PLATFORM STATION KEEPING STUDY

OCEAN THERMAL ENERGY CONVERSION PROGRAM

Final Report

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ABSTRACT

Station keeping considerations for the OTEC plant can have significant effect upon platform configuration and overall plant efficiency and cost. This study develops candidate configurations for dynamic positioning and mooring systems for the OTEC application, presenting parametric performance and cost data over a wide range of platform size and configuration and site environments. The data provide a general baseline, useful for OTEC system analysis, and generally applicable to future system configurations.

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SECTION 1

INTRODUCTION

1.1 General Statement of the Problem

Throughout the investigations of Ocean Thermal Energy Conversion (OTEC), major investigators have considered the ocean engineering aspects of platform station keeping as dictated by their specific configurations. Each station keeping scheme has been unique in some aspect, with variations in platform configuration and size, basic operating concept, and site environmental models making comparison and generalization difficult. Table 1-1 summarizes the essential aspects of station keeping schemes proposed to date.

The purpose of this study is to develop a general treatment of the OTEC station keeping problem, including but not restricted to the plant configurations proposed to date. Two general methods of station keeping are considered; namely, dynamic positioning and mooring. Station keeping configurations are developed and evaluated from the standpoint of performance and cost. All cost data presented in this report include estimates for raw materials and production fabrication. Purchase quantities are assumed to be adequate for initial outfitting of a commercial sized plant.

The results of the study are general, allowing application throughout the evolution of OTEC. The study provides data to be used in determining technical risk and identifies areas requiring future study.

1.2 Study Objectives

The broad objective of this study is to assess the impact of station keeping upon the OTEC program. This is accomplished by establishing a general technological and economic data base for OTEC station keeping. Areas requiring future development are identified to assist in continued OTEC program planning. Specific study objectives are:

**TABLE 1-1. STATION KEEPING CONCEPT PROPOSED BY PREVIOUS
MAJOR INVESTIGATORS**

INVESTIGATOR	PLANT NET POWER $\dot{W}_{net}$ (MWe)	PLANT CONFIGURATION	SITE ENVIRONMENT			STATION KEEPING CONCEPT
			WIND SPEED (m/s)	SURFACE CURRENT (m/s)	DEPTH Z (m)	
U. Mass ⁽¹⁾	400	Fully submerged catamaran hull	--	2.0	--	Single point mooring using cold water pipe as tension member.
Lockheed ⁽²⁾	160	Spar	93	1.5	762- 6096	Single point mooring using buoyant link chain.
TRW ⁽³⁾	100	Circular barge	52	1.0	914- 2743	Dynamic positioning using effluent thrust. Operates fixed or grazing depending upon environmental loads.
APL/JHU ⁽⁴⁾	100	Conventional barge	18	0.5	--	Dynamic positioning using four 750-HP thrusters. Operate in grazing mode at 0.26 m/s grazing speed.

1. Determine plant/platform operational and site environment requirements and constraints.
2. Determine the range of drag forces to be reacted by the station keeping system.
3. Develop a general data base describing seawater effluent thrust, external thrusters, and deep water mooring lines.
4. Establish several station keeping concepts. Evaluate these concepts from the standpoint of performance, cost, and system impact.
5. Recommend future development needs.

1.3 Study Approach

Figure 1-1 shows the general study work flow. Platform drag and station keeping system performance are analyzed separately, allowing independent application of data from either analysis. Numbers in each task block refer to sections of this report. The independent analyses are parametric in nature to provide the broadest possible use. Candidate systems, developed from the general station keeping system descriptions, reduce the number of variables to provide a closer look at specific capabilities. These candidates are used to assess performance and cost in the OTEC context.

SI units are used throughout this report. Appendix A summarizes the units used and provides conversions to the English system.

