text{cold}} = 40^{\circ}\text{F} \therefore T = 40^{\circ}\text{F}$$

$$\text{Case 2 } T_{\text{warm}} = 70^{\circ}\text{F} \quad T_{\text{cold}} = 40^{\circ}\text{F} \therefore T = 30^{\circ}\text{F}$$

From Fig. 4-25 the seawater pump power required ratio is 0.68. The unit flow work ratio is determined from Fig. 4.4-5 to be:

$$\frac{\Delta H_2}{\Delta H_1} = \frac{15.1}{19.65} = 0.768$$

The available flow is proportional to the pressure and inversely proportional to the square root of the absolute temperatures. The pressures are obtained from Fig. 4-27 and the temperature as 70°F for case 1 and 62.5°F for case 2. The available flow ratio is, then, represented as follows:

$$\frac{W_2}{W_1} = \frac{113}{128.8} \sqrt{\frac{530}{522.5}} = 0.884$$

PSD 253

Fig. 4-26 Isentropic Turbine ΔH

PSD 254

Fig. 4-27 Heat Exchanger Operating Pressures (psia)

The available power ratio is, then,

$$\frac{SHP_2}{SHP_1} = 0.768 \times 0.884 = 0.68$$

which is equal to the required power ratio.

This holds true if the turbine efficiency is a function of the ratio of rotational speed to flow velocity. The latter, in turn, is proportional to the square root of the isentropic enthalpy head. From Fig. 4-24 it is seen that the flow ratio and, hence, the pump speed ratio is 0.85. The flow velocity ratio is as follows:

$$\frac{V_2}{V_1} = \sqrt{\frac{15.1}{19.65}} = 0.877$$

Therefore, the pump efficiency will be only slightly degraded. This can be counteracted by sizing the turbine for the most extreme off-design condition and providing inlet controls to maintain the desired pump speed at other conditions.

Ammonia turbine pump drive arrangements and cost-performance trade studies were not completed to a depth sufficient to make any recommendations. A cursory examination of the alternatives, however, provided the following:

- Turbine and gear reduction units can be packaged within the horizontal shaft pump pod configurations.
- Ammonia pressure loss from the evaporator to the turbine and from the turbine to the condenser is extremely dependent upon the power module platform configuration and may be sufficiently high to eliminate the use of ammonia turbines for driving seawater pumps.
- Variable speed electric motors of the voltage control, frequency control, or dc type together with necessary conversion and control equipment appear to be higher in cost and lower in reliability than the variable pitch impeller.

- Variable speed couplings, eddy current or fluid, have low efficiency at pump rpms below the design point and tentatively introduce heat problems in the horizontal shaft pod arrangement.

4.4.2 Radial Ammonia Turbines

The radial inflow turbines have been designed with appropriate generators for both the 50-MW(e) CPM and the 10-MW(e) MAPS. A turbine specification is presented in Appendix B and supporting data in Appendix F.

4.4.2.1 The 50-MW(e) net or 70-MW(e) gross machine is made up of four turbine casings, and two 35-MW(e) generators. The turbine/generator arrangement is shown in Fig. 4-28 and installed in a segmented module in Fig. 4-29.

The generator weight is 196,000 lb and has an envelope 250 in. long, 158 in. high, and 133 in. wide. The mounting feet add approximately 20 in. to the width.

The individual turbine casings are estimated to weigh between 65,000 and 75,000 lb, and are approximately 124 in. overall length and 96 in. in diameter. The flange sizes are 60 in. at inlet and 72 in. at discharge.

The analysis and design for sizing the 70-MW(e) module is standard and the only requirement is torsional analysis where an expander casing is placed at either end of the generator. The latter configuration is one that has been developed for combined cycle plants and the alternator exciter is proportioned to allow a shaft to extend through the other end of the machine.

A company proprietary computer program (Rotoflow) has been developed to analyze this type of torsional vibration. The prime purpose of any calculations at this stage is to demonstrate that by adjustment of stiffness, the criticals can be chosen below and above the running speed leaving a window for safe running between the first and second torsionals.

4-83

Fig. 4-28 Rotoflow Turbine/Generator, 35 MW(e)

LMSC-D630248

Fig. 4-29 Turbine Location in a Segmented Module

4.4.2.2 The 10-MW(e) net or 14-MW(e) gross machine is a 100R radial inflow turbine with two back-to-back wheels in a single casing, connected to a 17,500 KVA alternator. Specifications for the alternator are shown in Appendix F. The generator is an electric machinery frame 6571, which weighs 118,000 lb and is 230 in. long, 114 in. high, and 133 in. wide. The turbine casing weighs 65,000 lb and with the main oil pump is 194 in. long, 128 in. wide, and 136 in. high including flanges. The casing width is 96 in. and is 134 in. at the greatest length. The shaft overall length is 124 in.

The turbine design point has been chosen using saturated ammonia conditions at inlet and exit at 69.5°F and 50.13°F , respectively. The 14-MW(e) gross unit, designed to yield a net 10-MW(e), has been designed around a 3600 rpm, 43 in. diameter turbine with two wheels back-to-back in a single casing. The expander wheel produces about 9,700 hp and the wheel efficiency is guaranteed to be 88 percent or better. From this must be subtracted bearing and generator losses to arrive at the electrical power generated. The bearing losses are about 80 hp and the generator efficiency is 96.9 percent at full load.

The total ammonia mass flow is 2.9 million lb/hr, the inlet pressure is 127.6 psia and the discharge 89.4 psia. The discharge gas has approximately 2.5 percent condensation liquid which has no effect on the radial inflow blading. This latter is true because the force vectors on a liquid droplet lie parallel to the blade of the wheel, and, therefore, do not impinge on the blading.

The design conditions have been used to determine the shape of the wheel, nozzle opening, and flange connections. Off-design performance is measured by considering the ratio of actual to design enthalpy drop across the complete turbine nozzle assembly, and also the ratio of design to actual discharge volume ratio. A short description of how this is done is shown in Appendix F.

Detail design methods for optimization of the expander are based on considerable expertise which is company proprietary. However, the expander wheel is approximately 43 in. in diameter with a discharge diameter of 35 in. The

wheel has 24 blades and there are 25 nozzles. The latter have relatively low inertia and can therefore be rapidly closed to reduce power in any emergency condition such as a full-load rejection. The blades on the wheel are approximately 1 in. thick at the base tapering off to 0.3 in. at the blade tip. Tip speed is 675 ft/sec. The vapor inlet flange diameter is 60 in. and the vapor discharge diameter is 72 in. The turbine is provided with a seal system to minimize any escape of ammonia from the main Rankine cycle loop. Lubrication oil such as Mobil Vectra is ammonia compatible and no copper alloys will come into contact with either oil or ammonia. Lubrication system and controls are based on standard API practices and other codes developed in over 700 installations for radial inflow turbines. Each generator turbine set will have its own lube skid and controls, so that independency is achieved in the event of shutdown of one of the turbines, and no oil system cross contamination can occur.

Construction codes will normally be specified by the customer; however, Rotoflow machinery is generally supplied to meet the following codes and practices:

- ASME for welding and pressure vessels
- CSA and NEC electrical codes
- Rotoflow Standards (similar to APO with some exceptions)
- TEMA for filter requirements
- OSHA for safety and noise

Stress and Dynamic Analysis Rotating Section. The relatively low tip speed (670 ft/sec) leads to low wheel stresses and little difficulty with the shaft critical speeds. It is simple to detail design the blading so that its interaction with the nozzle valves does not excite heavy blade resonance.

To establish design suitability, the following design areas defined:

- Shaft critical, and in this case, the stiff shaft concept is used (i.e., one that always runs below the first critical speed).
- Bearing criticals, where the bearings are considered in relation to the mass of the shaft. Design speed is below the first critical, and

also avoids half-speed gyration which occurs at the second bearing critical.

- The stress distribution pattern was calculated so that good cycle fatigue life is established. Blade stresses are calculated to establish optimum blade taper and lean, again for adequate cycle fatigue life. The blade taper ratio is also used to ensure that blade vibration is not excited by nozzle wake blade interference.

Shaft Critical. A Rotoflow developed, proprietary-computer program has been used to analyze the shaft lateral critical speed. The shaft critical comes to 12,793 rpm with a design running speed of 3,600 rpm. The computer program shows forward and backward gyroscopic precession frequencies and half-speed precessions. These are well damped. A readout is attached in Appendix F showing frequencies and deflections.

The shaft critical being over the running speed, there is no difficulty with half-speed whirl as this occurs at roughly twice the first bearing critical, and the short shaft construction prevents rotor sag problems and the necessity to bar the unit over after shutdown. The coupling between the turbine and the generator is a Bendix diaphragm unit which has no lubrication requirements, and which can absorb the axial loads from the generator motor in high sea states. The low mass of the coupling results in freedom from criticals due to the coupling.

Torsional Oscillations. The wheel weight is 1,180 lb, while the whole rotor is 4,880 lb (see Appendix F). The weight of the generator motor is about 6,477 lb. Essentially, the generator is a massive component with the turbine simply oscillating with the shaft. At 2,671 the calculation is shown in Appendix F.

When adding additional turbines, i.e., to the other end of the generator, a Rotoflow-developed, proprietary-computer program is available to solve for multiple mass and stiffness inputs, up to ten of each.

Wheel Stress. Maximum stress is of the order of 9,000 psi which will result in a minimum start-stop fatigue life of 20,000 cycles. The calculation allows for an 8-to-1 scatter factor and a yield strength of 35,000 psi which should be readily achieved.

Maximum blade stress is calculated at 4,851 psi which includes centrifugal and gas bending loads. This would permit at least 10^7 start-stop cycles based on available fatigue data for KO-1 (Alloy 201). Alloy A355 is also considered and can be cast in these larger sizes with similar tensile properties to KO-1. At these stress levels, a creep life of 10^9 hours is estimated.

Coupling. The coupling has to handle the normal design torque loads, the maximum short circuit torque loads, misalignment deflections, and axial loads from the alternator. With the 14.5 MW(e) alternator, the motor weighs 18,200 lb and, with some sea state condition and 0.5 g axial load, this can impose 9,100 lb on the coupling and the thrust bearings.

The thrust bearings can take up to 20,000 lb for the 9-in. bearing selected which is adequate. The coupling, then, must be sufficiently stiff so that the motor will not be continually hitting the bump stops provided in the alternator, but not so stiff that extreme axial loads might damage the bearings before the alternator motor hits the bump stops.

A Bendix G7E322 or Zurn Ameriflex size 19 couplings are possible candidates; the final choice will be selected at the final design stage. Both are well proven units; the Bendix unit is lighter, which may be necessary for vibration prevention. Both of these are diaphragm-type units with no lubrication requirements and both have been in service for many years. The Bendix units have been used in installations at 3600 rpm and up to 100,000 hp.

General. Lists of long lead-time items, spares recommendations, and support equipment, plus a brief system item description producibility analysis are given in Appendix F. Also in Appendix F are nonrecurring costs at \$250,000 for

the 14-MW(e) machine and specifications for the alternator. A summary of pertinent 10-MW(e) MAPS turbine data is as presented in Tables 4-21 and 4-22.

Shaft Seals. The turbine incorporates labyrinth seals provided with pressure regulation to maintain seal pressure drop at minimum value, regardless of back-pressure variation on the turbine wheel.

The design specifies a labyrinth seal system with the following characteristics:

- The seal is backed with nitrogen and a small amount of ammonia vapor is allowed to lead out and mix with the nitrogen (to preclude N_2 leakage in the condenser), and the mixture is led out via a flow control valve to a recovery system.
- The recovery system is a 5-hp refrigeration unit at $0^{\circ}F$ that chills the gas, liquefies the ammonia and returns it to the closed loop, and returns the residue N_2 to the seal gas loop.

The nitrogen also leaks from the labyrinth seal back toward the bearing, which acts as a high-pressure seal, and the high-pressure oil and nitrogen are drained via a pressurized draining system. The oil drains back to the tank via a Cochrane valve and the gas is vented from the top of the drainer. The vented gas is cleaned up with a mist eliminator and returned to the nitrogen seal gas loop. The end of the shaft is at atmospheric pressure and can be simply sealed with a 2-psi air seal system.

The nitrogen flow rate for each of the seals involved is approximately 156 scfm. This is calculated as the maximum rate and is used for sizing the supply. In practice, the use of multiple labyrinth steps will reduce the leakage by 1/3 to 1/2 of this. The ammonia flow rate is approximately 200 scfm. As with the nitrogen, this represents a maximum flow rate for sizing equipment. The actual quantity will be about 1/3 to 1/2 of this. When mixed with the nitrogen, the mixture will be approximately 45 percent nitrogen/55 percent ammonia, with a flow rate of approximately 356 scfm to handle.

Table 4-21 (B 10)

ROTATING MACHINERY PER BASELINE MODULE OF 10-MW(e)
MAPS

Item	Number	Type	RPM	Horsepower	Efficiency	Comments
Turbine	1	Radial Inflow	3,600	-	0.88	17.5 MW capability 14.4 MW
Generator	1	AC	3,600	-	0.985	
Warm Water Pump	4	Axial	150 10%	-	0.86	
Warm Water Pump Motor	4	Submerged	1160 $\pm$ 10%	565	0.90	
Cold Water Pump	4	Axial	150 $\pm$ 10%	-	0.86	
Cold Water Pump Motor	4	Submerged	1160 $\pm$ 10%	725	0.90	
Ammonia Feed Pump	3	Centrifugal	-	-	0.90	
Ammonia Feed Pump Motor	3	Var. Speed	1160	233	0.90	
Ammonia Distribution Pump	4	Centrifugal	-	-	0.90	
Ammonia Distribution Pump Motor	4	Var. Speed	1160	72	0.90	

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Table 4-22 (C10)
FLOW RATES AND CONDITIONS AT DESIGN POINT OPERATION
(MAPS)

Item	Reference Baseline ΔP	Optimized Geometry
<u>TURBINES</u>		
Flow Rate (lb/hr)	2.97×10^6	2.74×10^6
Inlet		
Pressure (psia)	127.88	127.66
Temperature (°F)	69.6	69.5
Velocity (ft/sec)	150	150
Quality (%)	99	99
Outlet		
Pressure (psia)	89.62	89.44
Temperature (°F)	50.25	50.15
Velocity (ft/sec)	75	75
Quality (%)	96.6	96.6
<u>PUMPS</u>		
Ammonia Feed		
Flow Rate (lb/hr) each	1.48×10^6	1.37×10^6
Pressure Rise (psi)	53	54
NPSH (psi)	5	5
Ammonia Distribution		
Flow Rate (lb/hr) each	1.48×10^6	1.37×10^6
Pressure Rise (psi)	15	15
NPSH (psi)	5	5
Warm Seawater		
Flow Rate (gpm) each	169,400	178,200
Pressure Rise (ft)	10	6.35
NPSH (ft)	12	8
Cold Seawater		
Flow Rate (gpm) each	147,250	160,400
Pressure Rise (ft)	15	10.17
NPSH (ft)	18	12

4.4.3 Axial Flow Turbine - Generator

The 50-MW(e) turbine was designed as two units each with a single stage (axial) turbine with the generator rotor on the same shaft (Fig. 4-30). The generator rotor is placed between the radial hydrodynamic oil lubricated bearings, while the exciter and the turbine are overhung on opposite ends. Between the radial bearings and the turbine are the dual-thrust bearing and the oil/ammonia seal. The 50-MW(e) axial turbine generator package can be arranged vertically or horizontally, depending on the development progress of the vertical generator. The vertical arrangement is the standard design for all hydroelectric power plants. The estimated weight of the turbine-generator is 200,000 lb.

Maintainability of the turbine-generator is assured by an access clam shell type of door arrangement downstream of the turbine, which allows inspection and/or removal of the turbine disks or blades and maintenance of the seal. The thrust bearing is also accessible since it is mounted in a horizontal split-bearing carrier. The reliability of the turbine-generator is in accordance with the OTEC specifications.

The efficiency of the 50-MW(e) is $\eta = 0.90$. Off-design trends are presented in Appendix F which offers more detailed preliminary design data. An ammonia-compatible generator, available for the 50-MW(e) turbine generator system, results in much simpler sealing requirements. The selection of ammonia compatible oil together with a reliable mechanical face type seal allows a small amount of ammonia leakage into the lube oil system, where they are separated and returned to the OTEC plant. The bearings are hydrodynamic journal bearings and pivot shoe hydrodynamic thrust bearings.

The 10-MW(e) single stage overhung axial flow reaction turbine is mounted on the two pole generator shaft. Figure 4-31 shows both overall dimensions, and the general arrangement, i.e., a horizontal rotor generator with integral self-excited field. The rotor is supported on journal-type, oil-lubricated radial bearings. Axial thrust forces due to sea-induced inertia loads and gas

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Fig. 4-30 AIRCO Turbine-Generator, 50 MW(e)

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Fig. 4-31 Axial Turbine Generator

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reaction pressures on the turbine are resisted by an oil-lubricated tilting pad thrust bearing located between the generator and the turbine assembly. Turbine working fluid (ammonia vapor) is prevented from leaking into the oil system and the atmosphere by a shaft seal tailored for this application. Oil-to-air shaft seals prevent loss of oil to the atmosphere on the generator side of the thrust bearing and at the radial bearings. The turbine includes an inlet scroll with nozzle vanes adjacent to the generator and a long exit diffuser duct with centerbody. The exit diffuser enhances the turbine overall efficiency by recovery of velocity energy from the gases leaving the turbine blading. A thrust balance system is provided on the downstream side of the turbine disk.

The basic design of the turbine-generator provides for ease of maintenance. The turbine exit diffuser has a short spool which provides access to remove and replace turbine disks. With the rotor removed, the turbine nozzle vanes are readily accessible. The thrust bearing and sealing assembly are accessible by removing half of the clamshell structure between the generator and the turbine inlet scroll.

The generator field excitation unit is overhung and accessible at the opposite end of the generator. Generator radial bearings are split horizontally. The upper half is easily removable for shaft journal and bearing inspection.

The high inherent reliability of the turbine-generator is due to the conservative temperatures and pressure associated with the OTEC solar energy scheme. The generator size, speed, and power ratings are based on many years of operating experience with these machines. Elimination of copper bearing materials known to be sensitive to ammonia will further enhance the reliability of the turbine-generator.

The single stage overhung turbine gives the maximum attainable performance. Readily replaceable blading enhances turbine performance as well as ease of maintenance. The straight high efficiency exit diffuser also enhances turbine performance. Mounting the turbine rotor directly to the generator shaft

reduces the number of bearings and shaft seals to a minimum. Flexible couplings between the turbine and generator are eliminated. The generator selection is based upon the availability of generator types. Experience in the industry has defined the configuration of the generator.

The inherent simplicity and reliability of the two-radial-bearing rotor systems have been avoided by industry because the analytical tools for predicting the shaft dynamics have not been adequate. As a result, the turbine manufacturer and the generator manufacturer have each provided radial bearings. An interconnecting shaft with flexible couplings provided the interface. In the case of a multiple stage steam turbine, it would not be feasible to overhang the turbine on the generator rotor. However, for the OTEC turbine-generator, a single stage axial flow reaction ammonia turbine is ideal. AIRCO has extensive background experience integrating compressors and turbines on a two-radial-bearing shaft. Appendix F presents more detailed data for the axial flow turbine.

4.4.4 Turbine Auxiliaries.

The turbine auxiliary subsystems include the turbine control subsystem and the lube-oil subsystem. The control logic is the same for both types of turbines. Actual turbine speed, sensed by a permanent magnet pickup, is compared with a speed reference and an appropriate signal is used to operate the inlet control. For the axial flow turbine, the inlet control consists of a large control valve located just upstream of the turbine inlet flange. For speed regulation, the valve is modulated as required. The inertia of this valve is not significantly larger than the inertia of variable stator blades which could be used for shutting down the turbine. However, the complexity of this arrangement is not warranted for the present design since the performance is nearly optimum over the range of operation and very little improvement can be achieved by varying the turbine guide vanes for the selected reaction turbine configuration.

The multiple nozzle system for the radial inflow turbine has little inertia and can be closed very rapidly. A typical time history of a shutdown, due to loss of load, is shown in Fig. 4-32. An acceleration sensor opens the nozzle actuator vent at 0.025 sec after load loss. The system reaction starts to close the nozzles at 0.1 sec with complete closure at 0.3 sec. The

O-534

Fig. 4-32 Shutdown Capability - Radial Turbine

overspeed is limited to approximately 17 percent, which is well within the overspeed safety limit for the turbine and generator.

For the case in which the power system is synchronized with another system, a torque sensor is added to the control loop. Changes in power demand are immediately sensed as changes in generator torque, and the inlet vanes are adjusted to preclude any variations in rotational speed.

For the axial configuration, the lube-oil subsystem is part of the generator lubrication system with the addition of a seal-oil subsystem. The seal-oil

subsystem consists of a seal-oil pump and a pressure controller which ensures that the seal-oil pressure is sufficiently high to lubricate the mechanical seal between the bearings and the ammonia atmosphere. Leakage of seal oil through the mechanical seal will be at a pressure somewhat lower than the ammonia leakage flow, which are combined and ducted to the cleanup system.

For the radial turbine, the lubrication subsystem is separately mounted. The lube console is arranged so that component changes, particularly filter changes, can be made without shutting down the generator. An additional lube pump mounted on the expander shaft provides the lube oil while the turbine is running and, in addition, provides oil during coast-down.

The oil reservoir is incorporated into the base of the lube console and incorporates the following features:

- Sixteen-minute residence time to allow adequate de-aeration
- Heaters to maintain minimum oil temperature under the worst conditions
- Bottom of reservoir slopes to permit cleaning and water removal
- Baffles to allow adequate settling prior to the pump suction

The oil pumps used are screw type, manufactured by DeLaval, which have proved to be very reliable and quiet with good suction capability.

Filters are 5-micron Hilliard-Hilco and will be arranged with delta pressure sensing to check for blockage. Elements can be changed while the unit is running. Normal pressure drop is 5 psi.

Greer accumulators provide an emergency reserve in case the oil pressure drops rapidly below some predetermined level. The accumulators also provide additional oil for the high-speed actuators under emergency conditions. Accumulator capacity is sufficient to allow for coastdown to a point where the pressurized oil flow is not required for cooling the bearings.

High-pressure drainers are designed to maintain a backpressure on the seal system so that nitrogen leakage is minimized and, at the same time, provide

adequate drainage for the oil. The drainer will handle anything from a trickle to a full oil flow via a nonclogging, multiple port shear valve. The level at which the unit determines backpressure is adjusted by the drainer vent system. The pressure can be made a function of the wheel discharge pressure to minimize nitrogen use and/or burden on a nitrogen recovery system.

The pressure in the seal system is controlled by sensing wheel discharge pressure. Discharge pressure is used as a reference for the ammonia and seal gas vent and also for the nitrogen supply pressure. With this pressure reference system, the leakage of gas from the closed loop system can be minimized. That gas which does escape is easily recovered in a refrigeration system.

4.4.5 Generators

The 10-MW(e) generator for either turbine configuration is a commercially available, two-pole (3,600 rpm), 60 Hz, brushless horizontal rotor, totally enclosed, air-water cooled unit. The air in the generator housing rejects heat to a water cooled heat exchanger inside the generator shell. The unit requires 400 gpm of water on a 95°F day. The rotor-stator windings reject only 2 percent of the shaft power as electrical losses which can be returned to the system via an ammonia/seawater heat exchanger.

The generator rotor is machined from a solid alloy steel forging. Longitudinal slots are filled with copper (or aluminum) bars that are conductors for the rotating two-pole electromagnet. The direct current for the rotor windings is generated by a small brushless alternator with rotating solid state rectifiers. The alternator is overhung from one end of the generator shaft. Field excitation (stator) for the brushless alternator controls the output voltage of the generator, and power for alternator excitation may be generated by a very small permanent magnet generator outboard of the alternator. Normally, an external power source is utilized (2 kW); however, for the OTEC system, the completely self-contained generator is preferred.

The generator rotor is mounted on two horizontally split self-aligning bab-bitted journal bearings which are oil lubricated. The bearing oil is prevented from flowing along the shaft by slingers machined in the shaft and gravity drains in the bearing housings alongside the bearing.

Cooling air circulation for the rotor and stator is provided by two large axial flow fans one at each end of the rotor. This air flow is circulated through the rotor air gap and the stator. Heat is rejected from the air flow to a tube heat exchanger mounted within the upper portion of the generator shell. The enclosed generator with heat rejection to a liquid coolant reduces the noise intensity emitted by the generator and minimizes the possibility of corrosion from the environment.

The generator armature is in the stator. The stator laminations are electrical-grade silicon steel. The cylindrical stack of laminations is contained in a series of steel clamping rings that support the radial magnetically induced rotating forces. A very rigid structure is required to minimize the 120-Hz hum.

The armature coils are preformed, insulated, and inserted in their slots. Copper is normally used for generator windings; however, consideration will be given for aluminum conductors. The methods for forming, insulating, impregnating, protecting, inserting, and containing aluminum wire coils are the same as for copper. Slightly larger diameter conductors are required for aluminum due to its lower electrical conductivity. Table 4-23 summarizes the generator characteristics. For ammonia pump data see the applicable paragraphs in section 4.5.

Table 4.23
GENERATOR CHARACTERISTICS

Voltage	13,800 V line-to-line
Speed	3,600 rpm
Phase	3
Frequency	60 Hz
Power Factor	0.8
Efficiency	0.98
Rating	14,400 kW
Voltage Suppression	15%
Excitation	Brushless permanent magnet (no external power required)

4.5 AUXILIARY SYSTEMS

This section of the report describes various auxiliary systems required for the operation of the power module. It comprises four separate subsections dealing, respectively, with ammonia cycle subsystems, operational support subsystems, ballast control system, and biofouling control system. A more detailed discussion of the various ammonia cycle and operational support systems is to be found in Appendixes G and H.

