OCEAN THERMAL ENERGY CONVERSION (OTEC)
TEST FACILITIES STUDY PROGRAM

Final Report, Volume II, Part B.

January 17, 1977

Work Performed Under Contract E(04-3)-1156

Lockheed Missiles and Space Company, Incorporated
Sunnyvale, California

MASTER
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Division of Solar Energy

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**OCEAN THERMAL ENERGY CONVERSION
(OTEC)
TEST FACILITIES STUDY PROGRAM**

VOLUME II PART B

FINAL REPORT

Prepared by
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17 JANUARY 1977

UNDER CONTRACT NO. E(04-3)-1156

PREPARED FOR
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DIVISION OF SOLAR ENERGY

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FOREWORD

Lockheed Missiles & Space Company, Inc. and its team members have completed an 8-month study to develop alternative non-site-specific OTEC facilities/platform requirements for an integrated OTEC Test Program which may include land and floating test facilities. The study was performed for the Energy Research and Development Administration (ERDA) under Contract E(04-3)-1156 initiated with a kick-off meeting on 8 January 1976.

This document, Volume II - Appendixes, is bound in three parts (A, B, and C). Together they comprise a compendium of the most significant detailed data developed during the study.

Volume I provides updated copies of the visual aids and a summary of the oral presentation given to ERDA representatives on 26-27 August at Sunnyvale, California. For the convenience of the reader, the principal section headings for Volume I are listed below:

A. Program Overview	K. Testing II Seabased Cycle Plant
B. Baseline Configuration	L. Testing II Seabased Test Facility
C. Test Plans	M. Data Acquisition and Control System
D. Energy Utilization	N. Cost Summary
E. Scaling	O. Schedule Summary
F. Testing I Requirements	P. Conclusions
G. Testing I Test Beds	
H. Testing II Requirements	
I. Testing II Landbased Cycle Plant	
J. Testing II Landbased Test Facility	

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- 5 **Annotated Test List**
- 6 **Component Tests**
- 7 **Systems Tests**

Appendix 5

ANNOTATED TEST LIST

5.1 INTRODUCTION

The Statement of Work requires that a listing be made of potential AR&T experiments with a brief description, an objective or rationale statement and identification of the testing resources required. As indicated, in the presentation on August 26 and 27, 1976, an additional category of testing is required to include tests which do not fall into the categories of AR&T, Component and Cycle Plant testing and which should be accomplished to assure that the OTEC plant will perform satisfactorily. This category is entitled Engineering Development Tests. In order to meet the requirements of the Statement, an annotated test list, covering these two categories, is presented below. The list includes identification of potential experiments, an objective statement, the model size or scale, testing resources required, and a generic facility description.

5.2 ENGINEERING DEVELOPMENT

5.2.1 Working Fluid Heat Transfer Coefficient

- Objective – Determine the working fluid side coefficient for evaporation and condensation.
- Model – Single tube, or small tube array baseline material, full scale diameter, length not critical.
- Facility – Thermodynamics laboratory with metered working fluid flow and adjustable heat source and sink.

5.2.2 Seawater Pump Inlet Flow Distribution

- Objective – Determine the pressure distribution at the inlet of each pump. Data will be used to develop screens for pump testing.

- Model - 1/50 scale model of platform and module flow path. Quadrant model satisfactory for evaporator pumps. Plenum model required for condenser pumps.
- Facility - Hydraulics laboratory with metered flow. Can be closed loop test if inlet flow provided from large plenum.

5.2.3 Heat Exchanger Inlet Flow Distribution

- Objective - Determine flow distribution at front face of heat exchanger and need for modifications to assure uniformity.
- Model - 1/25 scale model of heat exchanger with multiple pump flows and appropriate transition sections. Exit diffuser for upstream pump location and inlet contraction for downstream pump location.
- Facility - Hydraulics laboratory with metered flow. Can be closed loop test.

5.2.4 Seawater Efflux Reingestion

- Objective - Determine potential for reingestion of evaporator seawater efflux.
- Model - 1/50 scale quadrant model with simultaneous source and sink simulation incorporating heat source with sensitive inlet sensors or means of flow visualization.
- Facility - Hydrodynamics laboratory capable of simulating ocean current.

5.2.5 Cold Water Pipe Flow

- Objective - Determine pressure loss and exit flow uniformity in cold water pipe.
- Model - 1/100 scale cold water pipe with simulated joints and exit plenum.
- Facility - Hydraulics laboratory with metered flow. Can be closed loop test.

5.2.6 Platform Dynamics

Objective - Determine platform response to external stimuli.

Model - 1/100 scale dynamically similar model with scaled restraining or positioning system.

Facility - Hydrodynamics facility with deep well, current and wave simulation.

5.2.7 Electrical Transmission System

Objective - Determine mechanical and electrical integrity of transmission system components.

Model - Full-scale components and sections of cable.

Facility - Vendor test facility. Some tests require seawater environment.

5.2.8 Heat Exchanger Producibility

Objective - Evaluate state-of-the-industry methods with regard to items such as tube/sheet joints, multiple tube assemblies and materials.

Model - Full-scale diameter tube or small array with section of tube sheet.

Facility - Vendor or contractor manufacturing research laboratory.

5.3 ADVANCED RESEARCH AND TECHNOLOGY

5.3.1 Water Side Heat Transfer Enhancement

Objective - Determine increases in water side heat transfer coefficient using tube inserts and special tube surfaces.

Model - Single full-scale diameter tube of sufficient length to assure fully-developed flow.

Facility - Fluid dynamics laboratory with thermodynamic test capability.

5.3.2 Working Fluid Side Performance Improvements

- Objective – Increase working fluid heat transfer by maximizing tube wetting, using special tube surfaces or adding surfactants.
- Model – Small array of full-scale diameter tubes, length not critical.
- Facility – Thermodynamics laboratory with metered working fluid flow and adjustable heat source and sink.

5.3.3 Heat Exchanger Geometry Effects

- Objective – Determine performance improvements provided by geometric variations, such as tube orientation or spacing ratio and by metallic or plastic panel type heat exchangers.
- Model – Small array of full-scale diameter tubes, length not critical; small sections of panel heat exchangers.
- Facility – Thermodynamics laboratory with metered working fluid flow and adjustable heat source and sink.

5.3.4 Heat Exchanger Producibility

- Objective – Determine methods for reducing fabrication costs and increasing reliability with regard to tube/sheet joints.
- Model – Small array of full scale diameter tubes with section of tube sheet.
- Facility – Contractor or vendor manufacturing research laboratory.

5.3.5 Power Turbine Performance

- Objective – Evaluate performance gains provided by new power turbine configurations.
- Model – 1/5 scale turbine models including baseline configuration.
- Facility – Vendor turbine test rig modified for use with working fluid vapor. Liquid injection system required for flow quality variation.

5.3.6 Anti-fouling Techniques

- Objective - Evaluate methods proposed for the prevention and removal of heat exchanger fouling deposits.
- Model - Full-scale diameter tube of sufficient length to assure fully-developed flow.
- Facility - Support rig, located in ocean environment; capable of varying water flow rate and heat flux to simulate evaporator or condenser operation.

5.3.7 Electric Power Transmission

- Objective - Evaluate improved high voltage D.C. transmission systems.
- Model - Full-scale diameter transmission line of sufficient length to determine unit line losses.
- Facility - Vendor supported facility capable of generating high voltage d.c. power with transmission line located in ocean environment.

5.3.8 Energy Utilization

- Objective - Evaluate feasibility of advanced energy utilization scheme.
- Model - Small scale bench set-up.
- Facility - Chemical engineering laboratory with process and storage equipment.

Appendix 6

COMPONENT TEST

6.1 INTRODUCTION

Conclusions in regard to component testing are based on information from a number of sources: comments solicited from independent authorities, information supplied by vendors of typical equipment, analysis of fundamental phenomena, and planning and designing of facilities performed by the study team. This appendix presents a summary of such information in support of sections E, F, and G of the main report, arranged in the following sections:

- 6.2 Solicited Comments on Component Testing**
- 6.3 Machinery Scaling**
- 6.4 Summary of Pump and Turbine Supplier Contacts**
- 6.5 Turbine Testing Requirements: Test Parameters**
- 6.6 Typical Component Tests**
- 6.7 OTEC Land-Based Test Facility Component Test Beds**
- 6.8 Optional OTEC (Major Component) Evaporator and Condenser Test Article Configurations and Their Effect on Test Facility Requirements**

6.2 SOLICITED COMMENTS ON COMPONENT TESTING

Several sources were invited to submit comments on OTEC component testing requirements. The most relevant responses are quoted herein.

6.2.1 T. Y. Lin; T. Y. Lin International: concrete structures

"In view of our previous involvement with your company on this project, and in view of our expertise on concrete and large structures, we will concentrate our discussion on the use of concrete for the following components:

1. Platforms
2. Cold Water Pipe
3. Mooring System

"We believe that concrete, being the main material for OTEC plant, will require intensive and extensive studies in order to avoid critical problems and to arrive at an optimum design. Our suggested studies are listed in 5 groups as below:

- "(I) Applicable to all components is the matter of Concrete as a material. The following lists three concrete materials which need critical study. Note particularly the first one, whose use could seriously change the layout and proportions of the plant.
- a) Lightweight concrete – since this weighs 115 pcf as versus our baseline design using normal weight concrete of 155 pcf, a major change in plant design would result.
 - b) Polymer impregnated concrete (PIC) – this has the advantage of much higher strength and greater resistance against water penetration and abrasion. Its application to the cold water pipe could produce a critical change. Its development is well advanced, and can be made ready for large scale application.
 - c) Fibrous concrete – although still in the developmental stage, it may be applicable in confined areas, where placement of reinforcement is not practical.

The performance of concrete with specific reference to OTEC plants involves:

- a) Long term behavior of concrete in contact with ammonia vapor and liquid.
- b) Long term behavior of concrete under permanent thermal gradient condition; e.g., 40° cold water one side, and possibly 75° room temperature on the other side of the separating concrete.
- c) Resistance of concrete against water penetration for various types and strengths of concrete.
- d) Miscellaneous, such as shrinkage and creep problems in large scale construction and in the ocean environment.

Studies should be made concerning control of certain concrete properties which are critical to OTEC on account of its sensitivity to weight variations:

- a) Development of method for quality and weight control.
- b) Establishment of tolerance limits for certain variables, e.g., weights, out-of-roundness, concrete thickness, finishes, verticality of slip-forming, etc.

"(II) PLATFORMS

- a) A critical problem is the tendency of the platform to "go under" due to the vertical component of the mooring line pull. If the mooring line does not intersect the center gravity of the platform, the centricity will generate pitching movement of the platform undesirable especially for the long cold water pipe below. Thus, we need to neutralize as many disadvantages of this method as possible and refining the structural configuration before it goes to test. One possible refinement is to add a tub-like buoyancy platform on top of the central column, to give it the necessary buoyancy reserve to stop the down drag of the plant under strong lateral forces. The buoyancy platform will also add another margin of safety against excessive loss of buoyancy from an accident or other reasons.

- b) Construction procedure for the platform, including slip-forming, and building the platform on a combination construction and launch barge.
- c) Develop design criteria for unusually large openings in shell structures.
- d) Evaluation of wave forces on this unusual structural shape.
- e) Marine growth and its effect on buoyancy and drag coefficients.

"(III) COLD WATER PIPE

It is generally acknowledged that this is a critical component for OTEC plant, and requires careful design, evaluation, and testing. Some of the problems are as follows:

- a) Construction procedure of cold water pipe by buoyancy plug method.
- b) Model test for installation of cold water pipes.
- c) Methods of repairing and replacing cold water pipes.
- d) Dynamic response of cold water pipe and structure; evaluation of design criteria.
- e) Pitching movement of platform of CWP joints (fatigue problem).
- f) Design of CWP joints and testing.

"(IV) MOORING SYSTEM

The bottom anchored mooring system is more reliable, does not consume energy and is relatively free from maintenance. Concrete is a suitable material for the gravity anchor as well as for the chain links. Most of the following requires testing in addition to design and analysis.

- a) Design of anchor and links, using concrete.
- b) Optimum location of mooring line on platform.
- c) Platform response to weight variations of mooring line.
- d) Dynamic response of system (including differential periodic inertia force on cold water pipe).

"(V) SCALE FACTOR

It is generally agreed that testing concrete structures using a microscale is a very difficult problem and will yield only limited amount of information. The smallest scale used by the authors involved testing a hyperbolic parabolic circular concrete roof of 400-ft diameter, using an 8-ft model, i.e., a scale of 1/50. But this was only for the determination of static and elastic responses, and the test was not really too useful in that it simply reaffirms the computer results.

Of the many concrete bridge and building components tested by the authors, generally, a scale of 1/4 or 1/5 is the minimum. The nuclear reactor vessels tested by General Dynamic, with the author as a consultant, had a scale of 1/4. In the case of our OTEC plant a wall thickness of 20-in. would be reduced to 4 or 5 inches, which is the minimum wall thickness that can be constructed with space left to enclose steel and tendons.

It is suggested that, in this first test facility, two scale ranges be used. For testing hydraulic and dynamic responses, a smaller scale may be sufficient. For testing behavior of concrete structures, the scale must be no less than 1/4 or 1/5.

"On account of the complex and inter-disciplinary nature of the OTEC plant, involving hydraulic, mechanical, chemical and structural problems, the best approach is certainly to build a small prototype and test it at sea. It is possible that while the heat exchange aspects can be tested fairly well on a land-based facility, the structural, constructional, and dynamic aspects may require a prototype tested offshore or in a large basin."

6.2.2 Laskar Wechsler; HYDRONAUTICS, Incorporated: platform hydrodynamics

"There are many types of experiments, scale model testing, and full-scale testing which can be performed in existing facilities around the country, and there should be no overriding requirement for all testing to be accomplished in one location. For

example, pump manufacturers have test facilities for checking the performance of their pumps, and pump scaling laws are well known. If the design of an OTEC plant requires a pump of such size that it cannot be tested in one of the existing facilities, serious consideration should be given to testing the full-size pump in the prototype plant installation rather than at a test facility specially built for OTEC plants. Many of the features to be tested are site related. It is suggested that it will be necessary to develop limited test facilities at several sites to investigate effects of local fouling conditions or currents. In summary, it is suggested that maximum use be made of existing facilities for testing full-scale or reduced size components and systems, and that consideration be given to the construction of full-size prototype platforms in which modules of the full-size plant could be tested. In this manner a realistic test environment would be provided and the size of the power plant could be increased in increments to the full design capacity by the addition of modules. Alternatively, different designs of components could be tested simultaneously.

"With regard to specific AR&T experiments and test facilities required to carry them out, the following comments are offered:

- "1. At present, under ERDA contract, HYDRONAUTICS, Incorporated is conducting a theoretical investigation of the motion in the sea of various types of OTEC platforms. The relative motion of the cold pipe to the platform is also being considered. It will be necessary to conduct model experiments to verify the calculations and possibly to provide revised coefficients in the equation of motion in order to properly characterize the motions. In addition, measurements of strains in the cold pipe can be made.

"The HYDRONAUTICS Ship Model Basin (HSMB[®]) with the new extension incorporating a 40-foot deep, 10-foot diameter deep well is very well suited to tests of OTEC platforms with and without cold water pipes. The extended tank has a width of 25 feet, a nominal water depth (except in deep well) of 11 feet and a length of 400 feet, and is large enough to conduct zero speed tests at any wave heading angle for platform models having a length and/or beam of up to at least 10 feet without significant wall effects. Even larger

models can be tested in bow and stern seas at both zero and finite speeds. Tests can be conducted in both regular and irregular seas (Pierson-Moskowitz spectrum) with maximum wave amplitudes of up to one foot.

"For a 500 MW OTEC platform a model scale of 1/80 to 1/100 would be appropriate. The resulting model size (length or diameter of 5 to 10 feet) for representative platform designs would insure the absence of significant scale effects. The resulting model pipe length (20 to 25 feet) for a typical 2000-foot long prototype pipe could easily be tested within the 40-foot deep well. The model could be constrained by a weak mooring system, either to restrict drift motions and extend testing time or to measure drift forces acting on the platform and/or cold water pipe. For these model sizes it is possible to measure not only platform and pipe motions, but also the loads in the cold water pipe.

"A moored platform/pipe configuration could be tested using bottom moorings whose characteristics simulate those of an actual deep mooring. For typical OTEC site locations, a model mooring depth of 20 to 50 feet and a correspondingly longer mooring scope would be required to actually model the complete mooring - few if any available facilities could model such a mooring.

- "2. Another problem to be investigated is to study the flow fields in the ocean due to the presence of an OTEC plant. ERDA has been supporting a theoretical analysis by the Naval Research Laboratory leading to the development of a computer program to calculate the flow fields and thermal gradients in the vicinity of the OTEC plant. The resulting equations have coefficients which must be determined experimentally. Facilities as follows exist at HYDRO-NAUTICS, Incorporated to determine these coefficients:

- (a) A clear Plexiglass tank in the form of a six-foot cube which may be conveniently stratified with salt or temperature.

- (b) A 75-foot long by 5-foot wide by 5-foot deep water channel which is equipped with a surface wave generating paddle, a wind stream generating fan, an adjustable bottom, and a current generating system for producing recirculating currents and current gradients.
- (c) An 80-foot long channel, 2-feet wide and 2-feet deep, equipped with a surface wave generating paddle and a variable speed instrumentation carriage.
- (d) The HSMB[®] previously described, which has two towing carriages and is equipped to thermally stratify the water.
- (e) Instrumentation is available to measure velocity, pressure and temperature under dynamic conditions and to obtain visual records of the flow fields.

"The above facilities... can provide the required flow field data using models dynamically scaled on the basis of densimetric Froude number, and owing to the large model size approximate Reynolds number scaling is simultaneously achieved."

6.2.3 T. A. V. Cassel; Bechtel Corporation: visit to Allis-Chalmers Custom Pump Division

"The main points of discussion are summarized below:

- They [Allis-Chalmers] agree that a test facility dedicated to OTEC pump models is probably unnecessary in view of model test facilities currently available at A-C as well as at other major pump manufacturing plants. They prefer to engineer and test models at their own facility prior to prototype design and fabrication.
- At the West Allis facility, they have the capability of testing models with up to 16,000 gpm of flow. Measured test parameters include variable suction head, pressure differential across the pump (including casing), motor torque, motor speed, and flow quantities. Cavitation can be investigated using a stroboscope and plexiglas sight ports in the pump casing. To simulate critical velocity profiles at the pump inlet, flow deflecting vanes have been used. High flow

capacities for model tests can be obtained using a 50,000 gpm open pit with wier facility, but accuracy is sacrificed. A-C sees no problem in fabricating and testing the 2.5' diameter impeller models proposed for OTEC tests at the West Allis plant.

- A-C's hydro turbine model test facility at York, Pa., has a smaller capacity than the 16,000 gpm West Allis facility, through the instrumentation at York is more elaborate.
- Velocity profiles, turbulences and vortices at the pump inlet are especially critical to low-head impeller design and performance. Prior to pump design for the OTEC-pilot plant and baseline plant, the baseline intake structures (including seawater flow turning plenums) should be modeled and tested at a hydraulic lab. Alden Labs near Boston and the University of Colorado at Fort Collins have such hydraulic lab facilities (A-C does not). The measured velocity profile should then be simulated at the pump model test facility.

"Regarding the proposed baseline and pilot plant pump designs:

- They feel the baseline speed is much too low and should be increased to, perhaps, 100 rpm. This would drop efficiency from the 80% area of large, slow impellers to around 70% but would permit less costly gearboxes, pumps and motors. This tradeoff between efficiency-and-cost should be investigated further.
- They feel the proposed pilot plant impeller diameters are excessive for the design flows and that the speed is too low. 80-85% pump efficiency is probable at the present proposed design.
- Greater efficiencies (i.e. 80-85%) are possible with large impeller diameters (vs. 60-70% for smaller sizes), but some of this efficiency is lost in increased exit losses.
- Aluminum-bronze is their first material choice for impellers subject to seawater service. They would not guarantee either epoxy or neoprene coated steel and do not believe it is possible to satisfactorily coat impeller blade edges.
- Seals, bearings and gearboxes are seen as critical design problems in the proposed OTEC baseline pumps.

- Following receipt of a prototype order, A-C foresees one year required for design, fabrication and testing of a model followed by two years for design and fabrication of the prototype.
- A-C would prefer that a single total contract package including impeller, vanes, pod, motor, seal, gearbox and bearing be awarded to simplify engineering and manufacture.
- They recommend reversing the flow direction in the present pilot plant pump design. The pod support vanes should be designed to impart a reverse-whirl in the suction flow which would be subsequently straightened by the impeller action. No vanes would then be needed on the discharge side. This will reduce exit losses and is a recommended practice for constant low-head pumps. Also, the present diffuser structure could be reduced in length by one-third by reversing the flow."

6.2.4 Blake King; Clean Energy Research Institute, University of Miami: critical components and testing

"The most critical components of an OTEC plant will be the heat exchangers (most particularly the vapor generator), the cold-water pipe, and the positioning system (mooring or propulsion).

"We have the following comments to make on test configurations and setups:

- Heat exchangers. The most urgently needed information on the performance of heat exchangers can be gained from testing what might be called full-size mini-modules of proposed heat exchanger configurations. Fluid passages should be full-size in terms of flow area and length, but only small aggregations of these elements need be tested to evaluate biofouling effects, flow aberrations, and so on.
- Cold water pipe. It is considered that much valuable data could be obtained from the dynamic testing of scale-model water pipe configurations in a suitable water tunnel at appropriate Reynolds' numbers and with simulated currents. Static and dynamic structural testing of joint designs should also be undertaken.

- c. Positioning system. Testing the strength of a mooring system element should present no problem, and the dynamic testing of scale models of mooring systems should also be relatively easy. Evaluation of the long-term reliability and integrity of a mooring is the difficulty, and we have no quick-and-easy methods to suggest that might be employed to handle it.

"We consider that it is essential that the testing of heat exchanger components be done at a possible OTEC plant site. The major unknowns which must be studied - biofouling and corrosion effects on heat transfer - can only be evaluated realistically when "prototype" sea water is the primary side fluid. Testing of the cold water pipe and of positioning system concepts can be carried out at any convenient location.

"Our principal concern at the University of Miami has been with the problem of heat exchangers. We feel that the following information could be gained from a well-planned test series conducted at a site where natural seawater, of the sort that a full-scale plant would use, is available in quantity -

- a. The effect, as a function of time, of the growth of biofouling on heat exchanger surfaces on (1) heat transfer and (2) pressure drops through flow passages.
- b. The efficacy of various methods proposed for cleaning heat transfer surfaces and for chemically inhibiting the growth of fouling organisms.
- c. The effectiveness of corrosion-control measures, and the consequences on heat exchanger operation of the accumulation of corrosion deposits on or near surfaces.
- d. Identification of fouling species, and the types of corrosion which are experienced."

6.2.5 J. Hilbert Anderson; Sea Solar Power, Inc.: critical components and testing

"Heat exchangers - this subject should be divided into two classes: boilers and condensers. The problems are quite different, and the economic significance is so great that thorough testing is well worth the money.

"Boilers must be tested in models where all passages are full size, and modules must be in sizes large enough to have many passages. Single passage testing is practically worthless. The gas collection and removal system must be similar in size to that of an actual full-size module. While small-scale testing is valuable for research and development -- and should be pursued for a long time to come -- the important objective of this test facility should be to test actual components of usable size under actual operating conditions.

"Performance testing should be done with sea water, very close to actual plant temperatures. Working fluid must be the actual working fluid. A recirculating system can easily be designed, so that the same sea water can be used continually. This has the advantage of eliminating corrosion and fouling from the performance testing phase of the program.

"A separate test facility should be set up for corrosion and fouling testing. This must use raw sea water, and electric potential, galvanic action, and temperature conditions must be closely similar to those that would be encountered in an actual sea thermal power plant. On the other hand, size of the test module can be much smaller than for the performance testing.

"It is most important that the test facility be designed to accept exchanger modules that are pressurized on the outside, so that sea water pressure nearly equals vapor pressure of the boiling fluid. This type exchanger has many advantages in cost and operation and therefore must be provided for in the test facility.

"The condenser test facility can be similar to the boiler facility, but with gas flows properly oriented and sea water temperatures properly maintained. It is probably much easier to refrigerate and recirculate the sea water, rather than to try to supply sea water at naturally cold temperatures.

"It is important in a condenser facility to have length of gas flow path like that of a full-size condenser. It is not so important to have the number of gas flow passages equal to a full-scale condenser.

"Water pumps can be tested better in the manufacturer's plant, as is usual practice. There should be no need for an independent test facility. Scaling of water pumps is regularly done with sufficient accuracy to be acceptable for the purpose. Scaling of 1/10 to 1/5 full size should be quite satisfactory.

"Ammonia turbines are best developed and tested in the manufacturer's plant. However, considerable development work is justified to attain maximum efficiency. Therefore several manufacturer development programs would be justified.

"Platforms need to be tested both in wave tanks and in towing tanks. For towing tanks, the platforms must be large enough to be in a Reynolds number range that will produce characteristics similar to those of a full-size plant.

"Cold water pipes must be tested under drag conditions and under conditions corresponding to storm forces on the platform. This will require a relatively deep wave tank and possibly a towing tank. It is probably also advisable to test drags in a flowing tunnel.

"It is doubtful if mooring systems are worthwhile testing. They are so costly and so site-sensitive that each one becomes an individual engineering gamble.

"Dynamic positioning systems are already in regular use on floating drill rigs and drill ships. There is no need or use for a special test facility for such systems."

6.2.6 G. L. Dugger; The John Hopkins University, Applied Physics Laboratory:

"[The following material] describes our facility capabilities at APL. We have proposed to ERDA some heat exchanger experiments which would demonstrate our system in a closed cycle with heat exchanger models equivalent to about 30-kW (net), based on three 3-in. -O.D., multipass aluminum tubes each in the evaporator and condenser. Instead of taking out power with a turbine, we would use an ammonia chiller at the condenser outlet to cool the ammonia to the temperature desired at the evaporator inlet (the condenser would operate at temperatures about 20° F above the design value). In winter

months, when the water in our cold water reservoir would naturally be cold, we can run the system at true condenser temperatures but lower pressure and lower evaporator temperatures. This compromise saves about \$100K compared to using a turbine and installing enough water chilling capacity to run both evaporator and condenser at nominal conditions simultaneously.

"WARM WATER SUPPLY SYSTEM FOR OTEC EVAPORATOR TESTING

"The warm water circuit utilizes an existing water system comprising a 10,000-gallon reservoir equipped with three remotely-operated water pumps that will have a total flow capacity of at least 2500 gpm. (One additional pump which is on hand will have to be installed to provide this capacity.) In preparation for a given test run, the water in the 10,000 gallon reservoir will be preconditioned to a selected temperature in this range either by use of an existing cooling tower (in hot weather) or by adding warm water from the existing facility boiler (in cold weather). (The cooling tower can cool 500 gpm from 110° F to 85° F when the ambient dry-bulb and wet-bulb air temperatures are 95° F and 75° F, respectively.) The desired range of parametric variation of inlet water temperature to the OTEC evaporator is 75° F to 82° F.

"During a test run, the temperature of the water to the evaporator will be maintained at the selected temperature by the admission of steam from the existing steam accumulator. The boiler can generate steam at a rate of 5200 lb/hr, which for 986 Btu/lb is equal to 5.13 million Btu/hr heat release upon recondensation. Only 88% of this capacity is required to make up an evaporator water temperature drop of 6° F on a continuous basis.

"COLD WATER SYSTEM FOR OTEC CONDENSER TESTING

"The cold water circuit to feed a condenser for OTEC experimentation utilizes the existing PRL [Propulsion Research Laboratory] process water system which consists of a 55,000 gallon skimming basin and a connecting 70,000 gallon spray basin. The condenser water flow rate may be varied up to the existing pumping capacity of 4000 gpm. The water may be cooled in the spray basin by using a 750-gpm pump to recirculate the water through spray-cooling nozzles above the basin. In addition, the purchase of a chiller is planned to provide extra pre-cooling capability when weather conditions demand.

"The water in these basins will be preconditioned before each test run to a temperature which will permit the desired long run duration. This preconditioning can involve either pre-cooling or pre-heating, depending upon the prevailing weather conditions:

- (a) In cold weather, the water will be preheated to a temperature near the nominal condenser inlet value, 58.6° F. The water surface in the spray basin will be insulated by use of floating styrofoam blocks to minimize heat loss. Trimming control of the condenser inlet water temperature will be provided with the steam system, which has the capacity to heat a 1500 gpm water flow through a ΔT of 3° for approximately 8 hours for this service while also providing the nominal reheating ΔT of 5° for the warm water system. Larger ΔT 's could be provided for times less than 8 hours. During steady-state operation, the spray-cooling water system may be used as required to reduce the water temperature from that at the condenser outlet to that at the condenser inlet.
- (b) In warm weather, the water in the basins will be precooled by about 4.8° below the nominal 58.6° F for each hour of desired run time using the spray cooling system and the chiller. During the test run, the water temperature in the basins will increase with time. Steam will be used to trim the water temperature to the desired value at the condenser inlet. Since the water temperature in the spray basin increases by 4.8° per hour, the longest run time for the nominal 58.6° F inlet temperature is determined by the freezing point of water and is 5.5 hours neglecting other heat inputs. There is an ample supply of steam to run both the evaporator and the condenser for this time.

"In a typical test arrangement, evaporator conditions would be fully simulated. The 20° higher-than-nominal temperatures in the condenser would have a small effect on heat transfer, but with appropriate ΔT 's maintained, the performance data would still be applicable for full-scale condenser design. This arrangement will allow run durations up to 5 to 8 hours using existing water-circulating systems at APL.

"In cold weather, the temperature in the cold water basin drops to 32° F and a check of the operation with the condenser at correctly simulated conditions can be made by operating the evaporator at temperature approximately 20° F below normal operating temperatures. For this case, a run time of approximately 2 hours can be achieved.

"TEST INSTRUMENTATION

"The Propulsion Research Facility at APL is well supplied with pressure, temperature, and flow rate sensors and equipment for rapid and accurate measurement, recording, and analysis of data. The shop facilities and technician support for fabrication and handling of the equipment and instruments are available, as are many of the instruments themselves. The facility instrumentation section has developed an outstanding digital data and control system which uses real-time computer control to operate a 50-channel, high-speed (30 KHz) multiplexer and random addressable submultiplexers. The submultiplexers may be installed in the vicinity of the test setup to reduce the wiring and signal conditioning required.

"The data acquisition system is interfaced with the facility control system to provide synchronization during operation of the tests. The control systems are used to attain desired test conditions through use of appropriate feedback loops and to monitor critical variables for automatic actions, including shutdown if necessary to avoid equipment damage or development of hazardous conditions. This instrumentation will assure that the heat transfer data obtained from the experiment will be of high quality and in sufficient quantity for useful application in subsequent development phases in an ocean thermal energy plant-ship program.

"HEAT EXCHANGER TEST CAPABILITIES"

APL-JHU PROPULSION RESEARCH LABORATORY

1. Typical test conditions for 5-hr run

Warm water flow rate = 1500–2000 gpm

Water temp. at evap. inlet = 80° F

Water temp. at evap. outlet = 75° F

Cold water flow rate = 1500–2000 gpm

Water temp. at condenser inlet = 60° F

Water temp. at condenser outlet = 65° F

2. Cold Weather, 2-hr. low temp. run

Warm water flow rate = 1500–2000 gpm

Water temp. at evap. inlet = 60° F

Water temp. at evap. outlet = 55° F

Cold water flow rate = 1500–2000 gpm

Water temp. at condenser inlet = 40° F

Water temp. at condenser outlet = 45° F"

6.3 MACHINERY SCALING

6.3.1 Pump Scaling

The procedures and parameters for scaling, or sizing, pumps have been established by experience in the commercial field and the aerospace industry.