Figure 1-1. Station Keeping Study Flow

Section 1 — References

1. Heronemus, W.E. et. al., "Summary of University of Massachusetts Research of Gulf Stream Based Ocean Thermal Power Plants", Proceedings, Third Workshop on Ocean Thermal Energy Conversion (OTEC) May 1975.
2. Trimble, L.C., et. al., "Ocean Thermal Energy Conversion (OTEC) Power Plant Technical and Economic Feasibility Final Report, Vol 1 and 2", Lockheed Missiles and Space Company, Inc., NSF/RANN/SE/GI-C937, April 1975.
3. Douglas, R.H., et. al., "Ocean Thermal Energy Conversion Research on an Engineering Evaluation and Test Program — Final Report, Vol 1 - 5", TRW Systems Group, NSF/RANN/SE/GI-C958, June 1975.
4. Dugger, G.L., et. al., "Ocean Thermal Power Plants", APL Technical Digest, Vol 14, No. 1, Jan-March 1975.

SECTION 2

EXECUTIVE SUMMARY

This report includes the results of a one-year study of OTEC platform station keeping. The subject is one of the major ocean engineering issues affecting the OTEC system and has significant impact upon platform configuration, site selection, and overall system efficiency and cost. The study establishes a base of station keeping data for application to OTEC system studies, combining parametric data and analysis of specific configurations to assure broad applicability of the results.

The analysis includes two general station keeping concepts; namely, dynamic positioning and mooring. Both are assessed on the bases of performance, cost, and development needs. Data are presented in a format chosen to facilitate performance and cost comparison of station keeping concepts. The cost information is in terms of cost per unit of plant power (\$/kWe), allowing direct comparison with other OTEC subsystems.

The study uses a parametric platform model for drag calculations. The model defines five generic platform configurations including semisubmersible, circular barge, conventional barge, submersible, and spar shapes. A general size scaling relationship allows study of a plant net power range of 20 to 1000 MWe. Three OTEC site environments are considered. One site is typical of a relatively benign Gulf of Mexico location; a second site is based upon Miami data and represents a severe OTEC environment. The third environment, chosen as the Westinghouse baseline model, represents a compromise between the other two.

Drag loads on the OTEC platform vary from 1.7 MN for the smallest plant in the benign environment to 51 MN for the largest plant in the most severe environment. For a given plant net power and environment, platform configuration differences lead to drag forces varying by factors as high as 3. For example, drag forces for a 100-MWe plant in the benign environment vary from 2.9 MN for the spar to 6.2 MN for the semisubmersible or a factor of approximately 2.

The thrust required for dynamic positioning of an OTEC platform can be generated by conventional propulsion systems. Thrusters similar to those used in the offshore oil industry are adequate for plants to several hundred megawatts in a benign environment. The largest OTEC plants in the most severe environment could require propulsion systems with power equal to that of the largest aircraft carriers. Thrust generated by effluent flow from the OTEC seawater transport system can supplement thrusters for OTEC plants of moderate power. Dynamic positioning using effluent thrust, alone, is feasible for plants above 600 MWe.

The thrust required to dynamically position an OTEC plant is within the capability of reasonable hardware. The power required to generate this thrust is an important issue. At least one tenth of rated plant output is required for propulsion to hold a 100-MWe plant on station under storm conditions in the benign environment. In the same environment, a 20-MWe plant requires at least 50% of rated output for propulsion and a 500-MWe plant using effluent thrust requires about 1% of plant power. A 20-MWe plant in the severe environment requires more propulsion power than the plant can produce.

A two-component mooring line using the strongest synthetic rope and chain available could withstand storm loads for a medium power OTEC plant in a benign environment. Extension of mooring technology to the entire OTEC drag range requires development of an advanced mooring line construction. Buoyant chains have been suggested to meet this requirement. Data for such a concept are presented in this report. An advanced mooring design for a 1000-MWe OTEC plant located in 1500-meter deep water appears feasible. Performance data for a range of mooring configurations are presented in the report.

Cost estimates for thrusters and mooring lines are developed as a function of plant net power. The costs for these alternatives fall within the same range. The cost for either candidate for a 100-MWe plant in the benign environment ranges from 40 to 100 \$/kWe. The upper limit increases to 400 \$/kWe for the severe environment. The unit cost for a 20-MWe plant in the severe environment can be as high as 1000 \$/kW. Costs demonstrate a dramatic economy of scale. Unit cost decreases rapidly with increasing plant size.