4.5.1 Ammonia Cycle Systems

The ammonia cycle systems described below are required to support the operation and maintenance activities of the power system. The functions served by these systems include ammonia transport, ammonia storage and fill, noncondensable gas removal, ammonia cleanup, and ammonia unloading. These systems of the 10-MW(e) modular application are scaled from the 50-MW(e) module conceptual design with the same time duration target retained for each filling, unloading, and noncondensable gas removal operation. In a multimodule commercial installation, most of the equipment of the ammonia filling and unloading subsystems could be shared by several modules attached to a single platform. Therefore for the 50-MW(e) module, they are considered part of the platform facilities. However, to make the 10-MW(e) modular application self-sufficient, preliminary designs of the complete subsystems and all equipment to operate these subsystems have been specified. All equipment for the ammonia cycle systems is commercially available with no development work required.

4.5.1.1 Ammonia Vapor and Condensate Subsystem. The functions of this subsystem are:

- o To convey ammonia condensate from the condenser to the evaporator sump
- o To convey liquid ammonia from the evaporator sump to the evaporator spray headers
- o To convey ammonia vapor from the evaporator to the turbine

The Ammonia Vapor and Condensate Subsystem (AVCS) preliminary design for the 10-MW(e) MAPS was derived from and is analogous to the conceptual design prepared for a 50-MW(e) CPM. The AVCS is a functionally integral part of the OTEC and the 10-MW(e) power cycle and, it consists of pumps, piping and associated equipment used to convey ammonia liquid from the condenser to the evaporator and ammonia vapor to the turbine. Descriptions of the evaporator, condenser, and turbine are presented in Sections 4.3.1, 4.3.7 and 4.4 (4.4.2 and 4.4.3), respectively. The condensate pumps, distribution pumps, and ammonia piping designs are described in the following paragraphs.

As shown in the subsystem flow diagrams (Fig. 4-33 and 4-34), the condensate pumps transfer ammonia liquid from the condenser sump to the evaporator sump. Three 50 percent capacity pumps are used, two of which are normally in operation at any given time. The third pump, held in standby readiness, automatically begins operation whenever an operating pump fails. The pumps are conservatively specified to give durable long-life service, and the redundant pump provides a very reliable pumping capability.

Both vertical and horizontal centrifugal pumps were considered for the condensate and distribution pumping applications. Vertical Centrifugal pumps were chosen for the following reasons:

- o Vertical pumps with the correct flow-head relationships are more readily available from potential pump suppliers
- o Vertical pumps require only one shaft seal
- o A more compact arrangement is possible with the vertical pumps, resulting in smaller diameter machinery rooms
- o The required vertical distance below the heat exchangers is practically the same for the two pump types since it is determined mainly by the net positive suction head required to pump ammonia liquid near saturation conditions
- o Vertical centrifugal pumps of the same design are available for a wide range of pumping head, requiring only additional stages to achieve higher developed head. This offers the potential for interchangeability of parts between the condensate and distribution pumps

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Fig. 4-33 Ammonia Vapor and Condensate System, 50-MW(e)

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Fig. 4-34 Ammonia Vapor and Condensate Systems, 10-MW(e)

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A three-stage vertical centrifugal design for the condensate pumps was selected to attain the required pump outlet head and flow capacity. Electric induction motors were selected as the pump drives, based primarily on their proven reliability in similar service. This type of pump drive is adaptable to variable speed operation through the use of a variable frequency power supply. The vertical centrifugal pumps selected for the ammonia condensate pumps are adaptable to a compact machinery room arrangement, as shown in Figs. 4-35 and 4-36 for the 50- and 10-MW(e) modules, respectively.

The condensate flow demand varies over a considerable range, depending on the energy available from variable difference between the warm and cold seawater temperatures. Three methods of flow rate control to permit operation over the expected range of seawater temperature differences were considered:

- o Throttling control, where the pumps are driven by constant speed electric motors and the liquid flow rate is varied by throttling the pump discharge
- o Variable speed control, where the pump motor speed is varied using a variable frequency power supply to meet different flow rate demands over the full range of flows
- o Mixed mode control, where throttling control is used near maximum flow demand and variable speed control is applied for lower flow rates

The mixed mode control appears to be attractive when compared to pure throttling or full range variable speed control because, at higher flows, the throttling losses are smaller than the 15 to 20 percent losses incurred with the frequency control equipment. Conversely, at lower flows the throttling losses exceed the losses of the frequency controlled alternative.

An economic analysis of the benefits and costs was conducted for both a 50-MW(e) commercial power module and a 10-MW(e) modular application. This analysis is described in more detail in Appendix G of this report. In the

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Fig. 4-35 Machinery Room Arrangement, Condensate Pumps, 50-MW(e)

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Fig. 4-36 Machinery Room Arrangement, Condensate Pumps, 10 MW(e)

analysis, the benefit ratio (defined as the present value of electrical energy saved over a 30 year plant life divided by the present value of the capital investment for additional equipment) was determined for the variable speed and mixed mode flow controls. The results, given in Table 4-24, show that mixed mode flow control is economically the best of the three approaches for both the 50-MW(e) and the 10-MW(e) modules with benefit ratios of 3.20 and 2.99, respectively. Therefore, mixed mode flow control was adopted for controlling ammonia liquid flow rates in the condensate and distribution lines.

Table 4-24

COMPARISON OF FLOW CONTROL APPROACHES

Flow Control Approach	Benefit Ratio	
	50-MW(e)	10-MW(e)
Throttling	Base Case	Base Case
Variable Speed	2.41	2.12
Mixed Mode	3.20	2.99

An alternative variable speed drive, based on the use of wound rotor motors with variable secondary resistance, was considered but not economically evaluated. Although this system is more efficient than the variable frequency power supply, wound rotor motors are generally much larger than induction motors and also require cumbersome explosion proofing.

A second alternative for varying pump speed, to be evaluated in Phase II, is the use of hydraulic motor drives for each pump. These motors would be powered by a single variable displacement hydraulic pump, driven either from the main turbine generator shaft or by a separate electric motor. This system may prove to be both more efficient and less expensive than other flow control methods considered so far.

The condensate pump design requirements based on the most severe service conditions are shown in Table 4-25. The available net positive suction head (NPSH) is based on the elevation of the condenser sump liquid surface above the suction impeller, with allowances for suction line pressure loss and a 0.15 g vertical acceleration for operation at sea-state six.

Table 4-25

CONDENSATE PUMP DESIGN REQUIREMENTS

	50-MW(e)	10-MW(e)
Head	215 ft	241 ft
Capacity	26,200 gpm	5240 gpm
NPSH (available)	33.5 ft	18.7 ft
Fluid Temperature	50 F	50 F
Fluid	Ammonia	Ammonia
Number Required	3	3
Pump hp (ea)	1,046	233

The distribution pumps deliver liquid ammonia from the evaporator sump to the evaporator spray nozzles which irrigate the evaporator tubes. Four 33-percent-capacity pumps are selected for this service; three pumps operating, the other on standby. The standby pump starts automatically when one of the operating pumps fails. The pumps are driven by induction motors. The arrangement of the machinery room housing the distribution pumps is shown in Fig. 4-37 and 4-38 for 50-MW(e) and 10-MW(e), respectively.

Since the distribution flow is nominally 50 percent higher than the condensate flow, the flow per operating pump is the same in both the condensate and distribution streams. This design criterion led to selection of a single-stage version of the same (three-stage) pump model which was specified for the condensate pump. As a result, many of the condensate and distribution pump parts are interchangeable, and minimize the inventory of spare parts needed to service the pumps.

Fig. 4-37 Machinery Room Arrangement,
Distribution Pumps, 50 MW(e)

Fig. 4-38 Machinery Room Arrangement,
Distribution Pumps, 10 MW(e)

As is the case for the condensate pumps, the distribution pumps are required to operate over a range of flow rate demand, and mixed mode flow control was selected for the distribution pumps also.

The distribution pump design requirements based on the most severe operating condition are given in Table 4-26. The available NPSH is based on the same considerations discussed earlier for the condensate pumps.

Table 4-26

DISTRIBUTION PUMP DESIGN REQUIREMENTS

	50-MW(e)	10-MW(e)
Head	96 ft	75 ft
Capacity	26,200 gpm	5240 gpm
NPSH (available)	35.8 ft	19.6 ft
Fluid Temperature	57°F	57°F
Fluid	Ammonia	Ammonia
Number Required	4	4
Pump hp (max condition)	462	72

The major ammonia piping consists of runs (1) from the condenser to the condensate pumps, (2) from the condensate pumps to the evaporator sump, and (3) from the evaporator sump to the distribution pumps, (4) from the distribution pumps to the evaporator spray nozzles, and (5) from the evaporator to the turbine. The flow velocities of Table 4-27 were the primary bases for sizing all lines. The tabulated pump discharge and vapor line velocities were identified as optimum values during the Phase I, Task 1 conceptual design. However, larger lines and lower velocities were selected for the short line segments at the pump discharge to reduce pressure losses in the pump shutoff valves. Line sizes, selected for the major pipe runs, are presented in Table 4-28. All lines are fabricated of carbon steel. Welded construction is used except at certain points where flanges are used to allow component removal and where connections will be broken above seawater level to

Table 4-27

50-MW(e) OTEC LINE SIZING CRITERIA

Line Category	50-MW(e)	10-MW(e)
	Max. Flow Velocity (ft/sec)	Max. Flow Velocity (ft/sec)
Pump Suction Manifolds	3	3
Pump Suction Lines	5	5
Pump Discharge Lines	30	30
Vapor Lines to Turbine	160	160

Table 4-28

LINE SIZES FOR 50-MW(e) OTEC

Location	50-MW(e)		10-MW(e)	
	Pipe Diam. (in.)	Pipe Diam. (in.)	Wall Thickness (in.)	Schedule
Condensate Pump Suction				
Suction Manifold	84	36	0.625	20
Pump Inlet	48	24	0.375	20
Condensate Pump Discharge				
Pump Discharge	20	12	0.25	20
Discharge Manifold	30	12	0.25	20
Distribution Pump Suction				
Suction Manifold	84	48	0.75	-
Pump Inlet	48	24	0.375	20
Distribution Pump Discharge				
Pump Discharge	20	12	0.25	20
Discharge Manifold	36	16	0.312	20
Vapor Lines to Turbine (two Parallel Lines)	75	36	0.625	30

remove either the evaporator or the condenser while the remainder of the module is left in place.

Ammonia flow to the condensate and distribution pumps is provided via single penetrations of the condenser and evaporator shells which supply suction headers. The condensate and distribution pump suction lines are connected to these headers. Each suction line contains a temporary startup strainer, a stop valve, and an elastomeric expansion joint. The expansion joint is provided to minimize loads due to relative thermal expansion between the suction lines and the pump room walls. Each of the condensate and distribution pump discharge lines contains a stop-check valve and expansion joint just downstream from the discharge flange. The individual condensate pump discharge lines are collected into the discharge manifold which passes from the pump room into the condenser, exits the condenser shell, rises above the water line to flanges and valves, which permit isolation and removal of the condenser, and descends below the water line and enters the evaporator shell. Similarly the distribution pump discharge lines supply the distribution manifold which passes from the pump room into the evaporator where it connects with the evaporator spray nozzles. There are two ammonia vapor piping runs from the evaporator to the turbine, each of which carries half of the ammonia vapor.

Isolation valves in the evaporator vapor lines, in the turbine discharge line, and in the main condensate line are installed to permit removal of a condenser or evaporator. Butterfly valves are used throughout since they provide the most compact configuration for large diameter valving. The selected valves have a special seating geometry which provides reliable and tight sealing.

4.5.1.2 Ammonia Storage and Fill Subsystem. The overall functions of this subsystem, namely to store an adequate quantity of ammonia to fill the power system and to make ammonia vapor and liquid available for charging the system

and for makeup, are shared by module equipment and platform facilities for the 50-MW(e) module as follows:

- o Module equipment. Distribute the ammonia supply to fill power system components and control the ammonia makeup flow during plant operation.
- o Platform facilities. Store ammonia, fill the power system with ammonia, and supply makeup ammonia during plant operation.

For the 10-MW(e) unit, the ammonia storage and fill subsystem is an integral part of the power module, in order to render it self-sufficient.

A flow diagram for the module related parts of the 50-MW(e) subsystem is shown in Fig. 4-39. The 10-MW(e) power module subsystem is shown in Fig. 4-40. The equipment consists of the piping to distribute ammonia to the power system components and to control ammonia makeup flow rate.

Before the ammonia charging operation is initiated, the power system is prepared by vent and purge equipment in the Ammonia Unloading Subsystem. This is done in a series of evacuation and nitrogen dilution operations to reduce the potential for fire or explosion during the filling, by lowering the oxygen concentration inside the system. At the start of the ammonia filling sequence, the power system is evacuated to 1 psia with virtually pure nitrogen inside it.

The equipment for filling the power system consists of an evaporator, a superheater, and interconnecting piping as shown in the system flow diagrams, Fig. 4-39 and 4-40. The filling process consists of the following two steps:

- o First, the power system is charged with ammonia vapor to 90 psia. This is done to prevent thermal shock to the power system components, which could occur if liquid ammonia were to be introduced at low pressure and allowed to flash. For this step, ammonia is taken from storage as saturated liquid at 152 psia, throttled to 95 psia, and evaporated. This saturated vapor is then throttled to the pressure inside the power system (from 1 to 90 psia) and superheated to 50°F to permit its injection into the power system without thermal shock hazard.

Fig. 4-39 System Flow Diagram, 50-MW(e)

Fig. 4-40 System Flow Diagram, 10-MW(e)

Warm seawater is used as the heat source in both the evaporation and superheat processes which provides a valuable safety feature by limiting the temperature, and thereby the pressure, of the ammonia during this operation.

- o The second step in the filling operation consists of introducing the required amount of ammonia liquid into the power system to complete the ammonia charge. Part of this liquid charge also vaporizes until the saturation pressure at warm seawater temperature is reached; however, no significant temperature drop will result from this.

For the 50-MW(e) module (with platform equipment) this two part operation requires 15 hours, 14 of which are for vapor pressurizing; the remaining hour is used for liquid filling.

For the 10-MW(e) module (self-sufficient) the filling operation requires 5 hours, 4 of which are for vapor pressurizing with the remaining hour used for liquid filling.

At times a single component, such as a condensate or distribution pump, is emptied of ammonia to allow maintenance work. When the component is put back in service, it is charged with ammonia using the same ammonia filling equipment and the same procedure.

To maintain the proper working fluid inventory in the power system, liquid ammonia is furnished from storage to the condenser on command from a liquid level measurement in the condenser sump.

A list of major equipment items for the 10-MW(e) module only is given in Table 4-29.

4.5.2 Noncondensable Gas Removal Subsystem

The function of this subsystem is to eliminate noncondensable gases from the power system following system startup and during normal operation.

Table 4-29

MAJOR EQUIPMENT FOR AMMONIA STORAGE AND FILL SUBSYSTEM

Ammonia Storage Tanks

Quantity	4
Storage capacity (each)	45,000 lb
Dimensions:	
Diameter	8 ft-6 in.
Length	25 ft-0 in.
Material	Carbon steel
Design pressure:	250 psi

Evaporator

Quantity	1
Type	Shell and tube heat exchanger
Shellside fluid	Ammonia
Tubeside fluid	Warm seawater
Heat transfer surface	212 ft
Dimensions:	
Diameter	1 ft-10 in.
Length	9 ft-0 in.
Materials:	
Tubes	Titandim
Shell	Carbon steel
Shell pressure rating	250 psi

Superheater

Quantity	1
Type	Shell and tube heat exchanger
Shellside	Ammonia
Tubeside fluid	Warm water
Heat transfer surface	44 ft
Dimensions:	
Diameter	12 in.
Length	9 ft-0 in.
Materials:	
Tubes -	Titanium
Shell -	Carbon steel
Shell pressure rating:	250 psi

After filling the power system with ammonia, residual noncondensable gas (primarily nitrogen) is removed following start of power generation. In addition, nitrogen used in the turbine seals may also leak into the power system and require removal. Experience with steam power plants shows that noncondensable gases in the power system diminish the heat transfer effectiveness of the condenser. For example, it has been shown that 1 percent noncondensable gas causes a 45 percent reduction in the heat transfer coefficient in a typical steam plant condenser. Similar effects are anticipated with ammonia; therefore, noncondensable gases must be removed to attain full plant output capability.

During plant operation, the noncondensable gases migrate to the condenser which has a manifold to gather and remove a vapor stream from a region containing a relatively high concentration of noncondensibles. Piping in the 50-MW(e) module, as shown in Fig. 4-41, is used to transport the withdrawn stream of ammonia vapor and noncondensable gas to the platform to recover the ammonia before discharging the noncondensable gases. For the 10-MW(e) module, a compressor and heat exchanger, as shown in Fig. 4-42, are used to condense virtually all the ammonia from this vapor stream. The ammonia liquid is returned to the power system, and the noncondensable gas, with some residual ammonia, is vented to the ammonia purge stack for safe disposal into the air. The gas removal subsystem is sized to reduce the noncondensable gas concentration in the power system from 1 to 0.1 percent in 12 hours.

The noncondensable gas removal system is operated just after system startup to remove noncondensable gases remaining after the filling operation, and it is used during normal operation to remove gas which leaks into the power system from sources such as the turbine seal system. This latter operation is manually initiated when the signal from the noncondensable gas indicator in the condenser reaches a prescribed level.

Fig. 4-41 Flow Diagram Noncondensable Gas Removal System, 50-MW(e) Module

Fig. 4-42 Flow Diagram Noncondensable Gas Removal System, 10-MW(e) Module

A list of major equipment for the 10-MW(e) module is given in Table 4-30.

Table 4-30

MAJOR EQUIPMENT FOR 10-MW(e) NONCONDENSIBLE GAS
REMOVAL SUBSYSTEM

Ammonia Compressor

Quantity	2
Type	Reciprocating, pressurized crankcase
Capacity	85 cfm
Compression ratio	2:1
Discharge pressure	165 psig
Motor horsepower	50 hp

Condenser/Subcooler

Quantity	1
Type	Shell and tube heat exchanger
Duty	850,000 Btu/hr
Heat transfer surface	116 ft
Dimensions:	
Diameter	1 ft - 8 in.
Length	4 ft - 0 in.
Materials:	
Tubes	Titanium
Shell	Carbon steel

4.5.3 Ammonia Cleanup Subsystem

The function of this subsystem is to remove seawater which may leak into the ammonia working fluid because the presence of water causes a serious reduction in power generating capability.

During power plant operation, seawater leakage into any part of the power system collects in the evaporator. To remove this seawater, a sidestream of

liquid ammonia, amounting to 0.7 percent of the power system flow, is taken from the evaporator for treatment in the cleanup subsystem which is shown in Figs. 4-43 and 4-44. All the equipment for the Ammonia Cleanup Subsystem is considered as module dedicated rather than platform related since this subsystem may have to operate continuously in the event of seawater inleakage until the power system is shut down to repair the leakage. The 50-MW(e) and 10-MW(e) modules are similar in this case. The primary cleanup step is vaporization of 97 percent of the ammonia in a seawater heated evaporator. This purified ammonia vapor is compressed and returned to the power system where it rejoins the primary ammonia stream entering the condenser.

The liquid effluent from the evaporator is further treated to remove virtually all the ammonia from the water. The liquid is pumped through a second evaporator to recover energy from the compressed vapor stream. The remaining liquid from this second evaporator is treated in a distillation (stripping) column reducing the ammonia content to less than 400 ppm before disposal into the sea.

Precipitates such as calcium carbonate and magnesium hydroxide, formed by reaction of seawater and ammonia, are removed with the water effluent at the bottom of the distillation column thus purging those sludge-forming materials from the power system.

This ammonia cleanup system differs from the one described previously in the conceptual design (Task 1 of Phase I), with a flash vaporization process plus a distillation step used in place of the single-step distillation process of the earlier conceptual design. Compared to the single-step distillation, the net electrical output of the plant is increased significantly with this combination of processes as shown in Fig. 4-45. For seawater in leakage of 2 gpm and 0.4 gpm the 50- and 10-MW(e) modules, respectively. In this figure, the composite power loss curve is the sum of the turbine power loss and the cleanup system using both evaporation and distillation processes is considerably less than that required for distillation alone and the operation at the optimum point with the two-step system results in a 7.5-MW(e) net and

Fig. 4-43 Ammonia Cleanup System, 50-MW(e)

Fig. 4-44 Ammonia Cleanup System, 0-MW(e)

Fig. 4-45 Effects of Water on OTEC Power Loss

1.5-MW(e) net output higher, as compared to operation at the optimum point for distillation alone for the 50- and 10-MW(e) modules, respectively. A study to evaluate the available options for the cleanup process is described in Appendix F.

A list of major equipment items for the 10-MW(e) module is given in Table 4-31.

Table 4-31

MAJOR EQUIPMENT ITEMS FOR THE 10-MW(e) AMMONIA CLEANUP SYSTEM

Cleanup evaporator	
Quantity	1
Type	Shell and tube heat exchanger
Shellside fluid	Ammonia
Tubeside fluid	Warm seawater
Heat transfer surface	2100 ft ²
Materials	
Tubes	Titanium
Shell	Carbon steel
Shell pressure rating	250 psig
Ammonia Compressor	
Quantity	2
Type	Reciprocating, pressurized crankcase
Capacity	1200 cfm
Inlet pressure	56 psia
Discharge pressure	92 psia
Motor horsepower	175 hp
Ammonia Pump	
Quantity	2
Type	Centrifugal
Flow	2 gpm
Suction pressure	56 psia
Discharge pressure	92 psia
Motor Horsepower	1/2 hp
Heat Recovery Evaporator	
Quantity	1
Type	Shell and tube heat exchanger
Heat transfer surface	120 ft ²
Materials	
Tubes	Titanium
Shell	Carbon steel
Shell pressure rating	250 psia
Distillation Column	
Quantity	1
Type	Packed column
Dimensions	
Diameter	5 ft
Length	6 ft 0 in.
Material	Titanium internals
Reboiler	
Quantity	1
Type	Electrically heated
Heater rating	100 kW
Material	Titanium internals

4.5.4 Ammonia Unloading Subsystem

The functions of this subsystem are to remove the ammonia from the power system after shutdown, to purge it, and to fill it with air to allow safe access for maintenance and disassembly. This subsystem is also used for removing ammonia from the turbine or any of the condensate and distribution pumps, after isolation, to allow disassembly and maintenance. The removal of the ammonia from the power system is accomplished in three distinct operations:

- o Draining liquid ammonia from the power system and transferring it to storage
- o Recovering ammonia vapor from the power system, condensing it, and transferring the resulting liquid to storage
- o Venting and purging residual ammonia vapor from the power system, leaving the system filled with air at atmospheric pressure

For the 50-MW(e) module, the ammonia drain equipment, which is located in the module, provides the means for draining liquids from the power system using piping, collection tanks and transfer pumps. The collected ammonia liquid is transferred to the ammonia storage tanks on the platform. A drain collection tank is provided for the condenser segment of the power module, and another collection tank is provided for the evaporator segment. This arrangement is necessary because of the segmented nature of the module. Two parallel, 50 percent capacity transfer pumps are provided in each collection tank, so that the drain function is available even if one transfer pump fails. The collection tank capacity is sufficient to store the volume of one condensate or distribution pump. When a larger volume, for example the condenser sump, is to be drained, transfer pumps are activated to transport the excess ammonia to the platform storage tanks. The ammonia vapor recovery equipment is included in the platform support facilities to serve all modules. Each module includes piping to connect power system components to the ammonia vapor recovery equipment in the platform support facility. The flow diagram of this ammonia vapor piping is also shown in Fig. 4-46. As shown in the drawing, a vapor recovery line is connected to each power system component that contains a significant amount of vapor to be recovered. Each recovery line connects to

Fig. 4-46 Ammonia Unloading System, 50-MW(e)

a header leading to the platform support facility. Ammonia vapor is removed from the evaporator and condenser until a pressure of 60 psia is attained. Further ammonia removal and purging of these two large components is carried out on a service ship or at dockside.

The vent and purge equipment, which is part of the platform support facilities, is used to disperse quantities of ammonia which cannot be economically recovered and to fill the turbine, condensate pumps, and distribution pumps with air, after a nitrogen purge, to allow safe disassembly or maintenance. Each module includes piping to connect components to the ammonia vent and purge equipment in the platform support facility. A vent and purge line is connected to each component in the power module except the evaporator and condenser. All vent lines connect to a header leading to the platform support facility. All purge lines connect to a header which leads to supplies of nitrogen and clean atmospheric air which are furnished by the platform support facilities. Ammonia vapor is vented from a power system component and is diluted before disposal to a safe concentration in air by the platform facilities. After being vented to atmospheric pressure, the interior volume of the component is evacuated to 1 psia and refilled with nitrogen to 15 psia by platform facilities. This evacuation-nitrogen fill sequence assures that an explosive concentration of ammonia in air is never encountered. The final purging operation consists of forcing filtered air through the component until the ammonia concentration is low enough to permit safe disassembly of the component.

The 10-MW(e) module flow diagram for the Ammonia Unloading System is shown in Fig. 4-47. The system consists of drain tanks with transfer pumps located at low levels in the machinery rooms under the condenser and the evaporator. These receive liquid ammonia from the power system and transfer it to the storage tanks. The ammonia vapor recovery stream has compressors and condensers to recover the vapor left in the system after draining the liquid ammonia. Finally, the vent and purge function is served by a vacuum pump, storage volume for purge nitrogen, and blowers to move purge air and to admix dilution air at the stack.