A number of design constants derived by dimensional analysis have long been in use. These include pertinent pump operating and geometric parameters. The most common are:

$$\text{Specific speed} = N_S = \frac{N Q^{1/2}}{H^{3/4}} \quad (1)$$

$$\text{Suction specific speed} = N_{SS} = \frac{N Q^{1/2}}{NPSH^{3/4}} \quad (2)$$

$$\text{Specific head} = H_S = \frac{H}{N^2 D^2} \quad (3)$$

$$\text{Specific capacity} = Q_S = \frac{Q}{N D^3} \quad (4)$$

$$\text{Specific diameter} = D_S = \frac{D H^{1/4}}{Q^{1/2}} \quad (5)$$

$$\text{Specific power} = P_S = \frac{HP}{N^3 D^5} \quad (6)$$

$$\text{Head coefficient} = \psi = \frac{H}{U^2/g} \quad (7)$$

$$\text{Flow coefficient} = \phi = \frac{C_M}{U} \quad (8)$$

Application of the above design constants requires that the flow be dynamically similar and that they be applied only to the same type of pump from which they were originally derived. Then the similarity laws will hold true with an accuracy sufficient for practical purposes throughout a wide range of speeds and sizes. This approach represents the design procedure basis for all pumps. The performance of a known size and type of pump (axial, centrifugal, or mixed flow) can be extrapolated to much larger or smaller pumps. Since economy and facility limitations will dictate the size of the model that can be tested, the design constants can be applied to size the pump model where horsepower or facility flow capacity may be the limiting factors.

At present, it appears that the extrapolation of pump sizes is limited by experience only; there is no theoretical limitation to the use of the design constants if dynamic and geometric similarities are maintained. Pumps with diameters of 22 ft and capacities exceeding 1 million gpm have been designed by this process, e. g., the storage pump at the TVA Hiwassee Dam and the pump at Murphy, California.

Assuming that a large pump has been designed, and its subscale model sized and tested, what will be the performance of the large pump? The head, flow, and speed characteristics can be determined from the design constants, but the efficiency must be determined from one of the empirical formulas developed from the accumulation of much data and experience. The most widely used of these equations is Moody's (Ref. 1):

$$\frac{1 - h}{1 - h_M} = \left(\frac{D_M}{D} \right)^{1/4} \left(\frac{H_M}{H} \right)^{1/10} \quad (9)$$

Another is the German standard DIN 1944 (Ref. 2):

$$\frac{1 - h}{1 - \frac{f}{f_M} h_M} = \left(\frac{D_M}{D} \right)^{1/10} \left(\frac{N_M}{N} \right)^{1/10} \quad (10)$$

in which

$$\frac{f}{f_M} = \frac{1 - 0.546/D^{3/2}}{1 - 0.546/D_M^{3/2}} \quad (11)$$

If the model and the prototype have the same head rise, then the German DIN formula becomes

$$h = \left(\frac{f}{f_m} \right) h_M \quad (12)$$

This has the same form as the Csandy-Rutschi formula (Ref. 3):

$$h = \left(\frac{f}{f_m} \right)' h_M \quad (13)$$

in which

$$\left(\frac{f}{f_m} \right)' = \frac{1 - 0.708/D^{1.6}}{1 - 0.708/D_M^{1.6}} \quad (14)$$

The use of Eqs. (10) and (13) is complicated by the fact that Eqs. (11) and (14) are sensitive to absolute size. This prevents Eqs. (10) and (13) from responding to scale factor alone. As these equations were all developed from test data (and represent the

best curve fit of the data scatter), it is much simpler, and just as accurate, to use the Moody formula; although Csanady gives a procedure for a Reynolds number correction (Ref. 3). The Moody formula will predict efficiencies from 1 to 3 percent lower than the German DIN for 1/5 or less scale models and about 1 percent higher for models larger than 1/4 scale.

The choice of scale factor for seawater pump design testing depends on the approach chosen for a subscale cycle plant. If the cycle plant is required to have complete geometric similarity to the full-scale power module, then four pumps will be required for each heat exchanger and, for example, a 6-MW_e-cycle plant would have pumps of about 6-ft diameter. However, if a single pump per heat exchanger is used, the pump diameter will be 12 ft, which is much closer to the size of the pumps for the full-scale power module. In choosing the scale factor for pump models to verify the design of the cycle plant pumps (either 6- or 12-ft diameter), the pump vendor would select the best compromise between the desire for a relatively small jump in scale and the limits imposed by cost and test capability.

In the resulting pump development sequence, the small-scale test pump and the cycle plant pump provide two stages of verification of the final power module pump design. However, if cycle plant pump performance is to be used to verify the full-scale pump design, measures must be taken to ensure that the velocity profile, boundary layer thickness, and swirl at the cycle plant pump inlet are similar to the full-scale conditions and the NPSH is high enough in the cycle plant test to avoid cavitation.

In conclusion, a pump model is generally sized by the power or flow capacity of the vendor's test facility. Very seldom is a model size specified by a particular scale factor. There are data for all scales down to 1/9, and some pump developers will risk-guaranteeing performance based on a 1/10 scale model but would prefer a model 1/8 scale or larger. The limits appear to be based on practical aspects rather than academic. The use of the cycle plant as a test bed for further verification of pump design prior to building the full-scale pump depends on adequate simulation of full-scale power module pump inlet conditions.

6.3.2 Turbine Scaling

Preliminary design and scaling procedures for turbines differ very little from pump procedures in the use of the design constants for estimating performance. The difference appears when the working fluid becomes compressible, such as the ammonia vapor in the OTEC plant. The volume flow, Q , is then computed at a standard station in the turbine, the rotor exit, and is given in terms of cubic feet per second instead of gallons per minute; otherwise the design constants are the same.

The design constants, or similarity parameters, most widely used for turbine sizing and preliminary design are specific speed N_S , specific diameter D_S , Reynolds number R_N , and Mach number M . The Reynolds number is defined at the rotor exit:

$$R_N = \frac{VD}{\gamma} = \pi \frac{N}{60} \frac{D^2}{\gamma} \quad (15)$$

In this form it can be rewritten to suit the analytical procedures of the designer utilizing one or more of the design constants. Now a known turbine can be scaled up or down keeping geometric relationships constant, such as the blade tip clearance ratio ($\Delta l_t/l_t$), blade height ratio (h/D), hub diameter ratio (D_h/D), trailing edge thickness ratio (tB/h), and disk clearance ratio (Δ/D). As with pumps, extrapolation from radial type to axial type is not permitted, nor is it permissible to compare partial admission turbines with full admission turbines.

The full-scale turbine for the baseline plant is rated at 46 MW. A model of the turbine would be designed with the same head across the rotor and the mass flow would vary linearly with the gross power. Therefore the scale factor, by D_S , becomes

$$SF = \frac{D_M}{D} = \sqrt{\frac{Q_M}{Q}} = \sqrt{\frac{MW_M}{MW}} \quad (16)$$

If the turbine were to be sized for the cycle plant at a gross power of 6 MW,

$$SF = \frac{D_M}{D} = \sqrt{\frac{6}{46}} = 0.36 \quad (17)$$

and the turbine model diameter would be

$$D_M = 0.36 D = 3.44 \text{ ft} \quad (18)$$

The turbine model speed would be

$$N_M = \sqrt{\frac{Q}{Q_M}} \cdot N = 5887 \text{ rpm} \quad (19)$$

Since the minimum recommended (by experience) scale factor for turbine modeling is 0.20, the above sizes exceed this minimum and should yield data that can be safely applied to full-size turbine performance.

6.3.3 Practical Aspects of Model Testing

The feasibility of subscale testing and the general range of acceptable scale factors having been determined, the following aspects of modeling may be discussed:

- Necessity of model testing
- Effect of similarity parameters
- Size and power level of the model
- Optimum specific speed and turbine efficiency
- Systems considerations for the test loop
- Disposition of turbine output power

6.3.3.1 Necessity of Model Testing. At present, no technological problems are anticipated in building large-capacity chemical vapor turbines other than manufacturing problems resulting from the large size. Designing and predicting the aerodynamic

performance of turbines in the 40 to 60-MW range are well within the state of the art. However, the behavior of the working fluid within the range of state points considered for OTEC plants and its effect on turbine performance are not well known. These effects, in particular wetness of ammonia, can best be tested and evaluated by sub-scale testing of the turbine configuration. Testing a model will enable prediction of the performance of a large machine, and indeed manufacturers and industry experts recommend a model test prior to building the more expensive baseline plant machine.

6.3.3.2 Effect of Similarity Parameters. While the dimensionless similarity parameters, specific speed N_s , specific diameter D_s , Reynolds number R_e , and Mach number M , are generally used for turbine modeling, the constancy of R_e and M can be relaxed provided the following criteria are satisfied (Refs. 4, 5, 6):

$$R_e > 2 \times 10^5$$

$$M \leq 0.3$$

Within these limits, the parameters R_e and M have negligible effect and the parameters N_s and D_s have the dominant effect on the correlation between turbine geometry and turbine performance. In the proposed pilot-plant turbine design, the Reynolds number and Mach number at the turbine exit are about 10^6 and 0.25, respectively. Since in the proposed pilot plant turbine design the N_s and D_s are similar to those of the baseline plant turbine design, the model is expected to yield meaningful performance data.

Similarity of pressure and density ratios (turbine inlet to exit) must also be maintained (to within 5 percent; Ref. 4) for the model to yield data that can be used to predict the full-size turbine performance.

6.3.3.3 Size and Power Level of the Model. Constancy of a large number of geometric relationships is needed for scaling a turbine. Manufacturing and economic constraints of some of these relationships (e.g., blade tip clearance ratio, disk clearance ratio)

limit the scale factor. A scale factor of 1/5 is considered reasonable by the turbine manufacturers; i.e., to effectively model a turbine, it is generally not necessary to use a model with larger than one-fifth the dimensions of the full-scale turbine.

Based on this scale factor, the minimum gross power of the model turbine would be:

$$\sqrt{\frac{MW_m}{MW}} = 1/5$$

$$MW_m = \frac{MW}{25}$$

$$\approx 1.84 \text{ MW, when the baseline unit size is 46 MW}$$

Discussion with a turbine manufacturer (Ref. 4) reveals that the only consideration for selecting a larger capacity model is to be able to connect a 3,600-rpm induction motor directly to the turbine. However, the rpm of the model is related to the rpm of the baseline unit according to:

$$N_m = \sqrt{\frac{MW}{MW_m}} \cdot N$$

Thus, to have a 3,600-rpm unit, a 5.1-MW model is needed when the 46-MW baseline unit is designed for 1,200 rpm. If the baseline unit is an 1,800 rpm machine, an 11.5-MW model is needed.

Selection of the pilot plant capacity should consider the pilot plant objectives. If the pilot plant is intended for testing the turbine only, a 1.84-MW turbine is adequate. However, if the plant is intended to demonstrate the technical feasibility of all the critical components, the subscale testing of all tests components must be considered simultaneously and economics of the complete facility will dictate the selection of the turbine capacity.

6.3.3.4 Optimum Specific Speed and Turbine Efficiency. For the types of turbines considered for the OTEC plant, designs based on the specific speed range of 90 to 120 are believed to achieve maximum operating efficiency (Ref. 7). All the turbine designs proposed for the present OTEC plant are constrained within the optimum range without sacrificing efficiency for the sake of convenient rotational speed. This constraint was implicitly assumed in the discussion of the unit capacity and rotational speed of the model turbine.

6.3.3.5 Systems Considerations for Pilot Plant Test Loop. The two configurations of the pilot plant loop appear reasonable. The following aspects of modeling must be considered, however:

- Proper turbine inlet and exhaust conditions must be maintained to ensure similarity of pressure ratio and density ratio within acceptable limits.
- To predict off-design performance of the full-scale turbine, provision must be made to alter the turbine inlet condition. It would be ideal if the real ocean thermal conditions could be simulated.
- The test loop must be capable of maintaining any specific turbine inlet condition for extended periods to facilitate a test run.

6.3.3.6 Disposition of Turbine Output Power. In view of the intermittent operation expected in the pilot plant, it may not be desirable from an economic point of view and from the power company's standpoint to direct the turbine output into the utility grid via a synchronous generator. The turbine shaft power can also be removed by one of the following methods:

- A high-frequency generator may be used and the heat may be rejected in a resistive circuit with the help of air-blast coolers.
- An air compressor may be used to compress air and reject it at higher temperature.
- A water pump may be used and the water head may be destroyed with an expansion valve.

6.3.4 References

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6. C. G. Boone, Airco Cryogenics, Private Communication, 12 Apr 1976
7. L. L. Ambs and R. J. Veenema, Jr., "Power Turbines for Ocean Thermal Energy Conversion Systems," ASME Paper No. 75-WA/OCE-11, Presented at ASME Winter Annual Meeting, Houston, 1975

6.4 SUMMARY OF PUMP AND TURBINE SUPPLIER CONTACTS

A number of pump suppliers were contacted to determine what were the largest units that they had built and if they felt that units of a size and performance required by OTEC were feasible. The performance and size of the preliminary designs were given for the baseline 40-MW module.

6.4.1 Pump Vendors

Five pump vendors responded with information on the technical aspects of pumps for the OTEC application.

- A. Baylor Co., Houston, Texas. M. R. Fitzgibbons wrote that an existing tunnel thruster can act as a pump that will move 1.25×10^6 gpm of sea water with a 6.5-ft head. Its diameter is about 134 in. Back calculations based on static thrust showed an efficiency of 0.89 exclusive of the gear drive.
- B. Hitachi America, Ltd., sent literature that listed an axial flow pump with a 450,000 gpm flow rate at a 27.1-ft head with a 112-in. diameter bore, and a second large pump rated at 325,500 gpm with a head of 81.5 ft with a 134-in. diameter bore. The vendor claimed, over the phone, that they could meet our requirements.
- C. Allis Chalmers, Milwaukee, Wisc. W. W. "Doc" Weltmer maintained that Allis Chalmers could design and develop a pump to meet OTEC specs. He cited a number of large pumps by Allis Chalmers which were about the size required, high flows of low heads and diameters over 20 ft. His facility had a 9500 gpm capacity which is about 5.4 percent of OTEC's flow and would result in a 1/14 scale model. His preliminary design was a 3000 HP unit, tip diameter 16.5 ft, hub diameter 6.25 ft, with the motor inside the hub. He had confidence in extrapolating results from the small model because its Reynolds number would be about 2×10^6 .

- D. Byron Jackson, Los Angeles. Byron Jackson was contacted for delivery time of a 100k gpm pump with a 25 to 35 ft head. Bryon Jackson advocated a mixed flow pump and quoted 15 to 18 months; based on several already delivered. They have confidence and would welcome opportunity to do a 20-ft diameter pump with high flow and low head.
- E. Johnston Pump, Los Angeles. Johnston Pump was contacted to obtain the delivery time for a 100k gpm pump with a 25 to 35 ft head; Johnston Pump quoted about 15 months. Johnston Pump also stated that they could not make a 1000k gpm pump, their biggest size was just under 250k gpm.

PG&E provided information on the pumps (by Hitachi) used in the San Luis Reservoir pumped storage hydroelectric plant. When pump is backflowed, it serves as a turbine to drive a generator. The generator (by G. E.) also reverses roles and serves as motor to drive pump. Pump specs are: $Q = 617,000$ gpm at 290 ft head, $D_t = 96$ in., $N = 120$ and 150 rpm, mixed-flow impeller design.

6.4.2 Turbine Vendors

Turbine vendors surveyed generally agree that an ammonia turbine in the range of 30 to 60 MW_e is very much within the state-of-the-art despite the fact that none has ever been built. There are unresolved questions relating to an ammonia turbine of the configuration and size required for our OTEC system, and there are considerable industry differences of opinion on these questions. For example, the predicted design point efficiencies vary from 85 to about 94 percent and the values vary depending upon the individual assessment of such factors as disc friction, bearing losses, tip losses, inlet and exhaust losses, diffuser recovery, wetness effects, etc. Within this wide range, accurate prediction of design point efficiency can have a substantial effect on the size turbine required for a given output. There are similar questions concerning the unknowns of factors such as off-design performance, ammonia as a working fluid, scaling beyond about 50 percent capacity increase over prior designs, corrosion, and contamination/leakage of ammonia. These may or may not be problems. The ultimate

question is the degree of resolution required before we are ready for a full scale design. With some exceptions, the turbine manufacturers are interested and cooperative, but need funded study programs to proceed.

Following is a brief account of our recent turbine manufacturer contacts.

Airco. Airco data was used in the original 40 MW plant design: axial flow, $HP = 42,846$, $N = 1,800$ rpm, $N_S = 91.57$, $D_S = 1.42$, $h_T = 0.902$, $H = 20.63$, and $D_t = 9.16$ ft. Originally, Airco did not degrade efficiency for wetness but later decided it might be necessary. They also proposed a 60 MW_e turbine and expressed no concern over this degree of scaling. These were single stage axial turbines with a 160-in. diameter wheel for the 60 MW_e unit (about 85,000 hp).

Airco has not built any ammonia turbines and does not know of anyone who has. Although they do build turbines in the 85,000 hp class, these are multistage radial units. The largest axial turbine they have built is in the order of 9,000 hp and probably is not single stage.

Airco quotes a turbine efficiency of 90.2 percent, which includes a 2.0 percent gain for an exit diffuser. They do not take any wetness loss, and provided a theoretical analysis by their Dr. Carter showing why ammonia should not be subject to this loss, but know of no test data supporting Carter's analysis. They feel they can easily predict turbine efficiency within 1 to 2 percent, but could not comment on any corrosion problems with ammonia. They do not feel that seal leakage should be a problem. A development program would not be necessary for a 60 MW_e unit, but a design optimization program would be desirable if we want to get the last 0.1 percent efficiency.

Rotoflow. In their proposal of December 1974, Rotoflow recommended limiting the turbine size to about 20,000 hp, or four units per 60 MW_e module. This was a single stage radial inflow turbine with rotors "over 80 inches" in diameter. (Scaling this to 85,000 hp by the square rule would make the rotor diameter over 165 in.). They say

the turbine for a one- or two-unit 85,000 hp module would be so large that transportation and casting facilities may be problems. In a later letter they stated that cost per unit of power may bottom out at about 25-30,000 hp.

Despite their recommendations for a 20,000-hp size, they are in no way concerned over even a tenfold scaleup from their present 10,000 hp because it would not present a significant extension of their present state-of-the-art. Consequently, they say there would be no significant development cost for a radial turbine. Use of an impulse (axial) turbine, however, would entail a major development program, and when finished it would look much like their present unit.

In our recent contacts they have largely reiterated the above information plus some added comments. They say an axial turbine would be smaller and somewhat less efficient, especially under off-design conditions. However, they quote 85 to 87 percent efficiency, versus Airco's 90.2 percent. John Holm (Director of Sales) says you must be to the right of the saturation line at the turbine inlet (we are not - we're just to the left of it, with some condensation) and, after the demister, any remaining moisture is well suspended as fog. Their rotor has patented features which avoid any resulting efficiency loss and wheel erosion. They have an exit cone (diffuser) that improves efficiency by 1 to 2 percent which is included in their 85 to 87 percent value. They do not see the need for model testing since the technology is fully developed. Also, a model would have to operate at a much higher speed, requiring a reduction gear to get the correct generator speed.

Despite all the above assurances, Rotoflow has not built any ammonia turbines but they see no problem here because they have built turbines for hydrogen, helium, natural gas, oxygen, and nitrogen. However, these are limited to about 10,000 hp, and as far as could be determined, all are multistage units.

Westinghouse. Westinghouse has not built any ammonia turbines, nor any turbines with such a low working fluid temperature. They feel that speed tolerance might be a problem. They consider that an 85,000-hp turbine at our state points is well within

the state-of-the-art. It would probably be a single-stage axial turbine similar in design to the last blade row of utility steam or gas turbines or a large marine propulsion turbine. Special consideration would be required for the materials, the inlet and outlet configuration, and the shaft seals to assure compatibility with the ammonia, to minimize inlet and exhaust losses, and to prevent ammonia leakage into the atmosphere or contamination of the lubrication. The turbine would be sufficiently unique to require a significant development program, but no scientific breakthrough would be required. They are now developing a shaft seal which they believe would be suitable for an ammonia turbine. Normally, they would hesitate to scale prior designs by more than 50 percent capacity increase; however, based upon their past experience with turbines of the size required they believe the turbine need not be a major consideration in choosing the size of a pilot plant.

A pilot plant would not supply much information on statistical failure rates (only one point on a curve), but it would provide an immediate indication of any gross problem, and the test information would be valuable in an effort for design optimization.

Ishikawajima-Harima. Ishikawajima-Harima submitted costs for a 30-MW unit and then stated that a 60-MW ammonia turbine would cost 70 percent more. Efficiency was 84.5 percent.

AiResearch. AiResearch recommended two stages to attain a 0.92 efficiency (dry) and two 21,000 hp turbines driving one generator.

6.4.3 Turbine Researchers

Mr. Ambs (University of Mass.), quickly selects the axial turbine in preference to the radial type because of its improved off-design performance. (As noted above, Rotoflow takes the opposite position.) He did not use a wetness loss in his NSF report of May 1975, but deducted a 2.5 percent η in his ASME paper of December 1975. He (like Airco) says he does not believe the loss will occur with ammonia as it does with steam, but he received so much criticism for leaving it out in May that he added it in

December. He is not aware of any test backup data on wetness and believes some gain can be realized by an exit diffuser but needs more information to quantify it, so he took no credit for this improvement, keeping it as a reserve against the other unknowns such as disc friction, bearing losses, etc. He thinks we could achieve about 91 percent η . If large Reynolds numbers are used (as he has done) he believes the efficiency can be accurately predicted up to his studied size range (35 MW_e, approximately 50,000 hp) but he cautioned that 85,000 hp is a scaleup factor of about 4 for most manufacturers. This could put some unknowns in the efficiency calculations. There might even be some benefits – e. g. , blade tip losses might be less.

He is not aware of any existing ammonia turbines. The turbines used in petrochemical plants such as butane and propane would be the most similar to ammonia. He feels that a development program is required to quantify these factors and others such as corrosion, seals, etc. , and to determine scaling effects in the size range of our interest.

Bryn Beorse stated that the turbine was more critical than the heat exchanger and that extrapolating component performance into system performance is dangerous because unexpected interactions occur.

Kenneth E. Starling (Prof. , University of Oklahoma) states that the turbine test facility should have two-phase flow capability.

6.5 TURBINE TESTING REQUIREMENTS: TEST PARAMETERS

The detailed test criteria and turbine performance objectives will be determined in accordance with overall cycle plant performance goals. Target values of specific items will be included in the detailed test plan. A partial list of items is as follows:

- Turbine efficiency
- Nozzle performance
- Blade performance
- Mass flow
- Speed (rpm)
- Inlet profiles for pressure, temperature, and velocity
- Inlet quality
- Exit profiles for pressure, temperature, and velocity
- Exit quality
- Diffuser performance

The test data will be used to construct a turbine performance map. Typical instrumentation requirements are listed in Table 6-1. It is assumed that the turbine manufacturer will perform the initial development tests with a model whose rotor diameter is of the order of 1 foot. Also, for expediency these first tests may be performed with a working fluid other than ammonia vapor, probably air. In this case, an Engineering Development or AR&T program using a duplicate of the model test turbine should be conducted simultaneously to develop empirical expressions which relate the performance with air to that with varying quality ammonia vapor. These data should be available for the design and performance prediction of the large scale model to be tested in the ERDA test facility.

Table 6-1

**AMMONIA TURBINE FACILITY
INSTRUMENTATION REQUIREMENTS**

<u>Parameter</u>	<u>Location</u>	<u>Number Required</u>	<u>Range-Units</u>
Pressure	Turbine inlet, stream	36	0-200 psi
Temperature	Turbine inlet, stream	12	30-150° F
Pressure	Turbine inlet, boundary layer	16	0-200 psi
Pressure	Turbine inlet, wall static	6	0-200 psi
Temperature	Turbine exit, stream	12	30-150° F
Torque	Turbine drive shaft, outlet	1	0-9000 ft lb
Pressure	Turbine diffuser exit, stream	36	0-200 psi
Pressure	Turbine diffuser exit, wall	6	0-200 psi
Turbine speed	Turbine shaft	1	0-2000 rpm
Power	Compressor drive motor	1	0-31 MW
Compressor Speed	Compressor drive shafts	3	0-4000 rpm
Pressure	Demister exit	1	0-200 psi
Temperature	Demister exit	1	30-150° F
Pressure	Compressor outlets	3	0-200 psi
Temperature	Compressor outlets	3	30-150° F
Pressure Difference	Across NH ₃ metering orifice	1	0-50 psi
Temperature	NH ₃ at orifice	1	30-150° F
Pressure	Desuperheater, inlet	1	0-200 psi
Vibration Accelerations	Turbine	3	-
Temperature	Desuperheater, inlet	1	30-150° F
Pressure	Desuperheater, exit	1	0-200 psi
Temperature	Desuperheater, exit	1	30-150° F
Pressure	Desuperheater, cold water in		0-200 psi

Table 6-1 (Cont.)

<u>Parameter</u>	<u>Location</u>	<u>Number Required</u>	<u>Range-Units</u>
Temperature	Desuperheater, cold water in		30-150° F
Pressure	Desuperheater, cold water out	1	0-200 psi
Temperature	Desuperheater, cold water out	1	30-150° F
Water flow	Desuperheater, cold water out	1	-
Power	Desuperheater water pump	1	100 kW
Pressure	Liquid receiver	1	0-200 psi
Temperature	Liquid receiver	1	30-150° F
Pressure	Liquid ammonia pump	1	0-200 psi
Power	Liquid ammonia pump	1	23 kW
Liquid Ammonia Temperature	At ammonia atomizer liquid inlet	1	30-150° F
Liquid Ammonia Flow Rate	At ammonia atomizer liquid inlet	1	0-60 pps
Pressure	In ammonia atomizer chamber	1	0-200 psi
Temperature	In ammonia atomizer chamber	1	30-150° F
Vapor Ammonia Flow Rate	At ammonia atomizer chamber exit	1	-
Position Indi- cator Lights	Valves open/closed position	20	Color
Position Indi- cators	Remote operated valves	6	0-90°
Power	Pumpdown compressor	1	40 kW
Pressure	Ammonia system, at pumpdown compressor inlet	1	0-200 psi

6.6 TYPICAL COMPONENT TESTS

A number of tests have been postulated in addition to those presented in Section F, Testing I Requirements, of the main report. These are included here as additional material defining the scope of testing anticipated in OTEC development.

COMPONENT TEST LIST

<u>COMPONENT</u>	<u>ELEMENT</u>	<u>TEST</u>
AMMONIA REFLUX PUMP	FULL SIZE UNIT	CAVITATION, PERFORMANCE MAP
AMMONIA FEED PUMP	SCALE MODEL	CAVITATION, PERFORMANCE MAP
AMMONIA CLEANUP SYSTEM	SCALE MODEL	EFFECTIVENESS, POWER DEMAND
PLATFORM	WARM WATER CIRCUIT	VELOCITY PROFILES, INLET & PLUME
	COLD WATER CIRCUIT	VELOCITY PROFILES, PLENUM EXIT & PLUME
	PLANT MODEL	DYNAMICS
MOORING SYSTEM	ANCHOR LINE	DYNAMICS OF TRAPEZE, BRIDLE, SWIVEL
	ANCHOR	EFFECTIVENESS, EMPLACEMENT
COLD WATER PIPE	SCALE MODEL	DYNAMICS, LOADS, JOINT DEFLECTIONS

COMPONENT TEST LIST

The ammonia reflux pump recirculates about four percent of the evaporator flow; for the LMSC plant this is about 2700 gpm. The liquid ammonia at the bottom of the evaporator is at the saturated vapor temperature and as such, is prone to cause cavitation in the pump. Providing an adequate NPSH may be a problem, therefore a pump with an inducer may be required. Tests will be required to determine whether the inducer alone can meet the total pumping requirements or whether it can be used only to build up the inlet pressure for a succeeding stage. Since the head and flow requirements are low, it is suggested that these tests be conducted with a full size pump, using liquid ammonia at controlled inlet temperatures and pressure to determine the minimum NPSH.

Each ammonia feed pump delivers about 31,400 gpm of liquid (LMSC design) from the condenser to the evaporator. A cavitation potential exists for the same reasons as for reflux pump. The head rise requirements are about four to five times that for the reflux pump but still less than 400 ft; which implies that an inducer may be able to perform all of the pumping work. A scale model test is suggested because the two pumps described most likely will be of the same type, mixed flow. The reflux pump could serve as a model for the feed pump and would be tested at different inlet conditions. If the pump designs differ, then it is suggested that a full size feed pump be tested since a pump of this flow rate and head rise will have only about a 40 inch diameter.

The pump test facility would require a large ammonia storage tank, a cooler for removing the head added by the pump, electric power to drive the pump motors, and sufficient valving and plumbing to meter and control the liquid flow.

COMPONENT TEST LIST

<u>COMPONENT</u>	<u>ELEMENT</u>	<u>TEST</u>
TURBINE	SCALE MODEL ASSY	PERFORMANCE MAP
	EXHAUST DIFFUSER	RECOVERY & FLOW PATTERN
	SEALS	MATERIALS EVALUATION
	BEARINGS	LUBRICATION, LIFE
	VARIABLE IGV & NOZZLES	SPEED CONTROL
	FULL-SCALE ASSY	PERFORMANCE MAP
SEAWATER PUMP	SCALE-MODEL ASSY	PERFORMANCE MAP
	INLET GUIDE VANES	PRESWIRL
	EXIT DIFFUSER	RECOVERY & FLOW PATTERN
	BEARINGS	LUBRICATION, LIFE
	SEALS	MATERIALS, LIFE
	MOTOR & GEAR BOX	SPEED CONTROL, LIFE
	FULL-SCALE ASSY	PERFORMANCE MAP

COMPONENT TEST LIST (Cont.)

The ammonia cleanup system draws several MW of power assuming 10 lb/min of water removal. With assumptions on heat recovery, a net parasitic loss of about 0.55 MW occurs. A scale model of the cleanup system could be tested to assure that the measured heat transfer, power consumption, and vapor quality meet the estimated performance.

There are three separate elements of the platform that bear investigation. The hydrodynamics of the two sea water circuits and the dynamic response of the complete plant to ocean surface conditions and ocean currents.

The platform mooring system testing can be done in two setups. The loads and dynamics of the anchor line may be done with or without the above plant model in a land-based test facility. The anchor tests are best done in an environment which duplicates actual ocean bottoms composed of either pelagic clay, biogeneous deposits (siliceous or calcareous oozes), authigenic deposits, or terrigenous sediments, or combinations thereof.

The cold water pipe model can be tested in segments and as a complete unit. The segment tests can be done in a wind tunnel over a range of Reynolds numbers which correspond to the full size in various current velocities. The dynamics of the platform and current loads would be imposed on the full length pipe model to determine its dynamic response and deflections. These tests could be done in a large tow tank with a large diameter well.

A turbine scale model and the full size turbine may be tested in the facility described in succeeding charts. The wide angle diffuser from the turbine exit to the condenser can be tested in a low speed wind tunnel, using air instead of ammonia vapor. Orientation of flow splitters to prevent separation and reverse flow would be the primary variable. Certain turbine components can be bench tested separately, such as the seals and bearings. A breadboard of the speed control loop could be bench tested which would include the sensors, feedback, and the hardware for the variable inlet guide vanes and/or nozzles.