SECTION 3

CONCLUSIONS AND MAJOR FINDINGS

The data presented in this report are intended primarily for use in system analysis of the OTEC plant. Although the data are most useful for system tradeoff, conclusions may be drawn at this point to steer future efforts and simplify later applications. These conclusions and major findings follow:

- An analysis of five generic platform candidates, three site environments, and plant net power levels from 20 to 1000 MWe leads to an OTEC plant drag range of 1.7 to 51 MN.
- Cost of dynamic positioning systems is sensitive to both plant configuration and environment. From the standpoint of station keeping, the OTEC plant should be of low drag design with minimum exposure to wind and strong preference should be given to the more benign environments. Deviation from these guidelines can have serious consequences in overall OTEC cost.
- Station keeping costs demonstrate a dramatic economy of scale relationship. Unit costs for both dynamic positioning and mooring in the severe environment decrease by approximately one order of magnitude over the plant net power range from 20 MWe to 1000 MWe.
- Station keeping of a 100-MWe plant in Site A is feasible using either dynamic positioning or mooring. Unit costs of the two systems are comparable with thruster unit costs of 40 to 100 \$/kWe and mooring line costs (excluding anchors) in the same range.

- Capital cost for a station keeping system for a 20-MWe plant in Site B is very high. The unit cost (cost/plant net power) of thrusters, for dynamic positioning could be as high as 1000 \$/kWe; mooring line unit cost exceeds 600 \$/kWe excluding the cost of anchors.
- Seawater effluent thrust is attractive for plant sizes above approximately 100 MWe in the more benign environments.
- The exit velocity of the seawater system should not be increased significantly from the baseline value to augment effluent thrust.
- The unit capital cost to generate horizontal force for platform station keeping is nearly constant for thrusters and advanced mooring lines. This approximate cost is 2 \$/N for thrusters and 1.7 \$/N for mooring lines.
- The power required to generate horizontal force using conventional thrusters may be approximated using the relationship, $T/\dot{W}_t = 178 \text{ N/kW}$ where T is the design bollard thrust and $\dot{W}_t$ is the thruster power.
- Existing thruster technology satisfies the requirements for plants to approximately 200 MWe in a benign environment. Combined with effluent thrust for the larger plants, existing thruster technology meets the requirements for the 20 to 1000-MWe range in benign environments.
- OTEC moorings for the demonstration plant size and greater will require an extension of existing mooring technology. This extension could be in the form of advanced state-of-the-art nylon lines or hollow cylindrical links (HCL), an advanced buoyant chain concept. The HCL line is the only candidate satisfying the entire OTEC drag range.

SECTION 4

REQUIREMENTS AND CONSTRAINTS

4.1 Plant/Platform System Considerations

Prior OTEC investigations have led to several well developed plant concepts. The platform configurations associated with these concepts include a circular barge⁽¹⁾, spar⁽²⁾, submersible⁽³⁾, and conventional barge⁽⁴⁾. Following an ERDA recommendation, this study includes five generic platform configurations. Four are generalized versions of the configurations listed above. The fifth is a semisubmersible similar to an offshore oil drilling rig. General platform appearance and specific characteristics required for calculation of drag loads are presented in Section 5.1.

This study considers OTEC plants ranging in power output from 20 to 1000 MWe. Most of the earlier conceptual design work has focused upon the plant power range of 100 to 200 MWe. The following discussion develops scaling techniques which will be used to extrapolate platform characteristics over the entire OTEC range.

Displacement is one of the most fundamental measures of size of a seagoing structure. General relationships can be developed relating dimensions, drag, dynamic response, and cost of similarly shaped vessels to the vessel displacement. Figure 4-1 is a plot relating OTEC plant/platform displacement, Δ , to plant net power, $\dot{W}_{net}$. Data from several earlier investigations are included for comparison.

A horizontal line on this plot of $\Delta/\dot{W}_{net}$ versus $\dot{W}_{net}$ would indicate linear scaling of displacement, i.e., the displacement of a 1000-MWe plant would be ten times that of a 100-MWe plant. This is a conservative assumption. The curves in the figures represent a displacement/plant power relationship assuming some economy of scale. The baseline curve used for this study is defined by the relationship

$$\Delta/\dot{W}_{net} = 3210 (\dot{W}_{net})^{-0.18}$$

Figure 4-1. OTEC Plant Platform Total Operating Displacement

Mass flow rate and exit velocities of the seawater transport system must be defined to perform the analysis of effluent thrust presented in Section 5.3. Values for these variables are taken from the Westinghouse FY 76 study⁽⁶⁾. Cold and warm water volumetric flow rates of $4 \text{ m}^3/(\text{s} \cdot \text{MWe})$ are used. Condenser and evaporator outlet velocities of 2 m/s are assumed with this value reducing to 1 m/s if discharge diffusers are included.