4-130

Fig. 4-47 Ammonia Unloading System 110 MW(e)

LMSC-D630248

Ammonia unloading is accomplished in the following three operations:

- o At first, liquid ammonia is drained by gravity into a drain tank beneath each heat exchanger and is pumped from there to the ammonia storage tanks. The equipment is sized to drain and transfer all the liquid in one hour.
- o The second operation consists of removing vapor from the power system, compressing it, condensing it, and transferring the resulting liquid to storage. For the compression part of the operation, two parallel 50 percent capacity compressors are used so that the vapor recovery function can proceed even if one compressor fails. With both compressors operating, the ammonia pressure inside the power system is reduced to 15 psia in 9.5 hours.
- o The last of the operations consists of venting the residual ammonia vapor and replacing it with air. Due to the explosive and toxic hazards of ammonia, this operation is carried out in three steps. First, the ammonia pressure inside the power system is reduced to 1 psia using two parallel vacuum pumps. During this step, the effluent ammonia is mixed with air to a safe concentration before venting to a stack. Next, the power system is filled to 15 psia with nitrogen. The resulting mixture has 7 percent ammonia which precludes the possibility of an explosive mixture with air during the purging operations. (The lower explosive limit for ammonia in air is 16 percent.) The third step in this operation consists of ventilating the power system internals with filtered air until the ammonia concentration is less than the eight-hour exposure limit of 50 ppm. Centrifugal blowers are used to force fresh air through the power system and to vent the discharge air through the stack. At the stack inlet, air is added to the stream bringing the ammonia concentration of the stack discharge to a biologically safe level. This three-step purging operation is completed in 12 hours. Thus, the entire ammonia unloading sequence can be performed in 22.5 hours.

The vapor recovery operation described in the second unloading operation above leaves 1,900 lb of ammonia to be vented to the atmosphere. Recovery of this small amount of ammonia is not economically feasible. Present air discharge regulations place no restrictions on gaseous ammonia emission. However, if future regulations would establish gaseous discharge limits, the vacuum pumps and the vapor recovery compressors can be connected in series to recover ammonia down to 1 psia internal pressure. That would leave 127 lb of ammonia in the power system for discharge to the atmosphere.

Each of the major power system components (evaporator, condenser, turbine, and pumps) can be isolated and unloaded without removing the ammonia from the other components. The same unloading equipment is used in all cases. However, the vapor recovery operation is omitted when unloading a pump or the turbine because the small amount of vapor cannot be economically recovered.

A list of major equipment items for the 10-MW(e) module is given in Table 4-32.

4.5.5 Power Distribution Subsystem

The functions of this subsystem are as follows:

- o Connect the generator with the platform power conditioning equipment
- o Condition and distribute the generated ac power for driving pumps and other auxiliary loads
- o Provide conditioning and distribution of startup power from an external source
- o Provide instrument and electric power wiring in the module
- o Provide an uninterruptable power supply for critical control and safety equipment

The method for providing these functions is shown in block diagram form in Figs. 4-48 and 4-49 for the 50- and 10-MW(e) modules, respectively.

The 13.8 kV 60 Hz ac generator is tied to the platform electrical distribution system at the platform bus breaker which provides fault current protection for the generator. Power conditioning for transmission to shore is to be supplied by the platform.

Auxiliary power for the pumps and other loads is provided by a connection to the platform bus through an auxiliary transformer which converts the voltage from 13.8 kV to 4160V. The auxiliary transformer output is connected to a bus supplying power to the seawater pumps, ammonia condensate pumps, ammonia distribution pumps, ammonia cleanup compressor, and a transformer (4,160V/480V) for the balance of the auxiliary loads. The seawater pumps and the ammonia

Table 4-32

MAJOR EQUIPMENT FOR AMMONIA UNLOADING SUBSYSTEM

Drain collection tanks	
Quantity	2
Configuration	horizontal, cylindrical
Capacity	50 ft ³
Ullage	15 percent
Dimensions	
Diameter	5 ft - 0 in.
Length	10 ft - 0 in.
Material	Carbon steel
Design pressure	250 psi
Transfer Pumps	
Quantity	4
Type	Vertical, centrifugal sump
Flow	150 gpm
Discharge head	100 ft
Motor horsepower	10 hp
Ammonia compressor	
Quantity	2
Type	Reciprocating, pressurized Crankcase
Capacity	75 cfm
Compression ratio	11:1
Discharge pressure	180 psig
Motor horsepower	40
Vapor Recovery Condenser	
Quantity	1
Type	Shell and tube heat exchanger
Duty	1,750,000 Btu/hr
Heat transfer surface	113 ft ²
Dimensions	
Diameter	1 ft - 4 in.
Length	6 ft - 0 in.
Materials	
Tubes	Titanium
Shell	Carbon steel
Vacuum pump	
Quantity	2
Type	Rotating cylinder
Capacity	70 cfm
Vacuum capacity	1 psia
Motor horsepower	5 hp

Table 4-32 (contd.)

Blower

Quantity	2
Type	Centrifugal
Capacity	4800 cfm
Discharge pressure	15 in., water gage
Motor horsepower	15 hp

Exhauster

Quantity	2
Type	Low pressure centriugal
Suction pressure	-3 in. water
Capacity	100,000 cfm
Motor horsepower	100 hp

Nitrogen Storage Cylinders

Quantity	10
Dimensions	
Diameter	2 ft
Length	40 ft
Charged pressure	2000 psig
Capacity	30,000 lb

Stack

Dimensions:	
Diameter	6 ft - 8 in.
Height	100 ft
Material	Carbon steel

Fig. 4-48 Electrical Power Block Diagram, 50-MW(e) Power Module

Fig. 4-49 Electrical Power Block Diagram, 10-MW(e) Power Module

condensate and distribution pumps are driven by variable frequency motor speed controls to allow operation over the lower range of fluid flow rate requirements. These pumps are also supplied with 60 Hz ac current for operation in the upper range of flows.

Startup power from an external source is also connected to the 4160 V bus. This allows operation of any combination of auxiliary loads limited only by the capacity of the external startup power generator.

A transformer from the 4160 V bus to a 480 V bus provides power for all auxiliary loads except for the large pump and compressor motors mentioned above. The module 480 V bus is connected to the platform normal power and standby buses. The former supplies power for the platform auxiliary loads during normal operation, and the latter connects the module to the platform standby power supply system.

Continuous operation of certain critical control and safety equipment is ensured by a set of batteries and an inverter. The batteries are kept at full charge during normal plant operation and are automatically put into service when needed. This uninterruptable power supply allows operation of critical system on emergency power until a standby power generator can be started and brought on line.

4.5.6 Ancillary Subsystems

4.5.6.1 HVAC Air Sampling. All compartments having ammonia piping are monitored to ensure safe ammonia vapor concentration levels. Three ammonia concentration threshold levels of particular concern are

- o Industrial exposure level (50 ppm for 8 hour exposure)
- o Lethal concentration (1700 ppm for 1/2 hour exposure)
- o Explosive concentration (lower explosive limit - 16 percent)

The first and second levels are continuously monitored by infrared analyzers, and the third is measured by a continuously operating combustible gas detection

system. Each detector system has an electrical output proportional to the concentration of ammonia in air. The detector output signals are used as inputs to the power module control computer for recording and alarm functions. The concentration signals for the industrial exposure level measurements are recorded to provide permanent personnel exposure records. Any concentration which exceeds a preselected value triggers a warning alarm in both the control room and the module compartment where the high concentration is detected.

The locations and numbers of ammonia vapor detectors are given in Table 4-33.

Table 4-33
LOCATIONS OF AMMONIA VAPOR DETECTORS

	Number of Detectors		
	Industrial Exposure Level	Lethal Concentration	Explosive Concentration
Condensate pump room	2	1	1
Distribution pump room	2	1	1
Control room	1		
Auxiliary machinery room	1		
Auxiliary ammonia subsystems deck	2	2	2

4.5.6.2 Ballast Control Subsystem. The ballast system provides the means for altering the draft, heel and trim of the power module, and the means for maintaining positive stability. The system consists of fresh water ballast, ballast tanks in the machinery room, piping, valving, pumps and power for pumps, and associated metering devices. A fixed ballast system utilizing concrete is briefly described as an alternate approach. The system for the 10-MW(e) module is identical to that of the 50-MW(e) module, except for the ballast weights required for each.

50-MW(e) Power Module. In the unballasted condition, the module segments are statically unstable in the mating attitude as indicated by negative metacentric height. The addition of ballast 2584 LT for the evaporator, 1516 LT for condenser is required to provide stability and to increase submergence so that the heat exchanger centerline depth is 68 ft. Fresh water ballast will be pumped into the shell side of each heat exchanger prior to launch 2200 LT for the evaporator and 1200 LT for the condenser, or 57 and 39 percent of capacity of these spaces. A separate fresh water piping system will be required if a portion of the ammonia piping system cannot function in this capacity. Separate ballast tanks will be located in the machinery space of each segment to provide trim and heel control over a small range of angles.

Four tanks of 100 LT capacity each will be located in each corner of the machinery room. Each tank will extend the full 25 ft height of the room and will have a width and length of 15 by 15 ft. Cross connecting piping will allow for transfer of fresh water ballast between tanks and between each tank and the heat exchanger. Subsequent to mating of the module to the surface platform, the ballast will be pumped out of the heat exchanger to be utilized for other modules and/or for provisioning the potable water supply onboard the platform. At-sea transfer hose and beam will be utilized as required for the conveyance of ballast and electrical power for pumping. The net positive buoyancy of each heat exchanger will thereby increase platform buoyancy, while loss of static stability is not necessarily critical in the mated condition.

An alternative to the use of fresh water ballast in the heat exchangers is to attach concrete ballast to the module segment, with required volumes of 1900 yd³ for the evaporator and 1030 yd³ for the condenser. This large volume is best attached either below the pump room or around the heat exchanger shell. Ballast tanks located in the machinery room will be as described for the primary concept.

10-MW(e) MAPS. Total ballast required for draft control is 926 LT for the evaporator and 400 LT for the condenser. The evaporator module segment is fitted with 290 LT of external concrete ballast below the pump room, 110 LT in

the access trunk and 432 LT within the joining structure between shell and machinery room, for a total of 832 LT or 432 yd³ of concrete. Four ballast tanks, one in each corner of the machinery room, contain a total of 94 LT of seawater ballast. These tanks are cross-connected to provide trim and heel control during the mating operation. The condenser module segment fixed concrete ballast consists of 253 LT mounted externally below the pump room and 110 LT in the access trunk. Four tanks containing a total of 80 LT of adjustable ballast (seawater) are located one in each corner of the machinery room.

Interface Requirements. The ballast control system will interface physically with a portion of the ammonia system piping if the latter is utilized for fresh water ballast transfer. There will be a functional interface with the platform for submergence and attitude control mating operation requirements, with the DACS for control of fresh water pumping, and with the ammonia cleanup system for cleaning residual fresh water from the ammonia portion of power system.

4.5.7 Biofouling Countermeasures

Biofouling control methods will be the combination of chemical (low level continuous chlorination at 0.1 to 0.2 ppm) and mechanical (M.A.N. brushes). Data from DOE biofouling experiments at Keahole Point show that tubes exposed to 6 ft/sec flowing seawater can be brought back to clean tube conditions after 10 weeks exposure with only 11 cleaning cycles with a M.A.N. brush. In a review of the test data from both Keahole Point, Hawaii, and St. Croix, V.I. by Fedkovich ("Measurement of Biofouling in OTEC Heat Exchangers" Fifth Ocean Thermal Energy Conversion (OTEC) Conference 2-78), it was concluded that OTEC heat exchangers would have to be cleaned only once every 7-8 weeks to maintain R_f 50×10^{-5} (0.0005 hr ft² °F/Btu) or every week to maintain R_f 10×10^{-5} (0.0001 hr ft² °F/Btu).

Admittedly, these conclusions were based on a small data base but more recent results from February through August, 1978, at Keahole Point provide similar encouraging results indicating that brushing tubes exposed to 6 ft/sec seawater every few weeks can bring the tubes back to a near clean condition.

It is reasonable to assume that a continuous chlorination schedule of 0.1 ppm and brushing with M.A.N. brushes will provide biofouling control adequate to guarantee a thermal resistance no greater than $0.0002 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$. This chlorination schedule is not considered too high from a parasitic power standpoint since it uses only 1.5 percent of the power generated. This value may be further lowered as new test data are developed from DOE tests at Keahole Point, Panama City, and LMSC tests at San Diego during the remainder of 1978 and into 1979.

4.5.7.1 Chlorination Subsystem. The chlorination subsystem consists of Chloropac (trademark) generators, together with necessary piping, valves, storage tank, and controls. The Chloropac generator is a standard product of Engelhard Industries Division, Engelhard Minerals and Chemicals Corporation. The generators produce sodium hypochlorite on the module or platform using approximately 2.8 kW/lb of equivalent chlorine using 30 ohm-cm water (1.7 kW/lb in 10 ohm-cm water). The generating cells electronically decompose seawater to produce sodium hypochlorite. The cell anodes are designed for 5 to 7 years of life at maximum output. No periodic flushing or cleaning is necessary. Cells are arranged in series to provide the necessary output and the groups of cells can be assembled to fit the desired space. The cells are designed for operating at 150 psi at 180°F and will withstand test pressures of 1000 psi. All assemblies are tested to 300 psi.

The chlorination system for the 10-MW(e) power module will use two Chloropac units that generate a maximum of 85 lb/hr of chlorine that can treat 680K gal/min of warm seawater at a dosage of 0.5 ppm. The chlorination unit will be completely automatic in that it will have a residue chlorine sensor that will provide a feedback to the chlorine generating unit to maintain a maximum of 0.05 ppm residue chlorine and provide for diversion to the cold

seawater as required. The overall power requirement of the chlorine generator is 750 kW when operating at maximum capacity, but it is anticipated that 0.1 ppm chlorine (150 kW) in combination with M.A.N. brushes will be sufficient to maintain clean heat exchanger tubes.

4.5.7.2 M.A.N. Brushes. The M.A.N. (Fig. 4-50) brush consists of two cages, one brush with bristles, and two end caps. The cast plastic brush cage holds the brush in the downstream position. Waterflow reversal causes the brush, due to the end caps, to shuttle to the downstream end of the tube. The bristles are nylon and the titanium wire is used to hold the bristles. The effect of the M.A.N. brush installation must be considered in determining the total cost of the biofouling countermeasures system. Data, provided by the M.A.N. Company, indicate that the brush and retaining cages produce an increased pressure drop (through the heat exchangers) equal to 1 foot of water head. This incremental pressure drop was incorporated into the optimization program computations and the heat exchangers and turbine-generator resized to provide 10-MW(e) net power. The results of the comparison are shown in Table 4-34.

The effect on heat exchanger cost can be estimated from data provided by Foster Wheeler which indicate that the slope of the incremental cost curve versus tube length (L) has a value of approximately 2/3. The incremental heat exchanger cost can be estimated from

$$\frac{\text{Cost}}{\text{base cost}} = \frac{2}{3} \frac{L}{L_{\text{Base}}} - 1$$

Clean tubes are vital for high performance of surface condensers and heat exchangers. Conventional methods of tube cleaning usually make it necessary to take the equipment out of service and the manual work involved is apt to be formidable. And even then the success of hand cleaning is only temporary because heat transfer starts to deteriorate rapidly as new deposits build up in the tube. Tube cleanliness warrants maximum heat transfer and also helps to avoid tube corrosion.

Modern plant management will therefore rely on automatic cleaning to maintain 100 % tube cleanliness and eliminate manual chores.

Fig. 1: The M.A.N. on-load tube cleaning system uses brushes, the most effective and yet harmless type of cleaning equipment. Each tube is served by its own brush and has a plastic cage at each end in which the brush is parked between cleaning cycles. The special advantage of this method is that loose dirt is swept along ahead of the brush because the annular gap provided between the tube wall and the brush cap causes the brush to travel at a lower velocity than the water. This obviates abrasive wear of the protective metal oxide layer on the interior of the tube.

Fig. 2: Reversing the circulating water flow starts all brushes moving through the tubes. When normal circulating water flow is restored the brushes return to their initial position. Shuttling through the tubes, the brushes effectively clean all tubes and loosened deposits are flushed out of the condenser.

Fig. 3: The circulating water flow reversal valves are timed to gradually decrease the water flow in the tubes to zero and then start the water flow in the opposite direction. In the process a bypass is established to the condenser discharge side so that continuity of water flow is ensured and water hammer is eliminated. The whole cleaning cycle takes no more than 80 seconds. An automatic timer permits intervals between cleaning cycles to be matched to the degree of water pollution. There is also a manual override feature.

Flow reversal can be effected by any of several methods to suit local conditions. In Fig. 2 it is done by additional pipework and valves, preferably motor operated.

Fig. 4-50 M.A.N. Brushes

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

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

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

A million tubes are already being cleaned by this method.

A million tubes have been uprated for maximum heat transfer.

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Fig. 4-50 (cont.) M.A.N. Brushes

Table 4-34

EFFECT OF M.A.N. BRUSH SYSTEM

	<u>Optimized Geometry</u>	<u>M.A.N. Brush</u>
Condenser Tube Length (ft)	25.47	26.48
Evaporator Tube Length (ft)	25.81	26.95
Turbine-Generator (MW)	13.26	13.59

The cost of the turbine-generator is linear for the small change shown. From the above, it is seen that heat exchanger and turbine-generator costs will increase by 2.5 to 3 percent due to the M.A.N. brush system.

In November 1978, LMSC completed tube wear and brush life tests on 1.40-in. diameter brush in a 1.50-in. OD aluminum tube. A total of 11,740 cycles (23,480 passes) were made in a 5052 O-temper aluminum tube. No measurable wear was found in the test tube when it was compared to a nonused section of the same tube stock, even though the brush in addition has traveled a total of 194,376 ft. Photos magnified approximately 50 times show individual bristle ends to be worn in a random manner around the bristle rod circumference. The brush outside diameter (average over the length of the brush) did not show appreciable wear for the travel distance (no wear within 0.008 in. measurement tolerance). From these data, it is estimated that the brush will be good for 10 years usage at the three cycles/week rate. For MAPS costing purposes, however, a 3-year brush life was used. Brush details are on the following sheets.

The tube-to-tubesheet weld and M.A.N. brush cage design baseline configuration is to have the tube extend 5/8 in. beyond the tubesheet face and use a fillet-type weld for joining the tube to the tubesheet. The fillet-type weld, however, may require an additional machining operation on the tubesheet and extreme care to ensure that the thin-walled tube is not degraded. (See Welding Handbook, American Welding Society Chapter 69 Aluminum and Aluminum

Alloys). The backup for the fillet weld is the internal bore weld. The M.A.N. cage (see attached sheet) is the socket ended cap which will have a face matching the fin to provide a mechanical attachment to the tube. A bonding agent will be used to seal the crevice between the tube and the tubesheet (if necessary).

The backup position is to use an end weld for joining the tube to the tubesheet thus the tube end is flush with the tubesheet. The M.A.N. cage is the spigot-ended cage. This combination may require the tube end inside diameter to be increased to provide a smooth transition between the cage and the tube. An adhesive or an indentation in the cage (small tube ring) would be used for attaching the cage to the tube.

4.5.8 Corrosion Control Subsystem

The methods that will be used to control corrosion of the power systems will be:

- a. Review of all components of the system to ensure the selection of corrosion resistant materials and the avoidance of incompatible materials
- b. Application of proven marine coatings such as epoxy coal tar with inorganic zinc primers
- c. Cathodic protection

Detailed design review will include examination of all components exposed to seawater and their relationship to the remainder of the seawater system. This will ensure the best selection and combination of materials and will highlight those areas where functional design requirements mandate the use of combinations of materials that are not compatible. In these cases, paint coatings and cathodic protection will be applied to control corrosion.

The method of cathodic protection will be sacrificial anodes because it is considered the most reliable and requires no maintenance for a minimum of 10 years. It is also the most economical from the standpoint of initial capital outlay and does not require power to operate except for reference electrodes to monitor the potential of the structure being protected.

For the 10-MW(e) heat exchanger clad tubesheets, the use of 33 of the BTZ-27 zinc anodes made by Federated Metal Division of ASARCO (see attached Data Sheet) is planned. Assumptions were 3 ft/sec seawater velocity, 5-year life of anodes using 10 mA/ft². These anodes, BTZ-27, will virtually cover all the steel weld joints. The 1/2-in. diameter core rod is welded to steel tube-sheet. The composition in accordance with MIL-A-18001 is as follows:

<u>Element</u>	<u>Percent</u>
Lead	0.006 maximum
Iron	0.005 maximum
Cadium	0.015 - 0.025
Copper	0.005 maximum
Aluminum	0.10 - 0.50
Silicon	0.125 maximum
Zinc	Balance

For the exterior surface of the heat exchangers such as piping, machinery room, etc., a luminum anodes such as the high efficiency "Reynodes" produced by Reynolds or "Galvanlum" produced by Dow will be used. The chemical composition is proprietary so exact composition cannot be given.

A 10-year life requires 6,500 lb of anodes of various sizes to meet the configuration of the heat exchangers, piping, and their arrangement. An example of one that might be used is ASARCO C360, shown in Fig. 4-51.

ALUMINUM OFFSHORE ANODES

Available
in these
anodes

Galvalum[®]
or
Hi-Amp[™]

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

Anode No.	Net Al Wt.	WxH	La	Lc	X	Type	Steel Core	
								Schedule
A 375	325	6½"x6½"	8'	10'	—	A	2" schedule 80 pipe	
A 875	725	9½"x9½"	8'	10'	—	A	4" schedule 80 pipe	
B 385	325	6½"x6½"	8'	10'	—	B	2" schedule 80 pipe	
B 910	725	9½"x9½"	8'	10'	—	B	4" schedule 80 pipe	
							Internal	Legs-Pipe
C 360	325	6½"x6"	8'	7'	5'	C	2"x2"x¼" angle	2" schedule 80
C 405	370	6½"x6½"	8'	7'	5'	C	2"x2"x¼" angle	2" schedule 80
CT 840	725	9½"x8½"	8'	7'	5'	CT	"T" 4" D. x 5¼" W.	4" schedule 80

(ST4WF)

Special anode sizes can be furnished in the cross-sections shown, from 4' L to 8' L

The following nominal weights/ft. can be used for estimating:

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

NOTE—DIMENSIONS & WEIGHTS ARE NOMINAL

AMERICAN SMELTING AND REFINING COMPANY
FEDERATED METALS DIVISION
Cathodic Protection Dept.

120 Broadway, New York, N. Y. 10005
212-732-9500

Fig. 4-51 Aluminum
Anodes

Bulletin No. L729-5-5M

™ Trademark of
American Smelting and Refining Company

® Registered trademark of
The Dow Chemical Company

4-147

Printed in U.S.A.

BALLAST TANK

TZ-27: 1.4" x 1.4" x 48". Contains 1/2" diameter cast-in galvanized steel mounting core. Nominal weight, 27 lbs.; current rating, 1 amp-yr.

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

BTZ-50: 2" x 2" x 48". Contains 1/2" diameter cast-in galvanized steel mounting core. Nominal weight 50 lbs.; current rating, 2 amp-yrs.

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

PIER AND PILING ANODES

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

PZ-150: 4" x 4" x 36". Contains 3/4" diameter cast-in galvanized steel pipe core extending 3" beyond ends. This anode can also be made with 3/8" diameter galvanized steel rod core. Nominal weight, 150 lbs.; current rating, 6 amp-yrs.

PZ-50: 2" x 2" x 48". Contains 3/8" cast-in galvanized steel rod extending 4" beyond each end. Nominal weight, 50 lbs.; current rating, 2 amp-yrs.

CONDENSER ANODES

CZ-5: Current rating 1/4 amp-yr.; other sizes available on request.

4.6 CONTROL AND INSTRUMENTATION

Included in the power module design is a Data Acquisition and Control System (DACS). DACS is a supervisory control system satisfying the power module requirements for monitoring and controlling all operational functions, and module health and safety parameters. The DACS conceptual design, developed to satisfy the 50-MW(e) CPM requirements, is directly applicable on a smaller scale to the 10-MW(e) MAPS. The conceptual design was developed to a level of detail sufficient to establish procurement spans and obtain cost data.

DACS has been refined and expanded during the 10-MW(e) MAPS preliminary design task. The system design has a closed loop self-test capability to enable rapid malfunction identification and isolation. For the 50-MW(e) Power Module conceptual design, the monitoring and control process is shown in Fig. 4-52. A basic block diagram for DACS is given in more detail for the preliminary design of the 10-MW(e) MAPS in Fig. 4-53.

A preliminary performance specification for the DACS has been completed and is included in Appendix B (LMSC specification No. X5531908). The instrumentation and hardware description is presented in Appendix I.

4.6.1 Control Philosophy

The philosophy which has guided the DACS design is based on three goals:

1. To acquire and process all operational, housekeeping, and safety data in a uniform manner using standard industrial equipment whenever possible in order to minimize cost and potential for error, and to allow for expandability and adaptability of the system
2. To make all operating parameters immediately available for automatic control action or manual diagnostic procedures
3. To provide maximum flexibility, permitting convenient alteration of the computer algorithms for processing input data for display or control action

Fig. 4-52 Basic Block Diagram for Data Acquisition Control
and Instrumentation 50-MW(e) CPM

This led to the choice of an integrated Data Acquisition and Control System (DACS) based on a general purpose digital computer to sense and process system operational data, and to regulate system operation by commanding the settings of system control parameters. (Ref. Fig. 4-52 and 4-53). Through the DACS console the operator can choose the data display, select the system operating mode, manually override the automatic control, and, within limits of proper authority and built-in system safeguards, alter the data processing and control algorithms.

Fig. 4-53 Basic Block Diagram for Data Acquisition
Control and Instrumentation 10-MW(e) MAPS

4.6.2 Operational Instrumentation

Operational or control instrumentation is the minimum complement of instrumentation required to operate a Power Module continuously in a safe manner. The basic list of control parameters and associated sensors is shown in Table 4-35.

The control instrumentation enables the control system to adjust ammonia and/or seawater flow rates and turbine speed to obtain the performance desired. The parameters being measured in order to perform this function are the same for the 50-MW(e) Power Module, as for the 10-MW(e) MAPS, with the exception of seawater inlet temperatures which are provided through

Table 4-35

50-MW(e) POWER MODULE CONTROL PARAMETERS
AND GENERIC SENSOR SELECTION

Control Parameter	Selected Type of Sensor
Evaporator & Condenser Seawater Inlet Temperatures	Thermistor Probe
Evaporator & Condenser Sump Liquid Levels	Capacitance Probe
Generator Voltage	Volt Sensor
Turbine rpm (Generator Frequency)	Tachometer Generator
Evaporator & Condenser Pressures	Sensing tube to Pressure transducer
Vane Position	Turbine Inlet Potentiometer

the platform interface. A brief description of the selected instrumentation is presented in Appendix I.