COMPONENT TEST LIST

<u>COMPONENT</u>	<u>ELEMENT</u>	<u>TEST</u>
CONTROL SYSTEM	BREADBOARD	SIMULATED INPUTS
MALFUNCTION DETECTION SYSTEM	SENSORS	SENSITIVITY, CONTROL SYSTEM INTEGRATION

COMPONENT TEST LIST (Cont.)

At this time it is not recommended that a facility be made available for testing scale model sea water pumps. A full-scale test is described in later charts. The three elements which can be tested separately from the pump are the exit diffuser, the bearings, and the seals. The exit diffuser becomes critical if more than one pump is used to supply sea water, and if the pumps are upstream of the evaporators and condensers. Models of a multi-pump installation can be tested in a water tunnel to determine the length required for the mixing to provide a uniform radial and circumferential velocity distribution at the heat exchanger inlet. Bearing life and seal effectiveness can be determined by bench tests of various types of seals and bearings, with lubrication, because rubber bearings may not be successful candidates for a 20-year life.

The control system for plant operation can be breadboard tested with simulated inputs for the control of ammonia pumping, turbine speed, sea-water pumping, etc. to determine system interactions, response times, and amplitudes of excursions.

The malfunction detection system sensors can be laboratory-tested for sensitivity and response times. A means for visual and/or audible signals should be tested, in addition to determining the method for inputting the signals to the central control system.

AT-SEA WATER PUMP TEST

PERFORMANCE MAPPING LIMITS

- POWER = 1.0 TO 5.0 MW
- FLOW = 0.25×10^6 TO 1.8×10^6 GPM
- HEAD = 4 TO 10 FT
- SPEED = 25 TO 100 RPM
- NPSH-IMMERSION

FACILITY DEFINITION

- FULL-SIZE PUMP
- INSTRUMENTATION - 200 CHANNELS
- INFINITE RESERVOIR
- FLOATING PLATFORM
- VERTICAL OR HORIZONTAL SHAFT INSTALLATION
- RETRACTABLE
- THRUST TO 100,000 LB

AT-SEA WATER PUMP TEST

The test facility is intended to test full size sea water pumps with impeller diameters to 25 feet. Measurements of the power input, flow, speed, and head rise will provide data for mapping the efficiency in terms of these parameters. The pump test setup will be installed on a floating platform in an infinite reservoir. Since all tests will be made at steady state conditions, the pressures can be measured by commutated readings, such as from a scanivalve, to reduce the required number of channels. Because both horizontal and vertical shaft pump designs have been proposed, the facility must have the capability for testing both types. Thrust takeout of up to 100,000 lb, or thrust neutralizations, must be considered in the facility design.

AT-SEA WATER PUMP TEST ARRANGEMENT

AT-SEA WATER PUMP TEST ARRANGEMENT

The significant features of this arrangement are that:

The test section can be raised out of the water to provide a shirt-sleeve environment for installing or working on the pump.

The test section can be lowered to various depths for NPSH determinations.

The test section can be rotated to allow for testing of horizontal or vertical shaft pump designs.

Thrust neutralization is provided which relieves the need for additional platform anchoring.

AT SEA PUMP TEST SETUP INTERNAL LINES

NOMINAL VALUES:

$Q \approx 1.35 \times 10^6$ TO 1.8×10^6 GPM

$H \approx 10$ TO 4 FT

$T \approx 70,000$ TO $100,000$ LB

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AT-SEA PUMP TEST SETUP

INTERNAL LINES

The pump test section can test pumps with any diameter to 25 feet simply by modifying the pump spool piece with a different fairing and changing the venturi section to fit the new flow range. Provisions is also made for changing the screen upstream or the pump to simulate the incoming velocity profile to that which occurs in the plant installation. A flow straightener is provided downstream in order to shorten the length required for uniform flow to be established before entering the flow metering section. The Butterfly valve at the end indicates a variable resistance to the flow and provides a means of controlling the pump head.

SEAWATER PUMP DEVELOPMENT

SEAWATER PUMP DEVELOPMENT

As with the turbine, considerable data are available on the parametric performance of large volume flow rate pumps. However, the vendors contacted have indicated that development of seawater pumps, for OTEC requirements, would require one year of small-scale development effort with an additional two-year period for development of a full-scale prototype unit.

Personnel utilization, in the overall program, suggests that the efforts, on the development of the rotating machinery, be conducted sequentially. Therefore, requirements for the seawater pump development efforts will be established by the last Quarter of 1977 and a vendor selected by the end of the first Quarter of 1978. Similarly, requirements for the full-scale seawater pump test facility will be established by mid-1978 and construction initiated at the beginning of 1979.

Delivery of the prototype unit will occur in mid 1981 and, following installation, the test facility will be relocated to the test site. It is assumed that sufficient performance data will be obtained by the last Quarter of 1982 to permit fabrication of an adequate number of pumps for the full-scale module of the sea-based pilot plant.

LAND BASED SEAWATER HYDRODYNAMIC TEST

TEST DEFINITION

- SIMULATE WARM WATER CIRCUIT
- SIMULATE COLD WATER CIRCUIT
- MEASURE INLET & EXIT VELOCITY PROFILES
- MEASURE NEAR & FAR-FIELD FLOW PATTERNS
- MEASURE INTERNAL FLOW CHARACTERISTICS
- VARIABLE IMMERSION DEPTH
- 1 / 50 SCALE MODEL OF 1 / 4 SECTOR

FACILITY REQUIREMENTS

- LARGE WATER TANK
- 30 MODEL EXIT DIAMETERS OF FREE FLOW
- 20 MODEL EXIT DIAMETERS OF DEPTH
- UNDERWATER PHOTOGRAPHY
- FLOW VISUALIZATION
- INSTRUMENTATION – 200 CHANNELS

LAND-BASED SEA WATER HYDRODYNAMIC TEST

The warm water circuit from the inlet to its discharge must be modeled for each of the candidate OTEC plants. The velocity profile at the evaporator face must be measured and fixes developed to make it uniform.

The cold water pipe discharges into a plenum which supplies flow to each condenser. The flow in the plenum is very turbulent which may lead to unequal flows to each condenser, especially if the losses in each of the legs are not balanced. The flow distribution to each condenser, velocity profile at the condenser face, and pressure drops in the circuit must be determined and fixes developed to ensure steady state operation. Since the pumps are the flow regulating device, cycle periods and amplitudes must be determined so that the control system will power the pump to attenuate rather than amplify the unbalanced flow.

The near and far flow field patterns should be mapped to determine if the warm water inlet is reingesting any of the discharge. Available immersion depth will permit determining the magnitude of any surface effects induced by the plumes; or whether large objects with various drafts will affect the intake characteristics.

The test facility can be simply a large tank with dimensions such that wave reflections do not interfere with the data. Provisions should be made for visual and photographic observations of the intake and plume flow patterns. Analog data recording should be used to be able to document the velocity and pressure fluctuations.

OTEC PLANT SEAWATER HYDRODYNAMIC CIRCUIT

SEA WATER HYDRODYNAMIC CIRCUIT

A cross section of the LMSC OTEC plant illustrates that a quadrant of the warm water circuit can be modeled. The radial bulkheads at the inlet provide a natural separation of flow. Four miniature pumps would be installed and an orifice plate or screen would be located at the exit to simulate the evaporator pressure drop.

The model of the cold water circuit would include a short length of the cold water pipe and the four exits simulating the four condensers.

LAND-BASED SEAWATER HYDRODYNAMIC TEST ARRANGEMENT

LAND-BASED SEA WATER HYDRODYNAMIC TEST ARRANGEMENT

The hydrodynamics of the warm and cold sea water circuits could be tested in an above ground 40-ft diameter by 25-ft deep tank. The test would be conducted by personnel on the platform which also supports the control console and data recording equipment.

Three models could be tested - a 1/4 or 1/2 segment of the warm water circuit, a 1/2 or whole segment of the cold water circuit, and a plant model having complete circuits of both warm and cold water. Miniature pumps would be used to vary the flow rate through each leg of the circuit which could be measured by turbine type flow meters.

The warm water model would be tested at various immersion depths to observe surface effects generated by the intake and plume dispersion, with possible effects on the intake pattern.

The cold water model would include a segment of the cold water pipe of sufficient length to have no effect on the discharge plume. The cold water plenum chamber would be modeled and flow division to each of the four modules would be determined.

The complete plant model would be used to determine circulation patterns and plume interactions. The internal ducting would be duplicated to determine the pressure losses and the possible need for flow dividers.

Locating a model within 10 feet of the edge and with a discharge velocity of 1.0 to 3.0 fps would allow a 30 to 10 second data-taking time before wave reflections and flow dispersion at the tank wall started to interfere with the natural flow pattern. The available test time becomes important when considering models which are Froude-scaled or Reynolds-number scaled. Froude scaling relates the dynamic to gravitational affects and the velocity scales directly with the square root of the scale factor. Reynolds scaling relates the dynamic to viscous forces and the velocity varies inversely with the scale factor.

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6.7 OTEC LAND-BASED TEST FACILITY COMPONENT TEST BEDS

Test beds have been designed for a number of major OTEC components. The designs are based on specific design criteria and described in terms of a list of equipment with specified capacities. These designs are the basis for layouts and cost estimates presented in the report. The test beds included are

- Turbine Critical Component Test Bed
- Seawater Pump Test Bed
- Heat Exchanger Critical Component Test Bed
- 2.5-MW Heat Exchanger Component Test Bed
- Hydraulic Circuit Test Bed

OTEC LAND-BASED TEST FACILITY
TURBINE CRITICAL COMPONENT
TEST BED

BACKGROUND

A land-based facility dedicated to testing 1/16, 1/4 and full-size OTEC ammonia turbines is proposed by LMSC as part of an OTEC power plant development program. The facility consists of an ammonia expansion-recompression loop with a demister at the turbo-expander discharge and desuperheating equipment at the compressor discharge. Power from the test turbine is used together with supplementary power from an electric motor to drive the ammonia compressors. A preliminary turbine test bed configuration suggested by LMSC is shown in Figure 5.

DESIGN CRITERIA

LMSC specifies that the test bed be designed such that ammonia vapor conditions (e.g., temperature, pressure, vapor quality, etc.) at both the turbine inlet and outlet match those in the 64 MWe baseline OTEC power plant. Ammonia mass flow rates are 1/16, 1/4 or full-flow baseline rates depending on whether 1/16, 1/4 or full-size turbines are being tested. Turbine conditions from LMSC's baseline analysis (i.e., computer run of 3 March 1976, 1:17:2) provide the design conditions shown in Figures 6, 7, and 8 at locations "N1" and "N2".

Demisting is required because, according to the Elliott Company, the baseline turbine exhaust vapor quality of 0.97 is too wet for direct admission into the ammonia compressor. Liquid is removed in a demister to bring the quality up to the recommended minimum condition of 0.99. Liquid from the demister drain is reinjected with atomizing nozzles into the ammonia vapor pipe upstream from the turbine inlet and serves to regulate vapor quality at the turbine inlet. A water-cooled shell-and-tube heat exchanger provides the first stage of compressor discharge desuperheating prior to liquid injection.

Figure 5
ORIGINAL LMSC TURBINE TEST
BED ARRANGEMENT (SUPERSEDED)

Figure 6
TURBINE TEST 3ED SCHEMATIC
FOR FULL AND 1/4-SIZE TESTS

Figure 7
1/16-SIZE TURBINE TEST
BED SCHEMATIC

Figure 8
AMMONIA CYCLE DIAGRAM
FOR TURBINE TEST BEDS

TEST BED DESCRIPTION

Figures 6 and 7 show the test beds as engineered to accomodate 1/4 and full-size turbines and 1/16-size turbines respectively. A separate test bed is provided for the 1/16-size turbines to avoid turn-down problems which would otherwise be encountered with the full-size demister, motor, pumps, controls and instrumentation. Drawing P-502 shows plan and elevation views and a single-line electrical diagram for the turbine test facility including both full-size and 1/16-size test beds.

Following are equipment and piping requirements for the turbine test facility. Only major system components are listed. Structures, bulk electrical materials, lighting, HVAC equipment, small piping, insulation and other miscellaneous materials are not specifically identified.

OTEC LAND-BASED TEST FACILITY
TURBINE TEST BED

= EQUIPMENT LIST =

Tanks and Vessels

Ammonia demister (Full-size): 2,773 lb/sec NH_3 vapor at 54.65°F ,
97.42 psia, 0.99 quality at outlet; 0.97 quality at inlet; 400
in. I.D., 40 ft. seam-to-seam horizontal vessel; carbon steel;
108 in. diameter end inlet; two 76 in. diameter side outlets;
5 in. liquid drain connection.

Ammonia demister (1/16-size): 173.3 lb/sec NH_3 vapor at 54.65°F ,
97.42 psia, 0.99 quality at outlet; 0.97 quality at inlet; 108
in. I.D., 12 ft. seam-to-seam vertical vessel; carbon steel;
28 in. diameter inlet and outlet; 1-1/4 in. liquid drain connec-
tion.

Ammonia receiver (Full): 450 ft^3 capacity; 5.75 ft. diameter cylin-
drical bullet 18 ft. tangent-to-tangent length with hemispherical
heads; 250 psi design pressure; insulated, carbon steel.

Ammonia receiver (1/16): 12 ft^3 capacity, 1.62 ft. diameter bullet
6.5 ft. tangent to tangent with hemispherical heads; 250 psi design
pressure; insulated, carbon steel.

Desuperheater (Full): Shell-and-tube heat exchanger with NH_3 vapor
shellside and water tubeside; all carbon steel; 15,250 ft^2 tube
surface; 18,431 gal/min water flow; 1,386.5 lb/sec NH_3 vapor flow
(135 psia @ 93.8°F inlet and 131.95 psia @ 79.7°F outlet); 46.12×10^6
Btu/hr capacity; 2 required.

Desuperheater (1/16): As in full-size, above, except: 984 ft^2 tube
surface; 1189.5 gal/min water flow; 86.65 lb/sec NH_3 vapor flow;
 2.976×10^6 Btu/hr capacity; 2 required.

Pumps and Drivers

Desuperheating water pump (Full): 36,900 gal/min @ 17 psi TDH; with nominal 600 hp synchronous motor driver; 4160V, 3-phase, 60 Hz.

Desuperheating water pumps (1/16): 4,100 gal/min @ 17 psi TDH; with nominal 70 hp motor driver; 480V, 3-phase, 60 Hz.

Ammonia reflux pump (Full): 690 gal/min @ 49 psi TDH; with nominal 30 hp motor driver; 480V, 3-phase, 60 Hz.

Ammonia reflux pump (1/16): 43 gal/min @ 49 psi TDH; with nominal 2 hp motor driver; 480V, 3-phase, 60 Hz.

Compressors and Drivers

Ammonia compressors (Full): 3 casings, 2 gearboxes, and motor in tandem on common shaft; two casings required @ 187,000 actual ft³/min each, 97.42 psia inlet, 135.0 psia discharge, 2-stage centrifugal, 34,800 hp required for each at 2000 rpm; one gearbox required @ 2,550 rpm to 2,000 rpm reduction, 70,000 hp transmitted; one casing required @ 124,665 actual ft³/min, 97.42 psia inlet, 135.0 psia discharge, 2-stage centrifugal, 23,200 hp required @ 2,550 rpm; one gearbox required @ 2,550 rpm to 1,200 rpm reduction, 95,000 hp transmitted; motor driver required to deliver 41,470 hp @ 1,200 rpm, synchronous with exciter, 13.8kV, 3-phase, 60 Hz; motor shaft to be double-ended with one end flanged for coupling to test turbine; all compressors to have positive liquid film seals and complete lube and seal oil systems.

Ammonia compressor (1/16): 31,166 actual ft³/min, 97.42 psia inlet, 135.0 psia discharge; 2-stage centrifugal; 5,900 hp required @ 4,942 rpm; positive liquid film seals; complete with lube and seal oil system; one gearbox required @ 4,942 rpm to 1800 rpm reduction, 2,500 hp transmitted; with nominal 2,500 hp synchronous motor driver, 1800 rpm, 4.16 kV, 3-phase, 60 Hz; compressor shaft

to be double-ended with one end flanged for coupling to test turbine.

Instrument air compressor: Nominal 10 hp motor-driven air compressor with receiver, auxiliary systems and controls to supply compressed air as required for instrumentation and controls; 480 V, 3-phase, 60 Hz.

Service compressor: Nominal 50 hp motor-driven compressor for air evacuation and subsequent NH_3 pressurization of system at initial startup; 480 V, 3-phase, 60 Hz.

Special Equipment

Oil Separators: Strip oil from 94°F , 135 psia NH_3 vapor at discharge of compressors and drain back to lube oil system; 4 required @ 520 lb/sec through flow, 42 inch piping connections; 2 required @ 347 lb/sec through flow, 36 inch piping connections; 2 required @ 87 lb/sec through flow, 16 inch piping connections.

Filter-strainers: For NH_3 vapor cleanup prior to compressor suction valves; 2 required @ 1,040 lb/sec through flow, 66 inch piping connections; 1 required @ 693 lb/sec through flow, 54 inch piping connections; 1 required @ 173 lb/sec through flow, 28 inch piping connections.

Desuperheating spray nozzles (Full): Atomize NH_3 liquid @ 147 psia, 54.8°F , and spray into 66 inch pipe of superheated NH_3 vapor @ 132 psia, 79.7°F ; one set of nozzles required for 29.6 lb/sec liquid spray into 1386.6 lb/sec vapor stream; one set of nozzles required for 7.4 lb/sec liquid spray into 346.7 lb/sec vapor stream.

Desuperheating spray nozzles (1/16): Atomize NH_3 liquid @ 147 psia, 54.8°F and spray into 16 inch pipe of superheated NH_3 vapor @

132 psia, 79.7°F; one set of nozzles required for 1.85 lb/sec liquid spray into 86.65 lb/sec vapor stream.

Bridge crane: 100 ton capacity; 68 ft. span traveling bridge; 65 ft. lift @ 12 ft/min; floor operator; 480V, 3-phase, 60 Hz.

Safety Equipment

Fire protection: Two 2-3/4-ton capacity steel-housed refrigerated CO₂ storage units, 110V, single-phase, 60-cycle; include piping, control valves, fire detectors, nozzles, alarms, control panels and two manual 125 ft. hose reels.

Self-contained breathing apparatus: 45 ft³ capacity air cylinders with harness, backpack, regulator and full-face cover air mask; located at several convenient areas in test building.

Gas masks: Industrial size with full-face cover facepiece; seamless canister good up to 2% by volume toxic NH₃ concentration; located at several convenient areas in test building.

Vinyl-coated cotton personnel suit: With expanded back for air cylinder; complete coverage with full-length pressure tight zipper, replaceable gloves and lens assembly in head gear.

Water sprinkler system: Provides fire and ammonia spill protection; includes 30,000 gal. storage tank; 750 gpm @ 100 psig, 50 hp diesel pump; automatic sprinklers; hose stations w/50 ft. ea. of 1-1/2" hose.

Instrumentation and Control

Leak detection: pH analyzer with sensor electrodes in water pipe at inlet and outlet of each desuperheating heat exchanger; includes recorder and alarms; 120 V, 60 Hz.

Data processing computer: Provides monitoring, control, recording, and computational capability; includes capacity for analog measurement of up to 500 transducers (e.g., thermocouples, pressure and velocity

sensors, etc.), digital input/output, operator keyboard with CRT display, line printer, magnetic tape unit for mass data storage, central processing unit, multiplexer equipment and required software; 220 V, 60 Hz.

Cooling water control valve (Full): 36 in. diameter carbon steel butterfly valve with motor positioner; 2 required.

Cooling water control valve (1/16): 12 in. diameter carbon steel butterfly valve with motor positioner; 2 required.

Suction pressure regulators: Carbon steel butterfly damper at NH_3 compressor suction ports; 97.42 psia design operating pressure; modulation to 10% of design pressure required; with pneumatic positioner; 2 required @ 66 inch diameter; 1 required @ 54 in. diameter; 1 required @ 28 in. diameter.

Turbine throttle valve: Carbon steel butterfly throttle valve at turbine inlet ports; 132 psia design operating pressure; with pneumatic positioner; 2 required @ 66 inch diameter for full-size test bed; 2 required @ 16 inch diameter for 1/16-size test bed.

Desuperheating liquid control valve: Carbon steel gate valve for liquid NH_3 at 147 psia and 54.8°F design conditions; with level controller on NH_3 receiver; 1 required @ 5 inch diameter for full-size test bed; 1 required @ 1-1/4 inch diameter for 1/16-size test bed.

Electrical

Circuit breakers: 13.8 kV, 3-phase, 1 required @ 3000A rating and 1 required @ 250A rating; 4.16 kV, 3-phase, 1 required @ 1000A rating, and one required @ 100A rating; 480 V, single phase, 3 required @ 70A rating.

High voltage transformer: 13.8/4.16 kV, 5000 kVA, 3-phase, 1 required.

Mid-voltage transformer: 4160/480V, 500 kVA, 3-phase, 1 required.

Low voltage transformer: 480/240/120 V, 25 kVA, single phase,
3 required.

13.8 kV motor controller: Current limiting contactor-breakers for
41,470 hp motor.

4.16 kV motor controllers: Current limiting contactor-breakers for
600 hp and 2500 hp motors.

480 V motor controllers: Current limiting contactor-breakers for 2 hp,
10 hp, 30 hp, 50 hp, and 70 hp motors.

Communications: A public address system and intercom system with a cen-
tral console in the control room and remote stations on the test
floor is required.

Piping

See Figure 9 for pipe sizes. All ammonia piping is insulated carbon
steel. The use of any copper or copper-base alloys must be avoided
in the ammonia system. Carbon steel shut-off valves shall be installed
upstream and downstream of all major equipment items in the NH_3 system
to permit isolation.

NH_3 Inventory: 16,000 lbs. of liquid anhydrous ammonia is required by
the turbine test facility: 15,600 lbs in the full-size test bed and
400 lbs in the 1/16-size test bed.

Figure 9
SCHEMATIC OF MAJOR PIPING

Stream No. (See Fig. 9)	Description	Fluid	Pipe Mat'l.	Full-size test bed	Pipe Size 1/16 size test bed
1	Turbine outlet	NH ₃ vapor	CS	108 in. OD/0.5 t _m	28 in. OD/0.13 t _m
2	Demister outlet, ½ stream			76/0.37	n.a.
3	Demister outlet, full stream			108/0.5	28/0.13
4	Small compressor inlet			54/0.25	same as "3"
5	Large compressors supply			94/0.44	n.a.
6	Large compressor inlet			66/0.31	n.a.
7	Large compressor inlet			66/0.31	n.a.
8	Small compressor outlet, ½ stream			36/0.25	16/Sch. 10
9	Large compressor outlet, ½ stream			42/0.25	n.a.
10	Sm. & one lrg. compr. disch., ½ stream			54/0.37	n.a.
11	Large compressor outlet, ½ stream			42/0.25	n.a.
12	Desuperheater inlet			66/0.44	same as "8"
13	Desuperheater outlet			66/0.44	16/Sch. 10
14	Turbine inlet, ½ stream	NH ₃ vapor		66/0.44	16/Sch. 10
15	Liquid ammonia reflux, full stream	NH ₃ liquid		5/Std.	1.25/Std.
16	Liquid ammonia reflux, ½ stream	NH ₃ liquid		3/Std.	0.75/Std.
17	Water supply to pump	Water		56/0.5	20/0.37
18	Cooling water pump outlet			48/0.5	16/0.37
19	Desuperheater water inlet			36/0.5	12/0.37
20	Desuperheater water outlet	Water	CS	36/0.5	12/0.37

STARTUP

As indicated on the equipment list, a 50 hp compressor is provided for initial evacuation of air from the ammonia system and subsequent pressurization of the system following an ammonia fill. One inch carbon steel piping is provided from all major vessels and equipment in the ammonia system to the compressor suction port for pumpdown. Air from the compressor discharge is vented to the atmosphere during pumpdown.

The liquid receiver is isolated from the system during the pumpdown operation by shut-off valves on all piping connections. The receiver is filled with liquid ammonia and then is connected to the pumpdown compressor as required to evacuate air from above the liquid surface. Ammonia vaporizes to fill the former air space as pressure within the receiver drops. Receiver pumpdown is stopped when the air evacuation is complete.

Following evacuation of the system, the 50 hp compressor discharge is switched to one inch carbon steel piping connected to the system's vessels and equipment. Ammonia is bled from the receiver into the system and the 50 hp compressor is employed to bring pressures up to operating conditions.

CONTROL

Control of the turbine test conditions is maintained by the following devices:

- Desuperheater cooling water flow control valve
- Desuperheating liquid ammonia flow control valve
- Turbine throttle valves

- Compressor suction valves

UTILITY REQUIREMENTS

The turbine test beds require that both electrical power and 60°F water be supplied to them during normal test operation. Water is required only for the desuperheating heat exchangers; the compressors and motors are air cooled.

Electrical loads and water requirements are as follows for testing turbines of the three sizes specified by LMSC:

	<u>1/16-Size</u> <u>(2.625MW Turbine)</u>	<u>1/4-Size</u> <u>(10.5MW Turbine)</u>	<u>Full-Size</u> <u>(42MW Turbine)</u>
NH ₃ Compressors Motor	1,930kWe @ 1.0PF	9,090kWe @ 0.7PF	32,540kWe @ 1.0PF
NH ₃ Reflux Pump	2kWe @ 0.85PF	10kWe @ 0.6PF	25kWe @ 0.85PF
Desuperheater Water Pump	53kWe @ 0.85PF	74kWe @ 0.5PF	476kWe @ 1.0PF
Miscellaneous Auxiliaries	50kWe @ 0.9PF	65kWe @ 0.9PF	65kWe @ 0.9PF
Total kWe Requirement	<u>2,035kWe</u>	<u>9,239kWe</u>	<u>33,106kWe</u>
Total KVA Requirement	2,050kVA	13,223kVA	33,118kVA
Water Supply Requirement	2379 gpm @ 60°F	4070 gpm @ 60°F	36862 gpm @ 60°F
Water Discharge Handling Requirement	2379 gpm @ 65°F	4070 gpm @ 71.3°F	36862 gpm @ 65°F

VENDOR CONTACTS

The following vendors were consulted during the engineering and design tasks for the turbine test facility:

Allis-Chalmers Corp.
local representative

NH₃ compressors

Cardox Division, Chchemetron Corp.
local representative

fire protection equipment

Carrier Corp.
Syracuse, New York

NH₃ system

Elliott Company
Jeanette, Pennsylvania

NH₃ compressors

General Electric Company
local representative

large motors

Hewlett-Packard
Palo Alto, California

computer system

MSA Company
local representative

personnel safety equipment

Milton Roy Company
local representative

pH analyzer for leak detection

Peerless Mfg. Company
Dallas, Texas

NH₃ demister

SINGLE LINE DIAGRAM

[illegible]

OTEC SEA-BASED TEST FACILITY

SEAWATER PUMP TEST BED

BACKGROUND

A sea-based test bed for testing full scale high capacity OTEC power plant seawater pumps is proposed by LMSC as part of the OTEC development program. The objective of this test bed is to performance test the largest seawater pump being considered for use in OTEC power plant designs. Such testing is to be done at sea using test support facilities contained on an undefined surface vessel. The vessel will be selected and specified at a later date by others to meet requirements being developed in this OTEC Sea-Based Test Facility Study.

DESIGN CRITERIA

LMSC has specified that the pump test bed be designed to accommodate the largest anticipated OTEC pump. This largest pump is identified as having a pumping capacity of up to 1.8×10^6 gal/min and is about 25 ft in diameter.

The pump test bed is to have the capability of testing pumps in either a horizontal or vertical centerline orientation. Horizontal orientation corresponds to the LMSC pump installation concept. A vertical orientation corresponds to the TRW concept with the pump bellmouth up and seawater being drawn down through the pump to simulate the TRW plant warm water circuit.

Pump test installations include various additional equipment items necessary to facilitate, control and obtain data from the pump test. A bellmouth entry section

followed by a short duct, in which velocity profile screens can be mounted, will be installed upstream of the pump. A section of duct with an anti-swirl lattice for flow straightening will be installed downstream of the pump. The last duct section will be a flow metering section ending with a flow control valve followed by a thrust neutralizing device.

Power, instrumentation and control functions will be accomplished from a support vessel above the water surface. Umbilical cord power and control cables will connect between the test bed and the presently unidentified support vessel. Servicing and installation of the test hardware is to be accomplished in a near normal atmospheric environment not requiring divers or unusual life support systems.

TEST BED DESCRIPTION

The design philosophy applied to configuring the seawater pump test bed was to define structural framework within which all testing components can be mounted. This test assembly can then be submerged and operated in either a horizontal or perpendicular orientation of the pump centerline. This design concept is shown on Bechtel Drawings No's P-505 and P-506, enclosed with this write-up. Features incorporated into this test bed conceptual design or assumed to be available from surface support vessels are as follows:

- o The pump and equipment test support structure is an open rectangular framework 35 ft by 35 ft by 170 ft long. It is a welded structure fabricated from pipe and tubing.
- o The top of the support structure is braced with pin jointed cross bracing that is removed to permit installation or disassembly of large test equipment items.

- o The support structure is connected to a strong-back frame at the forward end by hinge pin joints and at the aft end by cables and two latch pins. These connections allow the aft end of the test support structure to be lowered to the vertical centerline orientation by cable winches mounted on the strong-back frame.
- o It is assumed that the strong-back frame is supported at its four end points by rigid jack-up legs from the surface support vessel. It is also assumed that the eventual design of the surface vessel and its jack-up legs operating mechanisms will be capable of lifting the test support structure free of the seawater to permit on-deck service and installation of test equipment.
- o A remotely operated restrictor valve is installed at the exit end of the pump test ducting. As shown on Drawing No. P-505 this valve is made up of multiple rotating slats that can be closed similar to a venetian blind for flow control. This type of valve distributes the restricted opening over the entire duct flow area to minimize flow distortion effects on the upstream pump being tested.
- o A thrust neutralizing nozzle is installed aft of the restrictor valve to minimize pump assembly thrust loads that would be transmitted to the surface support vessel. This device is outlined on Drawing No. P-505 as a conical deflection plate that would turn the axial water flow to exit radially over the full 360 degree circle.

EQUIPMENT REQUIREMENTS

Following are operational and service equipment requirements for this seawater pump test bed. These equipment items are assumed to be part of the test support vessel facilities. Only major equipment items are listed.

Umbilical Cables and Reels: Umbilical cables and cable reels are required to provide power, instrumentation and control connections between the submerged pump test equipment and the surface test support vessel. It is anticipated that two separate umbilical cables will be required; one for pump motor power supply; the second for instrument and control cables. Cable length requirement is approximately 300 ft maximum.

Hydraulic System: A hydraulic system is required for under-sea actuation of cable winches, flow restrictor valves and other control functions that may be incorporated for future tests. Approximate system requirement: 10 gal/min at 2,500 lb/in², 20 hp electric motor, 480 V, 3-phase, 60 Hz.

Service Air Compressor: Nominal 10 hp motor-driven air compressor with receiver, auxiliary systems and controls to supply compressed air as required for instrumentation, controls and shop services: 480 V, 3-phase, 60 Hz.

Test Pump Power Supply: The largest pump to be tested will have a capacity of 1.8×10^6 gal/min and a maximum total differential head of 10 ft. The power supply required for this capacity, with allowances for pump and motor efficiency as well as transformer and line losses, is estimated to be 4.27 MVA. This power requirement is assumed to be 4,160 V, 3-phase, 60 Hz.

Service Crane: The test support vessel will require a crane with a lifting capacity of about 100 tons. It is estimated that this crane would use a 480 V, 3-phase, 60 Hz motor of 60 hp capacity.