4.2 Site Environment

Numerous geographical sites have been considered for OTEC including waters off the continental United States, Hawaii, and Puerto Rico and open ocean sites in tropical waters. The diversity of sites under consideration has led to development of a baseline design environment for use in this study. Two ERDA-recommended design sites are also included: Site A, which is relatively benign and Site B, a location with a severe environment.

Wind, ocean surface currents, and current variation with depth are the predominant environmental considerations affecting station keeping. A station keeping system must react drag loads resulting from the worst case combination of these environmental variables.

A majority of the proposed OTEC sites are subject to the high wind velocities of tropical storms. All gulf coast sites considered are subject to hurricanes, resulting in a high probability that a gulf-located OTEC plant will experience very high wind velocities in its 40-year life. Figure 4-2 summarizes design wind velocities from a number of sources. The first entry reflects American Bureau of Shipping (ABS) design practice for offshore oil rigs, platforms which operate in several areas proposed for OTEC. Next is, the 52 m/s (101 knots) baseline storm wind used for this study. Values used by various investigators in earlier studies follow. Sites A and B include storm wind velocities of 43 and 45 m/s, respectively. The last three entries include actual storm data for reference only. The Hurricane Eloise data are included as examples of a well documented storm. The data were taken from National Data Buoys 04 and 10 which were in the storm track.

Wind velocity, only, is specified for the design environmental model. Wind heading is assumed to be random and the worst case wind direction will be assumed for drag calculations.

Figure 4-2. Design Wind Velocity

A design ocean surface current, of 2 m/s (3.9 knots), will be used as a baseline value for the present effort. Figure 4-3 shows the relationship among values used by various investigators, Sites A and B, and the Westinghouse baseline. Note that variation in surface current constitutes the major difference between Sites A and B, varying from 1.5 m/s (2.9 knots) to 3.4 m/s (6.5 knots), respectively.

Variation of ocean currents with depth must be specified for OTEC station keeping considerations. The cold water pipe, extending to depths of at least 300 m, is a major drag contributor. The decrease in current with depth will significantly decrease the cold water pipe drag compared with that for constant velocity along the pipe. Actual current/depth profiles vary significantly among various sites. The Westinghouse baseline profile, shown in Figure 4-4, is a compromise among data from a number of sites. Profile assumptions made by other investigators are included in the figure. The use of a dimensionless current/depth profile leads to simple characterization of ocean currents for various sites. Assuming a fixed depth profile, the design surface current is the only current parameter needed to characterize a specific environment.

The environmental design model presented is representative of several OTEC sites and is consistent with the general nature of this study. Drag results presented can be scaled using an appropriate square-law velocity factor for environments of varying characteristics; however, a site specific design should include new calculations based upon empirical environmental data for that site.

Environmental data from several sources have been used throughout the course of this study. Table 4-1 summarizes the pertinent characteristics and sources for these data. Early in this study, ERDA (DOE) recommended consideration of sites A and B as described in the table. These sites and the Westinghouse baseline design environment were used throughout the effort. The DOE environmental package⁽⁹⁾ was received later in the program and was used, primarily, to establish a baseline water depth model for mooring analysis. As seen in Table 4-1, Site B is similar to the Key West site from wind and current considerations and Site A wind and current data are quite close to the Punta Tuna site values.