The selected control instrumentation is the following;

- (1) Ammonia liquid capacitance probe (INVALCO, Model 400)
- (2) Turbine Speed Tachometer (Stricht Co., Model C, Tachometer generator)
- (3) Evaporator and Condenser pressure transducer (Druck Ltd., Model PDCR 22)

4.6.3 Diagnostic Instrumentation Description

Diagnostic or test instrumentation is provided to monitor and evaluate the power module performance during normal operation and during periods of "off design" performance required for special test. This complement of instrumentation interfaces through the DACS enables a comprehensive diagnostic capability in the Control Center in addition to monitoring the necessary house-keeping and/or safety functions for the power module. The basic parameters being measured to perform this function, the selected type of sensor and the

sensor specification are delineated in Table 4-36 for the 50-MW(e) module and Table 4-37 for the 10-MW(e) MAPS. A brief description of the various types of instrumentation selected is given in Appendix I.

The selected diagnostic instrumentation is the following:

- (1) Thermister (Yellow Springs Instrument-YSI-Model 44018 on Model 710 probe)
- (2) Differential Pressure Sensor (Annubar Primary Flow Sensor)
- (3) Pressure Transducer with multiple measurements (Scanivalve with Druck Ltd, Model PDCR22)
- (4) Fluid Analyser (Gam Rad, Model 150)
- (5) Accelerometer (ENDEVCO, Model 2221F)
- (6) Electrode Analyzer (ORION Series 1000 On-line Analyzer)
- (7) Gas Sensor (International Sensor Technology, 1st Model AG2200)
- (8) Photometric Analyzer (Teledyne Analytical Instruments, Model 514)
- (9) Oxygen Deficiency Detector (Gas Tech, Model 1220)
- (10) Flooding Sensor (Compact, Model 600 N)

Table 4-36

TEST PARAMETERS AND GENERIC SENSOR SELECTION
50-MW(e) POWER MODULE

Test Parameter	Selected Type of Sensor
Evaporator & Condenser Outlet Water Temperatures	Thermistor Probe
Evaporator & Condenser Water Flow Rates	Differential Pressure Sensor
Evaporator & Condenser Ammonia Pressures	Strain Gage Bridge Transducer
Evaporator & Condenser Ammonia Temperatures	Thermistor Probe
Ammonia Vapor Quality	Fluid Analyzer
Ammonia System Valve Status	Contact Closure
Turbine vibration	Accelerometers
Ammonia Outleakage Detection	Electrode Analyzer
Seawater Inleakage Detection	Near Infrared Photometric Analyzer
Turbine & Generator Lube Oil Temperatures	Thermistor Probes

Table 4-37

BASIC INSTRUMENTATION SPECIFICATION LIST FOR THE
10- MW(e) MAPS

Measured Parameter	Selected Type Of Ssnsor	Specification Requirements
Temperature	Thermistor in Stainless steel probe	Temperature range: -5 to +45°C Accuracy: 0.1°C Output stability: 0.02%/°C variation from calibration temperature
Pressure	Bonded strain-gage pressure transducer	Combined linerity & hysteresis: 0.06% from best straight line Excitation: 24 Vdc Output: 0 to 100 mV
Flow	Annubar primary flow sensor	Accuracy: 1% of actual value Repeatability: 0.1% of actual value
Level	Continuous reading Capacitance probe	Accuracy: 3% Temperature range: -20 to 160°F Maximum working pressure: 600 psi Input voltage: 115 Vac Output: 4-20 mA
Vibration	Piezoelectric Accelerometer	Maximum weight: 11 grams Case Material: stainless steel Temperature Range: -65 to 500°F Acceleration limits: Vibration, 10 ³ g, sinusoidal Shock, 3000 g Base strain sensitivity: 5 equivalent g, nominal at 250 strain Charge Sensitivity: 10 pC/g nominal
Ammonia Outleakage Detection	Electrode Analyzer	Range: 0.1 - 10 ⁴ ppm Accuracy: + 10% of reading Input: 115 Vac Output: 4-20 mA
Seawater Inleakage Detection	Near Infrared Photometric Analyzer	Accuracy: 2% F.S. Range: 0 - 10 ³ ppm Input: 115 Vac Output: 4-2 mA
Ammonia Vapor Quality	Fluid Analyzer	Range: 10 ppm to 50.10 ³ ppm Repeatability: 1% indicated value Input: 115 Vac, 60 Hz Output: 0-100 mVDC

4.6.4 Dacs Control Center

4.6.4.1 DACS Control Center Equipment Arrangement. All of the equipment delineated in paragraph 4.6.2 is located in the Control Room on the 01 deck of the 10-MW(e) MAPS. [Ref. Appendix C, Dwg. 5531921, General Arrangement, 50-MW(e), Power Module Immersed-Segmented, and Dwg. No. 5531922, General Arrangement, 10-MW(e), Power Module, Immersed-Segmented]. A preliminary equipment layout in the Control Room is shown in Fig. 4-54. It should be noted that dual CPU's and CRT's are provided. (Ref. items (1) and (6) of paragraph 4.6.2). The use of dual CPU's is necessary to achieve adequate overall system availability. One CRT is provided to monitor and control normal operations. The alternate CRT display/keyboard terminal will be employed to monitor and control diagnostic/test and housekeeping functions. The Control Center CRT's are functionally switchable which provides enhanced operational reliability.

Fig. 4-54 Preliminary Equipment Layout Control Center 10-MW(e) MAPS

4.6.4.2 DACS Remote Terminal Units (RTU). As shown in Fig. 4-55, RTUs are located in each machinery room and on the main deck (reference Appendix C, Dwg. No. 5531921 and Dwg. No. 5531922). The RTU is designed such that for the main deck it will be enclosed in a standard rack-mounted enclosure and for the 2 RTUs located in the machinery rooms, NEMA-4X enclosures will be used. Details of RTU design are delineated in Appendix B, LMSC specification No. X5531908, Preliminary Performance specification, DACS PSD-I, 10-MW(e), MAPS. The RTUs are electronic units used to perform the following functions:

- o Multiplex, condition, and quantize sensor signals at remote locations for transmission to the Control Center CPU
- o Provide out-of-limits checking at the instrumented site
- o Perform a CPU unloading function
- o Activate local controllers upon receipt of CPU commands
- o Avoid the running of an excessive number of instrument wire pairs back to the control center.

Figure 4-55 presents the preliminary RTU configuration. For each RTU, analog and digital I/O modules, signal conditioning and I/O termination paneling will be customized for the individual application.

The concept of distributed RTUs effectively reduces the number of cables that would be required to be brought into the Control Center to two (2) twisted shielded pair from each RTU. Without RTUs, literally hundreds of wires would have to be run from the sensors to the Control Center. This concept can be implemented using the latest state-of-the-art, EIA RS-422 interface specification using line drivers and receivers. This is by far the most cost-effective approach when compared with the use of modems to transmit the sensor data to the Control Center.

The RTU operates under the control of the CPU, providing a flexible interface to all types of sensing and control equipment. Interrogation of the RTUs by the CPU is typically done by polling, with all RTUs listening simultaneously to detect their unique station address, and a sync code used to establish synchronization for receipt of the transmitted message. Upon recognizing its own address, the RTU accepts the remainder of the message, checks the validity of the transmission, performs the commanded operation, and transmits a reply. Commands may be either requests for data (status, analog sensor, or accumulator value), or control action (contact closure, or set point adjustment). The response message will contain either the data requested, or confirmation of the action performed in the case of control commands. Figure 4-56 is a logic flow diagram of the RTU operating sequence.

Data and control signal transmission between the RTUs and the Control Center will be on a EIA RS-422 compatible link. EIA RS-422 is a balanced voltage digital interface specification for communication of digital data over distances of 4000 ft or data rates up to 10 megabaud using a twisted-pair transmission line. This balanced configuration is isolated from ground noise currents. It is also immune to fluctuating voltage potentials between system ground references and to common mode electromagnetic interference. A family of monolithic low-power Schotky quad line drivers and receivers now exists to

Fig. 4-55 DACS RTU Block Diagram

4-159

Fig. 4-56 RTU Operating Sequence

satisfy the foregoing requirements, and they have been incorporated into the preliminary design. Figure 4-57 depicts the two wire balanced system data communication technique.

Fig. 4-57 Two Wire Balanced System Data Communication Technique

4.6.4.3 DACS Hardware Selection. The DACS hardware consists of an operator/DACS interface, a digital computer-oriented control center, and multiple Remote Terminal Units (RTU) connected to the control center computer via RS 422 compatible communications circuits. Figure 4-58 defines this configuration.

Major hardware elements of the DACS that have been selected are of off-the-shelf, state-of-the-art field-proven design. The selected equipment has been used extensively in various data acquisition and control system applications where continuous operation and reliability are important. The DACS equipment list for the 50-MW(e) Power Module delineated in Table 4.6-1 is also the DACS equipment list for the 10-MW(e) MAPS. A brief description of the selected hardware is given in Appendix I.

Fig. 4-58 Power Module Data Acquisition and Control System
General Configuration

4.6.4.4 DACS Software Selection. Software for the DACS will be comprised of standard field proven systems software for the MODCOMP II computer in conjunction with a set of custom applications programs to provide the data acquisition, control, processing and display functions of the 10-MW(e) MAPS facility. The software system is illustrated in Fig. 4-59. It centers on the MODCOMP MAX III disc-oriented, multi-programmable real-time operating system and includes:

- (a) MAX III Real-Time Operating System disc-oriented multi-programming system
- (b) FORTRAN IV Compiler - highly efficient real-time FORTRAN

Fig. 4-59 DACS Real-Time Operating Software

- (c) Macro-Assembler Full macro capability with extensive conditional assembly capability
- (d) Link Editor for combining program modules and generation of overlay structures

Table 4-38
PRELIMINARY DACS EQUIPMENT LIST, 50-MW(e) POWER MODULE

Quantity	Item	Model
1	MODCOMP 11/201 Processor	11/201
	• Direct Memory Processor	3704
	• Real-Time Clock	3743
	• 64K Memory	3670
	• Executive Features	3737
	• Multiply/Divide	3503
	• Hardware Fill	3742
	• Power Fail-Safe/Auto Start	3739
	• Memory Parity	3806
	• System Protect	0781
	• Interrupt Levels for Exec.	3730
	• Hardware Floating Point	3512
	• Controller for Programmer's Console Terminal	3752-2
	• CPU Panel	2
2	Peripheral Controller Interface	4903
1	2.6 Million Word Disc	4128-1
4	Disc Cartridge	4140
1	Asynchronous Terminal Interface	4807
	• 16 Full Duplex RS232-C Interfaces	
	• Selectable Baud Rates (75 to 19.2K)	
1	Punched Card Reader	4411
	• 300 opm	
1	Line Printer	4213
	• 50 to 150 lpm	
	• 132 Columns	
	• 64 ASCII Character Set	
1	Magnetic Tape Drive and Controller	4145-1
1	Magnetic Tape Drive	4151
4	Standard 19-in. Cabinets	0001
1	Versatec Plotter	3000
	• 100 points/inch	
	• Full Graphics	
	• Printer Mode Operations	
	• 11-in. Paper With Program Selectable lengths	
1	Versatec Controller	4218
3	Intecolor Color Graphic Intelligent Terminals	8001
	• RS-232C I/O Interface	
	• 75 to 9600 Baud	
	• 48-Line x 80 Characters	
	• 8080 CPU	
	• Foreground/Background, 8 Colors	
	• 19-in. Diagonal CRT	
	• 16 Function Keys	
	• Graphic Plot (160 x 192 points)	
1	T.I. Keyboard Printer	743
	• 30 Character/Sec	
	• 8-1/2-in., 72 Char./Line	
	• Standard ASCII Code	
3	Remote Terminal Units	LEC
	• Standard RS-232C Interface	
	• Programmable (microprocessor oriented)	
	• Flexible Instrumentation Interfacing	
	• Water/Dust/Gas/Explosion Proof Enclosure	
1	Control and Monitoring Console	EMCOR
1	Programmer's Entry Console	EMCOR
2	Test Conductor Chairs	EMCOR
3	Mounting Isolators	Acroflex

- (e) Source Editor text editing capability for manipulation of source code
- (f) Disc Cataloguer for maintenance of load module files on mass storage
- (g) Debug Program - for on-line checkout
- (h) Program Loader - loading and execution of programs and overlays
- (i) Math and Utility Libraries library of available routines including real-time services
- (j) Processor, Memory and Peripheral Device Diagnostic Programs
Routines for all hardware devices

The software package for the MODCOMP II is designed to work as a highly efficient yet flexible system. The MAX III operating system supports the program generation software listed above as well as the customized software developed for the 10-MW(e) MAPS applications. As such, MAX III will be the nucleus of the real-time software package, providing numerous centralized services that are shared by all of the applications programs.

The computer system diagnostic routines provided will give extensive fault location capabilities. The maintenance operator can perform auto-testing of a subsystem, repetitive execution of a part of a test, or repetitive execution of a user defined diagnostic sequence. All options, including hold patterns and read-back of the operator inputs are provided. Selective testing, options and printing of error messages and operator instructions are controlled by CPU sensed switch settings.

Control Center Applications Software. The applications programs will perform all currently required functions and are modularly structured to facilitate expansion to include additional inputs and/or additional functions. The applications software will include several components that function on a real-time basis.

Required Functions. The overall requirements of the DACS dictate that the following functions shall be provided:

- a. Operator Interface DACS will provide an input/output/display capability for the power system operators to monitor and control the MAPS in an interactive mode.
- b. Data Acquisition The system will acquire system performance and diagnostic data through the RTUs during MAPS operations.
- c. Monitor and Alarm The system will monitor the operational data and determine alarm conditions and output alarm messages on operator console CRTs.
- d. Display The DACS will provide for the display of alarm, test events and graphic diagrams of significant test functions on color CRTs in various formats on demand or under timer control.
- e. Data Logging The DACS will provide a capability to log operational and diagnostic data on magnetic tape for analysis. Also, the capability to copy performance and diagnostic data magnetic tapes for analysis at other processing facilities will be provided.
- f. Plotting The DACS will provide real-time plotting on the graphics plotter of time history for selected parameters.
- g. Supervisory Control The DACS will provide the capability to output open and closed loop control over the MAPS facility and various subsystems.
- h. Quick Look Analysis The DACS will provide the capability to perform analysis and processing of operating/diagnostic data.
- i. Power Fail/Auto Restart The DACS will be implemented to provide for an orderly shutdown in the event of power system outage with automatic restart after power system recovery.

Applications Software Elements. Applications software elements (modules) are provided as follows:

- a. System Startup
 - o Initialization and Recovery
 - o Operator Control

b. Preoperation

- o Calibration of Sensors
- o Operations Parameter Input
- o Data Base Configuration
- o DACS Checkout

c. Operations

- o Parameter Initialization
- o Scan Initialization
- o Data Acquisition
- o Graphics Display
- o Data Plotting
- o Logging of Performance and Diagnostic Data and System Operations
- o Limit Checking, Status and Alarm
- o Closed and Open Loop Control
- o Printed Reports

d. Post-Operation

- o Data Delog
- o Data Tape Copy
- o Data Plotting
- o Recalibration of Sensors

RTU Software. The RTU software provided for the DACS remote data acquisition and control terminals will interface with the DACS Control Center computer over dedicated communication lines.

The RTU will accept from the Control Center computer:

- o Data base information
- o Point scan lists
- o Requests for point(s) status
- o Perform function commands
- o Reinitialize RTU commands
- o Loop test mode commands

RTU software will provide the following capabilities:

- o Communications interface to the Control Center
- o Periodic scan of sensor points in scan list
- o Status information on request
 - Data from all sensors
 - Change data only
 - Specific point data
- o Scan rate change command
- o Initiate or stop periodic scan on request
- o Perform control commands (point set and discrete) on request

In a normal operational environment, the RTU will accept an initial data base and scan list from the Control Center computer. Upon receipt of an initiate scan command from the Control Center, the RTU will acquire data from specified points and retain that data until the next periodic scan. Receipt of a request for scan data from CPU will result in the data requested being transmitted to the Control Center. Periodic scanning continues until the RTU is commanded to stop. Point set and discrete commands are processed upon receipt.

The RTU may be initialized with a new data base and scan list information upon request by the Control Center computer.

The RTU software will be comprised of two major elements:

- o Systems software
- o Applications software

RTU System Software Modules. Remote terminal unit (RTU) executive software modules will include the following:

- o System initialization
- o Real-time clock handler
- o Interrupt service routines
- o Task scheduling

- o Executive services
- o Input/output handler (includes error detection on input; error protection on output)

Applications Software Modules. The following applications software modules and their functions are included:

- (a) Message Validation and Routing Module
 - o Decodes messages from Control Center
 - o Validates messages from Control Center; sends error messages when applicable
 - o Acknowledges Control Center messages
 - o Routes messages to appropriate processor (via exec task queueing)
- (b) Data Base Management Processor
 - o Accepts data base input from the Control Center and builds data base
 - o Updates data base as per sensor scan
 - o Outputs change data to the Control Center
 - o Outputs data base to the Control Center on request
- (c) Point Scan/Limit Check Processor
 - o Performs periodic sensor scan
 - o Notes alarm conditions
 - o Updates data base via the Data Base Management Processor (DBMP)
 - o Outputs change data to the Control Center via DBMP
- (d) Point Set Processor
 - o Accepts and outputs point set commands to controllers
 - o Notes set or fail status of control commands, permitting retries and/or the issuance of command failure reports
- (e) Discrete Command Processor
 - o Accepts and outputs discrete commands
 - o Notes set or fail status of control commands, permitting retries and/or the issuance of command failure reports.

4.7 MATERIAL SELECTION

This section presents a discussion of the materials selected for the Commercial Size Power Module, the MAPS and the test articles. The heat exchanger material selection is the primary factor and dictates the selection of compatible materials for the seawater and ammonia subsystems.

4.7.1 Heat Exchangers

During the conceptual design and based on mechanical properties, producibility and corrosion resistance, aluminum was selected as the best material to construct the heat exchangers. These characteristics resulted in aluminum being the lowest cost material for a 30 year life OTEC plant.

The selection of aluminum as the heat exchanger material is based on corrosion test data presented in the conceptual design report plus additional corrosion test data on aluminum alloys 110, 3003, 3003 Alclad, and 5052 in the form of tubing. The additional corrosion data are from two separate test programs performed at the F.L. la Que Corrosion Laboratory of the International Nickel Co. (INCO) at Harbor Island, North Carolina. One program was for the Naval Ship Research and Development Center (NSRDC) at Annapolis from 1951 through 1954 and the other for the Aluminum Company of Canada (ALCAN). There are also data on the performance of aluminum tubing and piping that had eight years of service in Canadian Mine Sweepers.

The NSRDC test results reveal a maximum pit depth for 3003 tubing of 0.040 in. after 42 months exposure to continuous 10 ft/sec seawater. The Alclad 3003 had a maximum pit depth of 0.019 in. for 24 months at 10 ft/sec. It must be noted that these test tubes were supplied with seawater that had previously passed through copper-nickel components which may have contributed to corrosion of the tubing. The conclusions reached from the NSRDC Report (E.E.S. Report 040037A(s) dated 28 Feb. 1955) aluminum tubing corrosion tests are as follows:

- o The 3S (3003) and Alclad 3S (Alclad 3003) aluminum alloys have higher resistance to pitting than 52S (5052) aluminum alloy.
- o Resistance to general corrosion and to corrosion-erosion of the 3S, Alclad 3S and 52S is considered good.
- o The life expectancy of the 3S and Alclad 3S pipes is estimated to be at least 10 years under the conditions of test.

The aluminum alloy tubing tested by NSRDG at Harbor Island was standard 1 in. IPS pipe with a 0.133 in. wall thickness. The importance of the NSRDC tests is that the Alclad tubing had no evidence of pitting of the core material after 24 months with less than 50 percent of the cladding being corroded. This experience is similar to a Canadian Mine Sweeper report by Chinn & Rogers Royal Navy Scientific Service 19 (1) 3S (1963) wherein Alclad 3003 tubing and pipe were completely free of pitting of the core material after 4 years service and the cladding loss (0.013 in. thick) was quite low. As of the date of the Chinn & Rogers report (1963), the tubing and pipe had been in service for eight years without a failure. Examinations of removed and sectioned tubing revealed the absence of pitting of the core material even when most of the cladding had been consumed. Chinn and Rogers concluded "that the service life of the tubing and pipe in 5 to 10 ft/sec flowing seawater would be in the ten year range."

The ALCAN aluminum corrosion test performed at Harbor Island used ALCAN 1S alloy pipe varying in diameter from 1/2 in. to 3 in. in 10 ft lengths arranged in a single line with flow entering the 3 in. diameter end. Hard rubber streamlined fittings were used at each joint to minimize turbulence. The flow rate varied from 5 to 36 gal/min over a 3 year test period but was 15 gal/min for the majority of the test.

At the end of the 3 year test program, 5 ft of each diameter (1/2 in., 3/4 in., 1 in., 1-1/2 in., 2 in. 2-1/2 in., and 3 in.) was sectioned and pitting attack evaluated. The maximum pit depth in the pipe exposed to OTEC type seawater velocities (4 to 7 ft/sec) was 0.035 in. Using an extreme value analysis of the pit depth data (Gumbel - "Statistics of Extremes" Columbia Univ. Press 1958) it was concluded that pitting attack is at a

minimum at 4.5 ft/sec. It was further concluded (using the extreme value analysis) that there would be no more than one pit that exceeded 0.070 in. in 40,000 ft of tubing exposed to 4.5 ft/sec flowing seawater.

The above described corrosion test results lend strong support to the use of ALCLAD 3003 or 5052 tubing for OTEC heat exchangers. The use of ALCLAD 3003 with a wall thickness of 0.10 in. and a clad thickness of 0.10 in. to 0.01 in. should provide a minimum of ten years life with a failure rate of less than 3 percent.

The use of Alclad-3003 versus Alloy 5052 needs to be resolved on a heat transfer measurement basis since the corrosion products of the cladding may enhance (promote turbulence) or impede (insulate) heat transfer. Data from DOE bio-fouling and corrosion test programs at Keahole Point and Panama City and LMSC programs at San Diego should resolve this question.

The conceptual design report selected 5052 aluminum tubing with a wall thickness of 0.075 in. as adequate for a ten year life heat exchanger. The corrosion test data on tubing discussed in the previous paragraphs support the selection of a thicker walled tubing such as 0.10 in. to ensure a life of 10 to 12 years. Even at this increased thickness, aluminum heat exchangers result in the most cost-effective power system.

The 50-MW(e) module preliminary design maintains the 5052 aluminum alloy material. Inquiries to date have indicated that the tubing will be drawn and have a circumferential fin upset on the outside of the tube. If rolled and welded tubing is to be considered, manufacturing development must be performed.

The shell and tubesheets are 5083 aluminum alloy which has a relatively high strength and a good resistance to seawater. It was assumed that all the heat exchanger internals would also be the 5083 alloy. However, a detailed analysis of the support plates, ammonia piping, and other internals would probably show that a less expensive alloy with a lower strength would be acceptable.

For example, the base price of 3003 alloy is about 10 percent less than the 5083, and it is more readily available. It can be welded to the 5083 shell where necessary.

The 10-MW(e) MAPS preliminary design tubing material choice of titanium was directed by DOE. Because of this choice, it will be necessary to review the design of the entire seawater system to ensure materials compatibility and thereby prevent galvanic corrosion. Galvanic corrosion will occur (1) if materials such as aluminum, steel, or certain copper-based alloys are used in components exposed to the seawater flowing through the titanium heat exchanger, and (2) if these components are electrically coupled to the titanium. Galvanic corrosion can be controlled through the use of cathodic protection via sacrificial anodes or impressed current and/or paint coatings. Insulation (plastic or rubber gaskets, sleeves, washers, etc.) can also be effective, but this approach is not considered as reliable as the combination of cathodic protection and paint coatings. The design of the various components in the seawater system such as piping, valves, and pumps will be reviewed to select materials for these components that will be compatible with the titanium heat exchangers. Where this is not possible, the corrosion control measures previously discussed will be applied.

Materials of Construction: As directed by DOE, titanium tubing will be used in the Modular Application heat exchanger. Welded tubing per ASME SB-338 Grade 2 will be used since it is lower in cost than seamless. Welded tubing can be finned by the upset method by several vendors.

The tubesheet must be titanium or titanium clad steel in order to weld the tubes for a positive seal. For the selected tubesheet thickness, a solid titanium tubesheet would be about 15 percent less expensive than the clad tubesheet. However, the shell transition ring would then have to be clad since titanium cannot be welded to carbon steel. It is better technically to weld the thin wall tube to the clad with a small weld than to weld the heavier tubesheet to the clad transition ring without damaging the cladding. The thinnest titanium clad that is available in 8-ft widths is 3/16 in. Therefore,

the tubesheets are A516, Gr 70, carbon steel with a 3/16 in. detaclad titanium facing. No credit is taken for the titanium in the stress calculations. For a nominal extra charge, the clad tubesheets will be 100 percent ultrasonically tested by the vendor to ensure an acceptable bond between the clad and the base material. The tubesheet sections are limited to the 92 in. width which is controlled by the available size of titanium plate. The detaclad titanium tubesheet is a standard product which is produced by at least two vendors.

The shell plate and other pressure boundary materials are SA-516, Gr 70, carbon steel and will be certified to ASME standards. For internal components, such as support plates, the more economical SA-36 carbon steel was chosen.

4.7.2 Turbine-Generator

The radial inflow turbine material selection is identical for the 50-MW(e) and the 10-MW(e) systems. All recommended materials have been selected after careful study and appraisal of expected operating conditions. Table 4-39 shows the material recommendations as well as the possible backup selections.