Control Room and Instrumentation: A control room incorporating a data process computer is required. The process computer will provide monitoring, control and computational capability; including capacity for analog measurement of up to 250 transducers (e.g. thermocouples, pressure and velocity sensors, etc.) digital input/output, operator keyboard with CRT display, line printer, magnetic tape unit for mass data storage, central processing unit, multiplexer equipment and required software; 220 V, 60 Hz.

Main On-Board Generators: Two 2,500 KVA diesel engine generators; 4,160 V, 3-phase, 60 Hz; including all auxiliaries.

Main Transformer: 4,160/480 V, 150 kVA, 3-phase.

Low Voltage Transformer: 480/120/208 V, 60 kVA, 3-phase.

Circuit Breakers: 4,160 V, 3-phase, 2 required
@ 750 A rating; 480 V, 3-phase, 1 required
@ 140 A rating.

480 V Motor Controllers: Current limiting contactor-breakers for 10, 20 and 60 hp motors.

UTILITY REQUIREMENTS:

The maximum electric load requirement during normal test operation is estimated as follows:

<u>Function</u>	<u>Load, kWe</u>
Service Air Compressor	7.5
Service Crane	44.7
Hydraulic System Pump	15.0
Lighting, Controls, HVAC, & Misc.	45.0
Seawater Pump Testing	<u>4,270.0</u>
Total	4,382.2 kWe

The average electric power consumption during testing is expected to be about 50 percent of the above total (2,091 kWe).

6-82A

THRUST NEUTRALIZER AS REQUIRED BY TEST POSITION

SUPPORT VESSEL SINGLE LINE DIAGRAM

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED PUMP TEST, PUMP SUPPORT STRUCTURE		
JOB No.	DRAWING No.	REV.
11726	P-505	A

6-82B

LMSC/ERDA OTEC TEST FACILITY			
SEA BASED PUMP TEST- PUMP SUPPORT STRUCTURE, LIFTING STRONGBACK			
JOB No.	DRAWING No.	REV.	
11726	P-506	A	

11 X 14 1/2" 5132

OTEC LAND-BASED TEST FACILITY
HEAT EXCHANGER CRITICAL COMPONENT
TEST BED

BACKGROUND

A land-based facility dedicated to testing 1/256 - size OTEC ammonia/water heat exchangers is proposed by LMSC as part of an OTEC power plant development program. The test bed consists of an ammonia vapor compression heat pump cycle coupled with a closed circulating water loop for testing both ammonia evaporators and condensers. These heat exchangers are regarded as "black boxes" for which the test bed provides supporting platforms and flanged interfaces for ammonia and water piping. The heat exchangers are to be erected within an envelope 35 ft. long by 24 ft. wide by 24 ft. high. A preliminary heat exchanger test bed configuration suggested by LMSC is shown in Figure 1.

DESIGN CRITERIA

LMSC specifies that the test bed be designed such that state points (e.g., temperature, pressure, enthalpy, etc.) for water and ammonia streams at the evaporator interface match those in the full-size 64 MWe baseline OTEC power plant. Mass flow rates and the corresponding heat transfer capacity in the evaporator are required to be 1/256 ($=2^{-8}$) of the full-size baseline values. Evaporator conditions from LMSC's baseline analysis (i.e., computer run of 30 January 1976, 13:44:0) provide the design state points and are shown in Figures 2 and 3 at locations "A", "B", "H1", "W1", and "W2".

LMSC suggests that baseline condenser conditions not be considered as design criteria for the test bed. It is possible,

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Checked			TITLE		Model	
Approved					Report No	

HEAT EXCHANGER CRITICAL COMPONENT
TEST BED

HEAT EXCHANGER - ELEVATION

Rec'd 3-10-76

FORM LMSC 362B 1 11 ETZ 1030

Figure 1
ORIGINAL LMSC HEAT EXCHANGER
TEST BED ARRANGEMENT (SUPERSEDED)

Figure 3: Schematic Diagram
for Evaporator Test Bed Design

WATER FLOWS

W1: 26,375 gal/min @ 79.75°F

W2: 26,375 gal/min @ 76.54°F

W3: 26,375 gal/min @ 79.70°F

W4: 1,653 gal/min @ 60°F

EVAPORATOR & CONDENSER PERFORMANCE

H1: Evaporator, 42.37×10^6 Btu/hr, 6.317°F LMTD

H2: Condenser, 41.71×10^6 Btu/hr, 6.554°F LMTD

AMMONIA

Ammonia Flow = 1,356 lb/min (Refer to Figure 2 for ammonia state points.)

however, to test 1/256-size OTEC condensers at baseline state points with minor modification to the test bed design. This option will be discussed in later paragraphs.

TEST BED DESCRIPTION

Figures 2 and 3 show the ammonia cycle pressure-enthalpy diagram and a schematic diagram of the test bed as designed to provide the required baseline conditions for OTEC evaporator testing. An energy balance is achieved in both the closed circulating water loop and the ammonia heat pump cycle by the following ammonia cycle sequence:

- 1,356 lb/min of ammonia vapor from the test evaporator and the demister enters a 900 hp, 2-stage centrifugal compressor at 131.92 psia and 99% vapor quality at "B".
- The vapor is compressed at 75% adiabatic compressor efficiency to 183 psia at "C". Ammonia compression to 183 psia provides the necessary heat for balancing the circulating water loop and for ultimately obtaining a 5.1°F approach between the water and ammonia at the condenser exit.
- The compressed vapor is expanded slightly to 167.07 psia at "D" by passing through an isenthalpic pressure reducing valve. This pressure is required to obtain the corresponding 84.8°F condensing temperature and 6.55°F LMTD in the condenser.
- The ammonia vapor is condensed to saturated liquid at "E" in a condenser which is assumed to have a UA value of 6.364×10^6 Btu/hr °F. This corresponds to 1/256 of the OTEC baseline condenser UA value.
- Saturated liquid leaving the condenser is subcooled in a two-step process from 84.8°F to about 55°F at "G". Subcooling is required to meet the baseline conditions at the evaporator inlet. In the first stage, the ammonia subcools to about 67°F in a shell and tube heat exchanger using

1,653 gal/min of 60°F water, (60°F water was specified by LMSC as being available at the test site). The second stage completes the subcooling using an air-cooled, 130 kWe mechanical chiller to provide 93 tons of cooling capacity.

- The subcooled liquid pressure drops from 164.36 psia to the baseline evaporator inlet condition of 133.46 psia at "A" as it flows through a liquid feed valve prior to entering the evaporator.

Following are equipment and piping requirements for these test bed conditions. Only major system components are listed. Structures, bulk electrical materials, lighting, HVAC equipment, small piping material, insulation and miscellaneous material are not specifically identified.

Tanks and Vessels

Ammonia demister: Shall discharge 1,356 lb/min NH_3 vapor at 71.37°F, 131.92 psia, and 0.99 quality; 42 in. I.D., 11 ft. seam-to-seam; vertical vessel with elliptical heads, carbon steel; 18 in. vapor piping connections; 4 in. liquid drain piping connection.

Ammonia receiver: 250 ft.³ capacity; 66 in. I.D., 0.5 in. wall, 10 ft. seam-to-seam, horizontal vessel with elliptical heads; carbon steel; 6 in. liquid piping connections including 6 in. fill pipe.

Subcooling heat exchanger: 900 SF surface; 823 tubes; 0.5 in. diameter, 16 Ga, CS, 8.3 ft. tube length; "U" = 150 Btu/hr SF °F; shell size approx. 2 ft. dia. x 10 ft. long; transfers 1.655×10^6 Btu/hr.

Pumps and Drivers

Circulating water pump: 26,375 gal/min @ 40 ft. TDH; with nominal 400 hp motor driver; 4160 V, 3-phase, 60 Hz.

Ammonia reflux pump: 137 gal/min NH_3 capacity @ 34.3 ft. TDH; with nominal $1\frac{1}{2}$ hp motor driver; 480 V, 3-phase, 60 Hz.

Subcooling water pump: 1,653 gal/min @ 5 psi TDH; with 8 hp motor driver; 480 V, 3-phase, 60 Hz.

Compressors and Drivers

Ammonia compressor: 3,029 actual ft^3/min inlet @ 131.92 psia, 71.37°F ; 183 psia discharge; 2-stage centrifugal; 10,000 rpm; positive liquid film seals; complete with lube system and gearbox; with nominal 900 hp motor driver; 4160 V, 3-phase, 60 Hz.

Instrument air compressor: Nominal 10 hp motor-driven air compressor with receiver, auxiliary systems and controls to supply compressed air as required for instrumentation and controls; 480 V, 3-phase, 60 Hz.

Service compressor: Nominal 25 hp motor-driven compressor for air evacuation and subsequent NH_3 pressurization of system at initial startup; 480 V, 3-phase, 60 Hz.

Special Equipment

Oil Separator: Strips oil from NH_3 vapor at compressor discharge and drains back to lube oil system; 1,356 lb/min NH_3 vapor through-flow @ 183 psia; 12 in. NH_3 piping connections.

Filter-strainer: For NH_3 vapor cleanup prior to compressor suction valve; 1,356 lb/min through-flow @ 131.92 psia; 18 in. piping connections.

Ammonia subcooler: 93 tons (i.e., 1.116×10^6 Btu/hr) refrigeration capacity; complete skid-mounted assembly including air-cooled condenser, 130 kW motor-driven reciprocating

compressor, heat exchanger for chilling 265 gal/min liquid NH_3 , controls and auxiliaries; 480 V, 3-phase, 60 Hz.

Bridge crane: 100 ton capacity; 75 ft. span traveling bridge, 40 ft. lift; 60 hp; floor operator; 480 V, 3-phase 60 Hz.

Safety Equipment

CO_2 fire protection: Portable refrigerated CO_2 tank mounted on wheels; 750 lb. CO_2 capacity; 1/4 hp refrigeration unit; includes 50 ft. of hose with nozzle having 150 lb/min maximum discharge.

Self-contained breathing apparatus: 45 ft³ capacity air cylinders with harness, backpack, regulator and full-face cover air mask; located at several convenient test areas in test building.

Gas masks: Industrial size with full-face cover facepiece; seamless canister good up to 2% by volume toxic NH_3 concentration; located at several convenient areas in test building.

Water sprinkler system: Provides fire and ammonia spill protection; includes 30,000 gal. storage tank; 750 gpm @ 100 psig, 50 hp diesel pump; automatic sprinklers; hose stations with 50 ft. each of 1½" hose.

Vinyl-coated cotton personnel suit: With expanded back for air cylinder; complete coverage with full-length pressure tight zipper, replaceable gloves and lens assembly in head gear.

Instrumentation and Control

Leak detection: pH analyzer with sensor electrodes in 42 in. circulating water pipe at inlet and outlet of each heat exchanger; includes recorder and alarms; 120 V; 60 Hz.

Data processing computer: Provides monitoring, control, recording and computational capability; includes capacity for analog measurement of up to 500 transducers (e.g. thermocouples, pressure and velocity sensors, etc.) digital input/output, operator keyboard with CRT display, line printer, magnetic tape unit for mass data storage, central processing unit, multiplexer equipment and required software; 220 V, 60 Hz.

Circulating water control valve: 42 in. carbon steel butterfly valve with motor operated positioner.

Ammonia vapor pressure reducing valve: 12 in. carbon steel pressure reducing valve for normal operation at 183 psia high side and 167 psia low side; with pneumatic positioner.

Ammonia liquid feed valve: 6 in. carbon steel liquid regulating valve for normal operation at 164.4 psia high side and 133.5 psia low side; with pneumatic positioner.

Suction pressure regulator: 18 in. carbon steel butterfly damper at NH_3 compressor suction port; 132 psia design operating pressure; modulation to 10% of design pressure required; with pneumatic positioner.

Subcooling water control valve: 12 in. carbon steel butterfly valve with motor operated positioner.

Electrical

Circuit breakers: 4160 V, 3-phase, 1 required @ 2000A rating and 1 required @ 1200A rating; 480 V, 3 phase, 1 required @ 70A rating.

Main transformer: 4160/480 V, 750 kVA, 3-phase.

Low voltage transformer: 480/120/208V, 50 kVA, 3-phase.

4160 V motor controllers: Current limiting contactor-breakers for 900 hp and 400 hp motors.

480 V motor controllers: Current limiting contactor-breakers for 1/4 hp, 1-1/2 hp, 8 hp, 10 hp, 25 hp, 60 hp, and 175 hp motors.

Communications: A public address system and intercom system with a central console in the control room and remote stations on the test floor is required.

Piping (all insulated):

Circulating water: 36 in. square duct, CS, with reinforced 0.375 in. wall.

Subcooling water: 12 in. CS, Sch 40.

NH₃ compressor suction: 18 in. CS Sch 10.

NH₃ compressor discharge: 12 in. CS Sch 10.

NH₃ liquid: 6 in. CS Sch 40.

NH₃ reflux: 4 in. CS Sch 40.

Evacuation and pressurization: 1 in. CS Sch 40.

Shut-off valves: Carbon steel shut-off valves shall be installed upstream and downstream of all major equipment items in the NH_3 system to permit isolation.

NH_3 inventory: 9000 lbs of liquid anhydrous ammonia is required by the system.

Drawing P-501 shows plan and elevation views and a single-line electrical diagram for the heat exchanger test facility including optional equipment for baseline condenser testing.

STARTUP

As indicated on the equipment list, a 25 hp compressor is provided for initial evacuation of air from the ammonia system and subsequent pressurization of the system following an ammonia fill. One inch carbon steel piping is provided from all major vessels and equipment in the ammonia system to the compressor suction port for pumpdown. Air from the compressor discharge is vented to the atmosphere during pumpdown.

The liquid receiver is isolated from the system during the pumpdown operation by shut off valves on all piping connections. The receiver is filled with liquid ammonia and then is connected to the pumpdown compressor as required to evacuate air from above the liquid surface. Ammonia vaporizes to fill the former air space as pressure within the receiver drops. Receiver pumpdown is stopped when the air evacuation is complete.

Following evacuation of the system, the 25 hp compressor discharge is switched to one inch carbon steel piping connected

to the system's vessels and equipment. Ammonia is bled from the receiver into the system and the 25 hp compressor is employed to bring pressures up to operating conditions.

CONTROL

Control of heat exchanger test conditions is maintained by the following devices:

- Compressor suction valve.
- Ammonia vapor pressure reducing valve.
- Circulating water flow control valve.
- Subcooling water flow control valve.
- Ammonia liquid feed regulating valve.
- Reflux flow control valve.
- Chilling regulators in the mechanical subcooling machine.

UTILITY REQUIREMENTS

The heat exchanger test bed requires that electrical power and 60°F water be supplied to it during normal test operation. The circulating water system is a closed loop and the compressor and mechanical subcoolers are air cooled, thereby reducing utility water consumption. 1,653 gpm of 60°F water is required for the first stage of subcooling, which raises the water temperature 2°C at discharge.

The electric load requirement during normal test operation is as follows:

Ammonia compressor	706 kWe
Circulating water pump	298
Mechanical subcoolers	130
Subcooling water pump	6

Reflux pump	1 kWe
Miscellaneous auxiliaries (e.g., instrument air compressor, computer, lighting, HVAC, etc.)	45
	<hr/>

Total Electric Power Requirement 1,186 kWe

CONDENSER TEST OPTION

As mentioned earlier, the test bed shown in Figure 3 is designed to maintain baseline conditions at the evaporator. With some modification, the same test bed may be used for maintaining baseline conditions at the condenser as shown in Figures 4 and 5. For condenser testing, the following test bed modifications are required:

- The ammonia compressor requires a two-speed gearbox to provide 645 hp @ 8,755 rpm for condenser testing (versus 900 hp @ 10,077 rpm for evaporator testing).
- An additional 93.7 tons of mechanical subcooling capacity is required. This provides a total capacity of 186.7 tons to subcool the liquid ammonia from 54.5°F at the condenser exit (point "E" on Figure 4) to the 29°F required at the evaporator inlet (point "A"). Approximately 375 kWe is required for this total subcooling load.
- The water/ammonia subcooling heat exchanger is bypassed because the available 60°F water is too warm for cooling.
- Provisions are required to desuperheat the compressor discharge vapor using atomized liquid from the receiver. Additional equipment includes 3/4 inch CS piping from the receiver to the atomizer, a 1/2 hp liquid ammonia pump (13.4 gpm @ 90 ft. TDH), an atomizing spray nozzle on the compressor discharge piping, and a desuperheating liquid flow control valve.

Utility requirements for condenser testing include only electric power requirements. Bypassing of the subcooling

Figure 5: Schematic Diagram for
Optional Condenser Test Bed Design

WATER FLOWS

W1: 24,250 gal/min @ 49.96°F

W2: 24,250 gal/min @ 46.50°F

W3: 24,250 gal/min @ 49.90°F

EVAPORATOR & CONDENSER PERFORMANCE

H1: Evaporator, 41.99×10^6 Btu/hr, 6.64°F LMTD

H2: Condenser, 41.27×10^6 Btu/hr, 6.48°F LMTD

AMMONIA (Refer to Fig. 4 for State Points)

Ammonia Flow @ "A", "B", "C", and "F" = 1,287 lb/min

Ammonia Flow @ "D", and "E" = 1,356 lb/min

Ammonia Flow @ "G" = 69 lb/min

heat exchanger obviates the need for cooling water as required for evaporator testing. The electric power requirements are:

Ammonia compressor	506 kWe
Circulating water pump	275
Mechanical subcoolers	375
Reflux pump	1
Desuperheating liquid pump	1
Miscellaneous auxiliaries	45
<hr/>	
Total Utility Requirement	1,203 kWe

VENDOR CONTACTS

The following vendors were consulted during the engineering and design tasks for the heat exchanger test facility:

Allis-Chalmers Corp. local representative	NH ₃ compressors
Cardox Division, Chemetron Corp. local representative	fire protection equipment
Carrier Corp. Syracuse, New York	NH ₃ system, complete
Elliott Company local representative	NH ₃ compressors
Hewlett-Packard Palo Alto, California	computer system
MSA Company local representative	personnel safety equipment
Milton Roy Company local representative	pH analyzer for leak detection
Peerless Mfg. Company Dallas, Texas	NH ₃ demister
Wyatt Division, U.S. Industries Dallas, Texas	test procedures

6-98A

SINGLE LINE DIAGRAM

LMSC/ERDA OTEC CRITICAL COMPONENT TEST FAC. GENERAL ARRANGEMENT HEAT EXCHANGER TEST BED		
JOB No.	DRAWING No.	REV.
11726	P-501	B

6-98B

LMSC/ERDA		
OTEC CRITICAL COMPONENT TEST FAC		
GENERAL ARRANGEMENT		
HEAT EXCHANGER TEST BED/APL EXCH.		
JOB No.	DRAWING No.	REV.
11726	P-503	A

July 30, 1976

OTEC LAND-BASED TEST FACILITY
2.5 MW HEAT EXCHANGER COMPONENT
TEST BED

BACKGROUND

A land-based facility dedicated to testing 2.5 MW OTEC ammonia/water heat exchangers is proposed by LMSC as part of an OTEC power plant development program. The test bed consists of an ammonia vapor compression heat pump cycle coupled with a closed circulating water loop for testing both ammonia evaporators and condensers. These heat exchangers are regarded as "black boxes" for which the test bed provides supporting platforms and flanged interfaces for ammonia and water piping. The heat exchangers are to be erected within an envelope 43 ft. long by 24 ft. wide by 35 ft. high.

DESIGN CRITERIA

LMSC specifies that the test bed be designed such that state points (e.g., temperature, pressure, enthalpy, etc.) for water and ammonia streams at the evaporator interface match those in the full-size 70 MWe baseline OTEC power plant. Mass flow rates and the corresponding heat transfer capacity in the evaporator are required to be $1/25$ ($=10 \times 2^{-8}$) of the full-size baseline values. Evaporator conditions from LMSC's baseline analysis (i.e., computer run of 30 January 1976, 13:44:0) provide the design state points and are shown in Figures 2 and 3 at locations "A", "B", "H1", "W1", and "W2".

It is possible to test $1/25$ - size OTEC condensers at baseline state points with minor modification to the evaporator test bed design. This option will be discussed in later paragraphs.

001-9
-2-

Figure 2: Schematic Diagram
for Evaporator Test Bed Design

WATER FLOWS

W1: 263,750 gal/min @ 79.75°F
W2: 263,750 gal/min @ 76.54°F
W3: 263,750 gal/min @ 79.70°F
W4: 16,530 gal/min @ 60°F

EVAPORATOR & CONDENSER PERFORMANCE

H1: Evaporator, 42.37×10^7 Btu/hr, 6.317°F LMTD
H2: Condenser, 41.71×10^7 Btu/hr, 6.554°F LMTD

AMMONIA

Ammonia Flow = 13,560 lb/min (Refer to Figure 2 for ammonia state points.)

TEST BED DESCRIPTION

Figures 1 and 2 show the ammonia cycle pressure-enthalpy diagram and a schematic diagram of the test bed as designed to provide the required baseline conditions for OTEC evaporator testing. An energy balance is achieved in both the closed circulating water loop and the ammonia heat pump cycle by the following ammonia cycle sequence:

- o 13,560 lb/min of ammonia vapor from the test evaporator and the demister enters a 9000 hp 3 stage centrifugal compressor at 131.92 psia and 99% vapor quality at "B".
- o The vapor is compressed at 75% adiabatic compressor efficiency to 183 psia at "C". Ammonia compression to 183 psia provides the necessary heat for balancing the circulating water loop and for ultimately obtaining a 5.1°F approach between the water and ammonia at the condenser exit.
- o The compressed vapor is expanded slightly to 167.07 psia at "D" by passing through an isenthalpic pressure reducing valve. This pressure is required to obtain the corresponding 84.8°F condensing temperature and 6.55°F LMTD in the condenser.
- o The ammonia vapor is condensed to saturated liquid at "E" in a condenser which is assumed to have a UA value of 6.364×10^7 Btu/hr °F. This corresponds to 1/25 of the OTEC baseline condenser UA value.
- o Saturated liquid leaving the condenser is subcooled in a two-step process from 84.8°F to about 55°F at "G". Subcooling is required to meet the baseline conditions at the evaporator inlet. In the first stage, the ammonia subcools to about 67°F in a shell and tube heat exchanger using 16,530 gal/min of 60°F water, (60°F water was specified by LMSC as being available at the test site). The second stage completes the subcooling using a water-cooled 800 kW centrifugal chiller to provide 930 tons of cooling capacity.

- o The subcooled liquid pressure drops from 164.35 psia to the baseline evaporator inlet condition of 133.46 psia at "A" as it flows through a liquid feed valve prior to entering the evaporator.

Following are equipment and piping requirements for these test bed conditions. Only major system components are listed. Structures, bulk electrical materials, lighting, HVAC equipment, small piping material, insulation and miscellaneous material are not specifically identified.

Tanks and Vessels

Ammonia demister: Shall discharge 13,560 lb/min NH_3 vapor at 71.37°F , 131.92 psia, and 0.99 quality; 134 in. I.D., 16 ft. seam-to-seam; vertical vessel with elliptical heads, carbon steel; 38 in. vapor piping connections; 14 in. liquid drain piping connection.

Ammonia receiver: 2500 ft.³ capacity; 144 in. I.D., 0.75 in. wall, 22 ft. seam-to-seam, horizontal vessel with elliptical heads; carbon steel; 20 in. liquid piping connections including 20 in. fill pipe.

Subcooling heat exchanger: 9000 SF surface; 4553 tubes; 0.5 in. diameter, 16 Ga, CS, 15 feet tube length; "U" = 150 Btu/hr SF $^\circ\text{F}$; shell size approx. 5 ft. dia. x 18 ft. long; transfers 1.655×10^7 Btu/hr.

Pumps and Drivers

Circulating water pumps (3): 263,750 gal/min total @ 40 ft. TDH; with 3-1000 hp driver motors; 4160 V, 3-phase, 60 Hz.

Ammonia reflux pump: 1370 gal/min NH_3 capacity @ 34.3 ft. TDH; with nominal 15 hp motor driver; 480 V, 3-phase, 60 Hz.

Subcooling water pump: 16,530 gal/min @ 5 psi TDH; with 80 hp motor driver; 480 V, 3-phase, 60 Hz.

Compressors and Drivers

Ammonia compressor: 30,290 actual ft³/min inlet @ 131.92 psia, 71.37 °F; 183 psia discharge; 3-stage centrifugal; 3,600 rpm; positive liquid film seals; complete with lube system and gearbox; with nominal 9000 hp motor driver; 4160 V, 3-phase, 60 Hz. Starter motor 700 hp required.

Instrument air compressor: Nominal 10 hp motor-driver air compressor with receiver, auxiliary systems and controls to supply compressed air as required for instrumentation and controls; 480 V, 3-phase, 60 Hz.

Service compressor: Nominal 50 hp motor-driver compressor for air evacuation and subsequent NH₃ pressurization of system at initial startup; 480 V, 3-phase, 60 Hz.

Special Equipment

Oil separator: Strips oil from NH₃ vapor at compressor discharge and drains back to lube oil system; 13,560 lb/min NH₃ vapor through-flow @ 183 psia; 38 in. NH₃ piping connections.

Filter-strainer: For NH₃ vapor cleanup prior to compressor suction valve; 13,560 lb/min through-flow @ 131.92 psia; 60 in. piping connections.

Ammonia subcooler: 930 tons (i.e., 1.116×10^7 Btu/hr) refrigeration capacity; complete assembly including water-cooled condenser, 800 kW motor-driver centrifugal compressor, heat exchanger for chilling 2,650 gal/min liquid NH₃, controls and auxiliaries; 4160 V, 3-phase 60 Hz. Exterior cooling tower is required.

Bridge crane: 200 ton capacity; 100 ft. span traveling bridge, 45 ft. lift; 125 hp; floor operator; 480 V, 3-phase, 60 Hz.

Safety Equipment

CO₂ fire protection: Portable refrigerated CO₂ tank mounted on wheels; 2000lb. CO₂ capacity; two hp refrigeration unit; includes 100 ft. of hose with nozzle having 150 lb/min maximum discharge.

Self-contained breathing apparatus: 45 ft³ capacity air cylinders with harness, backpack, regulator and full-face cover air mask; located at several convenient test areas in test building.

Gas masks: Industrial size with full-face cover facepiece; seamless canister good up to 2% by volume toxic NH₃ concentration; located at several convenient areas in test building.

Water sprinkler system: Provides fire and ammonia spill protection; includes 90,000 gal. storage tank; 1750 gpm @ 100 psig, 100 hp diesel pump; automatic sprinklers; hose stations with 50 ft. each of 1½" hose.

Vinyl-coated cotton personnel suit: With expanded back for air cylinder; complete coverage with full-length pressure tight zipper, replaceable gloves and lens assembly in head gear.

Instrumentation and Control

Leak detection: pH analyzer with sensor electrodes in 10 ft. circulating water pipe at inlet and outlet of each heat exchanger; includes recorder and alarms; 120 V, 60 Hz.

Data processing computer: Provides monitoring, control, recording and computational capability; includes capacity for analog measurement of up to 500 transducers (e.g., thermocouples,

pressure and velocity sensors, etc.) digital input/output, operator keyboard with CRT display, line printer, magnetic tape unit for mass data storage, central processing unit, multiplexer equipment and required software; 220 V, 60 Hz.

Circulating water control valves (3): 66 in. carbon steel butterfly valve with motor operated positioner.

Ammonia vapor pressure reducing valve: 38 in. carbon steel pressure reducing valve for normal operation at 183 psia high side and 167 psia low side; with pneumatic positioner.

Ammonia liquid feed valve: 20 in. carbon steel liquid regulating valve for normal operation at 164.4 psia high side and 133.5 psia low side; with pneumatic positioner.

Suction pressure regulator: 38 in. carbon steel butterfly damper at NH_3 compressor suction port; 132 psia design operating pressure; modulation to 10% of design pressure required; with pneumatic positioner.

Subcooling water control valve: 38 in. carbon steel butterfly valve with motor operated positioner.

Electrical

Circuit breakers: 4160 V, 3-phase, 1 required @ 3000 A rating, 1 required @ 2000 A rating; and 7 required at 1200 A rating.

Main transformer: 4160/ 480 V, 750 kVA, 3-phase.

Low voltage transformer: 480 / 120 / 208 V, 150 kVA, 3-phase.

4160 V motor controllers: Current limiting contactor-breakers for 9000 hp, 1000 hp, and 700 hp motors.

480 V motor controllers: Current limiting contactor-breakers for 2 hp, 10 hp, 15 hp, 40 hp, 50 hp, 80 hp, and 125hp motors.

Communications: A public address system and intercom system with a central console in the control room and remote stations on the test floor is required.

Piping (all insulated):

Circulating water: 120 in. CS, with 0.625 in. wall.

Subcooling water: 38 in. CS, Sch 40.

NH₃ compressor suction: 38 in. CS Sch 10.

NH₃ compressor discharge: 38 in. CS Sch 10.

NH₃ liquid: 20 in. CS Sch 40.

NH₃ reflux: 14 in. CS Sch 40.

Evacuation and pressurization: 3 in. CS Sch 40.

Shut-off valves: Carbon steel shut-off valves shall be installed upstream and downstream of all major equipment items in the NH₃ system to permit isolation.

NH₃ inventory: 90,000 lbs of liquid anhydrous ammonia is required by the system.

Drawing P-507 shows plan and elevation views and a single-line electrical diagram for the heat exchanger test facility including optional equipment for baseline condenser testing.

UTILITY REQUIREMENTS

The heat exchanger test bed requires that electrical power and 60 °F water be supplied to it during normal test operation.

The circulating water system is a closed loop and the compressor is air-cooled, thereby reducing utility water consumption.

Waterflow of 16,530 gpm at 60°F is required for the first stage of subcooling, which raises the water temperature 2°C at discharge.

The electric load requirement during normal test operation is as follows:

Ammonia compressor	7200	kWe
Circulating water pumps	2400	
Centrifugal subcoolers	800	
Subcooling water pump	70	
Reflux pump	10	
Miscellaneous auxiliaries (e.g., instrument air compressor, computer, lighting, HVAC, etc.)	170	
	<hr/>	

TOTAL ELECTRIC POWER REQUIREMENT 10,650 kWe

CONDENSER TEST OPTION

As mentioned earlier, the test bed shown in Figure 2 is designed to maintain baseline conditions at the evaporator. With some modification, the same test bed may be used for maintaining baseline conditions at the condenser as shown in Figures 3 and 4. For condenser testing, the following test bed modifications are required:

Figure 4: Schematic Diagram for
Optional Condenser Test Bed Design

WATER FLOWS

W1: 242,500 gal/min @ 49.96 °F

W2: 242,500 gal/min @ 46.50 °F

W3: 242,500 gal/min @ 49.90 °F

EVAPORATOR & CONDENSER PERFORMANCE

H1: Evaporator, 41.99×10^7 Btu/hr, 6.64 °F LMTD

H2: Condenser, 41.27×10^7 Btu/hr, 6.48 °F LMTD

AMMONIA (Refer to Fig. 4 for State Points)

Ammonia Flow @ "A", "B", "C", and "F" = 12,870 lb/min

Ammonia Flow @ "D", and "E" = 13,560 lb/min

Ammonia Flow @ "G" = 690 lb/min

- o The ammonia compressor requires a two-speed gearbox to provide 6450 hp @ 3100 rpm for condenser testing (versus 9000 hp @ 3600 rpm for evaporator testing)...
- o An additional 937 tons of centrifugal subcooling capacity is required. This provides a total capacity of 1867 tons to subcool the liquid ammonia from 54.5°F at the condenser exit (point "E" on Figure 4) to the 29°F required at the evaporator inlet (point "A"). Approximately 1800 kWe is required for this total subcooling load.
- o The water/ammonia subcooling heat exchanger is bypassed because the available 60°F water is too warm for cooling.
- o Provisions are required to desuperheat the compressor discharge vapor using atomized liquid from the receiver. Additional equipment includes 3 inch CS piping from the receiver to the atomizer, a 5 hp liquid ammonia pump (134 gpm @ 90 ft. TDH), an atomizing spray nozzle on the compressor discharge piping, and a desuperheating liquid flow control valve.