Figure 4-3. Design Surface Current Velocity

Figure 4-4. Current vs. Depth Design Model

TABLE 4-1. SUMMARY OF OTEC SITE ENVIRONMENTAL DATA

SITE	LOCATION	DEPTH	STORM WIND VELOCITY	SURFACE CURRENT VELOCITY
		Z (m)	V_w (m/s)	V_s (m/s)
A(a.)	Gulf of Mexico (General)	> 3050	43	1.50
B(a.)	Miami, Florida	800	45	3.35
— (9)	Keahole Point, Hawaii	960	34	1.12
— (9)	Punta Tuna, Puerto Rico	1200	48	1.42
— (9)	New Orleans, Louisiana	1200	52	1.28
— (9)	Key West, Florida	1476	59	3.30
— (9)	Brazil	> 4000	31	0.26
Baseline	—	1500	52	2.00

a. ERDA (DOE) letter dated March 1, 1977.

Section 4 — References

1. Douglas, R.H., et. al., "Ocean Thermal Energy Conversion Research on an Engineering Evaluation and Test Program — Final Report, Vol 1-5" TRW Systems Group, NSF/RANN/SE/GI-C958, June 1975.
2. Trimble, L.C., et. al., "Ocean Thermal Energy Conversion (OTEC) Power Plant Technical and Economic Feasibility — Final Report, Vol 1 and 2", Lockheed Missiles and Space Company, Inc., NSF/RANN/SE/GI-C937, April 1975.
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4. Dugger, G.L., et. al., "Tropical Ocean Thermal Power Plants and Potential Products", American Institute of Aeronautics and Astronautics (A1AA) Paper NO. 75-617, April 1975.
5. "Rules for Building and Classing Offshore Mobile Drilling Units", American Bureau of Shipping, 1973.
6. Little, T.E., et. al., "Deep Water Pipe, Pump and Mooring Study, Ocean Thermal Energy Conversion Program", Final Report Westinghouse Electric Corporation, 1976, ERDA/C00-2642-3.
7. Withee, W.G. and Johnson, A., Jr., "Buoy Observations During Hurricane Eloise", Data Report of Environmental Sciences Div., Data Buoy Office, NOAA, 1975.
8. Colon, J.A., "Some Aspects of Hurricane Carla (1961)", Hurricane Symposium, Publication No. 1, American Society for Oceanography, 1966, p.1H.
9. DOE Informal report, "OTEC Demonstration Plant Environmental Package", July 14, 1977, Point of contact: Dr. Loyd Lewis.

10. Barr, R.A. et. al. "Theoretical Evaluations of the Seakeeping Performance and Resistance/ Propulsion Characteristics of Five Candidate OTEC Platforms" Hydronautics, Inc. T.R. 7513-1, ERDA/C00-2681-1, July 1976.

SECTION 5

RESULTS AND DISCUSSION

5.1 Candidate Plant/Platform Configurations and Scaling Techniques

Figure 5-1 provides descriptive data for the five candidate platforms considered in this study. Each platform is described by six coefficients which, used with appropriate scaling relationships, define the platform over the range of OTEC plant displacements.

The displacement coefficient, k_{Δ} is used in the relationship

$$\Delta = k_{\Delta} (\dot{W}_{\text{net}})^{0.82} \text{ (metric ton)} \quad (1)$$

to define plant displacement, Δ , as a function of plant net power, $\dot{W}_{\text{net}}$ (MWe), as discussed in Section 4.1. The baseline plant description uses the value $k_{\Delta} = 3210$ for all configurations. All other scaling relationships will use displacement as the independent variable; however, plant net power may be substituted if desired using Equation (1).

The submerged area coefficient, k_s , relates frontal area of the submerged portion of the hull, A_s , to displacement by:

$$A_s = k_s \Delta^{2/3} \text{ (m}^2\text{)} \quad (2)$$

The value of k_s is chosen for the platform heading giving minimum current drag, e.g., bow to current for the ship configuration. The drag per unit area due to current far exceeds that due to wind for the current-to-wind velocity ratios assumed for this study. Assume that the wind is 90° to the current. If the platform were turned into the wind, the increase in current drag would be much greater than the decrease in wind drag for most configurations.