Table 4-39
MATERIAL RECOMMENDATIONS AND BACKUP SELECTIONS

<u>Machine Part</u>	<u>Recommended Material</u>	<u>Backup Selection</u>
Casing	316L Stainless Steel	304L Stainless Steel
Shaft	17-4 PH Stainless Steel	316L Stainless Steel
Wheels	C.355 Aluminum	KO-IT6 Al Alloy
Bearings	6061 Al Alloy	
Nozzles	17-4 PH Stainless Steel	316L Stainless Steel
Lab. Seals	6061 Al Alloy	
O-Rings	Chloroprene	Nitrile

The above selection of alloys for casings, bearings, etc. is well established. The choice of material for the expander wheel is between Alloys 201, 206 and C355.

In this size of casting, the most important concerns are obtaining good ductility and good strength. The alloy used in the majority of such wheels up to this time is Alloy 201, otherwise known as K-01 aluminum. This alloy has been used as a base for further developments, and an alloy known as 206 is the result. Both are based on modifications to the French Alloy AU5GT-0T4, an alloy with 50 years of successful use in Europe.

The prime advantage of the 201 and 206 alloys is their good ductility and toughness when heavy chilling and rapid solidification to maintain maximum properties is not practical. Use of any other popular alloys, such as C.355, is possible with some attention to quenching to achieve needed properties. However, rotors up to 53 in. in diameter have been cast successfully in this material. Fifty inch expanders and compressors are being manufactured in C.355 for a large Air Frame Refrigeration Turbine application. The design and testing will be well established prior to OTEC going into service and the horsepower rating per turbine shell is very similar, i.e., 10,000 versus 9,700 hp design.

Water cooling tubes, cast into the wheel, are necessary in order to achieve elongations on the order of 2 to 4 percent.

The axial flow turbine material selection is identical for the 50-MW(e) and the 10-MW(e) systems, with possible variations in composition as the result of detailed stress analysis. Materials were selected with the operational environment and ammonia working fluid in mind.

Table 4-40 shows the recommended material for the various components.

4.7.3 Ammonia Cycle and Auxiliaries

4.7.3.1 Ammonia. Anhydrous ammonia will be the working fluid used in the OTEC power cycle. It is required that the power system be supplied with C-grade ammonia specified as "Minimum 99.5 percent pure" (by weight). Of interest to OTEC, ammonia is produced and available on the west coast (Helms,

Table 4-40
TURBINE-GENERATOR RECOMMENDED MATERIALS

<u>Item</u>	<u>Recommended Material</u>
Generator Housing	Carbon Steel
Generator Rotor	Carbon Steel
Winding Materials	Copper
Turbine Inlet Scroll	17-4 PH Stainless Steel
Turbine Stator Blades	Stainless Steel
Support Ring	Stainless Steel
Turbine Disk	Stainless Steel
Turbine Blades	Stainless Steel
Seal Cartridge	Carbon Steel
Bearing Cartridge	Carbon Steel (Babbitt faced pads)

and Richmond, California, for example) and on the Gulf Coast (Donaldson, Louisiana). To obtain the best price, ammonia will be purchased and delivered in tank-car (26 tons or larger) or tank wagon lots (approximately 20 tons). The price of 99.5 percent purity ammonia in these quantities as of June 1978, was \$130/ton on the Gulf Coast and \$160/ton on the West Coast, both FOB the above points of production. The difference between tank car and tank wagon for delivery is only in off-loading capability. Tank car delivery requires terminal facilities to pump the ammonia from the car, whereas tank-wagon carries its own pumping and transfer equipment. Both require storage facilities at the off-loading site with a capacity of about 20 ton minimum.

Grades of Ammonia. As reported by USS Agri-Chemicals, in producing ammonia the producer aims for optimum purity (99.999 percent) at all times, and 99.5 percent purity is considered the minimum acceptable output. Producer grades of ammonia go under various labels:

- o Minimum 99.998 percent pure — variously called premium, refrigeration, metallurgical, industrial, or electronic grade. Currently this grade sells for about \$360/ton, but there is indication it could be negotiated for as low as \$181/ton in tank car lots.

- o Minimum 99.9 percent pure -- anhydrous
- o Minimum 99.5 percent pure -- C-grade, commercial, etc.

In the procurement of ammonia, it is necessary to compare current price with the degree of purity (minimum) required and then to specify both purity and price due to the ambiguity of terminology and pricing policy. As stated, the producer aims for optimum purity; so depending on his production run and the market, if the minimum purity of 99.5 percent is specified, the delivered product may be anywhere in the range of 99.5 to 99.999 percent pure.

The term "commercial grade" and the specification of grades lower than 99.5 percent purity (hydrated ammonia) should be avoided for the following reasons:

Commercial grade is often used synonymously with "fertilizer grade" or agriculture grade, and may vary between 95 and 99.98 percent purity. If obtained directly from the producer, it may be the same as "C-grade" as described above. However, for agriculture purposes it is almost always diluted with water for agriculture handling, and often an extra charge is made for this dilution.

For OTEC power system operations, system performance is degraded as working fluid purity decreases. Purity less than 95 percent causes significant thermodynamic degradation and loss of power. To remove excess water from the ammonia, the power system incorporates an ammonia cleanup system. This cleanup system has the ability to maintain working fluid purity of 95 percent or better based on maximum in-leakage projections. Thus, the system has the ability to accept ammonia of lesser purity than specified above if cost and availability justify this action.

Contamination. The small amount of contamination of ammonia from a producer is primarily water with possibly some oil. Seawater in-leakage is expected to be the principal source of ammonia contamination in OTEC operations. A literature search and analysis of the possible effects of seawater contamination

was conducted principally to determine the chemical and physical effects on liquid ammonia. It appears that with the presence of even negligible quantities of water in ammonia, stress corrosion cracking of improperly treated materials may be enhanced. The presence of large quantities of water in the ammonia (in excess of 5 percent) encourages other types of corrosive action on susceptible materials. The analysis conducted, leads to the conclusion that the leakage of small amounts of seawater on the physical properties of liquid ammonia, e.g., density, freezing point, boiling point, etc., will not degrade OTEC performance more significantly than equivalent quantities of fresh water.

Seawater contains many constituents. It was desirable to determine if any of these could react adversely with liquid ammonia. The major constituents are chlorides and sulphates of alkali and alkaline earth metals, i.e., NaCl, KCl, $MgCl_2$, $CaSO_4$, $CaCl_2$, etc., as shown in Table 4-41. No interaction of these salts with liquid ammonia has been identified. The solubility of some of these salts and others are given in Table 4-42.

If liquid ammonia contains 2 percent seawater content (by volume), then it follows from Table 4-42, all the alkali metal salts will dissolve while alkaline earth metal compounds will not.

Although water is extremely soluble in ammonia, interaction is small. Aqueous solutions are commonly referred to as solutions of the weak base NH_4OH (ammonia hydroxide). There is no evidence that undissociated NH_4OH exists. Solutions of ammonia are best described as NH_3 (aq), with the equilibrium written as NH_3 (aq) + $H_2O = NH_4^+ + OH^-$; $K_{298\text{ K}}$ (equilibrium constant) with

$$K_{298\text{ K}} = \frac{[NH_4^+][OH^-]}{[NH_3]} = 1.81 \times 10^{-5} \quad (pK_b = 4.75)$$

Table 4-41

COMPOSITION CONSTITUENTS IN SEAWATER IN SEAWATER HAVING A
CHLORINITY OF 19 PERCENT

Elements	(g/kg)	(g/unit of chlorinity)
Chloride, Cl^-	18,890	0.99894
Sodium, Na^+	10.560	0.5556
Magnesium, Mg^{++}	1.273	0.06695
Sulfate, SO_4^{--}	2.649	0.1394
Calcium, Ca^{++}	0.4104	0.02106
	mg/kg	
Carbon (as HCO_3^- or CO_3^{--})	28	0.00735
Bromide, Br^-	65.9	0.00340
Strontium, Sr^{++}	8.1	0.00070
Boron, As II, BO_3	4.6	0.00137
Silicon, as silicate	0.01-4.5	
Fluoride, F^-	1.4	0.00007
Nitrogen, as NO_3^-	0.01-0.80	
Aluminum, Al^{+3}	0.5	
Rubidium, Rb^+	0.2	
Lithium, Li^+	0.1	
Phosphorus, As PO_4^{--3}	0.001-0.1	

Trace elements present in concentrations of 1 to 50 $\mu\text{g/kg}$: barium, iodine, arsenic, iron, manganese, copper, zinc, lead, selenium, caesium, uranium.

Trace elements present in concentrations of less than 1 $\mu\text{g/kg}$: molybdenum, thorium, cerium, silver, vanadium, lanthanum, yttrium, nickel, scandium, cobalt, cadmium, mercury, gold, tin, chromium, radium.

Table 4-42*

* Solubility given in moles of solute per 10 moles of NH_3 . * Solubility listed at -33.4° unless otherwise given.
 * K. Frenkel'tagen, *Z. Physik. Chem.*, **128**, 1, 239 (1927). * W. C. Johnson and O. F. Krumholz, *Z. Physik. Chem.*, **167**, 240 (1931).
 * G. S. Forbes and C. F. Norton, *J. Am. Chem. Soc.*, **48**, 2278 (1926). * H. Hunt and L. Boney, *J. Am. Chem. Soc.*, **55**, 3528 (1933).
 * M. Linhard and M. Stephan, *Z. Physik. Chem.*, **167**, 67 (1933). * H. Hunt, *J. Am. Chem. Soc.*, **54**, 3509 (1932).
 * P. C. Scherer, Jr., *J. Am. Chem. Soc.*, **53**, 3694 (1931). * M. Linhard and M. Stephan, *Z. Physik. Chem.*, **163**, 105 (1933).
 * E. C. Franklin and C. A. Kraus, *Am. Chem. J.*, **20**, 820 (1898). * G. Gore, *Proc. Roy. Soc. (London)*, **20**, 411 (1872).
 * A. M. Monastirskii and V. A. Pleskov, *Z. Physik. Chem.*, **156**, 176 (1931); *Zh. Fiz. Khim.*, **3**, 221 (1932). * B. Kuziloff, *Z. Physik. Chem.*, **25**, 107 (1896).
 * A. I. Shatenstein, "Liquified Gases as Solvents," Govt. Chemical-Tech. Pub. Office, Leningrad Div., 1934.
 * G. F. Smith and E. G. Koch, *Z. Anorg. Allgem. Chem.*, **223**, 17 (1935).
 * B. B. Vasil'ev, Ya. E. Seferovich, and V. M. Fridman, *Zh. Khim. Prom.*, **14**, 719, 818 (1937). * C. A. Kraus, *J. Am. Chem. Soc.*, **44**, 1216 (1922).
 * M. A. Portnov and A. M. Churavlev, *Z. Anorg. Allgem. Chem.*, **223**, 45 (1933). * E. C. Franklin, *J. Am. Chem. Soc.*, **37**, 2795 (1915).

* J.J. Lagonski and G.A. Moczygemba, "Liquid Ammonia," The Chemistry of Non-Aqueous Solvents, J.J. Lagonski, ed., Academic Press, New York, N.Y. 1967

In other words, with up to 2 percent water content, the formation of NH_4OH is negligible and the interaction products will mostly remain as $\text{NH}_3\text{H}_2\text{O}$ or $2 \text{ NH}_3\text{H}_2\text{O}$ (hydrated ammonia). Thus, it is concluded that seawater will have only minor physical effects on liquid ammonia, and will effect OTEC performance similar to a like quantity of fresh water.

4.7.3.2 Piping Material Considerations. In ammonia vapor and liquid ammonia systems, carbon steel and cast iron are generally suitable for anhydrous ammonia service. However, some uncertainties exist with air contaminated anhydrous ammonia and seawater contaminated ammonia. The corrosion and deposition effects caused by the seawater content are not clearly understood at this time. It is recommended that test programs be conducted to determine the effect of seawater inleakage on material selection.

Carbon steel is subject to stress corrosion cracking in air contaminated anhydrous ammonia. It is not likely that the main cycle piping and components will be subjected to air contamination except during initial startup, shutdown for maintenance, and restart, because air and nitrogen are purged from the system by the noncondensable gas removal subsystem. Stress corrosion cracking can be prevented by post weld stress relieving (at 1100°F minimum) parts containing high residual fabrication stresses. Another method of corrosion control used in ammonia systems is to add 0.2 percent water to the ammonia. In the OTEC power system, however, water is considered a liability for thermodynamic reasons. In addition, seawater concentrations in the ammonia system can be expected to vary from very low levels in the vapor and condensate pipe and the condenser to as high as 2 percent in the evaporator. Hence, post weld stress relieving is recommended for carbon steel piping.

Aluminum, particularly the 3000 and 5000 series alloys, is also suitable for ammonia service, the common application being in refrigeration systems and storage tanks. Laboratory tests have shown that pitting attack of aluminum coupled to steel is promoted if anhydrous ammonia is contaminated with air. These same tests indicate that water additions may reduce the galvanic attack. However, as discussed above, water additions are detrimental to the

operation of the OTEC cycle and, in fact, seawater contamination may promote galvanic attack. Any seawater inleakage test program should therefore consider corrosion effects on aluminum-steel couples.

In summary, aluminum, steel and titanium are suitable for anhydrous ammonia service, provided the ammonia is not contaminated with air or seawater. Air contamination is expected to be minimal with the system purging noncondensibles after startup. Seawater inleakage will be handled by the ammonia clean-up system. Concentrations of seawater should be extremely small throughout the system except in the evaporator and distribution pumps, piping, and valves. The conceptual design approach is therefore to use carbon steel piping with post weld stress relieving for all ammonia service with titanium specified in specific, high risk applications. Actual material selections are discussed in the sections describing specific systems.

4.7.3.3 Auxiliaries. However, a number of streams having seawater or ammonia/seawater mixtures are encountered in the ammonia cycle subsystems. Steel is not appropriate for these streams, mainly because higher water concentrations and operation at elevated temperatures create increased corrosion hazard. Titanium piping is specified in the ammonia cleanup subsystem for liquid streams having ammonia and seawater mixtures with water concentrations above 25 percent and for seawater at 320°F. For the ammonia cycle subsystems, lined carbon steel is specified for seawater service, and titanium tubing is chosen for subsystem heat exchangers having seawater on one side and ammonia on the other. In those areas where steel piping may come in contact with titanium in the ammonia cleanup system, the steel will be isolated from the titanium.

4.8 AVAILABILITY

The power system availability is the ratio of the time the power system is capable of service, whether or not it is in operation, to the total or base time. In the OTEC power system context it is the fraction or percent of hours per year in service or available for service. The availability is a summation of (1) the hours of operation over and above prescribed limits based on load and available ΔT , (2) the reliability or failure rate of equipment, time to repair failures and return of the power module to service, and (3) downtime for scheduled maintenance.

4.8.1 Performance Degradation

Several of the important functions which impair performance are biofouling, corrosion (causing early failure) and equipment wearout. The latter is discussed under maintenance.

4.8.1.1 Biofouling Countermeasures. The biofouling countermeasures system, for the heat exchangers, will consist of a chlorination system and the M.A.N. brush system. The chlorination system will electrolytically generate chlorine from seawater and will be designed to provide a maximum chlorine dosage capability of 0.5 ppm continuously to the warm seawater system. This chlorination system output will be diverted to the cold seawater system for a total of 30 minutes during each 24 hours. The intermittent chlorination schedule will be based on test data from DOE and LMSC biofouling countermeasures programs.

It is anticipated that 0.1 at 0.2 ppm continuous chlorination dosage, in combination with M.A.N. brushes being cycled a maximum of three times every 24 hours, will be adequate to maintain the thermal resistance due to biofouling below a value of $0.0003 \text{ (hr ft}^2 \text{ F)/Btu}$ in the heat exchangers.

The chlorine generator (chloropac) consists of polyvinylchloride (PVC) piping, fittings and valves with titanium electrodes that are guaranteed for five

years. Operation of the chlorinator is automatic and it can be remotely controlled. Chlorination schedules can be maintained via automatic monitoring of the discharge seawater. The results of this monitoring are fed back into the output of the generator to maintain a predetermined level of chlorine at the seawater outlet. The replacement of the titanium electrodes or PVC components of the chlorination system will require as little as five minutes to as long as a few hours. The M.A.N. brush system operates by simply reversing the flow of the seawater to move the brushes back and forth through the tubing. Retaining cages at each end of the heat exchanger tubes stop and hold the brush until the flow is reversed.

The M.A.N. brushes should provide more than 10 years of service. This is based on LMSC test data where a M.A.N. brush was cycled for a total of 194,376 ft of travel in natural seawater. The bristles of the brush show a rounding of the tip due to wear; however, when the diameter of the brush is compared to a new brush there is a difference of only a few thousandths of an inch. Further, an aluminum tube (5052 O temper) subjected to 11,740 cycles (23,480 trips) showed no wear when compared to an unused tube from the same piece of stock.

For a 55 ft-tube, where the brush is cycled three times per day, the brush showed a life of 1.6 years and for the more probable case of three cycles per week more than 11 years. The test brush had virtually the same resistance to travel through the tube after testing as it had at the start of testing indicating the same bristle diameter to tube inside diameter interference.

4.8.1.2 Corrosion Control Methods. The corrosion control methods to be used on the power module will consist of paint coatings and cathodic protection to prevent corrosion of the anodic materials such as the shell and tubesheet weldments. The paint coating material will be inorganic zinc primer with 2 epoxy coal tar top coats for a total dry film thickness of 17 to 20 mils. If properly applied, this coating should have a life of at least 5 years.

The cathodic protection system consists of aluminum sacrificial anodes fastened to the structure by welding at key defined points in the waterducts, around the machine rooms and on the heat exchanger shell. Design life estimates of the anodes are 10 years. There are no maintenance requirements for the anodes and, replacement of a depleted anode can be accomplished in less than one hours. (See Appendix D for brochure sheets on the sacrificial anodes.)

For the 10-MW(e) heat exchanger's titanium clad tubesheets, the plan is to use 33 of the BTZ-27 zinc anodes made by Federated Metal Division of ASARCO. Assumptions were 3 ft/sec duct seawater velocity, 5-year life of anodes using 10 milliamperes/ft². These anodes, BTZ-27, will virtually cover all the steel weld joints. The 1/2 in. diam. core rod is welded to steel tubesheet. The composition in accordance with MIL-A-18001 is:

	<u>Percent</u>
Lead	0.006 maximum
Iron	0.005 maximum
Cadmium	0.015 - 0.025
Copper	0.005 maximum
Aluminum	0.10 - 0.50
Silicon	0.125 maximum
Zinc	Balance

For the exterior surface of the heat exchangers such as piping, machinery room, etc., aluminum anodes such as the high efficiency "Reynodes" produced by Reynolds, or "Galvanlum" produced by Dow, will be used. The chemical composition is proprietary so anode composition cannot be given.

A 10-year life requires 6,500 lb of anodes of various sizes to meet the configuration of the heat exchangers, piping and their arrangement. An example of one that is presently proposed is ASARCO C360.

4.8.2 Maintenance Plan

The maintenance approach for the power modules includes achieving the required plant (module) availability by periodic maintenance and minimizing the time to

repair failed equipment. Seven levels of scheduled maintenance have been established, with the higher levels occurring less frequently and having more intensive maintenance activity. The frequency of occurrence and allowable plant downtime allocated to each maintenance level is such that the overall availability will be 90 percent or greater. Preventive or scheduled maintenance is performed to keep the equipment in a satisfactory operational condition by providing systematic cleaning, adjustment, replacement, and refurbishment of components before they are expected to fail. Repair maintenance carried out on a non-scheduled basis to restore a component to a satisfactory condition by providing immediate correction of a failure after it has occurred.

The seven levels of scheduled maintenance are shown in Table 4-43. The activities are arranged such that whenever a higher level is carried out, all lower level activities are included in the same maintenance period. The integrated plan availability with this maintenance schedule (assuming an instantaneous availability of 0, 0.75, or 0.90, respectively, while the plant is at 0, 75 percent or 90 percent output) is 0.910 (Table 4-43, Note 2), to 0.920 (Table 4-43, Note 1) over a ten year period including heat exchanger changeout.

If the OTEC plant contains several power modules, it may be economically attractive to stock a spare module, which is exchanged for an operational module that is due for its 36- to 40-month overhaul. The replaced module is then refurbished in a shipyard and becomes available as a spare. By staggering the 36- to 40-month exchanges, the 50 or 60 day overhaul downtime becomes an eight day changeout period. In this case the module availability would be 0.963.

The maximum allowable module power reduction will usually not occur when maintenance is actually carried out, for one of three reasons: (1) many major components are redundant and one of the set can be removed from service without affecting plant operation, (2) some systems operate intermittently and can be serviced when not operating, and (3) some systems are not critical to plant

Table 4-43
SCHEDULED MAINTENANCE LEVELS

LEVEL	DESCRIPTION	FREQUENCY	MAXIMUM MODULE POWER REDUCTION
LEVEL 1	FILL/CHANGE/DIAGNOSTIC TEST	WEEKLY/1 DAY DURATION	NO POWER REDUCTION
LEVEL 2	ADJUST/CLEAN	MONTHLY/2 DAYS DURATION	90% POWER FOR 12 HOURS
LEVEL 3	ADJUST/CHANGE	3 MONTHS/3 DAYS DURATION	0 POWER - 24 HOURS 75% POWER - 12 HOURS 90% POWER - 12 HOURS
LEVEL 4	MINOR REPAIR/REPLACE	6 MONTHS/5 DAYS DURATION	0 POWER - 3 DAYS 75% POWER - 1 DAY
LEVEL 5	REPAIR/REPLACE/REFIT	12 MONTHS/10 DAYS DURATION	0 POWER - 5 DAYS 75% POWER - 2 DAYS
LEVEL 6	OVERHAUL/REFURBISH	36 MONTHS	0 POWER - 50 DAYS (1) 0 POWER - 60 DAYS (2)
LEVEL 7	HEAT EXCHANGER CHANGEOUT	10 YEARS/90 DAYS	0 POWER - 90 DAYS

operation and can be bypassed or removed from service for the duration of maintenance with no impact on plant operation. As a result, availability could be higher than the minimum resulting from the schedule in Table 4-43.

Effectiveness of the Maintenance Approach: The workability of the plan depends on the effectiveness of preventive maintenance in heading off unscheduled plant downtime and the ability of maintenance crews to perform the work in the allotted period. The effectiveness of maintenance depends in turn on having adequate component reliability to span the duration between maintenance periods, and the ability to identify components requiring attention at each maintenance period, either by preplanning based on known component characteristics or by diagnostic tests made while the plant is in operation. Performing maintenance in the allotted time requires that the longest MTTR fit within the maintenance period, that adequate personnel be available to perform the required maintenance tasks in parallel, that spares be available, and that work space and equipment be adequate. All of these factors have been considered for the various subsystem descriptions as provided at the preliminary design level. The conclusion is that the maintainability approach is conservative and the requirement of 90 percent availability at 60 percent confidence level will be met or exceeded.

Maintenance Options: The basic schedule presented in Table 4-43 can be modified through the following options:

- o An automated operational concept for the power module where the module is operated unattended so that Level 1 is omitted and all necessary maintenance is accomplished at Level 2
- o A spare detachable power system module is used so that Level 6 activities are performed by exchanging modules allowing overhaul work to be done at a shipyard instead of at the platform; the downtime is, therefore, that required for module changeover only.

The automated option reduces operating cost (manpower), but has added expense for controls and signal transmission, and incurs the risk of plant downtime during the period for mobilization of a repair crew in response to an unscheduled repair call. This tradeoff can be resolved in the detailed design phase, provided that adequate definition of a plant operational scenario is available.

The effectiveness of the spare detachable module is an economic improvement resulting from cutting the 50 to 60 day downtime for the 36- to 40-month overhaul to an eight day changeover. Assuming that the module operates at full output following changeover, and electrical energy is sold at \$0.040/kWh, the added revenue over the thirty year life of a module is shown in Table 4-44.

Table 4-44

ADDED REVENUE PER 10-MW(e) MODULE RESULTING FROM
MODULE EXCHANGE INSTEAD OF OVERHAUL

<u>Downtime for Overhaul (days)</u>	<u>Added Revenue Over 30-Year Module Life</u>
50	\$18,000,000
60	\$22,000,000

With a turnaround time of 60 days for module transit and refurbishment in a shipyard, and changeout every 36 months, a single spare module can service a system of up to 18 operating modules. The added revenue is offset by the cost of the spare module, the dedicated shipyard, and transit equipment. If these costs total, for example, \$2000/kW, then a spare module is cost-effective in a system having more than five or six operating modules.

Maintainability Assessment: A system-by-system assessment has been made, with results tabulated in Table 4-45. The table shows that MTBFs and MTTRs of major components are consistent with the planned operating duration between maintenance periods and downtime during maintenance. Personnel, equipment,

Table 4-45
SYSTEM MAINTAINABILITY ASSESSMENT

System/Component	Estimated MTTR	Maintenance Level	Duration Between Maintenance	Duration of Maintenance Period	Critical to Plant Operation	Servicing Requires Plant Downtime
Heat Exchanger (Unit)	10 d	6	3 yr	50 d	Yes	Yes
Heat Exchanger Tubes (1)	8 hr (1)	3	3 mo	1 d	No	Yes
Evaporator Liquid Distribution Assembly	2 d	4	6 mo	4 d	No	Yes
Turbine	1 d	5	12 mo	5 d	Yes	Yes
Ammonia Flow Control	3 d	5	12 mo	5 d	Yes	Yes
Generator	4 d	5	12 mo	5 d	Yes	Yes
Ammonia Vapor Stop Valves	3 d	5	12 mo	5 d	No	Yes
Ammonia Piping	4 d	7	10 yr	90 d	Yes	Yes
Ammonia Liquid Pumps	1 d	4	6 mo	1 d	Yes	No
Ammonia Liquid Control Valves	1 d	4	6 mo	1 d	Yes	Yes
Ammonia Liquid Stop Valves	1 d	4	6 mo	1 d	No	Yes
Seawater Pump	3 d	6	3 yr	50 d	No	Yes

Table 4-45 (cont.)
SYSTEM MAINTAINABILITY ASSESSMENT

System/Component	Estimated MTTR	Maintenance Level	Duration Between Maintenance	Duration of Maintenance Period	Critical to Plant Operation	Servicing Requires Plant Downtime
Seawater Pump Housing	10 d	6	3 yr	50 d	No	Yes
Noncondensable Gas Removal	1 d	*	N/A	N/A	No	No
Ammonia Cleanup System	1 d	*	N/A	N/A	No	No
Ammonia Storage and Fill System	1 d	*	N/A	N/A	No	No
Ammonia Unloading System	1 d	*	N/A	N/A	No	Yes
Electrical System (by component)	2 d	2	1 mo	2 d	No	No
Instrumentation (by component)	1 d	2	1 mo	2 d	No	No
Control System (by component)	1 d	3	3 mo	3 d	No	No
Corrosion Control	5 d	5	12 mo	5 d	No	No
Biofouling	1 d	4	6 mo	4 d	No	No

- (1) 3 percent of heat exchanger tubes assumed to develop leaks in 10-year heat exchanger life. Two divers, one in each water box are assumed capable of detecting and plugging six tubes per hour. Approximately 100 leaking tubes must be detected and plugged every 3 mo. Eight divers, two in each water box, can do this in about 4 hours. An additional 4 hours is allowed for rest periods.
- (2) Back-flushing of ammonia distribution system will be required periodically to remove particulate matter from orifices.
- * Maintenance scheduled during system nonoperating periods.

and space required to perform schedule maintenance will be determined during detailed design.