Utility requirements for condenser testing include only electric power requirements. Bypassing of the subcooling heat exchanger obviates the need for cooling water as required for evaporator testing. The electric power requirements are:

Ammonia compressor	5160	kWe
Circulating water pump	2220	
Centrifugal subcoolers	1800	
Reflux pump	10	
Desuperheating liquid pump	5	
Miscellaneous auxiliaries	170	

TOTAL UTILITY REQUIREMENT	9365	kWe
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FLEXIBLE RANGE OF OPERATING CONDITIONS VS. REFERENCED BASELINE DESIGN STATE POINT TEST

Bechtel has qualitatively evaluated the changes in the cycle equipment to increase the test bed operating flexibility to accommodate warm water temperature ranging from 75°F to 85°F and cold water values from 40°F to 50°F and ammonia (evaporator) saturation temperatures from 67°F to 77°F and ammonia condenser saturation temperatures from 48°F to 58°F.

The initial preliminary contact with suppliers has indicated that limited changes in the cycle equipment will be required.

In conjunction with the LMSC suggestions, the above level of operating flexibility could be provided with the balance heat extraction accomplished by means of a plant cooling-water aftercooler installed between the NH_3 compressor discharge and the OTEC condenser for evaporator tests, supplemented by a refrigerated water aftercooler for the condenser tests. The present provision of a combined centrifugal and water-cooled subcooler between the receiver and the evaporator would be retained but chilled water cross connections (with control valves) would be provided so that chilled water could be applied alternately for either subcooling liquid NH_3 or aftercooling compressed NH_3 vapor.

It is probable that the addition of controllable after coolers in the compressor discharge line will remove the need for the liquid injection system presently shown between the receiver and the condenser inlet pipe. This subsystem (liquid injection) is therefore a candidate for deletion.

VENDOR CONTACTS

The following vendors were consulted during the engineering and design tasks for the heat exchanger test facility:

Allis-Chalmers Corp. local representative	NH ₃ compressors
Cardox Division, Chemetron Corp. local representative	fire protection equipment
Carrier Corp. Syracuse, New York	NH ₃ system, complete
Elliott Company local representative	NH ₃ compressors
Hewlett-Packard Palo Alto, California	computer system
MSA Company local representative	personnel safety equipment
Milton Roy Company local representative	pH analyzer for leak detection
Peerless Mfg. Company Dallas, Texas	NH ₃ demister
Wyatt Division, U.S. Industries Dallas Texas	test procedures

[illegible] [illegible]

The diagram is a detailed schematic of the Cryogenic Engineering Laboratory (CEL) layout. It shows a rectangular room with various equipment and piping. Key components include:

- Instrumentation & Control Room** (top left)
- Laydown Space** (left)
- APL Evaporator** (center left)
- 700 HP Booster Pumps** (center)
- APL Condenser** (center right)
- 1000 HP Circulating Water Pumps** (bottom right)
- UH₂ REC** (top right)
- UH₂ COMPRESSOR** (bottom center)
- UH₂ LIQ** (top right)
- UH₂ VAPOR** (top right)
- Mask Storage** (multiple locations)
- Exit** (multiple locations)
- Roll-up Door** (left and right)

The overall dimensions are 254' by 144'.

OTEC LAND-BASED TEST FACILITY SEAWATER
HYDRAULIC CIRCUITRY TEST BED

BACKGROUND

A land-based facility for testing 1/50 scale models of OTEC power plants for sea water hydraulic circuit flow characteristics is proposed by LMSC as part of an OTEC power plant development program. The test bed consists of a cylindrical seawater pool 40 feet in diameter and 35 feet deep with supports for mounting the test models within the pool. Also included with this seawater pool test facility is a pump and the piping necessary for draining of the pool as well as for controlled water circulation through the test models.

DESIGN CRITERIA

LMSC has specified that this seawater pool will be used for flow testing of 90 degree sector models reduced in scale from the prototype OTEC power plant configurations. These models will be tested to evaluate the seawater hydraulic circuit characteristics by providing a reasonable simulation of the pressure and velocity gradients at the seawater pump inlets. LMSC has further specified that this seawater pool is not required to simulate current, sea state or water discharge plume effects.

The OTEC platform model sectors will be tested at 1/50 scale of the prototypes. Some testing of 1/16 scale OTEC platform components is anticipated for this seawater test facility, however, configuration and sizing of the test facility is determined by 1/50 scale platform model sector requirements. LMSC 1/50 scale platform model reference dimensions are cold water pipe top inside diameter equals

2.52 ft (scaled from 120 ft) and heat exchanger shell diameter equals 1.46 ft (scaled from 72.75 ft). Additional elements that will be incorporated with the platform models for testing include water inlet sections, a spool section for simulation of intake screens, the pipe section, the pump section, a spool section for simulation of heat exchangers and the discharge section. Flow straighteners and distortion generators will be added upstream and downstream of the pumps.

Elements required in the seawater test facility are a cylindrical test pool with supports for holding test models. Means are to be provided for inserting or extracting the test models and for draining and filling the pool with seawater. Water temperature control is not required.

TEST BED DESCRIPTION

The following Figure 1 is a schematic diagram of the test pool and the seawater pumping systems required for filling and draining the pool.

Figure 1 Seawater Hydraulic Circuitry Test Bed Schematic

Features incorporated into the test facility design are as follows:

- o The seawater test pool is an in ground concrete cylinder of 40 ft inside diameter and 35 ft deep.
- o A bridge structure spanning the pool is provided for test model support and service access.
- o A building incorporating a 20 ton capacity bridge crane is provided to enclose the pool, control room and test model service space.
- o A vertical centerline submersible pump of 2500 gal/min capacity at 50 ft total differential head is provided for draining the pool or alternatively for circulating water through test models. This is a two-stage centrifugal pump driven by an extended shaft from an electric motor above the pool water level.

EQUIPMENT REQUIREMENTS

Following are equipment and piping requirements for this seawater hydraulic circuitry test bed. Only major system components are listed. Bulk electrical materials, lighting, HVAC equipment, small piping material, and miscellaneous material are not specifically identified.

Seawater Pool: To be in ground cylindrical reinforced concrete with 40 ft inside diameter by 35 ft deep; seawater capacity of 329,010 gallons.

Model Support Fixture: A portable bridge structure shall span the pool. This structure shall be sufficient to support models and instrument cables during testing, and to provide personnel access to the models. The bridge structure shall be capable of being hoisted and moved to the service area for model installation, instrumentation and servicing.

Facility Enclosure: A building shall enclose the test pool, control room and test model servicing area. This building shall incorporate support rails for a 20 ton capacity bridge crane. Building size shall be approximately 100 ft long by 70 ft wide by 50 ft high.

Seawater Pump: 2,500 gal/min @ 50 ft total differential head; submersible centrifugal pump with vertical centerline and extended drive shaft to the motor above the water surface. Motor requirement; 65 hp @ 1760 rpm, 480 V, 3-phase, 60 Hz.

Pipe: Approximately 300 ft of 12 in. carbon steel schedule 40 pipe.

Valves: Three 12 in. gate valves.

Bridge Crane: 20 ton capacity; 65 ft span traveling bridge, 80 ft lift; 15 hp; floor operator; 480 V, 3-phase, 60 Hz.

Service Air Compressor: Nominal 10 hp motor-driven air compressor with receiver auxiliary systems and controls to supply compressed air as required for instrumentation, controls and shop service; 480 V, 3-phase, 60 Hz.

Low Voltage Transformer: 480/120/208 V, 35 kVA, 3-phase.

Circuit Breaker: 480 V, 3-phase, 1 required @
170 A rating.

480 V Motor Controllers: Current limiting contactor-breakers for 10, 15 and 65 hp motors.

Communications: A public address system and intercom system with central console in the control room and remote stations on the test floor is required.

Data Processing Computer: Provides monitoring, control, recording and computational capability; includes capacity for analog measurement of up to 100 transducers (e.g. pressure, velocity and flow sensors, etc.) digital input/output, operator keyboard with CRT display, line printer, magnetic tape unit for mass data storage, central processing unit, multiplexer equipment and required software; 220 V, 60 Hz.

UTILITY REQUIREMENTS:

The seawater hydraulic circuitry test bed requires that electrical power, potable water and seawater be available to the facility. Seawater will be required only when the pool is being filled and water is assumed to be available at pressure from the central facility sea water system. Potable water is required only for facility housekeeping and is also assumed to be available from the central facility potable water system. Electrical power is required for test, maintenance and housekeeping operations.

The maximum electric load requirement during normal test operation is estimated as follows:

<u>Function</u>	<u>Load, kWe</u>
Service Air Compressor	7.5
Bridge Crane	11.2
Seawater Pump	46.7
Lighting, Controls, HVAC, & Misc.	30
	<hr/>
Total	95.4 kWe

The average electric power consumption is expected to be about 50 percent of the above total (48 kWe).

6-120A

LMSC/ERDA OTEC TEST FACILITY SEA WATER HYDRODYNAMICS TEST BED			
BECHTEL	JOB No.	DRAWING No.	REV.
	11726	P-504	B

6.8 OPTIONAL OTEC (MAJOR COMPONENT) EVAPORATOR AND CONDENSER TEST ARTICLE CONFIGURATIONS AND THEIR EFFECT ON TEST FACILITY REQUIREMENTS

6.8.1 Discussion Premises

- Premise 1

The facility cooling capacity (aftercooling and/or subcooling of NH_3) will remain unchanged and will correspond to either 0.25 MW_e or 2.5 MW_e (net). This amounts to either 623 Btu/sec or 6230 Btu/sec cooling capacity.

- Premise 2

The shell diameter is the significant dimension which cannot be scaled down for accurate modeling of shell-side performance

- Premise 3

Tube length (and shell length) is not required to be full scale since the water-side heat transfer is relatively invariable with length (except for local entrance effects). For this reason, shortening of the test article (tube length) has little effect on the shell-side performance, whereas all the essential interactions between tube pattern geometry, ammonia liquid falling film and vapor cross-flow are accurately reproduced.

6.8.2 Discussion

The current test facility study, which deals with heat exchanger major component testing, considers test articles that are abbreviated in the tube sheet area and are therefore not identified as scale models but rather as segments of a full-scale heat exchanger. These segments have been shown in Fig. F-24 as square bundles of 2-in. -diameter tubes of either 4 ft by 4 ft tube sheet dimensions (0.25 MW_e) or 12.7 ft by 12.7 ft tube sheet area (2.5 MW_e). The equivalent OTEC power indicated corresponds to full-scale tube length ($\cong 50 \text{ ft}$).

In order to obtain larger tube sheet area than indicated in Fig. F-24, it is feasible to trade tube length for tube sheet area. For example, if the normal tube length of 50 ft

were reduced to 5 ft, the corresponding tube sheet area could be increased essentially inversely to yield a heat exchanger model equivalent to 25 MW_e (40 ft square or 45 ft in diameter) in water flow area but requiring the facility thermal capacity of a 2.5 MW_e OTEC cycle. To use this capability, the heat exchanger test bed would have to be modified in order to recirculate 10 times the water flow rate that has been proposed for the 2.5-MW_e facility.

The example just cited could just as well have been taken to be the upgrading from 0.25 MW_e to 2.5 MW_e so that the test article shown in Fig. F-24 for 0.25 MW_e would in fact have a 12.7-ft-square tube sheet (in lieu of 4 ft) but would have tube and shell length of only 5 ft. The value of W₁, W₂, and W₃ tabulated opposite Fig. G-3 would be increased from 26,375 gpm to 263,750 gpm. Because of the previously mentioned water tube entrance (beneficial) effect on water-side local heat transfer conductance, the performance of the abbreviated tube length test article would be slightly higher than the corresponding full-scale heat exchanger. This disparity could be compensated for by lowering the water velocity slightly for the short-tube model. The small drop or rise in water temperature would also yield larger than full-scale LMTD's which could be adjusted in the controlled water-to-ammonia temperature differential.

Appendix 7

SYSTEMS TESTS

7.1 INTRODUCTION

Systems Tests are those tests conducted on an arrangement of power cycle equipment which is called cycle plant. Appendix 7 presents data pertaining both to land based and sea based cycle plant testing requirements and arrangements which corresponds to Report Sections H, I, and K. The contents of this Appendix is

- 7.2 Cycle Plant Definition
- 7.3 Cycle Plant Diagrams
- 7.4 Power Plant Thermal Resources
- 7.5 Land Based Cycle Plant
- 7.6 Sea Based Cycle Plant

7.2 CYCLE PLANT DEFINITION

The cycle plant diagram is a schematic representing the power cycle with key annotated state points. The cycle plant systems provide a functional as well as component breakdown to assist in test identification and definition.

CYCLE PLANT DIAGRAM

CYCLE PLANT DIAGRAM

THE CYCLE PLANT SCHEMATIC DIAGRAM SHOWS THE WORKING FLUID AND WATER FLOW STATE POINTS. THE ALPHA/NUMERIC NOTATIONS ENCIRCLED ON THE DIAGRAM REPRESENT WORKING FLUID AND WATER FLOW POINTS. THE ENCIRCLED ARABIC NUMERALS PERTAIN TO THE WORKING FLUID MAIN FLOW CONDITIONS STARTING AT DEMISTER OUTLET AND ENDING WITH DEMISTER INLET. THE ENCIRCLED ROMAN NUMERALS APPLY TO THE WORKING FLUID SECONDARY FLOW OF LIQUID EXTRACTED FROM THE VAPOR IN THE DEMISTER AND THE FLUID RECYCLED BY THE REFLUX PUMP. THE SEA WATER FLOW IS REPRESENTED BY THE FOUR ENCIRCLED ALPHAS WITH THE WARM WATER FIRST.

CYCLE PLANT SYSTEMS

FUNCTIONAL SYSTEMS

AMMONIA VAPOR
AMMONIA CONDENSATE
COLD/WARM WATER CIRCULATION
INSTRUMENTATION AND CONTROLS

MAIN COMPONENTS

EVAPORATOR
TURBINE/GENERATOR
CONDENSER
CONDENSATE PUMPS
COLD/WARM WATER PUMPS

AUXILIARY SYSTEMS

* AMMONIA CLEANUP
FOULING COUNTERMEASURES
AMMONIA STORAGE/TRANSFER
AUXILIARY COOLING WATER
CATHODIC PROTECTION
ELECTRICAL
LUBE OIL

COMPRESSED AIR
LEAK DETECTION/FLUID SAMPLING
EQUIPMENT HANDLING
FIRE PROTECTION
SAFETY EQUIPMENT
NITROGEN PURGE

* OPERATIONAL TESTS

CYCLE PLANT SYSTEMS

TO ASSIST IN CYCLE PLANT TEST DEFINITIONS, TESTING REQUIREMENTS AND TEST BED ARRANGEMENTS, THE CYCLE PLANT IS SHOWN IN FUNCTIONAL AND COMPONENT BREAKDOWNS AS WELL AS IT'S MAJOR AUXILIARY SYSTEMS. THE CYCLE PLANT FUNCTIONAL BREAKDOWN CONTAINS TWO AMMONIA LOOPS - VAPOR AND CONDENSATE OR LIQUID, THE WATER CIRCULATION LOOPS AND THE INSTRUMENTS AND CONTROL. FROM A HARDWARE STANDPOINT THE MAIN COMPONENTS ARE IDENTIFIED AS THE EVAPORATOR, THE TURBINE/GENERATOR, THE CONDENSER, THE CONDENSATE PUMPS AND THE COLD/WARM WATER PUMPS.

THE MAJOR AUXILIARY SYSTEMS LIST SERVE TWO PURPOSES. FIRST AS A GUIDE, AS STATED ABOVE, FOR DEFINING TESTING REQUIREMENTS SUCH AS NOTED BY THE AMMONIA CLEANUP AND FOULING COUNTERMEASURES. SECOND, THE LIST PROVIDES A BASIS FOR DEFINITION OF THE EQUIPMENT NECESSARY FOR THE CYCLE PLANT TEST ARRANGEMENT.

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7.3 CYCLE DIAGRAMS

Three sets of cycle diagrams are included in this section to supplement the typical example given in Section H and to show state point differences from April 1975. The first set contains the June 1976 baseline state points followed by the seven cycle plant diagrams for 1, 3, 6, and 9 MW_e and both titanium and aluminum tubes. The second set of diagrams are the proposed March 1976 baseline containing the proposed baseline and 3, 6, and 9 MW_e titanium tube evaporator and condenser. The third set contains the April 1975 baseline and 2 and 4 MW_e cycle plant diagram.

BASELINE POWER CYCLE

NET POWER OUTPUT	66.11	MW
GENERATOR GROSS	90.39	MW
TURBINE GROSS	91.76	MW
RANKINE CYCLE EFF	2.936	%
SEA WATER PUMP POWER	16.815	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	72.75	FT
TUBE LENGTH	48.16	FT
NUMBER OF TUBES	110,588	
TUBE WALL THICKNESS	0.034	IN.
CONDENSER DIAMETER	72.75	FT
TUBE LENGTH	52.46	FT
NUMBER OF TUBES	110,588	
TUBE WALL THICKNESS	0.029	IN.

2000 FT CWP, TUBE SPACING 1.25

STATION	⑧/⑨	①	②	③	④	⑤	⑥	⑦	⑧	A	B	C	D	I	II/III
PRESSURE (PSIA)	129.49	128.99	29.22	91.94	91.97	90.77	92.38	198.36	130.53						
TEMPERATURE (°F)	70.31	70.09	70.19	51.59	51.61	50.92	50.92	50.97	50.97	79.75	76.44	41.76	45.16		
ENTHALPY (BTU/LB)	603.73	624.03	624.03	608.06	608.06	98.93	98.93	99.49	99.49						
DENSITY (LB/FT ³)	0.458	0.438	0.439	0.324	0.324	38.96	38.96	38.95	38.95						
QUALITY (%)	95.00	99.00	99.00	96.67	96.67										
FLOW (CFS)										14,532	14,532	13,643	13,643		
FLOW (LB/SEC)	5677	5448	5448	5448	5448	5443	5448	5448	7039					229	1591
ΔQ (BTU/SEC)										2,857,665		2,773,753			
ΔP (PSF)										285.0		267.6			
DEMISTER LIQ FLOW (GPM)														2705	

CYCLE DIAGRAM - JUNE 1976 BASELINE

The cycle diagram provides a concise one-page statement of state points and pertinent configuration parameters. The alpha/numeric notations encircled in the schematic are the same as the tabular column headings. The encircled arabic numerals pertain to the working fluid main flow conditions, the four encircled alphas apply to sea water conditions, and the encircled roman numerals apply to working fluid secondary flow conditions in the demister and reflux pumps. In these diagrams it is assumed that the liquid extracted from the vapor coming out of the evaporator by the demister (I) was fed to the reflux pump.

Pertinent configuration parameters are tabulated at the right side of the diagram as well as below the schematic diagram. The "net power output" is the power available for use, that is the generator gross minus the sum of the cycle plant auxiliaries, the module and core support power requirements, and the internal losses.

The June 1976 Baseline diagram incorporates the 2000 ft cold water pipe, the evaporator and condenser tube spacing to diameter ratio of 1.25 with titanium tubes whose wall thicknesses reflect the operating conditions, and the evaporator and condenser heat fluxes reflecting the approximate 38°F water temperature differential.

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CYCLE PLANT BACKUP CHARTS

THE NEXT SEVEN CHARTS ARE THOSE WHICH COMPLETE THE EIGHT COMPUTER RUNS FOR DEFINITION OF 1, 3, 6 AND 9 MW CAPACITY AND TITANIUM AND ALUMINUM TUBES. THE EIGHTH CHART IS PRESENTED IN THE CYCLE PLANT DISCUSSION SECTION.

1 MW_e CYCLE PLANT - TITANIUM TUBES

NET POWER OUTPUT	1.00	MW
GENERATOR GROSS	1.257	MW
TURBINE GROSS	1.276	MW
RANKINE CYCLE EFF	2.925	%
SEA WATER PUMP POWER	0.207	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	8.88	FT
TUBE LENGTH	50.0	FT
NUMBER OF TUBES	1,647	
TUBE WALL THICKNESS	0.058	IN.
CONDENSER DIAMETER	9.30	FT
TUBE LENGTH	50.0	FT
NUMBER OF TUBES	1,805	
TUBE WALL THICKNESS	0.058	IN.

CONDITIONS BASED ON POWER MODULE CONFIGURATION

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	A	B	C	D	I	II/III
PRESSURE (PSIA)	129.35	128.85	128.97	91.86	91.38	90.68	92.08	199.71	130.39						
TEMPERATURE (°F)	70.25	70.03	70.08	51.54	51.56	50.87	50.87	50.42	50.92	79.75	76.36	41.76	45.01		
ENTHALPY (BTU/LB)	603.72	624.01	624.01	608.10	608.10	98.87	98.87	99.44	99.44						
DENSITY (LB/FT ³)	0.458	0.438	0.438	0.323	0.323	38.96	38.96	38.96	38.96						
QUALITY (%)	95.00	99.00	99.00	95.68	96.68										
FLOW (CFS)										197.3	197.3	198.7	198.7		
FLOW (LB/SEC)	79.2	76.0	76.0	76.0	76.0	76.0	76.0	76.0	98.2					3.2	22.2
ΔQ (BTU/SEC)										39,869		38,703			
ΔP (PSF)										317.5		278.2			
DEMISTER LIQ FLOW (GPM)														37.7	

CONDITIONS BASED ON POWER MODULE CONFIGURATION

NET POWER OUTPUT	6.00	MW
GENERATOR GROSS	7.532	MW
TURBINE GROSS	7.647	MW
RANKINE CYCLE EFF	2.929	%
SEA WATER PUMP POWER	1.233	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	22.4	FT
TUBE LENGTH	46.3	FT
NUMBER OF TUBES	10,480	
TUBE WALL THICKNESS	0.058	IN.
CONDENSER DIAMETER	22.4	FT
TUBE LENGTH	50.7	FT
NUMBER OF TUBES	10,480	
TUBE WALL THICKNESS	0.058	IN.

[illegible]

9MW_e CYCLE PLANT - TITANIUM TUBE

NET POWER OUTPUT	9.00	MW
GENERATOR GROSS	11.299	MW
TURBINE GROSS	11.471	MW
RANKINE CYCLE EFF	2.932	%
SEA WATER PUMP POWER	1.850	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	27.0	FT
TUBE LENGTH	48.0	FT
NUMBER OF TUBES	15,248	
TUBE WALL THICKNESS	0.058	IN.
CONDENSER DIAMETER	27.0	FT
TUBE LENGTH	52.7	FT
NUMBER OF TUBES	15,248	
TUBE WALL THICKNESS	0.058	IN.

CONDITIONS BASED ON PCWER MODULE CONFIGURATION

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	①	②	③	④	①	②/③
PRESSURE (PSIA)	129.49	128.99	129.20	91.97	92.00	90.83	92.37	198.67	130.53						
TEMPERATURE (°F)	70.31	70.09	70.18	51.61	51.63	50.94	50.94	50.99	50.99	79.75	75.60	41.76	45.04		
ENTHALPY (BTU/LB)	603.73	624.03	624.03	608.09	603.09	98.95	98.95	99.51	99.51						
DENSITY (LB/FT ³)	0.453	0.438	0.438	0.524	0.324	38.95	38.96	38.95	38.95						
QUALITY (%)	95.00	99.00	99.00	96.67	96.67										
FLOW (CFS)										1,905	1,905	1,771	1,771		
FLOW (LB/SEC)	710.5	682.0	682.0	682.0	682.0	632.0	682.0	682.0	881.2					28.6	199.2
ΔQ (BTU/SEC)										357,717		347,229			
ΔP (PSF)										295.1		278.0			
DEMISTER LIQ FLOW (GPM)														338.4	

1 MW_e CYCLE PLANT - ALUMINUM TUBE

NET POWER OUTPUT	1.00	MW
GENERATOR GROSS	1.223	MW
TURBINE GROSS	1.242	MW
RANKINE CYCLE EFF	2.920	%
SEA WATER PUMP POWER	0.173	MW
HEAT EXCHANGER TUBE MATERIAL	Al	
EVAPORATOR DIAMETER	8.33	FT
TUBE LENGTH	47.30	FT
NUMBER OF TUBES	1,449	
TUBE WALL THICKNESS	0.065	IN.
CONDENSER DIAMETER	8.33	FT
TUBE LENGTH	52.71	FT
NUMBER OF TUBES	1,449	
TUBE WALL THICKNESS	0.065	IN.

CONDITIONS BASED ON POWER MODULE CONFIGURATION

STATION	0/9	1	2	3	4	5	6	7	8	A	B	C	D	I	II/III
PRESSURE (PSIA)	129.58	129.08	129.20	92.09	92.11	90.41	92.31	199.95	130.62						
TEMPERATURE (°F)	70.35	70.13	70.18	51.67	51.69	51.00	51.00	51.05	51.05	79.75	76.09	41.76	45.56		
ENTHALPY (BTU/LB)	603.74	624.03	624.03	603.15	608.15	99.02	99.02	99.59	99.59						
DENSITY (LB/FT ³)	0.458	0.438	0.438	0.324	0.324	38.95	38.95	38.95	38.95						
QUALITY (%)	95.00	99.00	99.00	96.68	96.68										
FLOW (CFS)										178.4	178.4	165.8	165.8		
FLOW (LB/SEC)	77.2	74.1	74.1	74.1	74.1	74.1	74.1	74.1	95.7					3.1	21.6
ΔQ (BTU/SEC)										38,848		37,713			
ΔP (PSF)										295.1		280.1			
DEMISTER LIQ FLOW (GPM)														36.7	

3 MW_e CYCLE PLANT - ALUMINUM TUBE

NET POWER OUTPUT	3.00	MW
GENERATOR GROSS	3.669	MW
TURBINE GROSS	3.725	MW
RANKINE CYCLE EFF	2.935	%
SEA WATER PUMP POWER	0.523	MW
HEAT EXCHANGER TUBE MATERIAL	Al	
EVAPORATOR DIAMETER	14.1	FT
TUBE LENGTH	50.9	FT
NUMBER OF TUBES	4,122	
TUBE WALL THICKNESS	0.065	N.
CONDENSER DIAMETER	14.1	FT
TUBE LENGTH	56.2	FT
NUMBER OF TUBES	4,122	
TUBE WALL THICKNESS	0.065	N.

CONDITIONS BASED ON POWER MODULE CONFIGURATION

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	A	B	C	D	I	II/III
PRESSURE (PSIA)	129.72	129.22	129.40	90.08	92.11	90.91	92.41	199.30	130.76						
TEMPERATURE (°F)	70.41	70.19	70.27	51.67	51.59	51.00	51.00	51.05	51.05	79.75	75.91	41.76	45.75		
ENTHALPY (BTU/LB)	603.76	624.04	624.04	608.09	608.09	99.02	99.02	99.58	99.58						
DENSITY (LB/FT ³)	0.459	0.439	0.439	0.324	0.324	38.95	38.95	38.95	38.95						
QUALITY (%)	95.00	99.00	99.00	95.67	96.67										
FLOW (CFS)										554.8	654.8	608.4	608.4		
FLOW (LB/SEC)	230.5	221.2	221.2	221.2	221.2	221.2	221.2	221.2	285.8					9.3	64.6
ΔQ (BTU/SEC)										116,025		112,620			
ΔP (PSF)										313.4		295.6			
DEMISTER LIQ FLOW (GPM)														109.7	

6 MW_e CYCLE PLANT - ALUMINUM TUBE

NET POWER OUTPUT	6.00	MW
GENERATOR GROSS	7.324	MW
TURBINE GROSS	7.435	MW
RANKINE CYCLE EFF	2.932	%
SEA WATER PUMP POWER	1.032	MW
HEAT EXCHANGER TUBE MATERIAL	Al	
EVAPORATOR DIAMETER	22.4	FT
TUBE LENGTH	37.3	FT
NUMBER OF TUBES	10,480	
TUBE WALL THICKNESS	0.065	IN.
CONDENSER DIAMETER	22.4	FT
TUBE LENGTH	41.3	FT
NUMBER OF TUBES	10,480	
TUBE WALL THICKNESS	0.065	IN.

CONDITIONS BASED ON POWER MODULE CONFIGURATION

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	①	②	③	④	⑤	⑥/⑦
PRESSURE (PSIA)	129.70	129.20	129.40	92.11	92.14	90.94	92.49	198.99	130.74						
TEMPERATURE (°F)	70.40	70.18	70.27	51.69	51.70	51.02	51.02	51.07	51.07	79.75	76.73	41.76	44.39		
ENTHALPY (BTU/LB)	603.75	624.04	624.04	608.10	608.10	99.04	99.04	99.60	99.60						
DENSITY (LB/FT ³)	0.459	0.439	0.439	0.324	0.324	38.95	38.95	38.95	38.95						
QUALITY (%)	95.00	99.00	99.00	96.67	96.27										
FLOW (CFS)										1,290	1,290	1,199	1,199		
FLOW (LB/SEC)	460.6	442.0	442.0	442.0	442.0	442.0	442.0	442.0	571.1					18.6	129.1
ΔQ (BTU/SEC)										231,809		225,012			
ΔP (PSF)										243.7		229.2			
DEMISTER LIQ FLOW (GPM)														219.3	

9 MW_e CYCLE PLANT-ALUMINUM TUBE

NET POWER OUTPUT	9.00	MW
GENERATOR GROSS	10.995	MW
TURBINE GROSS	11.162	MW
RANKINE CYCLE EFF	2.917	%
SEA WATER PUMP POWER	1.556	MW
HEAT EXCHANGER TUBE MATERIAL	Al	
EVAPORATOR DIAMETER	24.99	FT
TUBE LENGTH	47.04	FT
NUMBER OF TUBES	13,046	
TUBE WALL THICKNESS	0.065	IN.
CONDENSER DIAMETER	24.99	FT
TUBE LENGTH	52.48	FT
NUMBER OF TUBES	13,046	
TUBE WALL THICKNESS	0.065	IN.

CONDITIONS BASED ON POWER MODULE CONFIGURATION

STATION	0/9	1	2	3	4	5	6	7	8	A	B	C	D	I	II/III
PRESSURE (PSIA)	129.49	128.59	129.20	92.14	92.17	90.97	92.53	198.66	130.53						
TEMPERATURE (°F)	70.31	70.05	70.18	51.70	51.72	51.03	51.03	51.08	51.08	79.75	76.09	41.76	45.56		
ENTHALPY (BTU/LB)	603.73	624.03	624.03	608.17	608.17	95.06	99.06	99.62	99.62						
DENSITY (LB/FT ³)	0.458	0.438	0.439	0.324	0.324	38.95	38.95	38.95	38.95						
QUALITY (%)	95.00	99.00	97.00	96.58	96.68										
FLOW (CFS)										1,606	1,606	1,492	1,492		
FLOW (LB/SEC)	695.1	667.1	657.1	667.1	667.1	667.1	667.1	667.1	861.9					28.0	194.8
ΔQ (BTU/SEC)										349,326		339,623			
ΔP (PSF)										293.7		279.1			
DEMISTER LIQ FLOW (GPM)														331.1	

CYCLE DIAGRAM - MARCH 1976 PROPOSED BASELINE

THE CYCLE DIAGRAM PROVIDES A CONCISE ONE PAGE STATEMENT OF STATE POINTS AND PERTINENT CONFIGURATION PARAMETERS. SEE THE CYCLE DIAGRAM, JULY 1976 BASELINE FOR FORMAT DETAILS.