	Displacement Coef k_{Δ}	Submerged Area Coef, Hull k_s	Submerged Drag Coef, Hull C_{d_s}	Wind Drag Area Coef k_w	Wind Drag Coef C_{d_w}	Draft Coef k_h
Spar	3210	1.7	0.7	0.13	1.3	2.4
Circ Barge	3210	0.95	0.5	0.41	1.3	0.69
Ship	3210	0.46	0.5	0.74	1.3	0.40
Submersible	3210	0.61	0.5	0.18	1.3	0.77
Semi-	3210	1.14	0.55	0.64	1.3	0.75

Cold Water Pipe Drag Coef., $C_{d_p} = 0.7$
Cold Water Pipe Depth = 600 m

Figure 5-1. Characteristics of Candidate Platforms

The proportionality of frontal area and displacement to the 2/3 power is based upon the assumption that all platform dimensions scale by a constant factor, i.e., that displacement scales as the cube of a linear dimension and that area scales as the linear dimension squared or displacement to the 2/3 power.

The submerged hull drag coefficient, C_{d_s} , is an average value based on the total submerged frontal area, A_s , of all hull elements. The total drag force, F_{d_s} , on the submerged hull is:

$$F_{d_s} = \frac{\rho_w}{2} C_{d_s} A_s V_s^2 \text{ (N)} \quad (3)$$

where V_s is the surface current velocity (m/s) and seawater density, $\rho_w = 1023 \text{ kg/m}^3$. A velocity profile correction is included in the drag calculations for platforms with draft exceeding 50 m.

The wind drag area coefficient, k_w , defines the platform area, A_w , exposed to wind and is taken for the axis resulting in maximum A_w . This coefficient relates area to displacement through the equation:

$$A_w = K_w \Delta^{2/3} \text{ (m}^2\text{)} \quad (4)$$

The choice of worst case exposed area results from the assumption of close hull/current alignment. Since the environmental model assumes that no correlation exists between wind and current headings, the worst case wind heading must be assumed.

The wind drag force is:

$$F_{d_w} = \frac{\rho_a}{2} C_{d_w} A_w V_w^2 \text{ (N)} \quad (5)$$

where C_{d_w} is the wind drag coefficient, V_w^2 is the design wind velocity (m/s) and the density of air, $\rho_a = 1.2 \text{ kg/m}^3$. A value of $C_{d_w} = 1.3$ is used for all configurations. Although this value is somewhat conservative, it is that used by the ABS for offshore oil rig design for general studies.

The hull draft coefficient, K_h , relates platform draft, h , and displacement through

$$h = k_h \Delta^{1/3} \text{ (m)}. \quad (6)$$

A cold water pipe maximum depth of 600 meters is assumed for all configurations. A cold water pipe drag coefficient, C_{d_p} , of 0.7 is assumed. The cold water pipe diameter, $d(m)$, is $1.59 \times (\dot{W}_{net})^{1/2}$ where $\dot{W}_{net}$ is the plant net power in MWe.

5.2 Platform Drag Analysis

Three major drag components are combined to give the total drag force on the OTEC platform. These are: cold water pipe drag, current drag on the submerged hull, and wind drag on the platform.

Since the relative headings of these three forces cannot be predicted for a general site, the total drag force is assumed to be the worst case combination of the three. The three drag components are calculated separately and these magnitudes added as scalars to obtain the total drag force.

Figure 5-2 is a plot of total drag force vs. displacement for the baseline environment. The data band represents the spread of drag forces due to platform configuration with the semisubmersible yielding maximum drag and the spar yielding minimum drag. The other configurations fall between these extremes. The plant net power labels are located using the baseline relationship between plant power and displacement. The Hughes Mining Barge (HMB) displacement of approximately 9000 metric tons is plotted for reference. The extension of the curves below the 20-MWe plant displacement is based on the platform models for this study, not specific calculations for the HMB. Figure 5-3 presents similar drag data for Sites A and B. Site B drag exceeds Site A drag by a factor ranging from 2.5 for the 1000-MWe semisubmersible to 4.3 for the 20-MWe spar.

Figure 5-4 is a plot of total drag force vs. plant net power for the baseline environment including data for all configurations. Figure 5-5 extends the data to include the upper drag limit for Site B and the lower limit for Site A. The general relationship among configurations holds for the three environments. The dark vertical band for the 20-MWe plant size represents the spread of drag forces for all configurations in site B. The 100-MWe vertical band is the spread for this size at site A. As in the earlier plots, the severity of Site B is apparent.