The table contains additional information useful to an assessment of maintenance impact on availability, with column headings explained as follows. Critical components are those required for plant operation, which, if not available, will cause plant shutdown. Where possible, such critical components will be redundant and the removal of one for servicing will have no operating time impact. Noncritical components are those which operate intermittently, have workaround opportunities, or can continue operation in a partially degraded condition without affecting plant output. Many of these can be scheduled for service on an opportunity basis between regular maintenance periods.

4.8.3 Reliability

The reliability of the power system rests on the probability that the power system will perform its intended function for the defined period of time under specified conditions. The factors which are not considered in reliability include: off design warm/cold water ΔT (seasonal or site dependent); platform functions not within limits, such as water supply not at given rates, or extreme platform motion, such as pitch, roll and heave; and the failure to follow a reasonable maintenance schedule.

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

Redundancy in four critical equipment areas provides a highly reliable power module.

- o Evaporator/condenser — tube contingency
- o Ammonia cycle — condensate and distribution pumps (standby redundant)
- o Control system — central processor and interfacing equipment
- o Seawater pumps — four pump configurations

The evaporator and condensor preliminary design incorporates approximately 7 percent excess tubes to provide margin for uncertainties or tolerances in tube wall thickness and material properties. During detail design, the percentage of excess tubes will be finalized when the tolerances for parts of the heat exchanger are established.

In the ammonia cycle, three 50-percent capacity pumps are used for condensate pumping (two will provide 100-percent flow); and in the ammonia distribution system at the evaporator, four 33 percent capacity pumps are provided (three will provide 100-percent ammonia flow to the evaporator). This redundancy ensures full flow capability in the ammonia liquid system with the loss of one pump in either system.

Where safety and life support systems are involved and/or where the loss of data or control may result in serious failures, critical control system componentry is made redundant. The control processor is redundant with automatic fail-over such that within a few seconds after failure the backup unit regains full surveillance and control. Key interfacing equipment is also redundant to ensure uninterrupted control.

The projected component failure rates are given in Table 4-46 and 4-47 for the CPM and MAPS, respectively. These data are gathered from many sources and are based on a mean failure rate. It should be noted that the mean time between failure rate incorporates a severity factor for application to the OTEC power system. Using generic failure rate distribution data and vendor supplied data on the turbines, an 11.5 year MTBF was determined. This value was degraded to 8 years for operation in the OTEC power system. The numbers also do not include all effects of redundancy incorporated into the design.

Table 4-46

50-MW(e) CPM CRITICAL COMPONENT FAILURE RATES

Item	MTBF (yr)	Failure Rate (failure/yr)	MTTR (days)
Evaporator			
- Tube	12	0.08333	3
- Shell	20	0.05000	6
- Tubesheet, Baffles	15	0.06667	4
- Liquid Distribution	15	0.06667	4
- Mist Extractor	20	0.05000	3
Condenser			
- Tube	12	0.08333	3
- Shell	20	0.05000	6
- Tubesheet, Baffles	15	0.06667	4
- Noncondensable Gas Removal	15	0.06667	4
Turbine	8	0.1250	1
Generator	10	0.1000	4
Ammonia Vapor			
- Piping	40	0.0750	3
- Valves/Controls	10	0.1000	3
Ammonia Liquid			
- Piping	40	0.0250	4
- Valves/Controls	10	0.1000	3
- Pumps	10	0.1000	3
Ammonia Auxiliaries	8	0.1250	2
Control System	8	0.125	1
Fouling Countermeasures	10	0.100	3
Corrosion Control	10	0.100	2
Auxiliaries			
- Personnel	-		2
Structure	50	0.020	10
		1.628	

Table 4-47

10-MW(e) MAPS CRITICAL COMPONENT FAILURE RATES

Item	MTBF (yr)	Failure Rate (failure/yr)	MTTR (days)
Evaporator			
- Tube	30	0.033	3
- Shell	30	0.033	6
- Tubesheet, Baffles	30	0.033	6
- Liquid Distribution	30	0.033	4
- Mist Extractor	30	0.033	3
Condenser			
- Tube	30	0.033	3
- Shell	30	0.033	6
- Tubesheet, Baffles	30	0.033	4
- Noncondensable Gas Removal	30	0.033	4
Turbine	8	0.125	1
Generator	10	0.100	4
Ammonia Vapor			
- Piping	40	0.025	3
- Valves/Controls	10	0.100	4
Ammonia Liquid			
- Piping	40	0.025	4
- Valves/Controls	10	0.100	3
- Pumps	10	0.100	3
Ammonia Auxiliaries	8	0.125	2
Control System	4	0.025	1
Fouling Countermeasures	6	0.167	3
Corrosion Control	10	0.100	2
Auxiliaries			
- Personnel	10	0.100	2
Structure	50	0.020	10
		1.701	

4.9 PRODUCIBILITY

The approach to the design of the commercial power system has been to apply current commercial practice throughout the production, assembly, and installation processes. The approach to the fabrication of the heat exchangers is described in Section 4.9.1. The fabrication of the rotating machinery is outlined in Section 4.9.2. The production of the auxiliary systems, specifically for use with ammonia, is described in Section 4.9.3. Assembly and installation of the components into completed power modules is discussed in Section 4.9.4. A discussion of various installation methods is presented in Appendix J.

4.9.1 Heat Exchangers

4.9.1.1 50-MW(e) CPM Heat Exchanger. The 50-MW(e) heat exchangers are about 79 ft in diameter and weight about 2,800 tons. Shops presently exist which are sufficiently large to contain the 79 ft diameter spheres and to handle the individual components during assembly. The 623 ton shell is the largest sub-assembly that could be lifted in most shops. It could be lifted when empty and placed on a fixture which could be moved with air, water, or casters. All final assembly of internals and tubes would be done on the special fixture which would also be used for a shipping support. The same general assembly procedure which is planned for the 10-MW(e) heat exchangers could be adapted for the larger units.

The shell plates are 6.5 in. thick. A quotation was obtained for cold forming the shell segments using existing equipment. Each sphere would require 72 segments which would weigh approximately 20,000 lb each before cutting to size. The plate for each segment would be about 84 in. by 372 in. by 6.5 in. thick. Delivery of shell sections for one evaporator and one condenser would start seven months after receipt of material and would be completed three months later.

The tubesheets and tube support plates would be purchased at 15 ft widths with the required length to form a full diameter section. The plates would be drilled and the sections welded together to form the completed tubesheets and support plates.

The transition rings, tube support plate rings and nozzles would be rolled in sections which would be welded end to end. The balance of the heat exchangers consists of tubing, pipe and smaller plate sections.

The finned tubing would require new production equipment to accommodate the 55 ft length. This would be set up at the final assembly plant to reduce shipping costs and costs of additional subcontractors.

A preliminary list of tools, jigs, and fixtures which would be required for a prototype set of heat exchangers has been prepared and is included in Table E-7 of Appendix E. The costs are based on 1978 prices and include the material and labor to fabricate the items listed. The costs are estimates based on similar items which are planned for the 10-MW(e) prototype fabrication. Some of the items are expendable and some could be used for production of additional heat exchangers of the same design.

4.9.1.2 10-MW(e) MAPS Heat Exchanger. The MAPS heat exchangers are about 40 ft in diameter so that final assembly can be performed in existing facilities such as in Foster Wheeler's Panama City, Florida, facility on the Gulf of Mexico. The main bay is 100 ft by 800 ft with two cranes having a total lift capacity of 550 tons with 50 ft under the hook. There is also a side bay of 35 ft by 800 ft and a steel fabrication bay of 40 ft by 175 ft. The high-bay cranes service a 70 ft by 150 ft barge slip. This facility can easily handle the 500-ton, 40-ft diameter heat exchangers.

Component parts could be fabricated by the main subcontractor or may be subcontracted. The shell segments can be formed and welded into quarter sections and tubes rolled, welded, and finned by subcontractors. No new facilities will be required for any of the fabrication work.

A manufacturing plan has been developed to identify all required steps; this plan is included in Appendix E. Assembly of the heat exchangers starts with the formed quarter shell sections which are supplied by a subcontractor. The sections are set up in positioning fixtures and welded to form a complete shell. Temporary "T" rails are used to roll the shell for optimum work positioning. Nozzle openings are flame-cut, and nozzle reinforcement pads and nozzles welded in place. All shell welds are 100 percent radiographed and stress relieved with local heaters.

The titanium-clad tubesheets are cut to shape, and the titanium stripped back 5/8 in. at all weld locations. The plates are then welded to form 15 ft by 30 ft half tubesheets. The tubesheets are stacked and drilled on the multi-spindle Moline drill. Final welds are made, and the row of holes closest to the seam reamed to remove weld distortion. The construction plan for the support plates is similar.

4.9.2 Rotating Machinery

Designs of the 50-MW(e) module turbines and ammonia pumps incorporated physical size as one trade-off. Limitations of size as a function of existing manufacturing capability and techniques and materials dictated the end configuration. The radial inflow turbine two-wheel unit physical size limitation defined both the power output and the fact that four 2-wheel units will be used for the 50-MW(e) power module. Production of the 24,000 hp size is available using existing facilities and equipments. Production of the axial flow turbine will require minor modification to equipment within existing facilities. These modifications are to accommodate the wheel dimensions.

Ammonia pumps for the 50-MW(e) module can be manufactured using existing facilities and equipment. The seawater pumps will require minor facility and equipment modifications depending on pump vendor and size of pump selected.

Designs of the 10-MW(e) module turbines have been performed to the preliminary design level and production facilities are available. Rotoflow has produced numerous turbo-expanders for the gas process industry that compare in size with the ammonia turbine that is required for the 10-MW(e) OTEC demonstration unit. Rotoflow has presented engineering design solutions resulting in compatibility of the turbine lube oil system with the ammonia working fluid for the proposed design. This design approach will be thoroughly evaluated on the Mini-OTEC program.

The size of the axial turbine hardware for the 10-MW(e) turbine-generator is such that current fabricating, handling and machinery equipment at contacted manufacturers are satisfactory for manufacturing the preliminary design presented. In addition, manufacturers appear to have available the required balancing and checkout test facilities for conducting the checkout tasks.

The LMSC approach of four pumps per heat exchanger not only requires minimum facility and equipment modifications but also provides for the highest reliability pump pod arrangement with individual parts sized within current transportation guidelines.

4.9.3 Auxiliary Systems

The 50-MW(e) and 10-MW(e) plant auxiliary components are all state-of-the-art and will not require developmental work. During the final design work, sizing of all components will be directed to allow the purchase of commercially available hardware and components.

4.9.4 Module Assembly

4.9.4.1 50-MW(e) Power Module The 50-MW(e) power system module assembly study investigated various methods that could be applied to produce the completed end article at a major shipyard. It was determined that a shipyard is best suited for the structural fabrication, component assembly and

installation, and final outfitting requirements inherent in each power system configuration that was considered, particularly in view of the logistics problems and the implications of potentially conflicting labor jurisdictions. Waterfront location, shops essential to support the production requirements, and all principal crafts with clearly defined labor jurisdiction agreements are found in major U.S. Shipyards. It was further determined that a major shipyard would not face any requirements for compliance with OSHA or other government regulatory agencies in the production of this article other than those associated with ship construction.

Heat Exchanger Handling. The extreme weight and bulk of the 50-MW(e) heat exchangers presents a special handling problem which exceeds state-of-the-art crane capabilities. The configuration and buoyancy properties of these heat exchangers, however, makes them ideally suited for transfer by flotation.

Recommended Facility. A graving dock facility similar to that of Fig. 1 with crane capacity dictated by optimum module size and seawater pump weight is considered to be the most cost effective facility. Heat exchangers and power module configurations are ideally suited to floating the heat exchangers into final position on the graving dock and constructing the power module around them without necessitating the physical lifting of these heavy heat exchangers. In Fig. 1, the graving dock is shown accommodating the submerged power module configuration, physically the largest of those considered in the course of this study. This facility also has the capability to accept a complete power module for repair or refurbishment.

Several U.S. shipyards already have the necessary facilities to construct the 50-MW(e) power modules under consideration. Among these yards is Newport News Shipbuilding and Dry Dock Company whose ideal power module handling facilities are illustrated in Fig. 2.

Construction Plans and Assembly Sequences. Construction planning must integrate the structural analysis, design, loading plan and schedule of prefabrications to permit the effective construction of these power modules.

Fig. 4-60 Graving Dock Facility (Submerged Power Module)

Fig. 4-61 Dock and Crane Facility at Newport News Shipbuilding and Dry Dock Company

Special attention must be provided to interface connections to ensure interchangeability between modules and platforms. Construction planning must closely monitor and control distortion. Emphasis must be placed on weld joint design and welding sequence.

Modular construction planning appears the best approach. Structure, along with component foundations, constructed to maximum size or weight (within the limits of lifting capabilities) would be fabricated. Interior piping and equipment would also be incorporated in the subassemblies, with make-up piping installed after subassemblies are fitted and welded together, as indicated in the typical assembly sequence for the 50-MW(e) immersed-unitized module configuration described in the following paragraphs.

Construction Plan. This power module requires a 180 ft by 320 ft graving dock (or dry dock) serviced by a crane or group of cranes having a minimum combined capacity of 175 tons (governed by seawater pump handling requirements). Actual crane capacity would be selected based on the optimization of structural assembly weights. No special equipment, tooling or handling techniques are required other than normal distortion control measures.

Assembly Sequence

- a. Structurally analyze and design equipment supports and temporary added structure
- b. Plan construction and preplan manufacturing
- c. Fabricate modular foundations, structure, and supports
- d. Float heat exchangers into position in graving dock (or dry dock) and dewater dock
- e. Build up "power module support structure," deck-by-deck, of pre-outfitted structural subassemblies
- f. Build-up heat exchanger support structure
- g. Load pre-outfitted "equipment area" subassemblies on the top of "power module support structure" and heat exchanger support structure. Schedule heavy equipment loading to minimize necessity for shipping holes

- h. Sequentially install access openings as required for direct load of major components (Due to structure removal for the direct loading of major components, specific areas of the power module may require major system and component installation completion to permit replacement of structure required to carry loads generated by the placement of other major components in other areas.)
- i. Prefabricate piping systems
- j. Install piping components or equipment that will be restricted due to power module's structure or major components and secure
- k. Sequentially install additional structure, foundations, and equipment
- l. Complete outfitting at dockside

The segmented power module is assumed to be similar to the unitized, except for its heat exchanger quick attach/detach feature. The segmented power module generally requires the same building facilities as the unitized. However, conventional building ways may be utilized to build this module without heat exchangers, these can later be towed to location and connected to the power module.

The submerged configurations are designed to operate in a vertical position attached to a spar; however heat exchanger handling problems preclude building in this vertical position.

The unitized module requires a 180 ft by 320 ft (minimum) graving dock (or floating dry dock) for horizontal assembly. This may require that equipment be mounted on its side if it cannot be mounted in subassemblies prior to loading.

A minimum crane capacity of 175 tons is dictated by seawater pump lifting requirements; however, due to the thickness of shell plating, a much larger crane would be desirable for handling heavy structural subassemblies and thus reducing fitting, welding, and outfitting time within the dock area.

Welding procedures must be developed and equipment acquired to efficiently form, fit and weld the relatively thick plating required.

4.9.4.2 10-MW(e) MAPS. The 10-MW(e) module final assembly starts with the completed shell set with the axis vertical. The upper end of the shell is flame-cut to fit the tubesheet/transition ring subassembly which is then inserted, welded, and stress relieved. The shell is turned 180 deg and the internals installed including the support plates, tie rods, support plate bracing, ammonia feed system, demister supports in the evaporator, and noncondensable gas system in the condenser. The second shell end is then flame-cut, the tubesheet/transition installed and welded, and the seam stress relieved. At this time, the heat exchanger is turned with the axis horizontal for tube installation and welding. The tubes are welded to the tubesheet, mechanically expanded, visually inspected, repaired as necessary, air/freon leak tested, and repaired as necessary. All shell openings are then closed, and the complete heat exchanger is hydrotested with rust-inhibited seawater. Finally, the unit is drained, washed down with potable water, air purged, vacuum dried, and filled with shop air. Any leaks are repaired and retested locally, and the unit is painted, preserved, or otherwise protected, as specified. For shipment, the unit is lifted by the two main shop cranes, moved through the door to the barge slip and loaded on a barge.

Shop tooling and mockups have been identified as required by each step of the manufacturing plan, and are listed in Appendix D.

The completed 10-MW(e) heat exchangers are presently estimated to weigh about 500 tons, which can be accommodated in existing facilities. Whenever possible, shop handling and loading on the barge will be done with available cranes; if necessary, the final lifts and loading will be done with air pallets. This method has been investigated and cost estimates have been obtained. Crane lifts will use the evaporator vapor outlets and the dummy trunnions on the condenser, which are located in the same relative position. These lift points will also be available for field handling including heat exchanger lifts and rotation. The heat exchangers will be transported from

the manufacturing plant to the assembly and/or operation site by barge. A 500-ton lift capability will be required to offload the barge.

Construction Support Equipment. There is no special construction support equipment required which is not currently being used in major U.S. shipyards.

Since crane capacities are not adequate to handle the 50-MW(e) heat exchangers, it is necessary to float heat exchangers into position onto a graving dock (or floating drydock) or transfer from a barge to a building ways by a transfer system such as a Hydranautics skidding system. The crane capacities are therefore dictated by seawater pump handling requirements, the heaviest being 175 tons which is well within the lift capacity of major shipyards. Finally, special equipment is required for the burning, handling, forming and welding of the thick plate associated with the submerged configuration. This equipment is available at major shipyard facilities.

4.10 COST ANALYSIS

This section of the report presents summaries of capital and tooling costs associated with 50-MW(e) and 10-MW(e) power systems of the immersed-segmented configuration; the first subsection, dealing with capital costs, also discusses heat exchanger costs in some detail. The costs for power systems of the immersed-unitized configuration are expected to be almost identical to those presented here, and consequently were not estimated separately. Previous work carried out by LMSC under this contract indicated that the capital costs of submerged power system configurations are 5 to 10 percent lower than those of immersed configurations. However, the platforms associated with these two configurations are markedly different and were excluded from the design effort presented here; consequently, costs for submerged power systems are not presented here either, as comparison with the costs of other configurations would not be meaningful.

4.10.1 50-MW(e) Capital Cost Summary

Table 4-48 summarizes the capital costs of the various components of 50-MW(e) power systems. Since backup information for each of the individual estimates is to be found in the tables of Appendix K, the discussion in this section is limited to highlighting the more important elements of Table 4-48. The total cost of production power systems (including heat exchanger replacement after 15 years) declines from about \$1,900/kW(e) for the first unit, to \$1,800/kW(e) for the eighth. Prototype unit costs are estimated at approximately \$2,150/kW(e).

The largest cost elements in the power system are the heat exchangers; their combined present costs, assuming replacement after 15 years, account for over half the total power system costs. Heat exchanger costs are discussed in more detail later in this section. Seawater pumps are the second costliest item, accounting for roughly one eighth total cost. LMSC pump cost estimates are taken from a parametric study carried out by the Worthington Pump Corporation, discussed fully in Section 4.4.1. This study indicates that, contrary to

Table 4-48

50-MW(e) POWER SYSTEM COST SUMMARY, [\$ /kW(e)]

	Prototype Unit	First Production Unit	Eighth Production Unit
Turbine	52	52	43
Generator	35	35	32
Warm Seawater Pumps	96	96	80
Cold Seawater Pumps	96	96	80
Ammonia Condensate Pumps	10	10	10
Ammonia Distribution Pumps	12	12	12
Ammonia System Piping	17	17	17
Ammonia System Valving	19	19	19
Control Systems	6	5	5
Chlorination System	23	23	23
Mechanical Cleaning System	13	13	13
Ammonia Support Systems	34	34	34
Electrical Equipment	112	112	112
Auxiliary Systems	13	13	13
Evaporator*	402	398	378
Condenser*	423	418	397
Mist Extractor	4	4	4
Structures	62	62	54
Component Installation	55	55	55
Subtotal	1,484	1,474	1,381
Contingency	297	147	138
Home Office and Engineering	223	147	138
Fees	148	147	138
Total	2,152	1,915	1,795

*Present worth including replacements. Assumptions: 15-year life; 6% escalation rate; 10% cost of money

expectations, seawater pumps are subject to diseconomies of scale; it also suggests that seawater pump costs are roughly 30 percent higher than originally expected. Clearly, there still appears to be considerable uncertainty concerning the costs of pumps large enough for OTEC applications. The only other cost element, accounting for over 5 percent of total cost is electrical equipment; roughly two thirds of this cost is associated with electrical cabling. Although in some ways they are more related to the platform than to the power system, the costs of the envisaged pump machinery rooms and access trunks, and of equipment installation, are included in Table 4-48 and, when combined, are slightly in excess of \$100/kW(e). The estimated costs given for controls and ammonia support (mainly startup and shutdown) systems are also platform-dependent, in that they are assumed to be shared among right modules in a 400-MW(e) power plant.

Before proceeding to the discussion of heat exchanger costs, it is desirable to comment on the rationale underlying the choice of allowances for contingency, home office and engineering, and systems contractor and integrator fees in Table 4-48. The exact distribution of these allowances for each component varies depending on the nature of the work involved, the practices of contractors, and the type of contract under consideration. For the purposes of estimating production unit costs, an allowance of 10 percent for each element was selected as giving an approximately correct total cost for the power system as a whole. In practice, the importance of these three cost categories is likely to be equal; their exact distribution is of only secondary interest for estimating purposes. Contingency allowances were doubled and home office and engineering allowances increased by 50 percent for prototype units, to reflect the added uncertainty and difficulty associated with prototype construction. These might also lead to an increase in the fees charged by systems integrators and contractors and to lower proportional increases in other allowances. However, the combined allowances chosen are believed to yield a reasonable estimate of prototype costs, regardless of the actual distribution of costs among the three types of charges for which the allowances are made.

Table 4-49 summarizes heat exchanger costs in the format requested by DOE. The estimate reflects direct shop labor rates of \$19 per hour for the prototype and first units and \$16 per hour for the eighth set. This reduction in labor rates is due to the expected effect of spreading overhead over a larger labor pool. In addition, a 6 percent reduction in labor hours is expected to occur in going from the prototype to the first production unit, after which labor hours are expected to decline following a learning curve with slope 0.97. Material costs are estimated on a weight basis at a unit cost of \$1.05/lb, and tubing costs are based on a unit cost of \$1.40/ft for finned 1.5 in. diameter tube. The subcontractor price for forming the shell segments is based on current quotation. Small decreases in material costs are expected to occur in the course of series production as a result of regular buying patterns. As was to be expected, the cost of aluminum tubing represents the largest component of total costs, accounting for roughly 55 percent of it. Roughly two thirds of this cost corresponds to the cost of the tubing proper, and the remainder to the cost of the enhancement process. The second largest cost element, and the largest labor item in the total, is the cost of tube bundle assembly, accounting for about half of labor costs and roughly an eighth of total costs. Shell material is the only other cost component accounting for more than 10 percent of total costs. Overall, material costs dominate, representing roughly three quarters of total costs. As a result, no drastic decreases in cost are likely to take place in the course of multiple-unit production, and the eighth production units are expected to be only some 10 percent less costly than the prototypes.

4.10.2 10-MW(e) Capital Cost Summary

This section is entirely analogous in spirit and structure to Section 4.10.1, which presented the capital cost summary for the 50-MW(e) power system. Table 4-50, similar to Table 4-48, summarizes the system's capital costs; individual component estimates are presented in Appendix K. The cost per kW of 10-MW(e) power systems is about 50 percent higher than that of the 50-MW(e) units, as a result both of using Ti-tubed heat exchangers and of failing to capture some available economies of scale. As observed in Section 4.4.1, large seawater

Table 4-49
HEAT EXCHANGER COST BREAKDOWN, 50 MW(e), \$/kW(e)

	<u>Subcontract & Material</u>	<u>Labor</u>	<u>Total</u>	<u>Subcontract & Material</u>	<u>Labor</u>	<u>Total</u>	<u>Subcontract & Material</u>	<u>Labor</u>	<u>Total</u>
Tubes	292*	75	367	292*	71	363	290*	54	344
Tubesheets and Rings	8	9	17	8	8	16	7	6	13
Tube Support Plates and Supports	20	40	60	20	37	57	19	29	48
Shell	62	6	68	62	6	68	58	5	63
Water Boxes	3	11	14	3	11	14	3	8	11
Nozzles	2	-	2	2	-	2	2	-	2
Ammonia Distribution and Noncondensable Removal System	2	4	6	2	4	6	2	3	5
Final Assembly	1	7	8	1	6	7	1	5	6
TOTAL	390	152	542	390	143	533	382	110	492

* Tubes at 1.00 \$/Ft basic + .40 \$/Ft enhancement

Table 4-50

10-MW(e) POWER SYSTEM COST SUMMARY (\$/kW(e))

	Prototype Unit	First Production Unit	Eighth Production Unit
Turbine	71	71	58
Generator	47	47	47
Warm Seawater Pumps	85	85	70
Cold Seawater Pumps	85	85	70
Ammonia Condensate Pumps	15	15	15
Ammonia Distribution Pumps	17	17	17
Ammonia System Piping	23	23	23
Ammonia System Valving	26	26	26
Control Systems	30	30	30
Chlorination System	23	23	23
Mechanical Cleaning System	28	28	28
Ammonia Support Systems	265	265	265
Electrical Equipment	140	140	140
Auxiliary Systems	34	34	34
Evaporator	592	584	551
Condenser	689	680	645
Mist Extractor	7	7	7
Structures	71	71	71
Component Installation	65	65	65
Subtotal	2,323	2,296	2,171
Contingency	462	230	217
Home Office and Engineering	347	230	217
Fees	231	230	217
Total	3,353	2,986	2,822

pumps appear to be subject to diseconomies of scale. Consequently, seawater pumps are slightly less costly for the 10-MW(e) power system; all other components are more costly for the smaller units. Production unit costs are estimated to range from just under \$3,000/kW(e) for the first to just over \$2,800/kW(e) for the eighth; prototype unit cost is estimated at roughly \$3,350/kW(e).