THIS DIAGRAM AND THE THREE CYCLE PLANT DIAGRAMS WHICH FOLLOW ARE BASED ON A MARCH 1976 CONFIGURATION WHICH WAS TO BE THE BASELINE CONFIGURATION FOR THE TEST FACILITY STUDY PROGRAM. IT WAS FOUND TO HAVE A FEW ERRORS AND THEREFORE WAS DISCARDED. IT IS INCLUDED HERE AS THE THREE CYCLE PLANTS (THE FOLLOWING THREE CHARTS) WHEN PLOTTED ON THE THERMAL RESOURCE CAPABILITY PLOT ARE WITHIN A FEW PERCENT OF THE "NET" POWER OUTPUT VERSUS THE THERMAL RESOURCE EVEN THOUGH THE TUBE WALL THICKNESSES ARE NOT CORRECT AND THE SIMULATED COLD WATER PIPE PRESSURE LOSS IS ERRONEOUS. THESE CONDITIONS POINT UP THE DOMINANCE OF THE WATER TEMPERATURE DIFFERENTIAL ON POWER CYCLE PERFORMANCE.

PROPOSED BASELINE POWER CYCLE MARCH 1976

NET POWER OUTPUT	54.15	MW
GENERATOR GROSS	84.501	MW
TURBINE GROSS	85.788	MW
RANKINE CYCLE EFF	2.593	%
SEA WATER PUMP POWER	16.818	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	72.75	FT
TUBE LENGTH	49.3	FT
NUMBER OF TUBES	120,000	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	72.75	FT
TUBE LENGTH	54.0	FT
NUMBER OF TUBES	120,000	
TUBE WALL THICKNESS	0.028	IN.

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	①	②	③	④	①	②/③
PRESSURE (PSIA)	132.22	131.72	131.95	97.92	97.96	96.75	98.35	200.79	133.25						
TEMPERATURE (°F)	71.50	71.28	71.38	54.93	54.94	54.28	54.28	54.33	54.33	79.75	76.50	46.22	49.59		
ENTHALPY (BTU/LB)	603.99	624.24	624.24	610.18	102.71	102.71	103.25	103.25							
DENSITY (LB/FT ³)	0.467	0.447	0.448	0.343	0.343	38.79	38.79	38.79	38.79						
QUALITY (%)	95.00	99.00	99.00	96.92	96.92										
FLOW (CFS)										15,606	15,606	14,558	14,558		
FLOW (LB/SEC)	6,028	5,785	5,785	5,785	5,785	5,785	5,785	5,785	7,474					243.1	168.9
ΔQ (BTU/SEC)										3,014,019		2,935,864			
ΔP (PSF)										278.4		262.9			
DEMISTER LIQ FLOW (GPM)														286.9	

3 MW_e CYCLE PLANT

MARCH 1976 PROPOSED BASELINE

NET POWER OUTPUT	3.00	MW
GENERATOR GROSS	3.908	MW
TURBINE GROSS	3.967	MW
RANKINE CYCLE EFF	2.601	%
SEA WATER PUMP POWER	0.836	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	15.75	FT
TUBE LENGTH	48.46	FT
NUMBER OF TUBES	5,625	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	15.75	FT
TUBE LENGTH	52.32	FT
NUMBER OF TUBES	5,625	
TUBE WALL THICKNESS	0.028	IN.

SIMULATED COLD WATER PIPE PRESSURE LOSS = 97.5 PSF

STATION	①/⑨	①	②	③	④	⑤	⑥	⑦	⑧	⑩	⑪	⑫	⑬	⑭	⑮/⑯
PRESSURE (PSIA)	131.94	131.44	131.24	97.98	98.00	96.79	98.38	140.26	132.98						
TEMPERATURE (°F)	71.38	71.16	71.07	54.96	54.97	54.30	54.30	54.36	54.32	79.75	76.51	46.22	49.45		
ENTHALPY (BTU/LB)	603.96	624.22	624.22	610.44	610.44	102.74	102.74	102.96	102.96						
DENSITY (LB/FT ³)	0.466	0.446	0.445	0.343	0.343	38.79	38.79	38.79	38.79						
QUALITY (%)	95.00	99.00	99.00	96.97	96.97										
FLOW (CFS)										738.4	738.4	715.1	715.1		
FLOW (LB/SEC)	284.3	272.8	272.8	272.8	272.8	272.8	272.8	272.8	352.5					11.5	79.7
ΔQ (BTU/SEC)										142,204		138,505			
ΔP (PSF)										280.0		281.1			
DEMISTER LIQ FLOW (GPM)														135.3	

7-22

NET POWER OUTPUT	6.00	MW
GENERATOR GROSS	7.812	MW
TURBINE GROSS	7.931	MW
RANKINE CYCLE EFF	2.613	%
SEA WATER PUMP POWER	1.670	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	22.3	FT
TUBE LENGTH	48.6	FT
NUMBER OF TUBES	11,250	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	22.3	FT
TUBE LENGTH	52.1	FT
NUMBER OF TUBES	11,250	
TUBE WALL THICKNESS	0.028	IN.

SIMULATED COLD WATER PIPE PRESSURE LOSS = 97.5 PSF

[illegible]

9 MW_e CYCLE PLANT - MARCH 1976 PROPOSED BASELINE

NET POWER OUTPUT	9.00	MW
GENERATOR GROSS	11.721	MW
TURBINE GROSS	11.899	MW
RANKINE CYCLE EFF	2.603	%
SEA WATER PUMP POWER	2.508	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	27.3	FT
TUBE LENGTH	52.4	FT
NUMBER OF TUBES	16.875	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	27.3	FT
TUBE LENGTH	48.5	FT
NUMBER OF TUBES	16,845	
TUBE WALL THICKNESS	0.028	IN.

SIMULATED COLD WATER PIPE PRESSURE LOSS = 97.5 PSF

STATION	0/9	1	2	3	4	5	6	7	8	A	B	C	D	I	II/III
PRESSURE (PSIA)	131.93	131.43	131.24	97.95	97.98	96.76	98.37	140.17	132.97						
TEMPERATURE (°F)	71.37	71.16	71.37	54.94	54.95	54.29	54.29	54.31	54.31	79.75	76.51	46.22	49.45		
ENTHALPY (BTU/LB)	603.96	624.21	624.21	610.42	610.42	102.72	102.72	102.94	102.94						
DENSITY (LB/FT ³)	0.466	0.446	0.445	0.343	0.343	38.79	38.79	38.79	38.79						
QUALITY (%)	95.00	99.00	99.30	96.97	96.97										
FLOW (CFS)										2,211	2,211	2,145	2,145		
FLOW (LB/SEC)	852.0	817.6	817.6	817.6	817.6	817.6	817.6	817.6	1,056.4					34.4	238.8
ΔQ (BTU/SEC)										426,201		415,106			
ΔP (PSF)										280.0		281.3			
DEMISTER LIQ FLOW (GPM)														405.7	

APRIL 1975

- 0000 FT CWP

NET POWER OUTPUT	40.00	MW
GENERATOR GROSS	62.420	MW
TURBINE GROSS	63.370	MW
RANKINE CYCLE EFF	3.26	%
SEA WATER PUMP POWER	18.80	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	72.75	FT
TUBE LENGTH	52.5	FT
NUMBER OF TUBES	120,000	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	72.75	FT
TUBE LENGTH	52.5	FT
NUMBER OF TUBES	120,000	
TUBE WALL THICKNESS	0.028	IN.

[illegible]

CYCLE DIAGRAM - APRIL 1975 BASELINE

THE CYCLE DIAGRAM PROVIDES A CONCISE ONE PAGE STATEMENT OF STATE POINTS AND PERTINENT CONFIGURATION PARAMETERS. SEE THE CYCLE DIAGRAM - JUNE 1976 BASELINE FOR FORMAT DETAILS.

THE APRIL 1975 BASELINE DIAGRAM CONTAINS THE DATA FOR THE CONFIGURATION PRESENTED IN THE APRIL 1975 REPORT (LMSC-DO56566). THIS CONFIGURATION INCORPORATES A 1000 FOOT COLD WATER PIPE, EVAPORATOR AND CONDENSER TUBE SPACING-TO-DIAMETER RATIO OF 1.20 WITH TITANIUM TUBES WHOSE WALL THICKNESS IS 0.028-INCH AND THE EVAPORATOR AND CONDENSER HEAT FLUXES REFLECTING A 32.25 WATER TEMPERATURE DIFFERENTIAL.

COMPARING THE JUNE 1976 DIAGRAM TO THE APRIL 1975 DIAGRAM SETS OUT THE STATE POINT CHANGES THAT OCCURRED WITH THE UPDATING OF THE BASELINE. THE MAJOR CHANGES ARE IN THE WATER FLOW RATES AND ASSOCIATED PRESSURE LOSSES IN THE EVAPORATOR AND CONDENSER, THE COLD WATER INLET TEMPERATURE, AND THE WORKING FLUID FLOW RATE. THE WORKING FLUID PRESSURE IN THE EVAPORATOR ALSO DECREASED.

NET POWER OUTPUT	2.00	MW
GENERATOR GROSS	2.623	MW
TURBINE GROSS	2.663	MW
RANKINE CYCLE EFF	2.609	%
SEA WATER PUMP POWER	0.576	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	12.86	FT
TUBE LENGTH	49.6	FT
NUMBER OF TUBES	3,758	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	12.86	FT
TUBE LENGTH	54.1	FT
NUMBER OF TUBES	3,750	
TUBE WALL THICKNESS	0.028	IN.

FEB 1975 COMPUTER RUN/1000 FT CWP 25 FT TOP ID/11 FT BOTTOM ID

[illegible]

CYCLE PLANT - 2 AND 4 MW

THE NEXT TWO CHARTS ARE BASED ON THE 1975 APRIL BASELINE. THESE
DIAGRAMS USE 0.028-INCH TITANIUM EVAPORATOR AND CONDENSER TUBES,
A 1000 COLD WATER PIPE WITH A 25 FOOT I.D. AT THE TOP AND A 11 FOOT
I.D. AT THE BOTTOM.

4 MW_e CYCLE PLANT - THIN WALL TITANIUM TUBE

NET POWER OUTPUT	4.00	MW
GENERATOR GROSS	5.255	MW
TURBINE GROSS	5.335	MW
RANKINE CYCLE EFF	2.600	%
SEA WATER PUMP POWER	1.160	MW
HEAT EXCHANGER TUBE MATERIAL	Ti	
EVAPORATOR DIAMETER	18.19	FT
TUBE LENGTH	50.0	FT
NUMBER OF TUBES	7,500	
TUBE WALL THICKNESS	0.028	IN.
CONDENSER DIAMETER	18.19	FT
TUBE LENGTH	55.9	FT
NUMBER OF TUBES	2,500	
TUBE WALL THICKNESS	0.028	IN.

FEB 1976 COMPUTER RUN/1000 FT CWP 25 FT TOP ID/11 FT BOTTOM ID

STATION	⑧/⑨	①	②	③	④	⑤	⑥	⑦	⑧	①	②	③	④	⑤	⑥/⑦
PRESSURE (PSIA)	132.28	131.78	131.58	98.52	98.54	97.33	98.93	140.57	133.32						
TEMPERATURE (°F)	71.52	71.31	71.22	55.25	55.26	54.60	54.60	54.62	54.62	79.75	76.50	46.52	50.02		
ENTHALPY (BTU/LB)	603.99	624.24	624.24	610.47	610.47	103.07	103.07	103.29	103.29						
DENSITY (LB/FT ³)	0.467	0.447	0.446	0.345	0.345	38.77	38.77	38.77	38.77						
QUALITY (%)	95.00	99.00	99.00	96.96	96.96										
FLOW (CFS)										989.4	989.4	888.5	888.5		
FLOW (LB/SEC)	382.7	367.2	367.2	367.2	367.2	367.2	367.2	367.2	474.4					15.4	107.2
ΔQ (BTU/SEC)										191,311		186,337			
ΔP (PSF)										290.1		258.0			
DEMISTER LIQ FLOW (GPM)														182.2	

7.4 POWER PLANT THERMAL RESOURCE

The "Once Through Cooled Power Plants" figure in Section H presented cooling water rated conditions for 15 plants. This section contains the remaining two figures, not presented because of time, which contains the survey results for "Once Through Cooled Typical Nuclear Power Plants" and "Selected Cooling Tower Plants".

ONCE-THROUGH COOLED TYPICAL NUCLEAR POWER PLANTS

PLANT	COOLING WATER RATED CONDITIONS	
	ΔT (°F)	FLOWS INTO
ARKANSAS NUCLEAR 1	15°	RESERVOIR
PILGRIM 1	29° (FLOW LESS THAN REQUIRED)	CAPE COD BAY
BRUNSWICK 1 & 2	12-18°	ATLANTIC OCEAN
DRESDEN 1, 2, 3	9-10°	RESERVOIR
ZION 1, 2	10°	LAKE MICHIGAN
INDIAN POINT 2	16°	HUDSON RIVER
OCONEE 1, 2, 3	12-21°	RESERVOIR
MAINE YANKEE	26°	9-10 TIDAL FLOW, BACK RIVER
9 MILE POINT	29-30° (FLOW LESS THAN REQUIRED)	LAKE ONTARIO
MILLSTONE 1, 2	22°	LONG ISLAND SOUND
SALEM 1, 2	22°	DELAWARE RIVER
POINT BEACH 1, 2	18-33°	LAKE MICHIGAN
TURKEY POINT	9-15°	COOLING CANAL

ONCE-THROUGH COOLED TYPICAL NUCLEAR PLANTS

Once-through cooled nuclear power plants, which generally have large cooling water flows (sufficient for an OTEC cycle plant application), usually have low temperature rises in their condensers. This is caused by recent thermal pollution restrictions limiting the rises to less than was allowed in older fossil-fired units. The only two cases in the slide where the temperature difference approaches the 30°F required by an OTEC cycle plant have flows less than the 300,000 gpm minimum (for a 3 MWe net cycle plant). Once-through cooled nuclear plants, therefore, do not appear to be a promising source of thermal resource for an OTEC cycle plant.

SELECTED COOLING TOWER PLANTS

PLANT	COOLING WATER RATED CONDITIONS	
	ΔT (°F)	FLOW (GPM)
MECHANICAL DRAFT		
CENTRALIA 1 & 2	30.5	448,000
COLSTRIP 1 & 2	31.7	452,000
JIM BRIDGER 1, 2 & 3	34	480,000
BROWNS FERRY 1, 2 & 3	31.7	1,650,000
NATURAL DRAFT		
HARRISON 1, 2, & 3	30.9	1,755,000
3-MILE ISLAND 2	31.4	416,000
TROJAN 1	37.3	425,000
LIMERICK 1 & 2	33.5	952,000
ARKANSAS 1	30.7	404,000
WATTS BAR 1 & 2	38	820,000
VOGTLE	34.5	475,000
SUSQUEHANNA	33.8	478,000

SELECTED COOLING TOWER PLANTS

As illustrated by the preceding slides, the number of once-through cooled power plants providing a suitable flow and temperature difference for an OTEC cycle plant is limited. Consequently, a brief survey of cooling tower plants was made. A recent American Power Conference paper entitled "Application Considerations for Central Station Cooling Tower Installations" lists almost all of the natural and forced draft power plant cooling systems installed up to 1975. The slide lists those selected plants which meet the requirements of a OTEC cycle plant. It should be noted that considerably more of this type of plant meet the requirements as once-through cooled plants. The cooling tower plants are less inhibited by environmental restrictions since the cooling water is contained in a closed loop. The higher temperature rises result primarily from economic considerations. However, the normal seasonal variations in cooling water conditions and part load operation of a power plant will impose constraints on operating a combined conventional/OTEC power plant that may not be acceptable to utility companies.

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7.5 LAND-BASED CYCLE PLANT

The development of and the characteristics of a selected cycle plant for a land test facility are presented in this section. This section is the basis for the last part of Section I. Included at the end is a brief statement of constraints/risks for the land based cycle plant.

SELECTED CYCLE PLANT

1. Introduction

The information given below summarizes the development of and the characteristics of a selected cycle plant for a land based OTEC test facility. This selected cycle plant is not necessarily an "optimum" selection. As discussed later, the unknowns associated with a non-specific site situation can only result in a design that identifies the important criteria rather than the optimum design.

The development of the selected cycle plant took place in two stages. The first stage (Tasks 1.2 and 1.4) was concerned with the consideration of two alternate cycle plant concepts and variations of those concepts. Technical and cost information for these tasks was developed and transmitted to LMSC in April, 1976. The second stage (Task 1.5) was concerned with the selection of one cycle plant concept based on the developed technical and cost information. The selected cycle plant was then delineated with its own layouts, P & ID, specifications, etc. This submittal briefly reviews the Cycle Plant Concepts and then presents the results of the Selected Cycle Plant task.

2. Cycle Plant Concepts

In February 1976, LMSC and Bechtel agreed on two cycle plant concepts for Bechtel to investigate further. These concepts were called the "bathtub" configuration and the "grade" configuration. System descriptions, specifications and drawings were produced for these configurations for 3 MWe, 6 MWe and 9 MWe "net" output cycle plants. Net output refers to the net output which the plant would

have if it were located in an OTEC plant in the ocean. For the land based cycle plant, additional pumping power requirements actually result in auxiliary power requirements greater than the turbine/generator gross output.

The two cycle plant configurations are summarized below:

2.1 Bathtub Configuration

The bathtub configuration, shown in Figure 1, originated from the desire to isolate the circulating water pumps from the intake structure pumps i.e. to have an OTEC cycle plant operating solely between upstream and downstream open pools, totally isolated from external hydraulic effects. The depth of the dry "bathtub" between the inlet and outlet pools was necessary to satisfy the anticipated high NPSH requirements of the circulating water pumps and to avoid vortex formation by these pumps. Also, the bathtub allowed the installation of optional vertical heat exchangers (shown in Figure 1) with little or no additional pumping power requirements by using the bathtub depth in combination with siphon recovery into the outlet pool. The disadvantage of this configuration is the high civil/structural costs associated with the deep bathtub, primarily in the concrete walls and floor necessary to withstand external earth and hydrostatic pressures.

2.2 Grade Configuration

The grade configuration, shown in Figure 2, was chosen as an alternate since it became apparent that the testing of large scale OTEC pumps in the circulating water loops may not be necessary, obviating the requirement for the dry bath-

tub for pump submergence reasons. Circulating water pumps are shown in Figure 2, however. They can be tested by increasing the intake structure pump head to satisfy the circulating water pumps NPSH requirement and then dissipating this additional head by using an orifice plate downstream of the heat exchangers. The pumps can be replaced by spool pieces or passive swirl-producing sections. The outlet pool is retained to achieve siphon recovery when the circulating water pumps are not installed. The installation of vertical heat exchangers requires additional intake structure pump head and support structure for the heat exchangers; these being the major economic disadvantages of this configuration.

2.3 Alternate Cantilever Wall Construction

The civil/structural designs for the cycle plants shown in Figures 1 and 2 were based on gravity dam concrete construction for the framework. When cycle plant cost estimates were prepared, it became apparent that the civil/structural costs utilizing this approach were out of line with regard to the overall plant estimate. The civil/structural design was therefore subjected to re-analysis and an alternate design, utilizing cantilever wall analysis, was developed. A sketch of the revised design for the 6 MWe net bathtub design is shown in Figure 3.

2.4 Cost and Schedule

Based on the information described above, (i.e. equipment specifications, system descriptions, layout drawings, revised civil/structural design, etc.) for 3 MWe, 6MWe and 9MWe bathtub and grade configuration cycle plants, cost and schedule information was developed and submitted to LMSC on April 23, 1976.

3. SELECTED CYCLE PLANT

3.1 General

Based on the information described in Section 2 and on other studies concerning component scaling, a 3 MWe "ducted open water loop" configuration was selected. This is similar to the grade configuration in that there is no upstream open pool. However, the floor elevation was not specified beyond it being "compatible with the heat exchanger envelopes minimizing pumping power in the cycle plant." Section 3 discusses the considerations in selecting this elevation.

Also specified was that the cycle plant should be capable of testing various kinds of vertical heat exchangers in addition to the LMSC baseline horizontal shell and tube design. This information was contained in an LMSC specification dated May 7, 1976. Subsequent to receipt of this specification, LMSC requested that the selected cycle plant equipment specifications be based on the 3 MWe net computer runs dated June 7, 1976 15:47:0.

3.2 Tradeoff Considerations

Tradeoff studies utilizing the data developed in Task 1.2 and 1.4 were not within the Bechtel scope of work. However, the parameters effecting the determination of the test bed floor elevation were reviewed. Major cost factors include:

- o civil/structural cost associated with the excavated and concrete lined test bed area
- o structural cost associated with supporting the heat exchangers on the test bed

- o initial cost of intake structure pumps and piping.
- o energy cost due to pumping head.
- o initial cost of piping due to transient vacuum conditions.

These cost factors in turn are affected by more basic parameters not known at this time. These include:

- o specific site soil conditions and groundwater situation
- o method of construction applicable
- o type and dimensions of all heat exchangers to be tested
- o utilization factor for each of these heat exchangers.
- o site dependent relative elevations (water sources, grade, water sinks) including level variations
- o cost of energy
- o selected size of cycle plant
- o appropriate capitalization factor
- o availability of capital

For the LMSC baseline horizontal heat exchangers, there is little pumping power gain to offset the large civil/structural costs of a deep bathtub configuration since the discharge elevation is just below grade in either case. Vertical heat exchangers however, would require considerably more pumping power for a grade configuration. As the utilization factor of vertical heat exchangers increases, energy cost would tend to lower the test bed floor optimum elevation. As the utilization factor of vertical heat exchangers decreases, however, civil/structural costs tend to raise this elevation. For purposes of this study, an intermediate elevation was chosen, closer to the grade configuration, with a floor elevation of -30'-0.

3.3

Layout

The basic layout of the 3 MWe net cycle plant with LMSC baseline horizontal heat exchangers installed is shown in Bechtel Drawing 11726-P-1001. An intake structure with 3-4000 HP pumps is shown for the cold circulating water system. A similar intake structure exists for the warm circulating water system. A single line diagram is shown for the plant including the auxiliaries located in the Cycle Plant Auxiliary Equipment Building shown in Bechtel Drawing 11726-P-1002.

The plant utilizes the cantilever wall construction discussed previously. A major change from the earlier grade configuration design is the removal of the discharge pond. The relatively expensive civil/structural cost of this pond has been eliminated by the use of an orifice/standpipe/control valve arrangement. Diameter of the standpipe has been tentatively set at the 10 feet diameter of the circulating water system pipe. However, transient hydraulic analysis is necessary to finally select its size. Although less information is available regarding the various OTEC vertical heat exchangers that may be tested, sketches have been produced showing the 3 MWe net cycle plant test bed with typical vertical heat exchanger installations (dimensions from LMSC). These are shown in Bechtel Drawings 11726-P-1011, 1012, 1013 and 1014. The tallest arrangement is for the LMSC vertical panel heat exchanger, with a maximum elevation of 70 feet above grade.

3.4 Hydraulic Design Considerations

The cycle plant will include two parallel circulating water systems: one for the warm and the other for the cold water. It is assumed that both the warm and cold flows will be obtained from open-water sources and will be returned to same. Under this assumption, the warm (or cold) circulating water system may be sub-divided into five main hydraulic components, as follows:

- o intake structure and discharge manifold
- o supply conduit from the intake to the cycle plant
- o cycle plant and vicinity
- o discharge conduit from the cycle plant to the discharge structure
- o discharge structure

The hydraulic design considerations presented below are applicable to both the warm and cold circulating water systems.

3.4.1 Intake Structure and Discharge Manifold

The structure is similar to most pump intakes for power plant cooling water systems. It includes, in the downstream order, stoplog guides, trash racks, traveling screens, vertical pumps, pump discharge valves and pump manifold. Conceptual design of the structure is highly preliminary because of its dependence on site selection. However, the structure will be designed with the following hydraulic considerations.

- o The intake will be designed with an average intake velocity of about 0.5 fps and with fish escape openings to avoid or minimize fish-kill problems.
- o The intake will be designed with acceptable flow conditions approaching the pump to prevent any formation of surface vortices and the possibility of resulting pump vibrations. Depending upon site selection and structural layout, hydraulic model studies may be required.
- o If possible, the structure will be located and designed to prevent or minimize silting problem.
- o The pump discharge valves will have vertical shafts. This minimizes unbalance torque acting on the valve. Under certain conditions, throttling of the pump discharge valves may be required to maintain desirable hydraulic gradelines.
- o The pump discharge manifold will be designed to prevent unstable flow conditions immediately downstream from the manifold, especially during two-pump operations. An arrangement with two successive converging bifurcation is tentatively proposed (see Bechtel Drawing 11726-P-1001). Tests using small scale hydraulic models may be required to confirm or modify the designs.

3.4.2 Supply Conduit

The supply conduit will be designed to withstand any possible positive and negative pressures due to transients, such as during pump start-ups, pump power failures, pump shutdowns and other valve movements. The hydraulic gradeline of the system will be relatively low, therefore, it is quite likely that the conduit will be subjected to full vacuum pressure in the event of pump power failure. This negative pressure would be a governing factor in pipeline design and material selection. Steel and FRP pipes of 10 feet in diameter and of economical classes generally would not withstand negative pressures down to full vacuum.

A hydraulic transient study will be conducted in the preliminary design stage to form the bases for specifying the pipeline and other components in the

system (such as the heat exchangers) as well as to establish schemes for the pump discharge valve operation. A final transient study will also be made using the final pipe and pump data to confirm the validity of the preliminary design.

3.4.3 Cycle Plant and Vicinity

The cycle plant will be designed to accommodate testing of both horizontal and vertical heat exchangers. There will be no basins at the inlet and outlet sides of the heat exchangers. However, a standpipe of identical diameter to the outlet pipe has been provided at the downstream end. The standpipe will serve to minimize the propagation of any downstream disturbances. More importantly, the standpipe will effectively isolate the downstream piping from the upstream piping and reduce the severity of transient pressures in the entire system. The water surface elevation in the standpipe (and hence the pressure in the heat exchanger) and the system discharge will be regulated by a butterfly valve downstream. It will be necessary to ensure that the butterfly valve will throttle the required head without causing any cavitation problem.

One isolation valve each has been provided at the upstream and downstream end of the cycle plant, for maintenance purposes. The downstream valve will be interlocked with the pumps such that:

- o The valve will be at its open position immediately prior to pump start-ups
- o The valve will not close when the pumps are in operation

The interlock is designed to prevent the heat exchanger from being subjected to pressures equal to or higher than the pump shutoff head. Interlock of the upstream control valve with the pumps will also be necessary if protecting the upstream piping from experiencing the pump shutoff condition is also desirable.

Although testing of bulb type circulating water pumps in the cycle plant is probably not necessary a transition section has been provided for possible pump installation.

The cycle plant can be used to test vertical type heat exchangers. Bechtel drawing 11726-P-1014 shows an open type vertical heat exchanger and drawings 11726-P-1011, 1012 and 1013 show enclosed types. Because of the increase in the height of these arrangements, an orifice plate has been included to provide the necessary head in the heat exchangers. Headloss across the orifice plate will be manually adjustable, ranging from about zero for the horizontal heat exchanger arrangement to about 60 ft for the highest vertical heat exchanger. The orifice plate will be located at least 10 pipe diameters upstream from the standpipe to provide sufficient distance for flow expansion.

The closed-type vertical heat exchangers can be run under a vacuum pressure down to about -25 ft of water column to save pumping cost. Scavenging pump connections will be provided at the maximum elevation points to remove air released from the water and to maintain a desirable vacuum pressure. Due to tight transitions in all vertical heat exchanger layouts, guide vanes will be needed in the approach sections and, possibly, downstream from the heat exchangers. This is necessary since poor flow distribution would increase the system loss, and, more importantly, affect the heat exchanger performance.

Hydraulic model tests will be conducted to develop guide vane designs for the selected vertical heat exchanger configurations.

3.4.4 Discharge Conduit

The discharge conduit is the piping downstream from the control valve. Although its design is very site dependent, it is expected that the discharge pipe will be low pressure and/or open channel flow. Other than providing sufficient venting and ensuring no blowback phenomena, no unusual hydraulic design problems are anticipated.

3.4.5 Discharge Structure

No elaborate discharge structure other than a point discharge is anticipated. Care will be taken to prevent flow recirculation from the discharge to the intake structure.

3.5 OTEC Test Facility Operation and Control

3.5.1 General

The cycle plant will be provided with a centrally located control room from which essential plant equipment and systems will be controlled. The plant operator will be provided with adequate information and controls in the control room to start-up, run at steady state and shutdown the main power cycle and major equipment. The operator will be able to override selected automatic plant controls and operate the power cycle and selected system components at "off normal"

conditions. No operator action will be required during plant transients and no system or equipment protective function will be overridden.

The operating status of all machinery and indication of all system parameters essential for safe and continued plant operation will be displayed. The abnormal conditions and trips or failures of normally operating equipment will be annunciated in the control room. All control room mounted instrumentation will be electrical or electronic. All essential system variables will be displayed at the location of control devices (e.g. pump start hand switch - pump discharge header pressure indicator). The warm water and cold water circulating water flows will be measured using ultrasonic devices and smaller flows will be measured using the conventional head drop method across orifices or nozzles. The ultrasonic flow measurement technique measures the mean flow velocity in the pipe using ultrasonic pulses simultaneously transmitted at an angle into the flow by two facing transducers. The system determines the flow velocity along four chordal paths and integrates to give total discharge. Accuracies of better than 1% are attainable using ultrasonic flow measurement in large pipes.

The turbine/generator supplied controls as well as other packaged systems control and instrumentation (compressed air etc.) will be integrated into the overall plant design and control.

The cycle plant controls will be designed to minimize system transient loads, to protect the systems and associated equipment from water hammer, and to protect the major pumps from cavitation, prolonged operation at shut off heads and reverse rotation.

The cycle plant operation and control including plant start-up, normal and abnormal operation has been reviewed. It should be noted however, that the procedures developed are preliminary and may be changed as a result of future detailed plant design and transient analysis. The characteristics of available equipment could very well impose specific requirements leading to alterations in the operation and control description.

Bechtel drawing 11726-P-1003 gives the basic piping and instrumentation diagram of the plant and serves as a basis for the following discussion.

3.5.2 Pre Start-up

Prior to cycle plant start-up, the conditions listed below must be satisfied:

- o An outside power source (the utility grid) must be available to start all plant pumps and auxiliary systems
- o A pre-operational check must be performed on all pumps, equipment, valves and controls and instrumentation
- o All circulating water piping flooded with water and air expelled out of the system
- o All nitrogen purged out of the ammonia cycle piping, condenser and evaporator
- o Power cycle piping charged with ammonia to a pre-determined inventory level
- o All make up and reject valves for ammonia condensate fully closed
- o Condensate pump recirculation valve full open
- o All auxiliary systems operational
- o Main turbine bypass valves closed
- o Main turbine tripped

3.5.3 Start-up

The following plant start-up procedure is suggested after all pre start-up conditions have been met.

- a. With external power available, one, two or three warm water and cold water circulating water pump pairs are started (depending upon final system and pump design). Subsequently, the temperature of the evaporator will start rising and the liquid ammonia in the condenser will cool.
- b. When a reasonable temperature difference is reached, one condensate pump is started and is followed by start-up of one reflux pump.
- c. As ammonia evaporation continues in the evaporator, the pressure in the system will increase. At this point the turbine bypass valves can be opened and by gradual opening of the turbine control vanes the ammonia flow can be transferred from the bypass system to the main turbine. The turbine can be gradually brought up to its rated speed, the generator synchronized to the grid and appropriate load applied to the cycle plant. Station power can then be increased by starting remaining warm and cold water intake structure pump pairs (depending upon final system design), and by further opening the turbine governing vanes to its rated load capacity.