Figure 5-2. Drag Force vs. Vessel Displacement for Baseline Environment

Figure 5-3. Drag Force vs. Displacement

Figure 5-4. Plant Drag vs. Plant Net Power Baseline Environment

Figure 5-5. Plant Drag vs. Plant Net Power

The relative contributions of wind, submerged hull, and cold water pipe drags are shown in Figure 5-6. Each vertical bar represents 100 percent of the appropriate total drag force. This plot is useful in assessing the sensitivity of total drag to a variation in either environmental or configuration parameters. For example, the baseline submerged hull drag coefficient for the ship configuration is 0.5. Consider the effect of reducing this value to 0.25 for a 100-MWe plant. From Figure 5-6 the ship hull drag constitutes about 15 percent of the total plant drag. Decreasing the drag coefficient by a factor of 2 would decrease the hull contribution by the same factor yielding a 7.5 percent reduction in total drag. For the same configuration, doubling the surface current velocity would increase the hull drag by a factor of 4 (drag is proportional to the square of velocity) yielding a total drag increase of approximately 45 percent. Refer to the discussion of scaling techniques in Section 5.1 for functional relationships required to assess changes in other parameters.

5.3 Seawater Effluent Propulsion

As suggested by several earlier investigators, the large horizontal momentum changes associated with the OTEC seawater transport systems generate significant forces (thrust) which could be used for station keeping purposes. The seawater transport systems are similar in construction to large water jet propulsion units; however, the implementation and design philosophy of the pumping system must be modified to generate useful thrust for some OTEC applications. Efficiency considerations for an ideal seawater transport system, designed for support of the OTEC plant cycle only, lead to use of a final diffuser (see Figure 5-7) to reduce the effluent velocity to the minimum value practical⁽¹⁾. Since effluent thrust is directly proportional to effluent velocity, the best design, for pumping only, leads to minimum thrust capability. Two options are available to increase thrust of the baseline pumping system. Effluent thrust, T , is proportional to the product of mass flow rate, $\dot{m}$, and the increase in horizontal water velocity or, for a zero inlet velocity and effluent velocity, V_9 :

$$T = \dot{m}V_9$$

This expression is valid for the cold water system since the horizontal component of the intake velocity will be very small.

Figure 5-6. Relative Drag Contributions - Baseline Environment

Baseline Model

Effluent Thrust Model

Figure 5-7. Effluent Thrust - Hydraulic Model

The mass flow rate will be considered constant for two reasons:

1. Variations in $\dot{m}$ would affect the thermodynamic cycle resulting in interaction between thrust requirements and the power cycle.
2. An increase in velocity throughout the system would result in undesirable system losses.

Effluent thrust augmentation is best achieved by increasing the effluent velocity. This is accomplished by decreasing the exhaust orifice area, as shown in Figure 5-7, which increases the pressures at points 8, 7, and 6 in the figure. An increased pump head results. This increase in head is plotted against nozzle velocity in Figure 5-8. Pump power (proportional to pump head) is proportional to the square of the nozzle velocity while thrust is directly proportional to the velocity. This relationship is important in comparing effluent thrust with conventional external thrusters which yield thrust approximately proportional to power input.

Figure 5-9 is a plot of effluent exit velocity, V_9 , versus thrust to plant net power ratio, $T/\dot{W}_{net}$. Data are given for systems using cold water only and a combination of cold and warm water effluent. Calculations for combined cold and warm water assume that the horizontal component of the warm water inlet velocity is zero. All calculations assume that cold and warm water flow rates, $\dot{m}$, are each $4 \text{ m}^3/(\text{s} \cdot \text{MWe})$. The baseline exit velocity of 2 m/s is indicated on the plot.

Figure 5-10 shows the relationship between additional pump power required to generate thrust and thrust generated. Note that some thrust is available at no power penalty since losses for the baseline exit velocity are attributed to the plant cycle, i.e., even the ideal seawater system generates some thrust.