Heat exchanger costs, again dominant, are discussed in more detail in following paragraphs; of course, Ti-tubed heat exchangers are not expected to require replacement in the course of the 30-year power system life. It will be noticed that the cost of controls and ammonia support systems per kW(e) are much higher than in the previous case, as a result of the 10-MW(e) power module (1) being entirely self-contained, (2) incorporating some dedicated equipment that was assumed shared among several modules for the 50-MW(e) power systems. The platform-dependent costs of access trunks and pump machinery rooms and of component installation have again been included in the total, and jointly amount to roughly \$120/kW(e).

A cost breakdown for the 10-MW(e) heat exchangers is shown in Table 4-51. The prototype costs are based on the Phase II proposal with the numbers for current 1978 costs before escalation; otherwise, the cost estimation procedure is entirely analogous to that outlined in Section 4.10.1. The shop labor shows a reduction in hours for first and eighth units due to a learning curve with slope 0.97 and also a reduction in hourly rate from \$19 to \$16/hr due to spreading the overhead over a larger number of manhours. The material costs show little reduction, as the quantities for one unit are substantial and little further quantity discount can be expected.

It should be mentioned that, for a production run of eight sets, Foster Wheeler might consider setting up a tube welding and finning line at Panama City. This could give a potential \$900,000 additional cost saving for the eighth set which is not included in the table.

Table 4-51
HEAT EXCHANGER COST BREAKDOWN, 10 MW(e), \$/kW(e)

	Production			First Production			Eighth Production		
	Subcontract and Material	Labor	Total	Subcontract and Material	Labor	Total	Subcontract and Material	Labor	Total
Tubes	907*	131	1038	907*	123	1030	900*	94	994
Tubesheets and Rings	58	20	78	58	18	76	54	14	68
Tube Support Plates and Supports	12	45	57	12	42	54	11	33	44
Shell	48	22	70	48	20	68	45	16	61
Water Boxes	3	14	17	3	13	10	3	10	13
Nozzles	3	-	3	3	-	3	3	-	3
Ammonia Distribution and Noncondesible Removal System	2	5	7	2	4	6	2	3	5
Final Assembly	1	10	11	1	10	11	1	7	8
TOTAL	1034	247	1281	1034	230	1264	1019	177	1196

*Tubes at \$2.25/ft basic plus 0.80/ft enhancement

Of course, the Titanium tubing is by far the dominant cost component, accounting for roughly three quarters of total cost; some 25 percent of tubing costs represents the cost of heat transfer enhancement. Other cost elements of some importance are tube bundle assembly, accounting for some 10 percent of total cost, and tube supports and tubesheets, the relatively high cost of the latter is due mainly to their Ti cladding. As before, the effect of series production on costs is quite small, and the eighth production set of heat exchangers is only some 7 percent less costly than the prototype units.

4.10.3 Estimate of Nonrecurring Costs

The possible desirability of building OTEC power system components in dedicated facilities has been investigated briefly. Turbines, seawater pumps, and heat exchangers immediately suggest themselves as obvious candidate components for manufacture in dedicated facilities. While some other power system components are also not available at present as off-the-shelf items, their manufacture when needed does not appear to pose especially difficult or new problems. Consequently, investigations in this area focused on the three types of components mentioned above.

In the course of discussions with turbine manufacturers, it was found that currently existing industrial facilities are adequate for OTEC ammonia turbine manufacture and, consequently, that dedicated facilities are unnecessary. While the size of OTEC turbines is quite large in relation to their output, in absolute terms it is not beyond the current state of practice. While no specific estimate was obtained for production tooling costs, these appear to be in the neighborhood of \$250,000. The situation concerning seawater pump manufacture is somewhat similar. While the size of contemplated OTEC pumps is beyond the experience of some manufacturers, several others appear capable of producing OTEC pumps in existing facilities. Initial tooling pattern costs for pumps in the 1000 cfs range are estimated by the Worthington Pump Corporation at between \$100,000 and \$200,000.

The question of heat exchanger producibility was investigated in some depth, and it was concluded that heat exchangers sized for 50-MW(e) can be built in existing facilities in the industrial complex. The required tooling cost, however, is quite considerable, amounting to approximately \$4,000,000 for the prototype heat exchanger, \$2,000,000 for the first production units, and roughly \$800,000 apiece for succeeding heat exchanger sets. A listing of contemplated tooling can be found in Appendix D.

It should be stressed that manufacture of OTEC power system componentry being economically competitive without need of dedicated facilities does not necessarily imply that such facilities are not necessary or desirable for the production of complete OTEC plants (including the platform system). The related issues of facilities required for final assembly of the power system and integration with the platform system are discussed in Section 4.9.4 and Appendix J of this report.

4.11 ENERGY RESOURCE ANALYSIS

A net energy analysis of the contemplated 10-MW(e) power system was carried out to determine the energy intensity of this system. While this analysis is necessarily incomplete, excluding as it does platform, mooring and energy delivery system as well as operation and maintenance activities, it does serve to highlight the net energy yield of the power system. The analysis draws heavily on the methodology developed at the Institute for Energy Analysis of Oak Ridge Associated Universities, as reported in Refs. 1 and 2.

Table 4-52 summarizes the power system net energy balance. Each individual component was assigned to the most appropriate industrial sector of the input-output table for the U.S. economy, and energy coefficients for these sectors were obtained from Ref. 2 in terms of Btu/ton. In some cases, no obviously appropriate sectors could be found; in these cases, components were assigned to catch-all sectors, such as "General Industrial Machinery." In the case of electrical components, three sectors appeared to be appropriate: Power Transmission Equipment, Transformers, and Switchgear; consequently, the average energy content of products falling in these three categories was used. The estimated energy content given in (1) for Titanium tubing was used, as no entry was found in (2) covering Titanium products. Component energy contents were obtained by multiplying the above-mentioned energy coefficients by component weights; the latter were estimated, in some cases, from incomplete information. However, weight estimation inaccuracies are believed to represent only a minor source of error by comparison with that resulting from using the published sectoral energy content coefficients based on the 1967 industrial census.

The component energy contents, shown in Table 4-52, have been modified as suggested in Ref. 1, by including in them the energy cost of producing and delivering the consumed energy itself. Table 4-53 shows the factors by which individual types of consumed energy must be multiplied to take these energy costs into account. In particular, it will be noticed that the factor given for electric energy implies a total primary energy cost of 13,860 Btu/kWh of

Table 4-52
10-MW(e) POWER SYSTEM NET ENERGY BALANCE

Component	No.	Sector Name	Energy Inputs			
			Electrical 10 ⁶ Kwh	10 ⁹ Btu	Thermal 10 ⁹ Btu	Total Primary 10 ⁹ Btu
Turbine	38.11	Al Castings	0.18	2.52	4.18	6.70
Generator	53.09	Motors, Generators	0.21	2.89	7.85	10.74
Seawater Pumps						
Ammonia Distribution Pumps	49.01	Pumps, Compressors	0.48	6.66	20.34	27.00
Ammonia Condensate Pumps						
Ammonia Systems Valving	42.08	Pipe Fittings	0.26	3.54	3.06	6.60
Ammonia System Piping						
Ammonia Support Systems						
Seawater Pump Set Shrouds	40.06	Fabricated Plate Work	2.60	35.84	139.67	175.51
Heat Exchanger Steel						
Structures						
Mechanical Cleaning System	32.04	Miscellaneous Plastics	0.12	1.59	4.53	6.12
Control Systems	51.01	Computing Machinery	0.06	0.78	2.10	2.88
Auxiliary Systems						
Chlorination System	49.07	General Industrial Machinery	0.04	0.49	1.53	2.02
Electrical	(1)		2.52	39.77	92.91	127.68
Subtotal			6.47	89.08	276.17	365.25
Fabrication (20%)			1.29	17.82	55.23	73.05
Ti Tubes and Cladding	(2)		13.11	181.64	37.24	218.88
TOTAL			20.87	288.54	368.64	657.18

NOTES: (1) Used average energy content values for sectors 49.05, 53.02, 53.03

(2) Used overall energy content of 576.10⁶ Btu/ton, from [2]

Table 4-53

PRIMARY ENERGY INPUTS FOR PRODUCTION AND DELIVERY OF FUELS
AND ELECTRICITY

Energy Carrier	Btu Spent per Btu of Product (a)
Electricity (b)	4.06
Coal (c)	1.01
Refined Petroleum Products (d)	1.22
Natural Gas (e)	1.12

Notes:

- (a) Direct and indirect inputs, including capital investment and including energy content of product.
- (b) Electricity delivered to customers.
- (c) Coal at minehead.
- (d) Output of refinery, excluding delivery.
- (e) Gas delivered to customers.

delivered electricity. The energy cost of component installation and power system integration can be estimated approximately by adding 20 percent to the total energy content of all power system components except heat exchanger titanium tubing and cladding, as described in Ref. 1. Finally, adding this fabrication energy cost to total component energy content yields the total primary energy inputs required to produce a 10-MW(e) power system, approximately 650 billion Btus. Of this total, roughly 55 percent corresponds to direct thermal energy inputs, and the remainder to electric energy consumption.

Titanium is by far the most energy-intensive material used in the power system. The titanium contained in the heat exchangers accounts for approximately one third of total required primary energy inputs, and for over 60 percent of electric energy requirements. Steel plate work and electrical equipment are two other major contributors to the system's energy

requirements, representing roughly one quarter and one fifth of the primary energy total, respectively.

The total electrical output of a 10-MW(e) OTEC power system operating at 90 percent capacity factor over thirty years is approximately 2,365 million kWh. To evaluate the net energy contribution of the power system, various ratios can be calculated involving this electrical output, or its primary energy equivalent, and the energy input totals given in Table 4-52. Perhaps the most useful such measure is the thermal energy subsidy required to produce one net kWh of electric energy. To obtain this measure, the required electric energy subsidy is first subtracted from the power system's total lifetime output to obtain its net electric energy production, and the required thermal energy subsidy is then divided by this quantity. Performing this calculation, it is seen that the required net unit thermal energy subsidy is roughly 157 Btu/kWh; this figure is in good agreement with that derived in Ref. 1. While the net energy analysis described here is admittedly incomplete, comparison of this quantity with that given earlier for conventional electricity generation (13,860 Btu/kWh) illustrates clearly OTEC systems' highly favorable net energy balance.

REFERENCES

1. A. M. Perry, G. Marland, L. W. Zelby, "Net Energy Analysis of an Ocean Thermal Energy Conversion (OTEC) System," ORAU/IEA (M)-77-1, Institute of Energy Analysis, Oak Ridge Associated Universities, Oak Ridge, Tennessee, February 1977.
2. D. B. Reister, "The Energy Embodied in Goods," ORAU/IEA(M)-77-6, Institute of Energy Analysis, Oak Ridge Associated Universities, Oak Ridge, Tennessee, February 1977.

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Section 5

HEAT EXCHANGER TEST ARTICLES

The heat exchangers will provide cost, performance, and operating experience data with the most critical component in the OTEC power system in a cost-effective manner. Compatibility of the test articles with the OTEC-1 capabilities was a key design and sizing requirement. The velocity of ammonia vapor has a strong influence on the structural requirements for tubes and tube supports, on the entrainment of liquid in the evaporator discharge, and on the efficiency and wetting of the tubes. This led to the design of a reduced scale evaporator with a means for supplementing the flow rate of the ammonia vapor. The heat exchangers are equipped with dual tube bundles (Appendix A7) so that a bundle of smooth tubes can be tested simultaneously with a bundle of enhanced tubes under similar conditions. The tube bundles also contain two different aluminum alloys in order to obtain in situ data. Most conceptual features of the commercial power system have been incorporated in the test article as described in the following sections.

5.1 TEST ARTICLE DESCRIPTION

The test article heat exchangers are reduced scale models of those designed for the 50-MW(e) commercial power system. Sizing of the heat exchangers recognized the OTEC-1 compatibility, resulting in a nominal 0.2-MW(e) net output based on projected OTEC-1 head losses and the performance of the test article enhanced tube bundle. The combined capacity from the smooth and enhanced tube bundle will be 0.37-MW(e); however, each half can be operated independently. The test article configuration incorporates many of the features of the commercial power system such as: (1) use of aluminum in the spherical shell, the tubesheet, and the tubes, (2) use of heat transfer enhancement on the tube exterior, (3) incorporation of the mist extractors and ammonia sumps, (4) spherical shell, and (5) inclination of the condenser to improve drainage of ammonia condensate.

The evaporator and condenser are contained in spherical aluminum shells of 17.4 ft O.D. The 8.6-ft diameter aluminum tubesheet is 2 in. thick. The tube bundle is divided into halves by a vertical divider plate as shown in Fig. 5-1. One half of the evaporator contains 2386, 7/8 in. diameter smooth aluminum tubes; the other half contains 2386 tubes enhanced with external fins. The condenser contains 2850 smooth tubes and 2850 enhanced tubes in its respective halves. Hook and vane mist extractors are located inside the shell at the two evaporator vapor outlets.

To simulate the ammonia vapor velocities over the tubes of a 50-MW(e) commercial power module, a system for supplying supplementary ammonia vapor is provided with the evaporator. This vapor is injected from perforations in the central divider. Chlorination is used to control the growth of biofouling, and M.A.N. brushes are installed in each tube to provide mechanical cleaning. Sacrificial anodes are located at the aluminum test article and carbon steel OTEC-I piping interfaces. The distribution of liquid ammonia in the evaporator is provided by irrigation. tubes located at the top of each segment of seawater tubes. Valves located outside the shell allow local control of the distribution of the liquid ammonia in the bundle. The shells are mounted on legs for installation on OTEC-1. The length of the legs (or spacer pads) can be varied to allow operation at off-design angles or inclinations. The test articles are equipped with a complete set of instrumentation for measurement of thermal and hydraulic performance. The test article system configuration is depicted in Fig. 5-2. Section 5.2 discusses the rationale underlying the selection of test article size, and Sections 5.3 through 5.7 treat some important aspects of the test system design and instrumentation. Finally, Sections 5.8 and 5.9 present a brief discussion of heat exchanger producibility and cost, respectively.

5.2 SELECTION RATIONALE

5.2.1 Cost

In order to ascertain the test article configuration, a comparison was made using a 1-MW(e) size between the full and half-bundle arrangement, and between

5-3

Fig. 5-1 Heat Exchanger Test Article Showing Smooth and Enhanced Tube Bundles

5-4

Fig. 5-2 Heat Exchanger Test Article Configuration and Functional Diagram

the 7/8 in. and 3/4 in. diameter tubes, to determine the most cost-effective way to get full scale vapor velocities across the tubes. The half-bundle arrangement concentrates the vapor flow into one-half of the bundle periphery, and increases flow velocity at the periphery by $\sqrt{2}$, as compared with a full circle bundle of the same capacity; this arrangement also allows the augmentation of vapor crossflow from the central plenum. Other bundle arrangements were considered (Appendix A5) but were rejected as too complex or costly with no offsetting advantages. The smaller tube size increases vapor velocity by packing 25 percent more heat transfer area per unit tube length into the same bundle diameter. The costs estimated for the three cases considered are shown in Table 5-1. The half-bundle arrangement was selected as most cost effective when the full bundle circle diameter is maintained and the number of tubes is cut in half.

Table 5-1

ALTERNATIVE 1-MW(e) TEST ARTICLE CONFIGURATIONS

Case	Bundle Arrangement	Bundle Diameter (in.)	Tube Diameter (in.)	Number of Tubes	Cost (\$)
1	Full Circle	119	7/8	Condenser 7,900	372,000
				Evaporator 5,750	304,000
2	Half Circle	161	7/8	Condenser 7,900	423,000
				Evaporator 5,750	355,000
3	Full Circle	119	3/4	Condenser 11,333	467,000
				Evaporator 8,130	366,000

Second, the advantage of reduced size was explored. The smaller sizes chosen were 0.2 MW(e) as a reasonable lower limit and 0.6 MW(e) as the arithmetic mean between 0.2 and 1.0. The three sizes (0.2, 0.6 and 1.0) were compared as to their thermal, fluid dynamics, and mechanical simulation of the 50-MW(e) heat exchangers, and cost.

For thermal simulation, a tube bundle 30 to 50 tubes high is necessary to get a fully developed ammonia liquid distribution. This number of tubes will fit into the 0.2-MW(e) size. However, since the tube bundle is divided vertically into five zones with ammonia feed tubes at the top of each zone, the height of each zone is less than 30 tubes. The test requirement of 30 to 50 tubes below the feed level is met by shutting off the flow to certain feed tubes, and feeding the zones by a trickle-down method from above.

Fluid dynamic simulation requires developing a vapor flow field in the tube bundle which is locally similar to that in the 50-MW(e) design. In the 0.2-MW(e) size (as well as in the larger sizes) the augmented vapor crossflow will allow velocities in the range encompassing the full-scale conditions.

Mechanical simulation is concerned with tube vibrations induced by vapor crossflow, pressure-shell structural design, and manufacturing costs. The vibration simulation requires proper crossflow velocities, provided as described above, and proper mechanical constraints on the tubes. A three-space tube design (two internal tube support baffles) provides all the constraint conditions found in the full-scale heat exchanger. The structural simulation is satisfied by making the pressure shell a geometrically similar subscale model of the full-scale design incorporating all necessary items such as mist extractor, sumps, and noncondensable gas removal. The relative costs of the three sizes are shown in Table 5-2.

Because the 0.2-MW(e) heat exchanger test article assembly can provide all the required engineering data and is lowest in cost by a considerable margin, it was the clear choice. A further reduction in size would place questions on the validity of test results.

5.2.2 OTEC-1 Compatibility

A schematic of the 0.2-MW(e) heat exchanger test loop is depicted in Fig. 5-3. The 0.2-MW(e) test hardware consists of contractor-furnished evaporator, mist extractor, condenser, evaporator ammonia distribution, ammonia vapor augmentation, seawaterside mechanical cleaning devices (M.A.N. brushes), and

O-511

Fig. 5-3 0.2 MW(e) Heat Exchanger Test Loop

Table 5-2

TEST ARTICLE COST/SIZE

<u>Capacity [MW(e)]</u>	<u>Cost</u>
0.2	Baseline
0.6	1.75 x baseline
1.0	2.4 x baseline

related instrumentation. All other equipment, ammonia circulation loops, test facility, instrumentation, turbogenerator (or power generator simulation), seawater piping and interface with OTEC-1 auxiliary subsystems, and the DACS are to be supplied by the OTEC-1 Systems Integration Contractor. Not enough information is presently available on the OTEC platform to be able to show how the 0.2-MW(e) heat exchangers will fit into the platform.

Test Article/OTEC-1 Interfaces. Modifications to OTEC-1 include routing seawater to the Test Article with separate chlorination provisions, and then interfacing with the DACS through an appropriate RTU.

Supply lines must be provided for ammonia, nitrogen gas, and chlorination. Noncondensable gas removal and ammonia/water cleanup systems will also utilize the central system provided by the OTEC-1 test facility. No changes will be required for the noncondensable gas removal system and the ammonia/water cleanup system other than the addition of appropriate valves and transfer lines to service the 0.2-MW(e) test loop. The OTEC-1 ammonia storage system capacity (assumed at 13,800 gallons) has to be shared with the 0.2-MW(e) heat exchanger requirements.

The nitrogen purge system should be increased in capacity (by about 20 percent) to provide the nitrogen required to purge both the 1-MW(e) and the 0.2-MW(e) systems at least twice. Adding nitrogen storage bottles to the existing system could provide the needed gas.

Adequate seawater supply should be available for simultaneous testing of the early 1-MW(e) heat exchangers and the PSD-I Test Article heat exchangers. The test article heat exchangers require reversible seawater flow to demonstrate the performance of the M.A.N. brushes. The flow requirements for the PSD-I test article heat exchangers is shown in Table 5-3.

Table 5-3

TEST ARTICLE HEAT EXCHANGER
SEAWATER REQUIREMENTS

Seawater	0.2-MW(e) Requirements	
	Flow (gpm)	Head (ft)
Cold	10,000	15
Warm	11,750	10

5.2.3 Thermal/Fluid Dynamic Performance

The test article will be used to verify and extend the current power system design data base that has been developed through lab oriented testing and computer-supported analytical studies.

Heat Transfer Measurement. As part of its Independent Research activities, LMSC has conducted experiments to determine the ammonia-side heat-transfer coefficient as a function of ammonia feed rate for a single isolated tube. These tests, using the feed system proposed for the commercial size, modular application, and test article heat exchangers, indicate that a value of 3,000 Btu/hr-ft²-°F is reasonable for the type of tube-side enhancement selected for the test article. One of the primary objectives of the test article experimental project is corroboration of these laboratory results and extension of testing experience to include interactions which occur within a multitube bundle.

Verification of Analytical Models. The analytical methods used for the evaluation of the thermal and fluid dynamic heat exchanger performance are straightforward and have been implemented through several computer programs. The procedures and related computer programs, which have been used to define the full-scale power system module performance, will be used to predict the thermal and fluid dynamic performance of the test article heat exchangers. The corresponding experimental data resulting from the OTEC-1 test program will provide a means of verifying the analytical procedures or indicating areas for modification. The basic equations used in performance prediction and the measurements necessary to verify the equations are summarized below.

The predicted heat flux is calculated using

$$Q = UA \cdot \Delta T \quad (1)$$

where

U = Overall heat transfer coefficient, Btu/hr-ft²-°F

A = Area on which U is based, ft²

ΔT = Log mean temperature difference, °F

The actual thermal flux can be determined from the seawater flow rate and temperature difference or from ammonia flow rates, temperatures, and quality upstream of the mist extractor.

Predicted seawater side-pressure drop is obtained by summing the entry loss, internal friction loss, and exit loss. The entry loss is assumed to be equivalent to a sudden contraction loss and is determined by Binder* to be

$$\Delta P_I = \left[0.34 - 0.4106 \left(\frac{A_2}{A_1} \right)^2 \right] q_2 \quad (2)$$

*Binder, R. C., Fluid Mechanics, Prentice-Hall, Englewood Cliffs, N.J., 1955

where

$$\begin{aligned}\Delta P_I &= \text{Inlet loss, psf} \\ A_2 &= \text{Flow area in tubes, ft}^2 \\ A_1 &= \text{Flow area in header, ft}^2 \\ q_2 &= \text{Dynamic pressure in tubes, psf}\end{aligned}$$

The internal friction loss is calculated from

$$\Delta P_F = f (L/D) q_2 \quad (3)$$

where

$$\begin{aligned}\Delta P_F &= \text{Friction pressure loss, psf} \\ f &= \text{Friction factor} \\ L &= \text{Tube length, ft} \\ D &= \text{Tube ID, ft}\end{aligned}$$

The friction factor is based on considerable flow test data in circular ducts. Since the kinematic viscosity of water is relatively constant, the friction factor can be related to the Reynolds number analogy by

$$f = \frac{0.01905}{(V_2 D)^{.21}} \quad (4)$$

where

$$V_2 = \text{Tube velocity, ft/sec}$$

The exit loss is equal to a sudden expansion of one dynamic head. In addition to the above, a pressure loss of 64 psf is attributable to the M.A.N. brush installation.

Predicted working fluid pressure drops within the vapor and condensate lines are determined from Eq. (3) with the friction factor related directly to the flow Reynolds number. The pressure drops within the heat exchangers are

computed from Eqs. (6-13) and (6-13b) as determined by McAdams.* Equation (6-13) is

$$P_F = 4f N G^2 / 2g\rho \quad (5)$$

where

N = Number of rows

G = Mass velocity at minimum area, lb/sec ft²

g = Gravitational constant, 32.2 ft/sec²

ρ = Density, lb/ft³

For the proposed staggered tube array, the friction factor can be determined from Eq. (6-13b) as

$$f = \left[0.23 + \frac{0.11}{(x_T - 1)^{1.08}} \right] \left[\frac{D_o G}{\mu} \right]^{-0.15} \quad (6)$$

where

x_T = Ratio of pitch to diameter

D_o = Tube outside diameter, ft

μ = Absolute viscosity, lb/sec-ft

The predicted parasitic power losses are derived from the seawater and ammonia flow rates, related pressure drops, and the efficiencies of the pumps and motors provided by the vendors. The comparison of the predicted parasitic losses with actual losses is dependent upon the OTEC-1 configuration. Incorporation of a turbine-generator in the power cycle will permit measurement of gross power. Measurement of the power to drive the ammonia pumps and a prorated portion of the seawater pump drive will permit evaluation of the parasitic power requirements.

*McAdams, W. H., Heat Transmission, McGraw-Hill Book Co., New York, N.Y., 1954

Fouling will be monitored on the heat exchanger test articles by maintaining constant seawater and ammonia flow conditions over extended periods and determining overall heat transfer performance degradation. The effect of fouling can then be computed. Operation of the fouling countermeasure schemes should return the heat exchangers to a "clean" performance capability. This method is based on the preliminary test results presented at the 5th OTEC workshop.