3.5.4 Normal Plant Operation

During normal plant operation, the turbine/generator power output will vary as the thermal gradient of the circulating water systems changes. The turbine control vanes should be adjusted to the optimum position for the operating flow rate.

The condenser and evaporator levels are automatically maintained by make-up and reject valves and by a control valve at the discharge of the condensate pumps.

A recirculation control valve is provided at the condensate pump discharge to assure a minimum flow when the evaporator level control valve is closed thereby avoiding pump overheating.

The standpipe level control valve and the intake structure pumps discharge valves operate to maintain the desired circulating water flow and heat exchanger pressure.

3.5.5 Abnormal Operation

a. Turbine Trip or Load Rejection

Following a turbine trip or load rejection the condensate pumps and reflux pumps are tripped. If the operator determines that the situation requires an extended shutdown, the circulating water pumps would be manually shutdown.

b. One Warm Intake Structure Pump Trip

The associated pump discharge valve will rapid close to approximately 15% valve position and then slowly to full close position (in accordance with the transient analysis results). The plant output will be reduced due to two pump operation.

c. Two Warm Intake Structure Pumps Trip

The associated discharge valves rapid close to 15% valve position (in accordance with the transient analysis results) and then fully close gradually. The main power cycle must be tripped and the remaining intake structure pump is manually tripped.

d. Three Warm Intake Structure Pumps Trip

All associated discharge valves must remain open and allow the system to come to an equilibrium state by itself.

e. One Condensate Pump Trip

Start second pump automatically if available or trip the ammonia cycle.

f. One Reflux Pump Trip

Start second pump automatically if available or trip the ammonia cycle.

3.6 Civil/Structural Considerations

As discussed in Section 2.3, the initial civil/structural assessment of the OTEC cycle plant resulted in a design utilizing mass concrete gravity retaining walls for the basic structure. The resulting volumes of concrete necessary to fulfill design requirements appeared excessive, and added considerably to the conceptual cost estimate.

Consequently, the civil/structural design was re-evaluated, with reinforced concrete retaining walls replacing the mass concrete gravity walls. The overall size of the walls and the volumes of concrete required were reduced considerably. Included in the retaining wall analysis were overturning moments caused by lateral soil pressure. Based on the maximum lateral pressure, a safety factor of 1.5 was retained. The factor of safety against sliding was low, but still above 1.0. However, sliding of an individual wall should not be a problem, as the walls face each other in a box configuration and sliding pressures will transfer as compressive forces in the slab connecting the individual retaining walls. Hydrostatic uplift pressures on the entire structure are a strong consideration. For the seventy foot deep bathtub configuration, bending pressures

from a maximum uplift of 2000 psf based on partially water filled soil resulted in a thickness of 7 feet in the slab. For the selected design, with the top of slab thirty feet below grade, a thickness of 5 feet will resist hydrostatic uplift across the test bed floor. This thickness is based on bending forced by the uplift pressure and assumes 5000 psi concrete with normal reinforcement.

Site considerations and soil parameters will play an important part in the final design of the cycle plant, as all civil structural design is highly dependent on the actual conditions encountered during construction. Marshy areas, for example, may require increasing base slab thickness to resist hydrostatic force, while relatively dry conditions will result in bending and shear forces from installed equipment dictating the design. The site conditions and soil parameters will also affect the retaining wall design, and may result in sizes different from those presented in this conceptual design, where one-half full hydrostatic pressure was assumed on the retaining walls.

The 30 foot depth of the cycle-plant test bed walls represents a cost compromise between excavation and sub-surface construction cost versus superstructure construction and pumping cost. Sub-surface construction in an area with a high water table may require extensive additional shoring, cofferdaming and dewatering practices to be employed. An extensive trade-off study, based on specific site conditions for a selected location, would be required to optimize costs.

Additional civil/structural elements which must be considered during future segments of the design effort include:

- o dynamic analysis of the supporting structure based on pump and machine vibration loads.
- o seismic analyses, based on operating conditions, for the site selected.

3.7 Cycle Plant Equipment Specifications and System Descriptions

Based on the June 7, 1976 LMSC cycle plant computer run, equipment specifications and system descriptions were developed as follows:

Major Equipment Specifications (Attached as Appendix A)

- o Baseline Evaporator
- o Baseline Condenser
- o Circulating Water Pumps (Pod Type)
- o Turbine/Generator
- o Liquid Ammonia Pumps
- o Ammonia Storage Tanks

Major Systems (Attached as Appendix B)

- o Ammonia Vapor System
- o Ammonia Condensate System
- o Warm Circulating Water System
- o Cold Circulating Water System

Auxiliary Systems (Attached as Appendix C)

- o Condenser Water Injection
- o Nitrogen Purge
- o Ammonia Cleanup

- o Lube Oil Purification
- o Compressed Air
- o Auxiliary Cooling Water
- o Cathodic Protection
- o Control Systems
- o Tube Cleaning
- o Area Drainage
- o Ammonia Transfer
- o Circulating Water Air Removal
- o Electrical
- o Fire Protection and Safety Equipment
- o Equipment Handling

The auxiliary systems descriptions were not developed in detail and are only intended to identify the required systems and provide estimates of their auxiliary power requirements. The auxiliary systems represent only a small fraction of the cycle plant costs.

Since the layout drawings were produced based on equipment specifications developed from prior LMSC computer runs, the layout dimensions do not always agree exactly with the latest equipment specification dimensions based on the June 7, 1976 computer run. However, the differences are not considered significant for layout purposes.

3.8 Equipment List

The cycle plant equipment together with pertinent size data and comments are listed in Table 3.8.

TABLE 3.8
OTEC 3 MWe NET CYCLE PLANT EQUIPMENT LIST

<u>ITEM</u>	<u>(PUMPS)</u>	<u>QUANTITY</u>	<u>SIZE (EACH)</u>	<u>COMMENTS</u>
Cold Circulating Water Pumps		1	400 HP	Pod type, 8.11 ft diameter impellor
Warm Circulating Water Pumps		1	500 HP	Pod type, 8.26 ft diameter impellor
Cold Intake Structure Pumps		3	4000 HP	Vertical wet pit type, replaceable bowl and impellor
Warm Intake Structure Pumps		3	4000 HP	Vertical wet pit type, replaceable bowl and impellor
Ammonia Condensate Pumps		2	125 HP	100% capacity each
Ammonia Reflux Pumps		2	25 HP	100% capacity each
Condenser Water Injection Metering Pump		1	1/2 HP	Positive displacement type
Ammonia Cleanup Pump		1	2 HP	
Auxiliary Cooling Water Pumps		2	5 HP	100% capacity each
Amertap Tube Cleaning Recirculating Pump		1	30 HP	
Sump Pumps		2	5 HP	
Ammonia Transfer Pump		1	5 HP	
Startup Mechanical Vacuum Pump		1	150 HP	25 scfm at 1" Hg. abs.
Circulating Water Air Removal Vacuum Pump		2	20 HP	600 cfm at 7" Hg. abs.
<u>ITEM</u>	<u>(TANKS)</u>			
Ammonia Storage Tank		1	8 ft diam x 44 ft L	Sufficient for 2 cycle plant inventories
Condenser Water Injection Storage Tank		1	500 gal.	
Nitrogen Storage Tanks				Racked nitrogen cylinders mounted in framework
Compressed Air Storage Tank		1	30 cu.ft.	
Auxiliary Cooling Water Surge Tank		1	500 gal.	
Circulating Water Air Removal Air Tanks		2	20 cu.ft.	

TABLE 3.8 (Con't)
OTEC 3 MWe NET CYCLE PLANT EQUIPMENT LIST

<u>ITEM</u>	<u>(HEAT EXCHANGERS)</u>	<u>QUANTITY</u>	<u>SIZE (EACH)</u>	<u>COMMENTS</u>
Ammonia Evaporator		1	47 ft L x 17 ft diam	titanium tubes, carbon steel shell
Ammonia Condenser		1	51 ft L x 17 ft diam	titanium tubes, carbon steel shell
Ammonia Cleanup Cooldown Heat Exchanger		1	7500 lbs/hr NH ₃	regenerative type
Auxiliary Cooling Water Heat Exchanger		1	0.5 x 10 ⁶ Btu/hr	
Ammonia Transfer Vaporizer		1	10,000 lbs/hr NH ₃	Uses 6000 lbs/hr 150 psig. steam
<u>ITEM</u>	<u>(MISCELLANEOUS)</u>			
Turbine/Generator		1	approx. 12 ft x 25 ft	3.71 MWe gross power output, 227.5
Turbine/Generator Auxiliary Skid		1	approx. 6 ft x 12 ft	lbs/sec NH ₃
Cold Intake Structure Traveling Screens		6	9 ft w x 48 ft high	
Cold Intake Structure Bar Screens and Rakes		6	9 ft w x 48 ft high	
Warm Intake Structure Traveling Screens		6	9 ft w x 48 ft high	
Warm Intake Structure Bar Screens and Rakes		6	9 ft w x 48 ft high	
Ammonia Demisters		2	34 sq ft	Mounted inside evaporator vapor domes
Refrigerated CO ₂ Unit		1	1/2 ton	
Portable CO ₂ Cylinders		10		
Breathings Air Packs		20		
Protective Personnel Suits		10		
Ammonia Cleanup Column		1	20 " diam x 20 ft high	
Air Compressors		2	20 HP	100 cfm/100 psig each
Compressed Air Dryer		1		100 cfm/100 psig each
Gantry Crane		1	250 ton	
Amertap Tube Cleaning Mechanisms		2		Located adjacent to condenser and evaporator tubesheets
Ammonia Transfer Compressor		1	25 hp	Includes aftercooler or liquifier
Ammonia Cold Trap		1		Vent gas compressor and water/dry ice/regrigeration system ammonia condensation unit.

The 3 MWe cycle plant requires that electrical power and 60 F cooling water be supplied to it during normal operation. Nitrogen and ammonia storage tanks are provided for system purge and fill. Depending on future trade off studies considering actual consumption of nitrogen and ammonia during startup, the size of the ammonia and nitrogen storage tanks may be either increased or supplemented by temporary rail-car tanks.

Water consumption (apart from the main circulating water requirements) is primarily for turbine/generator unit cooling. 200 GPM of 60 F cooling water is considered adequate. Electrical power requirements can vary depending upon the type of heat exchangers being tested. The intake structure pump heads may vary between 40 feet and 120 feet. An electrical system will be installed with sufficient capacity to handle the maximum requirement.

The following continuous requirements are anticipated:

Cold Intake Structure Pumps	2610 - 7830 kWe
Warm Intake Structure Pumps	2810 - 8430
Ammonia Condensate Pumps	93
Ammonia Reflux Pumps	18
Miscellaneous Auxiliaries	50
(e.g. air compressor, cooling water pumps, circulating water air removal vacuum pump, etc.)	
<hr/>	
Total	5581 - 16421 kWe

FIGURE 1

BATHTUB CONFIGURATION

GRADE CONFIGURATION

7-59

FIGURE 3

ALTERNATE CANTILEVER WALL CONSTRUCTION

7-60

OTEC CYCLE PLANT BASELINE EVAPORATOR SPECIFICATION

GENERAL SERVICE REQUIREMENT:

To evaporate liquid ammonia using warm water.

GENERAL CONFIGURATION:

Shell-and-tube design with single tubesheet. Ammonia evaporation occurs on shell side of tubes. Shell may be inclined up to 15 degrees from horizontal. Shell and tubesheet to withstand internal ammonia operating pressure with external water pressure at full vacuum. Evaporator sump capacity to provide storage for 30 seconds operating time. A reflux pump (see Table AP-1) will recirculate saturated liquid over the tubes. Boiling will be in the film regime rather than the pool regime.

The unit consists of shell, tubesheets, installed tubing, tube supports/baffles, sump, liquid ammonia spray dome and nozzles, vapor discharge domes (two) with demisters installed in each, and necessary vent, drain, blowdown, and instrument connections.

MATERIALS:

Tubes shall be welded titanium tubing. Tubesheets shall be low carbon steel (Option 1) for rolled tube to tubesheet joints or low carbon steel with titanium cladding on the water side (Option 2) for rolled and welded tube to tubesheet joints. Shell shall be low carbon steel. Tube supports/baffles shall be of material resistant to ammonia.

DESIGN SUMMARY:

Design conditions and evaporator dimensions are tabulated in Table E-1 for the 3 MWe net power output cycle plant. A sketch of the evaporator is given in Sketch HX-1.

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TABLE E-1

OTEC CYCLE PLANT BASELINE EVAPORATOR DESIGN SUMMARY

<u>Cycle Plant Net Power</u>	<u>3MWe</u>
Tube size	2" O.D. x 0.058" wall
Tube length (ft)	46.3
Number of tubes	5,240
Tube surface (sq. ft)	123,399
Tube bundle diameter (ft)	15.8
Shell diameter (ft)	17
Shell length (ft)	47
Sump volume (cu. ft)	175
Ammonia pressure (psia)	129.8
Ammonia temperature (°F)	70.4
Ammonia flow (lb/sec)	228
Water flow (cu. ft/sec)	655
Thermal load(Btu/sec)	119,347
Estimated tube weight(lbs)	170,000
Estimated total weight(lbs)	501,000

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OTEC CYCLE PLANT BASELINE CONDENSER SPECIFICATION

GENERAL SERVICE REQUIREMENT:

To condense ammonia vapor exhausting from an ammonia turbine by use of cold water.

GENERAL CONFIGURATION:

Shell-and-tube design, with single tubesheet. Ammonia condensation occurs on shell-side of tubes. Shell may be inclined up to 15 degrees from horizontal. Shell and tubesheet design to withstand internal ammonia operating pressure with external water pressure at full vacuum. Condenser sump capacity to provide storage for 60 seconds operating time.

The unit consists of shell, tubesheets, installed tubing, tube supports/baffles, sump, turbine exhaust dome, and necessary vent, drain and instrument connections.

MATERIALS:

Tubes shall be welded titanium tubing. Tubesheets shall be low carbon steel (Option 1) for rolled tube to tubesheet joints, or low carbon steel with titanium cladding on water side (Option 2) for rolled and welded tube to tubesheet joints. Shell shall be low carbon steel. Tube supports/baffles shall be of material resistant to ammonia.

DESIGN SUMMARY:

Design conditions and condenser dimensions are tabulated in Table C-1 for the 3 MWe net power output cycle plant. A sketch of the condenser is given in Sketch HX-1.

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TABLE C-1

OTEC CYCLE PLANT BASELINE CONDENSER DESIGN SUMMARY

<u>Cycle Plant Net Power</u>	<u>3MWe</u>
Tube size	2" O.D. x 0.058" wall
Tube length (ft)	50.8
Number of tubes	5,240
Tube surface (sq. ft)	135,277
Tube bundle diameter (ft)	15.8
Shell diameter (ft)	17
Shell length (ft)	51
Sump volume (cu. ft)	350
Ammonia pressure (psia)	91.4
Ammonia temperature (°F)	51.3
Ammonia flow (lb/sec)	228
Water flow (cu. ft /sec)	608
Thermal load (Btu/sec)	115,851
Estimated tube weight (lb)	186,000
Estimated total weight (lb)	419,000

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OTEC CYCLE PLANT CIRCULATING WATER PUMPS SPECIFICATION (OPTIONAL)

GENERAL:

The OTEC cycle plant can be used to test one warm circulating water pump and one cold circulating water pump installed at the inlet to the cycle plant evaporator and condenser respectively. Each pump could be installed in series with its respective intake structure pump and discharge through a transition piece to its associated heat-exchanger. The pumps shall be of a streamlined "pod" design as indicated in Sketch CWP-1. The pump pod shall include a drive motor, reduction gear, necessary accessories, and it shall carry the impellor. The pump pod will be horizontally mounted on struts in a pump shroud having an inlet bellmouth section and an outlet diffuser section. Inlet straightening vanes and outlet guide vanes shall be part of the pod/shroud assembly and may double as pod mounting struts. Exterior water pressure may vary from 60 feet to total vacuum (during transients). The interior of the pod will be at atmospheric pressure. Static seals at the propellor shaft and any other pod penetrations shall prevent in-leakage with up to a 60 foot head of water external to the pod.

PUMP DESIGN PARAMETERS:

Design parameters and approximate pump dimensions are tabulated in Table CWP-1 for the 3MWe net power output cycle plant.

SCOPE OF SUPPLY:

Each warm and cold circulating water pump package shall include: pump pod with impellor, drive motor, reduction gear, bearings, lubrication system, and seals; pump shroud with inlet bellmouth section, outlet diffuser section, inlet straightening vanes, outlet guide vanes, and pod mounting struts.

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OTEC CYCLE PLANT CIRCULATING WATER PUMPS SPECIFICATION (OPTIONAL) CONT'D.

MATERIALS:

The pump impellers shall be of cast steel with an epoxy or other synthetic coating for water service. Other pump materials shall be of a material suitable for water service.

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TABLE CWP-1

OTEC CYCLE PLANT CIRCULATING WATER PUMPS DESIGN PARAMETERS

3MWe Cycle Plant Net Output

<u>Pump Performance:</u>	<u>Warm CWP</u>	<u>Cold CWF</u>
Flow (cfs/gpm)	655/293,300	608/272,200
Head (psf/ft)	288/4.6	269/4.3
Motor Power (MWe/hp)	0.333/446	0.290/389
Power Supply	- 4160 volt, 60 Hz, 3 phase -	
Pump Speed (rpm)	123.4	121.3
Specific Speed	21,200	21,200
Pump Efficiency, %	85.5	85.1
Gear Efficiency, %	98.0	98.0
Motor Efficiency, %	94.0	94.0
Overall Efficiency, %	78.8	78.4
Pump Dimension and Weights:		
A - Impellor dia. (ft)	8.26	8.11
B - Pod dia. (ft)	6.56	6.56
C - Pod length (ft)	30	30
D - Diffuser outlet dia. (ft)	15	14.5
E - Bell Mouth inlet dia (ft)	8	7
F - Shroud length (ft)	32	32
Impellor Weight (lb)	9,870	9,380
Total Weight (lb)	212,000	205,000

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OTEC CYCLE PLANT AMMONIA TURBINE/GENERATOR SPECIFICATION

GENERAL:

The OTEC cycle plant will include one ammonia turbine and electrical generator. Ammonia vapor will be supplied to the turbine from the cycle plant evaporator via demisters. The turbine will exhaust via a diffuser to the cycle plant condenser. The turbine/generator may be of either vertical or horizontal shaft configuration, direct coupled.

TURBINE/GENERATOR DESIGN:

Turbine operating conditions and design parameters are tabulated in Table TG-1 for the 3MWe net power output plant cycle. Typical dimensions and weights are included for both vertical and horizontal turbine/generator units. A sketch of a vertical unit is given in Sketch TG-1.

SCOPE OF SUPPLY:

The turbine shall be complete with normal and emergency lube oil systems, seals and leak-off system, protective devices, adjustable inlet throttle/stop vanes, and exhaust diffuser section. The generator shall be complete with excitation system.

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TABLE TG-1

OTEC CYCLE PLANT TURBINE/GENERATOR DESIGN SUMMARY

<u>Cycle Plant Net Power</u>	<u>3MWe</u>
<u>Turbine</u>	
Working fluid	Ammonia
Ammonia inlet total press/temp. (psia/F)	129.2/70.2
Ammonia diffuser exhaust static press/temp. (psia/ F)	92.0/51.6
Ammonia flow (lb/sec)	227.5
Turbine gross power (MWe)	3.825
Turbine speed (RPM)	3600
Inlet flange diameter (inches)	42
Outlet flange diameter (inches)	48
Total to static wet turbine efficiency, %	88
<u>Generator</u>	
Generator gross power (MWe)	3.710
Power factor	.85
Voltage (KV)	13.8
Speed (RPM)	3600
<u>Turbine/Generator Skid</u>	
Dimensions (ft) (horizontal shaft)	12W x 25L
Dimensions (ft) (vertical shaft)	18H x 12D
Weight (lb)	50,000
<u>Auxiliary Skid (oil-controls)</u>	
Dimensions (ft)	6 x 12
Weight (lb)	3,000

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OTEC CYCLE PLANT LIQUID AMMONIA PUMPS SPECIFICATION

GENERAL:

The OTEC cycle plant will require two each of the following 100% capacity pumps to circulate liquid (anhydrous) ammonia:

Condensate Pump - takes suction from the cycle plant condenser sump and discharges through piping to the evaporator.

Reflux Pump - takes suction from the evaporator sump and pumps back to the evaporator liquid inlet to provide up to 50% recirculation of liquid in the evaporator.

PUMP DESIGN:

Operating conditions for the condensate and reflux pumps are tabulated in Table AP-1 for the 3 MWe net cycle plant with the baseline horizontal heat exchangers. Depending upon the actual requirements of vertical heat exchanger configurations, the pumps could be made larger head and throttled for horizontal heat exchanger operation, operated via variable speed drive or replaced as each heat exchanger type is tested.

Each pump shall be of steel construction with stainless steel trim, complete with mechanical shaft seals, electric motor drive, and pump/motor coupling. Pump and motor shall be mounted on a common rigid steel base. Power supply to all motors is 460 volt, 60 Hz, 3-phase.

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TABLE AP-1

OTEC CYCLE PLANT LIQUID AMMONIA PUMPS

<u>Cycle Plant Net Output</u>	<u>3MWe</u>
<u>Condensate Pump:</u>	2 Pumps - each 100% capacity
Flow per pump (lb/sec)	228
(cfs)	5.85
(gpm)	2,623
Temperature ($^{\circ}\text{F}$)	51
Suction pressure (psia)	96.2
Discharge pressure (psia)	156
Suction static head (ft)	20
Estimated motor size (HP)	125
<u>Reflux Pump:</u>	2 Pumps - each 100% capacity
Flow per pump (lb/sec)	114
(cfs)	3.06
(gpm)	1,376
Temperature ($^{\circ}\text{F}$)	70.4
Suction pressure (psia)	130
Discharge pressure (psia)	150
Suction static head (ft)	20
Estimated motor size (HP)	25

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OTEC CYCLE PLANT

AMMONIA STORAGE TANKS SPECIFICATION

PURPOSE: To store liquid anhydrous ammonia under pressure for use in the power cycle of the OTEC cycle plant.

GENERAL: The OTEC cycle plant will require a liquid ammonia storage tank sized to store sufficient ammonia for two complete charges (fills) of the cycle plant condenser and evaporator sumps and condensate piping. The ammonia will be stored under pressure (not refrigerated) in the tank located outdoors near the cycle plant.

DESIGN: The storage tank shall be of horizontal cylindrical configuration with bumped heads, and shall be constructed of steel. The tank shall be complete with the necessary liquid and vapor valves, safety pressure relief valves and appropriate pressure, level, and temperature indicating devices. Storage quantity, volume, and tank design parameters are tabulated in Table AST-1 for the 3MWe net cycle plant. The safety pressure relief valves shall discharge into a closed container.

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TABLE AST-1
 OTEC CYCLE PLANT
 AMMONIA STORAGE TANK DESIGN

Pilot Plant Net Power	3MWe
Fluid Stored	liquid anhydrous ammonia
Total Ammonia Storage (lb)	70,000
Max. Storage Temp. (°F)	100
Storage Pressure (psia sat.)	212
Total Storage Volume (ft.)	2020
Number of Storage Tanks	1
Tank Diameter (ft)	8
Tank Overall Length (ft)	44
Tank Design Press. (psig)	250
Tank Wall Thick. (inch)	.75
Est. Empty Tank Weight (lb)	34,000

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SKETCH HX-1

HEAT EXCHANGERS

SKETCH TG-1

TURBINE GENERATOR

OUTLINE DIMENSIONS (FEET)

SYMBOL	NET POWER OUTPUT		
	3 MW	6 MW	9 MW
A	10.2	11.2	11.8
B	12.5	13.1	13.8
C	2.0	2.0	2.0
D	8.2	9.2	9.8
E	7.9	8.3	8.7
F	7.1	7.4	7.5
G	1.9	2.0	2.1
H	1.0	1.0	1.2
J	5.0	5.3	5.4

SKETCH CWP-1

BULB-TYPE PUMPS

AMMONIA VAPOR SYSTEM - PIPING

In the cycle plant, saturated ammonia vapor is generated by evaporation of liquid ammonia on the shell side of the baseline horizontal shell-and-tube evaporator. The ammonia vapor, containing about 5% moisture, passes through demisters which are internal to the evaporator shell to reduce entrained moisture to 1% or less. Liquid ammonia captured in the demisters drains to the evaporator sump. Vapor leaves the demisters through two demister outlet lines which join into one main vapor line transporting the vapor to one ammonia turbine. The vapor expands through the turbine, driving an electric generator, and exhausts through a diffuser and transition duct to the shell side of the baseline horizontal shell-and-tube condenser.

The cycle plant evaporator, turbine/generator, and condenser requirements are given in separate specifications. Following is a tabulation of the pilot plant ammonia vapor piping design conditions, pipe sizes, and quantities for the 3 MWe net cycle plant. Also tabulated are design requirements for the ammonia vapor demisters.

AMMONIA VAPOR PIPING

Cycle Plant Net Output

3MWe

Conditions

Ammonia flow (lb/sec)	228
Ammonia pressure (psia)	130
Ammonia temperature (°F)	70.4
Ammonia sp. volum (cu. ft/lb)	2.2
Ammonia vapor flow (cu. ft/sec)	498

Piping

- All Grade B Seamless Carbon Steel, Std. Wt. -

Main Vapor Line:

nom. pipe size (inch)	30
length (ft.)	80
fittings	2-90 elbows

Demister Outlet Lines:

nom. pipe size (inch)	20
total length (ft)	50
fittings	2-90 elbows, 1 - Tee

Turbine Bypass Line:

nom. pipe size (inch)	20
length (ft.)	20
fittings	2-90 elbows, 2 shutoff valves

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AMMONIA VAPOR PIPING (CONT'D.)

Demisters - 2 banks of demisters per evaporator (one bank in a vapor release dome on each side of evaporator)

Conditions:

Ammonia vapor density (lb/cu. ft)	0.4581
Ammonia flow per demister (lb/sec)	114
(cu. ft/sec)	249
Ammonia velocity (ft/sec)	7.25
Flow area per demister (sq. ft)	34

Demister Design: - Vertical, plate-type (hook & vane), for horizontal vapor flow

Material - Stainless Steel -

Pressure drop (inches of water)	2
Inlet vapor quality (%)	95
Outlet vapor quality (%)	99
Demister thickness (inch)	8
Est. weight per bank (lb)	2,500

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AMMONIA CONDENSATE SYSTEM - PIPING

Ammonia vapor exhausting from the cycle plant ammonia turbine is condensed on the shell side of the baseline shell-and-tube condenser. Condensed liquid ammonia drains to the condenser sump and is pumped to the evaporator by one of two 100% capacity condensate pumps. A recirculation line is provided from the discharge of the condensate pumps back to the condenser sump to preclude vapor binding or dynamic imbalance of the pump at low flow rates to the evaporator.

Continuous recirculation of liquid ammonia from the evaporator sump back to the evaporator sprays is provided by one of two 100% capacity reflux pumps. Up to 50% reflux (50% of condensate flow) can be achieved with one reflux pump in operation.

The cycle plant evaporator, condenser, ammonia turbine, condensate pumps, and reflux pumps requirements are described in separate specifications. Following is a tabulation of the design conditions, sizes, and quantities of piping for the liquid ammonia condensate and reflux pumping systems.

AMMONIA CONDENSATE SYSTEM - PIPING

Condensate Pump Piping

Cycle Plant Net Output 3MWe

Condition:

Ammonia flow (lb/sec)	228
(cu. ft /sec)	5.85
(GPM)	2623

Piping:

- all Grade B carbon steel, seamless -

Suction line size (inch) 14

wall thickness (inch) 0.25

length (ft) 45

fittings 3-90 elbows, 1-Tee, 2 shutoff valves

Discharge line size (inch) 12

wall thickness (inch) 0.18

length (ft) 145

fittings 6-90 elbows, 2-Tees, 2 shutoff valves,
2 check valves, 1 control valve, 1 venturi
flow meter nozzle

Recirc. line size (inch) 6

wall thickness (inch) 0.134

length (ft) 30

fittings 2-90 elbows, 1 shutoff valve,
1 control valve.

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AMMONIA CONDENSATE SYSTEM - PIPING (Cont'd.)

Reflux Pump Piping

Cycle Plant Net Output 3MWe

Conditions:

Ammonia flow (lb/sec)	114
(cu. ft /sec)	2.93
(GPM)	1312

Piping:

- all grade B carbon steel, seamless -

Reflux line size (inch) 8

wall thickness (inch) 0.148

length (ft) 100

fittings 8-90 elbows, 2-Tees, 4 shutoff valves,
2 check valves, 1 control valve, 1
venturi flow meter nozzle.

WARM CIRCULATING WATER SYSTEM

In the cycle plant, it is assumed that the warm circulating water will be supplied to the evaporator from an open water source, at unknown elevation, via an intake structure and supply conduit. The actual warm circulating water supply system will vary considerably from one site to another, depending on the water source, water level elevations, soil conditions, water quality, distance to water, etc. Although, the horizontal heat exchangers have been selected for the baseline cycle plant, vertical heat exchangers may also be tested requiring considerably more pump head. Based on assumptions as to water location relative to grade and preliminary vertical heat exchanger information, a 40 to 120 feet range in intake structure pump heads has been assumed to give an indication of the pumping power range which may be encountered. Changes in pump head can be accomplished by either a variable speed drive or by changing the pump bowl and/or impellor possibly in conjunction with a two speed motor. Studies of several alternate pump options will be made when the vertical heat exchanger types and tests are better defined.

The warm circulating water system can be divided into three main sections as follows: a) intake structure and supply conduit, b) cycle plant area and c) discharge conduit and structure.

Intake Structure and Supply Conduit

The intake structure (see Bechtel Dwg. 11726-P-1001) includes stop log guides, bar screens and rakes, traveling screens and intake structure pumps, control valves and piping. Pump discharge piping is manifolded into a single buried supply conduit.

Cycle Plant Net Output3MWe

Warm Intake Structure Pumps:

Number	3
Flow (cfs/gpm) each	(218/98,000)
Head (ft)	40-120
Motor Power (MWe/hp) each	(0.94-2.81/1254-3761)
Pump Efficiency (%)	88
Traveling Screen/Bar Screen Width (ft)	9
Traveling Screen Height (ft)	48
Number of traveling Screens/Bar Screens	6
Pump Discharge Line Size (inches)	60
Supply conduit diameter (ft)	10
Supply conduit water velocity (ft/second)	8.3

Cycle Plant Area

Warm circulating water is supplied directly to the warm circulating water pump (if installed) by the intake structure pumps via the supply conduit and appropriate spool piece. Sufficient suction pressure is maintained to avoid WCW pump cavitation. The water discharges through the horizontal heat exchanger slightly below grade elevation into an outlet conduit. The horizontal run is supported by a structural steel framework which can be modified to allow for critical components to be removed, rearranged or interchanged. All conduit in the immediate area is lined steel pipe. The outlet conduit contains an orifice plate to dissipate surplus head when vertical heat exchangers are installed or when pump head is increased to provide sufficient suction head for the WCW pump.

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A 10 feet diameter standpipe and butterfly flow control valve is provided in the outlet conduit to adjust heat exchanger water pressure and to provide a free water surface to minimize transient problems. Shutoff butterfly valves are also provided in the supply conduit and outlet conduit. Inlet and outlet conduit diameters are both 10 feet for the 3MWe cycle plant.

Discharge Conduit and Structure

Discharge conduit diameter is 10 feet for the 3 MWe cycle plant. The outlet structure is undetermined at this time but would be of diverging cross section to provide a low exit velocity into the receiving body of water.

Rev. 2, September 10, 1976

COLD CIRCULATING WATER SYSTEM

The cold circulating water system supplies water to the condenser from an open water source. Except for the intake structure pump flows and horsepowers given below, the system is identical to the Warm Circulating Water System described above.

Cycle Plant Net Output

3MWe

Cold Intake Structure Pumps:

Number	3
Flow (cfs/gpm) each	(203/91,000)
Head (ft)	40-120
Motor Power (MWe/hp) each	(0.87-2.61/1164-3492)
Pump Efficiency (%)	88

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CONDENSER WATER INJECTION

In the commercial OTEC plant, the performance of the ammonia cycle is adversely affected by in-leakage of water due to tube leaks in the condenser. In the cycle plant, in-leakage of water will not occur because the external circulating water pressure will be much less than the internal ammonia pressures. In order to simulate the effect of water in-leakage on the cycle plant performance, a condenser water injection system will be provided. The injection system will be capable of raising the concentration of water in the ammonia cycle from zero to 5% within one hour.

The condenser water injection system consists of one 500 gallon water storage tank, one fractional horsepower/10GPM/positive displacement metering pump developing 150 psi head, 50 feet of 1" std. wt. pipe, and 3 valves.

NITROGEN PURGE

Nitrogen will be used to purge air from the cycle loop prior to filling with ammonia and to purge ammonia vapor from equipment and piping to atmosphere prior to maintenance. Portable fans will be used to ventilate equipment after a nitrogen purge. Nitrogen will be supplied from high pressure tank trailers. Piping will be approximately 200 feet of 150 psig 2" diameter steel tubing with 8 valves and 1 pressure regulator.

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AMMONIA CLEANUP

The cycle plant will require one ammonia cleanup system to test the baseline ammonia cleanup system concept and to remove water introduced into the ammonia cycle by the condenser water injection system. The cycle plant ammonia cleanup system is one-quarter the scale of the baseline system and is fed from the reflux pump discharge line.

The system will handle 7,500 lb/hr of ammonia with 2% water content, and consists of: one 25 GPM/2HP ammonia pump; one ammonia distillation column 20" dia. x 20 ft. tall with an electric heating coil of 1200 KW capacity; one small cooldown heat exchanger with a duty of 38,000 BTU/hr; 100 ft. of 4" dia. std. wt. steel discharge pipe; 100 ft. of 3" dia. std. wt. steel feed pipe; and 6 valves.

LUBE OIL PURIFICATION

The turbine/generator lubricating oil will require frequent cleanup to ensure adequate bearing life in the rotating equipment. A 5GPM, double filtration oil purifier package will be provided which will take suction from and return clean oil to the turbine/generator oil sump. Filling and draining of the oil sump will be accomplished with a portable air-driven pump and 50 gallon steel drums.

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COMPRESSED AIR

The cycle plant will require dry compressed air for various instrumentation and maintenance functions. Two 100 cfm/100 psig/20HP oilless reciprocating air compressors (one spare) will be provided, with a common 30 cu. ft receiver tank. One common regenerative-type air dryer will also be provided. Air distribution piping will consist of 400 feet of 1/2" copper tubing and 30 valves.

AUXILIARY COOLING WATER

The auxiliary cooling water system provides cooling water to the cycle plant generator, the turbine/generator lube oil system, the air compressors, ammonia liquifier, and other miscellaneous equipment. The auxiliary cooling water is freshwater which flows in a closed loop consisting of a 500 gallon surge tank, heat exchanger, two 5HP/200 GPM/40 ft. head pumps (one spare), and 200 feet of 3" std. wt. piping with 20 valves. Heat is rejected at approximately 500,000 Btu/hr through the heat exchanger to the plant cooling water system. Inhibitors will be used in the auxiliary cooling water heat exchangers.

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CATHODIC PROTECTION

A DC current, constant potential, automatically regulated cathodic protection system consisting of a distribution of anodes, the control system, and associated wiring will be used to limit galvanic corrosion of cycle plant carbon steel components in the warm and cold circulating water system. Components to be protected include steel circulating water piping and the warm and cold circulating water pumps. Power consumption of the cathodic protection system is estimated at 1 KW.

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CONTROL SYSTEMS

The major systems of the cycle plant will be controlled by remote control devices. Remote control devices will be located in a common central control room, along with critical system alarms and indicators, in order to facilitate both normal operation and safe and controlled plant startup or shutdown. The controls will provide the capability to operate the power cycle and selected components at "off-design" conditions.

Major system and equipment to be controlled from the central control room include:

- o Ammonia turbine/generator (turbine trip, reset, throttle vanes, bypass valve, generator excitation, synchronization, breaker control).
- o Ammonia condensate and reflux pumps and control valves.
- o Warm and cold circulating water pumps (if installed).
- o Warm and cold water intake structure pumps, and flow control valves.
- o Downstream pressure/level control valve.
- o Circulating water air removal pump
- o Ammonia evaporator and condenser level controls
- o Ammonia condensate pumps recirculation control

The following auxiliary systems will be locally controlled (at the equipment), with critical alarms duplicated in the central control room: Compressed air, auxiliary cooling water, ammonia transfer, nitrogen purge, ammonia cleanup, condenser water injection, tube cleaning (Amertap), lube oil purification, cathodic protection, and fire protection.

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CONTROL SYSTEMS (Cont'd.)

Considerable additional instrumentation may be required for specific tests to monitor and evaluate the performance of various components of the cycle plant. Such devices will be specified and estimated by LMSC.

TUBE CLEANING

The evaporator and condenser will each have a modified "Amertap" tube cleaning mechanism similar to the design used in the baseline OTEC study. The cycle plant Amertap system will recirculate the special sponge balls with a flow of about 500 GPM, using a 30 HP ball recirculating pump, for use with both the evaporator and condenser.

AREA DRAINAGE

Since the floor of the concrete pit area for the cycle plant is 30'-0" below grade, it will not be possible to gravity drain the area to the test facility drainage system. Hence, a sump pit and two 150 GPM sump pumps will be required to keep the area dry in the event of minor leakage and spillages and also to pump drainage from the heat exchanger water loop. The pumps will be equipped with automatic sump pit level controls and will discharge through 100 feet of 3" pipe into the test facility drainage system.

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AMMONIA TRANSFER

One 5HP/50 GPM ammonia transfer pump will be used to transfer liquid ammonia from the storage tank to the evaporator or condenser during system fill or makeup. Piping consists of 300 feet of 4" std. wt. steel pipe, 2 shutoff valves, and 1 control valve.

One steam heated ammonia vaporizer will be used to pressurize the condenser and evaporator with superheated ammonia vapor during system filling. Consumption of 150 psig saturated steam is 6000 lb/hr for an ammonia fill rate of 10,000 lb/hr. Steam supply will be from a portable steam generator. Piping from vaporizer to condenser and evaporator consists of 75 feet of 14" std. wt. steel pipe and 3 valves.

One 25 HP/100 CFM ammonia compressor and water cooled ammonia liquifier will be used to transfer ammonia vapor from the condenser or evaporator to storage during system draining. Piping consists of 75 feet of 8" std. wt. steel pipe with 3 valves. A condensate pump can be used to transfer ammonia condensate to storage via the reject line.

One 150 HP (25 SCFM @ 1" Hg abs.) mechanical vacuum pump will be used to evacuate the condenser, evaporator, and associated ammonia piping prior to ammonia fill. Piping consists of 75 feet of 8" std. wt. steel pipe and 3 valves. This vacuum pump can also be used to create a vacuum in the circulating water pipe if necessary on startup.

An ammonia cold trap will be used to reduce the amount of ammonia lost during venting of the non-condensable residual nitrogen after an ammonia fill has taken place and startup is being initiated. The cold trap may include a vent gas compressor and a water/dry ice/ refrigeration system ammonia condensation unit.

Rev. 1, June 25, 1976

CIRCULATING WATER AIR REMOVAL

Warm and cold circulating water is transported to/from the cycle plant evaporator and condenser via large diameter piping. A vacuum may exist at the high points in the circulating water lines due to siphon effect of outflowing water. Some of the air dissolved in the incoming circulating water will come out of solution under vacuum and will collect at the high point in the piping, restricting the circulating water flow. To remove air evolved in the circulating water piping, a circulating water air removal system will be provided to remove air from the warm and cold circulating water lines.

The circulating water air removal system consists of: two 20 HP vacuum pumps (600 cfm at 7" Hg. abs.); two 20 ft air release/level control tanks; 150 ft. of 6" std. wt. steel pipe; four 6" valves; and two 2" control valves.

Rev. 1, June 25, 1976

ELECTRICAL

A single-line diagram of the cycle plant electrical system is shown on Bechtel Drawing 11726-P-1001. The system includes a 3 MWe turbine-generator that delivers power to a 13.8 KV bus. Plant auxiliaries are fed from a 4160V load center. This load center can also provide power to a warm and a cold circulating pump if required in the future. Motors from 1 to 250 hp will be served by a motor control center. Fractional horsepower loads and other miscellaneous loads will be served from a 240/120 volt distribution panel. Redundant loads will be connected to separate motor control centers for greater plant reliability. A separate ammonia heater panel will serve the cleanup column heating load.

The step-down transformer and load center will be of the outdoor metal clad type. The 480V step-down transformer for the motor control center will be an outdoor type. The motor control center will be an indoor type installed inside the cycle plant auxiliary equipment building.

Rev. 1, June 25, 1976

FIRE PROTECTION AND SAFETY EQUIPMENT

The cycle plant will be connected to the test facility firewater system which includes a 90,000 gallon storage tank and a 1000 GPM/120 psig/75 HP diesel fire pump. 10 hose stations for 50 ft. lengths of 1-1/2" hose will be connected to the 6" main header in the cycle plant area. The fire water system can also be used to dilute and wash away any major ammonia spills.

Approximately 10 portable 20 lb. dry chemical or carbon dioxide cylinders will be located throughout the cycle plant area.

A 1/2 ton refrigerated carbon dioxide unit will be provided for the cycle plant generator.

Approximately 20 breathing air packs with protective face masks, and 10 protective personnel suits will be located throughout the cycle plant area for use in the event of an ammonia spill.

Four wind socks will be visibly located to indicate the upwind direction for personnel escape in the event of a major ammonia spill.

Rev. 1, June 25, 1976

EQUIPMENT HANDLING

Over the life of the cycle plant it may be necessary to remove the major components for maintenance or to replace a component with alternate designs. Major components which might be removed and reinstalled include the evaporator, condenser, turbine/generator, warm and cold circulating water pumps or flow distribution elements, condensate pumps, and reflux pumps. The physical arrangement of the cycle plant is such that a single traveling gantry crane spanning the evaporator and condenser could service all these major cycle plant components. A permanent cycle plant gantry crane as described below and capable of lifting the heaviest cycle plant component (evaporator) will be provided.

Cycle Plant Gantry Crane: 160 feet span, 60 feet rail to girder height, 250 ton rated load capacity, double leg, electric, traveling gantry crane, supported on multi-wheel rail tracks. The crane shall have two equal capacity, independently controlled trolley hoists. All bridge, trolley, and hook motions to be controlled from a single operator's cab supported from one of the crane bridge girders and capable of traversing the span of the girder. Power supply to the crane will be 460 volt, 3-phase, 60 Hz via single conductor cables on take-up reels.

Rev. 1, June 25, 1976

LAND BASED CYCLE PLANT CONSTRAINTS/RISKS

- WATER TEMPERATURE DIFFERENTIAL/SEASONAL VARIATION
- WATER FLOW RATES SEASONAL VARIATION
- WATER QUALITY
- ENVIRONMENT-WEATHER
- TRANSPORTATION
- OTEC EQUIPMENT SCHEDULE
- ABILITY TO ACCEPT ALTERNATIVE COMPONENT CONFIGURATIONS

7-98A

GRAPHIC SCALE 1" = 20'

LMSC/ERDA JTEC TEST FACILITY PILOT PLANT		
GENERAL ARRANGEMENT OPEN WATER LOOP VERTICAL HEAT EXCHANGER 6MW _e		
JOB No.	DRAWING No.	REV.
11726	P-201	A

7-98B

GRAPHIC SCALE 1" = 50'

LMSC/ERDA OTEC TEST FACILITY PILOT PLANT		
GENERAL ARRANGEMENT		
CLOSED WATER LOOP		
HORIZONTAL HEAT EXCHANGER 6 MW _e		
200 IN.	DRAWING NO.	REV.
11726	F-301	A

7-98D

1. REFRIGERATED CO₂ STORAGE UNIT
2. CONDENSER WATER INJECTION METERING PUMP
3. CONDENSER WATER INJECTION STORAGE TANK.
4. AUXILIARY COOLING WATER SURGE TANK
5. AUXILIARY COOLING WATER PUMP
6. AUXILIARY COOLING WATER HEAT EXCHANGER
7. CIRCULATING WATER AIR REMOVAL VACUUM PUMP
8. AIR COMPRESSOR
9. N₂ RECEIVER
10. REGENERATIVE AIR DRIER
11. MOTOR CONTROL CENTER
12. AMMONIA VAPORIZER
13. AMMONIA COMPRESSOR
14. AMMONIA COLD TRAP
15. AMMONIA FILL VACUUM PUMP
16. AMMONIA LIQUIFIER
17. AMMONIA CLEANUP PUMP
18. AMMONIA CLEANUP DISTILLATION COLUMN
19. AMMONIA CLEANUP COOLDOWN HEAT EXCHANGER
20. PROTECTIVE CLOTHING & BREATHING AIR

0 5' 10' 20'
GRAPH C SCALE

LMSC/EZDA OTEC TEST FACILITY		
CYCLE PLANT, AUX EQUIPMENT BUILDING		
JOB No.	DRAWING No.	REV.
11726	P-1002	A

7-98E

GRAPHIC SCALE 1" = 20'

LM.S.C./ERDA OTEC TEST FACILITY		
CYCLE PLANT, LAND BASED. VERTICAL SHELL & TUBE HEAT EXCH. WITH VERTICAL UP FLOW		
JOB No.	DRAWING No.	REV.
11726	P-1011	A

7-98F

7-98G

LMSC/ERDA OTEC TEST FACILITY		
CYCLE PLANT, LAND BASED VERTICLE SHELL & TUBE HEAT EXCH WITH VERTICAL DOWN FLOW		
JOB No.	DRAWING No.	REV.
11726	P-1012	A

H86-7

LMSC/EZDA OTEC TEST FACILITY		
CYCLE PLANT, LAND BASED. LMSC VERTICAL PANEL HEAT EXCH WITH VERTICAL DOWN FLOW		
JOB No.	DRAWING No.	REV.
11726	P-1013	A

7-981

LMSC/ERDA
OTEC TEST FACILITY

CYCLE PLANT, LAND BASED,
A.P.L. HEAT EXCH. WITH
VERTICAL DOWN FLOW

JOB No.	DRAWING No.	REV.
11726	P-1014	A

GRAPHIC SCALE 1" = 20'

TEST FACILITY CONSTRAINTS/RISKS

THE LISTED LAND-BASED TEST FACILITY CONSTRAINTS/RISKS HAVE BEEN IDENTIFIED. THE MAJOR CONSTRAINT IS THE WATER RESOURCES FOR THE CYCLE PLANT. THE LAND TOPOGRAPHY, WHICH INCLUDES SURFACE VARIATIONS AS WELL AS SOIL CONDITIONS, IS A MAJOR COST-SENSITIVITY CONSIDERATION. EXTREMES OF GRANITE OR MARSHLANDS WILL INCREASE THE COST OF THE TEST FACILITY CONSIDERABLY.

THE WEATHER AND ITS INFLUENCE ON SUCH THINGS AS BUILDING COSTS AS WELL AS THE LOCAL ASPECTS OF TRANSPORTATION, UTILITY AND SUPPORT SERVICES, PUBLIC ACCEPTANCE, AND AVAILABLE LABOR POOL ARE ALL CONSTRAINTS WHICH MUST BE RESOLVED IN THE SELECTION OF THE TEST FACILITY SITE.

THE FINAL CATEGORY OF CONSTRAINTS/RISKS PERTAIN TO THE LOCAL PUBLIC UTILITIES IN TERMS OF THE POTENTIAL RESTRICTIONS ON WATER OR ELECTRICAL POWER SUPPLY, UTILITY OPERATING PROCEDURES OR CONDITIONS AND INTERFACING PROBLEMS AND/OR PENALTIES ASSOCIATED WITH CONNECTING TO AND USING FROM OR SUPPLYING TO THE UTILITY. THIS INCLUDES THE EPA IMPACT.

7.6 SEA BASED CYCLE PLANT

The conceptual plant design study for the sea based cycle plant is presented in this section. This section is the basis for Report Section K and is brief as the test platform is discussed in Report Section L. Included at the end of a brief statement of the constraints/risks for the sea based cycle plant.

SEA-BASED OTEC CYCLE PLANT

TEST FACILITY

1.0 INTRODUCTION

A sea-based subscale test facility for the OTEC cycle plant has been proposed by LMSC as a potential step in the OTEC plant development program. The sea-based facility (as opposed to the land-based facility defined previously) is intended to demonstrate the technical feasibility of the power cycle components in an actual ocean environment utilizing the existing ocean thermal gradient. This report presents the results of a conceptual plant design study for the sea-based test facility.

A sea-based test facility will not only provide an actual ocean environment for the thermodynamic cycle, but will also permit evaluation of operational and maintenance problems associated with this environment. Operability and maintainability are expected to be among the most important factors in establishing feasibility of OTEC power plants. Another advantage of a sea-based test facility is its flexibility regarding site or location. The plant can be occasionally relocated to run tests under different ocean conditions.

The Bechtel effort for the sea-based test facility design study is summarized by the following activities:

- o Review LMSC sketches (LMSC provided Bechtel with sketches showing a preliminary conceptual arrangement).
- o Develop the power cycle equipment arrangement for a 6 MWe net cycle plant using the baseline LMSC horizontal heat exchangers.
- o Define the auxiliary equipment and include these in the arrangement.

- o Mark-up the 6 MWe net cycle plant arrangement to 3 MWe and 9 MWe plant dimensions.
- o Develop conceptual cost and schedule estimates for the 3, 6, and 9 MWe plants.
- o Develop 6 MWe net cycle plant arrangements showing vertical heat exchangers installed.

Section 2 of this report outlines the assumptions and ground rules used in the study. Section 3 briefly describes the main and auxiliary systems of the cycle plant, and Section 4 gives a brief description of the plant arrangement.

2.0 ASSUMPTIONS AND GROUND RULES

The following assumptions and ground rules specified by LMSC are used in the study:

- o The power cycle equipment is to be installed on an equipment base. The equipment base should be capable of being mounted or carried on a vehicle such as a ship, a barge, or a drydock. Since no specific vehicle has been selected at this time, the equipment base should be as flexible as possible.
- o The method of mounting the equipment base to a vehicle is not included since this will depend on the type of vehicle selected.
- o The power cycle equipment for the sea-based plant is to be the same as those of the land-based plant.
- o The plant should be capable of operating in both a dry and a wet environment (both below sea water level) since specification of a dry only or a wet only environment would otherwise put a constraint on the type of vehicle that could be used. Capability of operation in either environment provides more flexibility.
- o A surface support vessel will accompany the main vehicle. The auxiliary equipment directly associated

with the power cycle will be located on the cycle plant equipment base. All other auxiliary equipment will be located on the support vessel. It should be recognized here that for certain types of main vehicles, a support vessel may not be necessary since the main vehicle may be able to accommodate all the auxiliary equipment.

- o Generator excess energy will be dissipated through an electrical load bank.
- o Start-up power will be supplied by diesel-generators.

3.0 PLANT MAIN AND AUXILIARY SYSTEMS

3.1 MAIN SYSTEMS

Definition of the sea-based plant cycle diagram was provided by LMSC. The cycle is essentially the same as for the land-based cycle plant. The main and auxiliary subsea equipment is similar to the land-based cycle plant equipment except that, where applicable, it is enclosed in water-proof, pressurized containers to eliminate contamination by sea water. The main systems for the test facility include the following:

- o Ammonia Vapor
- o Liquid Ammonia
- o Sea Water

3.1.1 Ammonia Vapor System

The ammonia vapor system consists of the basic vapor handling and power conversion portion of the test facility. It includes the evaporator, demisters, turbine, generator, and associated piping and valves.

Anhydrous liquid ammonia is delivered to the evaporator, sprayed over the tubes, heated by sea water and vaporized. Since all of the ammonia does not immediately vaporize, the remaining liquid is collected in the evaporator sump. Reflux pumps are used to recirculate this fluid to the spray section for continued evaporation.

The ammonia vapor discharged from the evaporator is passed through demisters to remove the excess moisture. The vapor discharged from the demisters has a quality of about 99%. Surplus liquid is drained from the demisters and returned to the evaporator.

The essentially dry and saturated ammonia vapor is ducted to the turbine-generator. The 6 MWe net cycle plant turbine-generator has a nominal rating of approximately 7.5 MWe. The turbine exhaust is fitted with a diffuser for recovery of some of the kinetic energy of the discharged vapor. The diffuser is ducted into the condenser.

The generator has a 60 Hertz, 4160 volt output.

3.1.2 Liquid Ammonia System

The liquid ammonia system includes the condenser, the condensate pumps, reflux pumps, and piping and valves.

The condensed ammonia is collected in the condensate sump, then pumped to the evaporator by one of two 100% capacity condensate pumps. A level control valve at the discharge of the condensate pumps controls level in the evaporator sump.

Level in the condenser sump will be controlled by reject and makeup lines to and from ammonia storage.

The ammonia system is complemented by ammonia clean-up, make-up and storage auxiliary systems.

3.1.3 Seawater System

The seawater systems provide the heat source and sink necessary for the generation of thermal power. In these systems, warm water from the surface and cold water from the subsurface are introduced to the plant and pumped through the respective heat exchangers.

The warm seawater system consists of the warm seawater pump, ducting, and screen. Warm water drawn from about 65 ft below the sea surface flows through the screen and to the pump. The water is pumped through the evaporator and is then discharged.

The cold seawater system consists of the cold seawater pump, the ducting, and screen. Cold water is drawn from about 1100 ft below the sea surface through a screen and a duct which projects downward from the platform. The water is then pumped through the condenser and is discharged.

3.2 AUXILIARY SYSTEMS

The following auxiliary systems for the sea-based cycle plant have been identified:

- o Generator Power Dump
- o Startup Power Supply
- o Cooling Water
- o Turbine Oil Purification
- o Ammonia Clean-up
- o Leak Detection and Fluid Sampling
- o Electrical
- o Ammonia Make-up and Storage
- o Nitrogen Purge
- o Heat Exchanger Tube Cleaning
- o Hydraulic Pressure
- o Seawater Pump Pod Pressurizing
- o Compressed Air
- o Control and Instrumentation
- o Cathodic Protection
- o Communication
- o Fire Protection
- o Equipment Handling and Removal

Many of the above auxiliary systems are similar to those of the land-based cycle plant and the full-scale baseline plant which are described elsewhere. However, the Generator Power Dump System and the Startup Power System are unique to the sea-based cycle plant. Brief descriptions of these systems are given below.

3.2.1 Generator Power Dump System

A generator power dump system is provided to dispose of the net plant power output. Since the cycle plant is not expected to operate continuously for long periods, it is not economical to connect the generator to a utility grid. Moreover, the cycle plant is expected to operate at several locations under various ocean environments. Therefore, the plant net power is dissipated to the ocean through an electrical load bank.

The power dissipation system is schematically shown in Figure 1. The electrical load bank is essentially a set of electrical resistors operated at 480 volts. Heat generated in these resistors is removed by fresh water circulating in a closed loop which, in turn, rejects its heat to an open seawater loop through a heat exchanger. A unique feature of the fresh water closed loop is that fresh water is vaporized in the load bank and condensed in the heat exchanger. This feature reduces the size and cost of the heat exchanger equipment and is widely used in the industry.

The generator output voltage of 4160 volt was selected based on economics and generator size considerations. Since about 80% of the total power is dissipated in the load bank at 480 volt, it would apparently seem logical to generate power at this voltage. However, this is impractical for two reasons. Firstly, a 480 volt generator would be more expensive and very large in size. Secondly, the load bank is located at a distance such that 480 volt transmission would be uneconomical.

FIG. 1
GENERATOR POWER DUMP SYSTEM

3.2.2 Startup Power Supply System

This system is provided to supply auxiliary electrical power during plant start-up and to provide standby power during extended periods of plant shutdown. A block diagram of the electrical system showing the diesel generators of the startup power system is shown in Figure 2. The total auxiliary power requirements for the plant are expected to be approximately 1650 kW. For flexibility and reliability of operation, two standard size 900 kW, 4160 volt, diesel generators are provided. Auxiliaries requiring 4160 volts are connected directly to the 4160 volt bus. Power to the 480 volt auxiliaries is supplied through a 4160/480 volt step-down transformer. A separate 4160/480 volt step-down transformer is provided for the load bank because it is not economical to transmit 6 MWe at 480 volts over a long distance.

4.0 PLANT ARRANGEMENT

Based on the main and auxiliary systems discussed in Section 3, equipment dimensions (based on the land-based cycle plant) and plant arrangement engineering sketches were developed to determine space requirements and to form a basis for platform definition. The sketches are attached as Bechtel drawings 11726-P-2001 and 11726-P-2002. These drawings show the arrangement of the 6 MWe test facility.

The approach taken, as outlined in Section 2, is to develop an arrangement flexible enough so that the power cycle equipment of the test facility can be accommodated on any vehicle available for the purpose. Also, since it is not known whether the plant will be operating in a dry environment or a wet environment, the cycle plant and the equipment base are defined such that they can be installed

FIG. 2
ELECTRICAL SYSTEM

and operated in either environment. Since the specific vehicle is not known, the cycle plant equipment was arranged into a compact installation to minimize the preclusion of any available vehicles which may be considered.

The cycle plant equipment can be divided into two groups: power cycle equipment and auxiliary equipment. In general, the power cycle equipment is installed on the equipment base and most of the auxiliary equipment is installed on a support vessel. Some auxiliary equipment which requires close proximity to the power cycle equipment (e.g., turbine oil purification system), is also installed on the equipment base. Bechtel drawing 11726-P-2001 shows the general arrangement of the power cycle equipment on the equipment base. Drawing 11726-P-2002 shows approximate deck space requirements for the auxiliary equipment on the support vessel.

Sizes of power cycle equipment are the same as those of the land-based cycle plant. To avoid recirculation, the warm and cold seawater discharges are located on the same side of the equipment base. Submergence of the 6 MWe cycle plant equipment is governed by an estimated minimum required pump submergence of 65 feet to avoid cavitation. This results in an equipment base elevation of 93 feet below sea water level and a top-of-turbine elevation of 2 feet above sea level.

Arrangement of the auxiliary equipment in drawing 11726-P-2002 is developed for the 6 MWe case to the extent needed to determine approximate deck space requirement. No specific support vessel is identified. Therefore, this auxiliary equipment should be rearranged in accordance with the available deck space on the selected support vessel. Major equipment and facilities to be installed on the support

vessel include control room, load bank for generator power dump, auxiliary transformers, switch-gears, air compressor, diesel generators, ammonia storage and transfer equipment.

The support vessel is connected to the main vehicle by an umbilical cord. This cord will carry power cables, instrument and control cables, compressed air lines, ammonia fill and dump lines, and other fluid lines for some of the auxiliary systems.

Bechtel drawings 11726-P-2003 and 11726-P-2004 show the arrangements of the power cycle equipment for 3 MWe and 9 MWe net cycle plants, respectively. These drawings are marked-up versions of the 6 MWe base case arrangement and are intended to portray the space needed for these cases.

Bechtel drawings 11726-P-2005, 11726-P-2007 and 11726-P-2008 show the outline and space requirements of the major power cycle equipment for the 6 MWe case using various vertical heat exchanger configurations.

7-112A

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT 6 MWe NET		
NO.	REVISION NO.	REV.
11726	P-2001	A

7-112B

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT, SURFACE AUXILIARY EQUIPMENT, DECK BARGE SPACE ALLOCATION		
JOB No.	DRAWING No.	REV.
11726	P-2002	A

7-112C

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT 3 MWe NET		
JOB No.	DRAWING No.	REV.
11726	P-2003	A

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT 9 MWe NET		
JOB No.	DRAWING No.	REV.
11726	P-2004	A

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT 60MW _{NET} WITH APL HEAT EXCHANGER		
JOB No.	DRAWING No.	REV.
11726	P-2005	A

LMSC/ERDA		
OTEC TEST FACILITY		
SEA BASED CYCLE PLANT		
6 MWe NET		
WITH VERT TUBE HEAT EXCHANGER		
JOB No.	DRAWING No.	REV.
11726	P-2007	A

LMSC/ERDA OTEC TEST FACILITY		
SEA BASED CYCLE PLANT 6MWe NET WITH VERTICAL PANEL HEAT EXCH.		
JOB No.	DRAWING No.	REV.
11726	P-2008	A

SEA-BASED CYCLE PLANT CONSTRAINTS/RISKS

- SEA SURFACE CONDITIONS
- SHORE SIDE SUPPORT
- LOGISTICS – SITE / PORT DEPENDENCY
- PERSONNEL
- OTEC EQUIPMENT SCHEDULE
- ALTERNATE COMPONENT CONFIGURATIONS
- PLATFORM INDEPENDENT

SEA BASED CYCLE PLANT CONSTRAINTS/RISKS

THE AT-SEA CYCLE PLANT TEST OPERATIONS, TEST CONFIGURATION MODIFICATIONS AND MAINTENANCE ACTIVITIES ARE LIMITED BY WEATHER CONDITIONS SPECIFICALLY SEA STATES AT THE TEST SITE OR BETWEEN THE TEST SITE AND THE SHORE FACILITY. THE AT-SITE OPERATIONS IS DEPENDENT UPON SUPPORT FROM THE SHORESIDE FACILITY OR ACTIVITY BASED ON THE CAPABILITY AND SHORESIDE FACILITY CONDITIONS SUCH AS SIZE, SERVICES, AND DOCKSIDE ARRANGEMENTS.

THE LOGISTICS ASPECTS WHICH CONSTRAIN THE AT-SEA CYCLE PLANT ARE PRIMARILY ONE OF DISTANCE - HELICOPTER RANGE AND NUMBER AND SIZE OF SUPPLY OR SUPPORT VESSELS. PERSONNEL MAY BE A CONSTRAINT BASED ON TYPES OF WORKERS REQUIREMENT (LABORERS, TECHNICIANS, ENGINEERS, ETC.), THE ON-BOARD WORK SHIFTS (8 HOUR SHIFTS, 12 HOURS ON - 12 HOURS OFF) AND THE WORK ROTATION CYCLE, (TWO WEEKS ON - TWO WEEKS OFF OR FOUR WEEKS ON - FOUR WEEKS OFF OR ?).

A POTENTIAL CONSTRAINT IS THE COMPATIBILITY OF THE OTEC PROCUREMENT SCHEDULES WITH THE AT-SEA TEST PLATFORMSCHEDULE AS THE TEST PLATFORM IS NOT DEFINED. FINALLY THE COMPATIBILITY OF SPACE, VOLUME AND WEIGHT OF THE VARIOUS OTEC COMPONENTS, THE AUXILIARY AND ANCILLARY SYSTEM AND EQUIPMENT, AND THEIR FOUNDATIONS WHICH MAY BE CONSTRAINED BY THE TEST PLATFORM TYPE AND SIZE.