Accurate measurement of fouling and fouling rate can be made external to the heat exchanger using a device under development at Lockheed Ocean Laboratory, San Diego, California. The device basically consists of an electrically heated tube where inlet seawater and wall temperatures are measured along with flow rate. Accurate determination of heater power, temperatures, and seawater flow rate will allow prediction of fouling factor to within approximately $\pm 2 \times 10^{-5}$ hr. ft² °F/BTU. Using this device just upstream of the heat exchangers will allow assessment of biofouling for a variety of tube materials, cleaning methods, and flow rates. These results can be compared with less accurate fouling rate determination using heat exchanger tube wall temperature measurements stated above.

The effect of internal velocity on liquid entrainment will be evaluated by sampling the evaporator efflux for flow quality and measuring the liquid runoff from the mist extractor. The measured efflux flow rate and quality, together with the mist extractor runoff rate, will provide values for flow quality upstream of the mist extractor as a function of vapor velocity.

The cost/performance effectiveness of the selected heat transfer enhancement approach will be evaluated by direct comparison of enhanced and smooth tube performance data obtained from the test article heat exchangers. The configuration has equal numbers of enhanced and smooth tubes, respectively, in either half of the divided shell. The relative heat flux in the halves can then be obtained at identical thermal resource conditions. The seawater flow rates will be equivalent and the seawater temperatures will be measured at several locations upstream and immediately downstream of the tubes. Working fluid

parameters will be monitored in the individual flow systems for performance evaluation.

5.2.4 Mechanical Performance

Mechanical Design Features. The general arrangement of the test article heat exchangers is shown in Figs. 5-4 and 5-5. The heat exchangers will be designed and fabricated in accordance with ASME Boiler and Pressure Code Section VIII, Division 1. They are designed for a maximum ammonia design pressure of 250 psig. The heat exchangers are 17-ft diameter spheres and weigh approximately 25 tons each.

The test articles are prototypical of the 50-MW(e) reference design as far as practical. The general arrangement is similar to the 50-MW(e) design except that the partial inner tubesheet has been eliminated because it is unnecessary with the smaller diameter, and the tubesheet has been increased from a 1.5- to 2-in. thickness due to the higher design pressure. Also, the tube diameter has been reduced from 1.5 to 0.875 in. The test articles will be all aluminum and will use the same types and tempers of material as the reference design.

Tube Vibration Considerations. Crossflow induced tube vibration is the only mechanical damage mechanism in the otherwise static heat exchanger structure. The vibration can be excited by vortex shedding, turbulent buffeting, and fluidelastic whirling. Methods have been developed to predict each of these phenomena, and results have been tested against field observations in TEMA and HTRI Vibration Data Banks.

The heat exchanger tubes vibrate at discrete frequencies that can be calculated from the tube bundle geometry, tube material, and fluid inside the tubes. If the flow-induced vibration match the natural frequency of the tube, energy can be fed into the tube and can cause large amplitude tube vibrations. Mechanical failure can occur by (1) adjacent tubes impacting against each other, (2) tubes being cut by the baffles, (3) fatigue of the tube material, or (4) tube joint failure. The tubes at the periphery of the tube

5-15

Fig. 5-4 Test Article Evaporator, General Arrangement

LMSC-D630248

Fig. 5-5 Test Article Evaporator, Details

bundle are most prone to failure because the vapor velocity is highest in this region.

In vortex shedding, flow across the tube produces a series of vortices downstream of the tube. The Strouhal number is proportioned to the vortex shedding frequency and tube diameter, and is inversely proportional to crossflow velocity. It also varies with tube arrangement and spacing. If the tube starts to vibrate, the vortex shedding frequency can synchronize with the natural frequency of the tube. The vortex shedding frequency can be calculated. Because a free flow field downstream of the tube is required for vortex formation, this phenomenon is likely to affect only the outermost tubes in the bundle.

Turbulent buffeting is caused by the fluctuating forces which are due to the extremely turbulent flow of the shell-side fluid. The turbulence has a wide range of frequencies distributed around a dominant frequency which is proportional to crossflow velocity. The tubes can extract energy from the turbulence, particularly when the natural frequency is near the dominant frequency. The central frequency can be predicted.

Fluidelastic whirling produces an orbital tube motion due to lift and drag displacement of the tube at its natural frequency. The critical crossflow velocity where fluid elastic whirling starts can be calculated.

Although each of the above phenomena can be accurately calculated under idealized conditions, it is not possible to predict whether the phenomena will act together to amplify or dampen the intensity of vibration. In addition, the geometry of the tube bundle varies depending on the direction of the radial vapor flow relative to the triangular tube pitch pattern. The system mechanical damping is also difficult to predict. For these reasons, experimental data are necessary to demonstrate the characteristics of the tube bundle.

As a basis for design, conservative assumptions have been made regarding damping and the mutual reinforcement of vibration inducing phenomena. The preliminary vibration analyses showed that there is a wide range of possible vortex shedding frequencies, and the natural frequency of the tubes lies in the lower end of this range. The test article heat exchangers have been designed with tube support spacing to limit resonant tube amplitude to less than one-half the distance between tubes, and the vibrating stresses to less than the endurance limit for an infinite number of cycles. The enhanced tubes have smooth lands at the tube support plates, and the lands have a thickened wall for increased wear resistance.

Test Article Data Extrapolation

A basic objective of the heat exchanger test articles test program is to obtain data which is to be used to either verify the prediction algorithms, or indicate how the algorithm should be modified to provide the correlation to full size heat exchangers.

- (a) Water side ΔP and h . Entrance and exit loss coefficients are essentially a function of the contraction and expansion coefficients with a very weak Reynolds number dependence. Using the same tube pitch-diameter ratio will make the entrance and exit coefficients independent of tube diameter. The frictional loss will be modeled by Reynolds number and roughness to diameter ratio. The seawater heat transfer coefficient is modeled by Nusselt and Reynolds number and is only a weak function of tube diameter.
- (b) Fouling Rate. The fouling rate should be primarily a function of wall shear stress and not wall diameter. The influence of diameter can be assessed using several tube diameters in the external measurement of fouling described under Item 2. For the same seawater velocity, a two-fold change in diameter produces approximately a 15 percent change in wall shear stress for fully developed turbulent flow in a smooth tube.
- (c) Tube Vibration. Tube vibration is modeled by Strouhal number and geometric arrangement. The test article will generally have the same

pitch diameter ratio as the full scale heat exchanger although the Strouhal number for both vortex shedding and turbulent buffeting are relatively weak functions of pitch diameter ratio for 30 and 60 degree tube orientations. The primary consideration for tube vibration is the tube span-diameter ratio which can be sized to provide the worst condition for the prototype heat exchanger.

- (d) Ammonia Side P and h. All tests, performed to date at LMSC's Palo Alto Laboratories, with ammonia have indicated that the shell side heat transfer coefficient is not a function of tube diameter for plain and enhanced tubes. Fin enhanced tubes are sensitive to fin height when this is reduced below that required by the liquid film thickness. The minimum fin height is presently felt to be on the order of 15-20 mils. Pressure drop resulting from momentum change and friction can be calculated using conventional tube bank pressure drop equations. Partitioning the tube bundle will allow fairly accurate determination of vapor velocity and calculation of the local friction factor. Friction factor is principally a function of Reynolds Number which allows its extension to other conditions of velocity and diameter.
- (e) Tube Irrigation. The irrigation tube flow rate is determined using a computer program which takes into account the friction, momentum, and gravitational losses. No problem is envisaged in predicting the flow characteristics of the distribution tubes. Preliminary studies indicate that the problem of liquid deflection by vapor cross-flow is primarily a function of vapor velocity and tube geometry. Maintaining the same geometry in the test article will generally serve to make the results applicable to the full scale heat exchanger.
- (f) Droplet Carryover. The amount of carry-over will depend on the degree of liquid jet breakup in the distribution system and vapor-liquid entrainment within the tube bundle. The distribution tubes are designed to eliminate any atomization of the liquid jets and have been experimentally determined to be effective. Vapor-liquid entrainment is not completely understood at this time with the

exception of knowing that it is a function of vapor velocity and also, to some extent, liquid loading. If the droplet size for liquid falling between tubes does not vary significantly with tube diameter at the same liquid loading, the percentage carry-over should not be a significant function of tube diameter.

- (g) Inlet Erosion. Inlet corrosion should be governed by seawater velocity and contraction ratio. The contraction ratio is a function of the pitch diameter ratio which will be essentially the same for the test articles and full scale heat exchanger. A significant change in inlet corrosion with a diameter change on the order of two would not be expected for the same seawater velocity.

5.2.5 Material Selection

A review of results from two test programs performed at the F. L. LaQue Corrosion Laboratory indicate that 3003 aluminum alloy provides good resistance to general corrosion and to corrosion-erosion in seawater. These results (contained in Naval Ship Research and Development Center - E.E.S. Report 040037A(s) dated 28 February 1955 and Chinn and Rogers Report Royal Scientific Service 19 (1) 35 (1963), indicate the Alclad 3003 to have higher resistance to pitting than 5052. Further discussions with Kaiser Aluminum and Gifford Hill and Company representatives indicate Alclad 3004 to be superior to the Alclad 3003 in corrosion resistance and strength.

A cost comparison of 3003/3004 and 5052 tubing indicates the 3003 to be about 60 percent that of the 5052.

To evaluate these various tube aluminum tube materials both Alclad 3004 and 5052 tubes will be used simultaneously in the heat exchanger test articles to obtain fabrication, cost, performance and corrosion data under the same conditions.

Further, for evaluation of tube corrosion and biofouling countermeasure effects, every three or four months during heat exchanger test article

operation, four (4) or six (6) tubes will be removed from each heat exchanger for metallurgical examination and analyses. This sample, over the four (4) year period, plus data from current DOE and LMSC tests and obtained from other sources, will provide a base on which to accept or reject the aluminum alloys for heat exchangers in the OTEC environment.

5.2.6 Biofouling Effects

The heat exchanger test articles are to be instrumented to ascertain the effects of biofouling on performance. The instrumentation cost for each heat exchanger for reasonable accuracy appears to be quite high. Several different devices for the accurate measurement of biofouling rates and effects have been designed and fabricated. Several of these devices can be used with each test article heat exchanger for the express purpose of providing a standard against which the biofouling-heat exchanger performance and biofouling countermeasures can be correlated.

5.3 EVAPORATOR

The general arrangement of the test article evaporator is shown in Fig. 5-4 and details are shown in Fig. 5-5. The evaporator will be designed and fabricated in accordance with ASME Boiler and Pressure Code Section VIII, Division 1. It is designed for mounting on the surface of the OTEC-1 test platform with a maximum ammonia pressure of 250 psig. The evaporator is a 17-ft diameter sphere and weighs approximately 25 tons.

The test articles are prototypical of the 50-MW(e) reference design as far as possible. The general arrangement is similar to the 50-MW(e) design except that the partial inner tubesheet has been eliminated due to the smaller diameter, and the tubesheet thickness has been increased from 1.5 to 2 in. due to the higher design pressure. The tube diameter has also been reduced from 1.5 to 0.875 in. The test articles will be all aluminum and will use the same types and tempers of material as the reference design.

5.3.1 Tube Configuration

The evaporator is divided into equal halves by a double divider plate on the vertical centerline. Both divider plates are welded to the shell and tubesheets to form a thin central compartment supplemental vapor distribution as explained in Section 5.2.

The tubes in one-half the bundle are 7/8 in. OD with a smooth inside surface and a circumferential fin on the outer surface. The tube wall thickness is 0.060 in. minimum at the root of the fins, and the fins are 0.025 in. high with 30 fins per inch. The tubes will have smooth lands at the tubesheets and the tube support plates. The balance of the tubes are also 7/8 in. OD but have a 0.083 in. thick wall with a smooth inside and outside surface. All the tubes are on a 1.094-in. triangular pitch with the shortest distance between tubes in the vertical direction. The tubes will extend beyond the tubesheet with a fillet weld between the tubes and tubesheet.

The two halves of the tube bundle are identical (except for the tubes) so that the performance of the enhanced tubes can be compared directly to that of the smooth tube by operating the two halves simultaneously. The evaporator has 2,386 tubes of each configuration. Identical ammonia liquid and vapor connections are provided for each half of the tube bundle so that either side can be operated independently, or the two sides can be operated simultaneously. The half bundle with finned tubes is estimated to produce the equivalent of 0.2 MW(e).

5.3.2 Structural Considerations

The test article structural design was scaled from the 50-MW(e) reference design without modifying the computer model and making additional runs. A more rigorous analysis will be made as part of the detail design. Hand calculations have been made to size all pressure boundaries to conform to ASME requirements based on the 250-psig design pressure.

The tubesheets are 2 in. thick to take the design pressure. It does not appear that the partial second tubesheet is required with the small tubesheet diameter, but this must be confirmed in the detail design stage. The tubesheets are a single plate with no welds to interrupt the tube pattern. A 2.5-in. thick transition ring around the tubesheet is the interface with the shell.

The tube support plates are made in two halves because of the presence of the center divider plate. The support plates are retained in half rings which are supported from the shell with pipe spacers. Tie rods are threaded into one tubesheet, and axial spacing is maintained with tubular spacers between support plates and the tubesheet. Only three tube spans are provided in the test article. This is the minimum number required to obtain representative tube vibration data.

The 2.5-in. thick spherical shell is made of 12 "orange peel" sections which are cold-formed and welded together. The vapor nozzles on the evaporator and similar dummy nozzles on the condenser will be used as lift points for handling the heat exchangers in the shop and in the field.

5.3.3 Tube-to-Tubesheet Joint

It would be easier to attach the brush cages to the outside of the tubes if the finned section extended beyond the face of the tubesheet. The baseline design tube has a smooth land at the tubesheet and additional finning on a section of tubing projecting beyond the tubesheet face. The tube-to-tubesheet weld is a fillet weld at the outside face of the tubesheet. A special torch is required to fit the 0.22-in. space between the tubes. Also, if the tube wall is too thin, the fillet weld will burn through the tube and produce an unacceptable weld. The tube wall at the land will depend on the tube finning technique, but it should be about 0.080 in. to obtain the 0.060 in. wall under the fin. It is planned to develop a satisfactory fillet weld with a wall that is over 0.070 in. thick, requiring development tests. The more conservative proven end weld, although selected as the reference design for preliminary

design, will also require development tests early in the detail design should the more desirable fillet weld not be used.

The tube-to-tubesheet weld detail is shown as an end weld in the preliminary design drawing. This backup weld which was demonstrated during the conceptual design phase using a 1-in. OD aluminum tube with a 0.065-in. wall. If this design is used, the M.A.N. brush system cages must be so installed to be retained by the inner surface of the tube.

5.3.4 Ammonia Distribution

Duplicate liquid ammonia ports on the bottom of the evaporator are for the ammonia from the condenser and for the outlet from the ammonia sump. The return connection from the recirculation pumps is near the centerline of the shell.

The liquid ammonia feed system has five levels of distribution headers to duplicate the reference 50-MW(e) design. Because there are fewer tubes in the test articles, the number of water tubes in a column is reduced to 10 to 15 rather than the 35 to 68 in the larger reference design. The feed system operation is identical to the 50-MW(e) design as described in the preceding section.

The divider plate in the center of the evaporator is a double plate with 1.5 in. between the plates. The plates are perforated in a pattern that will be developed in the detail design. Some fraction of the ammonia vapor exiting from the evaporator can be recirculated through the 18-in. pipe header, enter the space between divider plates through slots, pass into the tube bundle through holes in the divider plates, and exit through the vapor ports. The recirculated vapor will be added to the freshly vaporized ammonia to provide a higher velocity at the outer edge of the tube bundle. The augmented velocity will more nearly simulate the 50-MW(e) design which has about twice the velocity passing over the outer row of tubes due to the greater volume of tube bundle and longer vapor flow path.

The flow through the demister will also increase, and the demister will not be as effective with the additional recirculated vapor flow. Demister performance could be evaluated for the higher flows to determine if the face area could be reduced in the larger evaporator design.

5.4 CONDENSER

The general arrangement of the test article condenser is shown in Fig. 5-6. The condenser configuration is similar to the evaporator with the same size shell, tubesheet, number of support plates, and support structure.

The condenser tubes are identical to the evaporator tubes. The condenser has 2,830 tubes of each configuration compared to 2,386 tubes in the evaporator. The larger number of condenser tubes fit within a 82.2-in. diameter outer tube limit compared to the 95.4-in. diameter for the evaporator. This is due to the deletion of evaporator tubes for ammonia feed manifolds and for tube columns with small numbers of tubes. However, the same shell size was selected for the evaporator and condenser to save tooling costs. The condenser legs are 15 deg off the centerline of the shell to provide the same condensate drain feature as the reference design.

The baffle plate between the finned and bare tube sections of the tube bundle is a single plate that is partially welded to the shell and tubesheets. It is cut off at the bottom, below the sump surface level, so that the same sump outlet connection can be used for either side of the tube bundle. When one side of the condenser is operating, pressure will be equalized across the baffle plate; therefore the plate thickness can be only 0.5 in.

A noncondensable gas removal system is provided for each half of the tube bundle. The center portion of the tube bundle is enclosed by a partial shroud. Half sections of riser pipes are welded to each side of the divider plate, and separate connections are provided through the top of the shell. The half pipe sections are used to minimize the number of tubes which must be

Fig. 5-6 General Arrangement of Test Article Condenser

deleted to make room for the risers. A larger shroud may be required when the noncondensable flow rate is specified.

The condenser has one ammonia vapor inlet for each half to accept vapor coming from the turbine. An impingement plate at each inlet provides for better vapor distribution in the tube bundle.

5.5 MIST EXTRACTOR

The evaporator has a vapor outlet for each half. A hook and vane type mist extractor inside the shell will provide low moisture ammonia to the turbine. The mist extractor will be of the same design as proposed for the 50-MW(e) design and will have the same ammonia flow per square foot of face area.

The mist extractor is about 9 in. thick and has a series of baffles to change the direction of vapor flow. At each directional change, the liquid droplets are captured by a small channel. The design is based on the standard configuration which is produced by Peerless Manufacturing Company.

The mist extractor performance is based on the product of the vapor density and the square of the velocity. A value of 25 is marginal and 15 is good based on density in lb/ft^3 and velocity in ft/sec . For the selected design of 15, the inlet moisture can be 8 percent for an outlet of 0.1 percent. An inlet quality of 50 percent will produce a discharge of 1 percent.

The liquid ammonia mist extractor drain line runs through the shell wall and back to the sump so that flow can be measured. By monitoring the ammonia quality in the turbine inlet line, the mist extractor performance can be determined and the quality entering the mist extractor can be calculated. This information can be directly extrapolated to the 50-MW(e) design by maintaining v^2 equal.

5.6 ANCILLARY EQUIPMENT

This section discusses the ancillary equipment required for operation of the heat exchanger test articles and which will be supplied as part of the deliverables for use onboard the OTEC-1 Test Facility.

5.6.1 Ammonia Distribution System

The ammonia distribution system for the evaporator incorporates a 15-hp pump capable of providing a maximum flow rate of 60 lb/sec with a head rise of 25 psi. This flow rate allows a 100 percent excess flow over the design flow rate. The distribution system incorporates five levels of feed tubes. Each level is provided with an external valve and metering system to permit individual "on" and "shut-off" capabilities to evaluate the effect of feed rate and the number of tubes in a column. The arrangement is provided for each half of the evaporator tube bundle assembly. In addition, the distribution system provides a small quantity of liquid ammonia to a desuperheater which is part of the supplemental vapor augmentation system described below.

5.6.2 Vapor Augmentation

The vapor augmentation system, for use in fluid dynamic and mechanical testing, incorporates a 25-hp blower capable of recirculating up to 50 percent of the vapor efflux from one side of the evaporator. If simultaneous fluid dynamic performance and mechanical testing is required, the capacity of the blower will be increased accordingly. An adjustable valve and flow meter are incorporated for flow regulation. In addition, the system incorporates a desuperheater which permits the addition of a small quantity of condensate to counter any enthalpy addition from the blower.

5.7 INSTRUMENTATION

To evaluate thermodynamic performance, fluid dynamic performance and mechanical performance of the heat exchanger test articles appropriate instrumentation is being provided and is delineated in Table 5-4.

Table 5-4

TEST ARTICLES INSTRUMENTATION LIST

Measurement Parameter	General Location	No. of Measurements	Sensor Type	Performance Data
Vibration	Condenser Tubes	12	Accelerometer	Mechanical
	Evaporator Tubes	12	Accelerometer	Mechanical
Pressure	Evaporator Vapor Gallery	12	Pitot Tube/ ΔP Xducer	Fluid Dynamic
	Evaporator Vapor Extraction Ports	6	Pitot Tube/ ΔP Xducer	Fluid Dynamic
	Condenser	6	Pitot Tube/ ΔP Xducer	Thermodynamic
Temperature	Evaporator	12	Thermistor Probe	Thermodynamic
	Condenser	6	Thermistor Probe	Thermodynamic
Stress	Evaporator Shell	12	Strain Gage	Mechanical
	Condenser Shell	8	Strain Gage	Mechanical
Flow	Evaporator Mist Extractor	1	Flow Sensor	Fluid Dynamic & Thermodynamic

A preliminary instrumentation schematic (Dwg. No. 553924, Appendix C) has been prepared which reflects the general arrangement of sensors with relation to the heat exchangers. Two J-Boxes will be provided and installed on each heat exchanger for sensor conditioning, power requirements, termination, and Data

Acquisition System (DAS) interface. In addition to providing a normal interface with the government furnished data acquisition system for monitoring tests at the Engineering Control Center, LMSC will provide suitable calibrated instrumentation with adequate sensitivity for local readout of test parameters while under test, if required (e.g., vibration monitoring).

The sensors selected for the heat exchanger test articles are the same types as selected for the 10-MW(e) MAPS. A complete description of the selected sensors appears in Section 4.6.

5.8 PRODUCIBILITY

The aluminum heat exchanger test articles utilize existing fabrication processes and available facilities. The test articles consist of the evaporator and condenser components with instrumentation and pumping and control equipment attached to each, along with the supplementary ammonia vapor system for use with the evaporator. It is planned to attach, install, wire, terminalize, and check out all the measurement and control components after delivery of the completed and tested heat exchangers. The ammonia flow distribution pumps will then be installed along with the flow control valves for the distribution system. The supplemental ammonia vapor system will be fabricated, filled, and delivered for installation on the evaporator when it is installed in OTEC-1.

The 10-MW(e) heat exchanger manufacturing plan was altered to fit the test articles. The major sequences are the same, but some intermediate steps can be deleted due to the material change and smaller size. The modified plan was used to identify tooling, development models, and shop hours for fabrication.

5.8.1 Test Articles

The shell segments will be cold-formed from 2.5-in. thick aluminum plate. The shell assembly will be similar to the sequence for the 10-MW(e) units. The tubesheets will be ordered as a single, circular piece and will not require

any weld seams. The tube support plates will be half sections which will terminate at the divider plate. The final assembly sequence will be the same as for the larger heat exchangers with the addition of the divider plates.

Specific tooling has been identified for the heat exchanger fabrication. A list of tooling and estimated cost is shown in Table E9 of Appendix E. Welding procedures will be qualified for the tube-to-tubesheet weld, other TIG welding, and MIG welding. Manufacturing development mockups will be used for tube-hole repair, tubesheet plugging, tube expansion, plasma cutting tubesheet opening in shell, determining shell seam-welding sequence, and tube-to-tubesheet preshift qualifications.

The completed heat exchangers will weigh about 25 tons and will be about 17 ft in diameter. The heat exchangers can be handled in the shop easily with the existing facilities. Crane lifts during fabrication will use the evaporator vapor ports and the condenser trunnions. The legs will be used for support during shipment.

The overall fabrication cycle from start of work on shell to shipment is 14 months. The schedule is based on a 5-day work week with three shifts per day. Additional front-end time is required to order material, prepare process sheets for the shop, design and fabricate tooling, and perform development tests.

The most critical item is the shell. Aluminum shell plate material is quoted at 12-to-16 weeks delivery from some sources, and one major manufacturer is completely sold out for 5 months in advance. The shell forming and partial assembly is quoted at 26 weeks. The tubesheets and support plates are less critical because no subassemblies are required. The 3-month bundle assembly time is based on two-teams tubing and two-teams welding which is probably the largest number that can work effectively on this size heat exchanger. All material must be requisitioned during the detail design effort, and almost all material orders must be placed prior to the completion of detail design.

5.8.2 Manufacturing and Fabrication Demonstration

Fabrication techniques planned for the commercial heat exchangers will be demonstrated in the test articles, including the fabrication and assembly of the spherical shells, the tube sheets, and the tube-to-tube sheet attachment. The fabrication method related to the enhanced heat transfer tubes will confirm the processes, sealing methods, and production costs. Inspection methods will be demonstrated on the test articles.

5.9 COSTS

Table 5-5 summarizes cost information pertaining to the 0.2-MW(e) heat exchanger test articles. The overall cost of the unit is around \$4,000 /kW(e). Of course, these test articles are in no way intended to demonstrate commercially competitive economics. Due to the nature and design of these heat exchangers, their cost structure is quite different from OTEC heat exchangers. Thus, roughly one-half of total costs are accounted for by shell material and manufacture, and less than 10 percent by the cost of the tubing itself. In passing, it is perhaps well to remark that half the tubes in these test articles are plain tubes, the other half being enhanced-surface tubes.

Table 5-5

HEAT EXCHANGER COST BREAKDOWN: 0.2-MW(e) UNITS
(All Costs in \$/kW(e))

Item	Subcontract & Material	Labor	Total
Tubes	365*	590	955
Tubesheets and Ring	75	185	260
Tube Support Plates and Supports	90	355	445
Shell	1600	485	2085
Water Boxes	NA	NA	NA
Nozzles	20	10	30
Ammonia Distribution and Noncondensable Removal System	30	90	120
Final Assembly	10	235	245
Total	2190	1950	4140

*Tubes at 0.53 \$/ft basic + 0.43 \$/ft enhancement