Consider, for example, dynamic positioning a 100-MWe plant in the baseline environment using effluent thrust. Use a nominal drag force of 6 MN (from Figure 5-2 or 5-4). The thrust to power ratio is 60 kN/MWe giving an additional pump power ratio of 0.46 for cold water only or 0.23 for cold plus warm water. This means that 46 percent or 23 percent, respectively, of the net plant output power would be required to generate sufficient thrust to hold the plant against the assumed storm drag load. Furthermore, the seawater pumps would have to be sized

Figure 5-8. Nozzle Head Loss

Figure 5-9. Effluent Exit Velocity vs. Thrust

Figure 5-10. Power for Effluent Thrust

to absorb this additional power. Assuming a baseline cold water pumping power of 11 percent⁽¹⁾, the system using cold water only would require a pump with 5 times the power capability of the baseline design.

This effluent thrust data should be used with care. It is based upon assumptions of zero inlet velocity and ideal pump and convergent nozzle performance (100% efficiency). Marked reductions in this performance, especially for off design pump operation, are likely in actual applications. The curves represent ideal effluent thrust performance.

5.4 Auxiliary Thruster Propulsion

Conventional thrust producing devices applicable to dynamic positioning of the OTEC platform may be divided into six general categories as depicted in Figure 5-11. Application of these configurations depends upon performance goals, thrust vector control requirements, and hull configuration. The study will examine the thruster candidates for similarities and differences, developing a general thruster model based upon the similarities. This model will present thruster performance and cost in general terms of thrust to power ratio, $T/\dot{W}_t$ (N/kW) and cost to thrust ratio, C/T (\$/N) without reference to the specific requirements of the OTEC platform. These data will be used to evaluate specific OTEC station keeping system concepts in Section 6 of this report.

The following list of thruster configurations includes a brief discussion of the normal applications and attributes of each. (Refer to Figure 5-11.)

- **Fixed Screw**

This configuration is normally used to propel conventional hull shapes and will be defined as any layout including a straight shaft connecting an external propeller to an internal prime mover. The direction of thrust is fixed with respect to the hull (unless modified by a rudder or steering nozzle) and represents the simplest of the six configurations. Installations absorbing above 52000 kW (70000 hp) per screw are feasible with 4 screw systems (about 280,000 hp) in service.

FIXED SCREW

TUNNEL

RIGHT ANGLE DRIVE, EXTERNAL SCREW

CYCLOIDAL

WATER JET

POD

Figure 5-11. Thruster Configurations

- **Tunnel Thruster**

Any thruster including a propeller located in a straight duct mounted entirely inside the hull will be classed as a tunnel thruster. These units are used to generate athwartships thrust in combination with fixed screw main propulsion, a combination widely used in dynamically positioned drill ships. This combined configuration results in relatively low hull drag for transit operation and good directional control at low speed; however, the vessel must hold a heading near the vector sum of the drag forces to efficiently utilize the main propulsion system. Most tunnel thruster configurations utilize a right angle drive to transmit power to the screw with minimum flow interference. Tunnel thruster installations to about 2240 kW (3000 hp) are within the state-of-the-art.

- **Right Angle Drive, External Screw**

These thrusters may be configured as shown in the figure with a single right angle drive and vertical prime mover or may include a second right angle unit, inside the hull, to allow horizontal prime mover mounting. The propeller may be fixed in heading or may rotate. This latter, azimuthing configuration yields excellent thrust vector control with thrust heading change rates limited by the mass and hydrodynamic loads of only the outboard unit, not the vessel as with a fixed, main screw system. Totally self-contained right angle drive thrusters are becoming a standard building block for oil-rig dynamic positioning systems offering operational flexibility and ease of maintenance. Units using right angle bevel gears as a final drive are currently limited to about 2240 kW (3000 hp). Some extension of this value appears feasible within the OTEC demonstration plant time frame.

- **Cycloidal Propeller**

The cycloidal propeller is characterized by a thrust vector perpendicular to the axis of rotation. Cyclic variation of blade angle of attack yields a trainable thrust vector which can assume any heading. Changes in thrust azimuth can be very rapid, a characteristic highly desirable for tugs, ferries, and similar vessels. The degree of thrust control exceeds that of other designs significantly, exceeding most requirements for open water dynamic positioning.

- Water Jet Propulsion

These systems, characterized by internal screws mounted in a converging duct, are normally used to deliver thrust along the vessel longitudinal axis. Water jets are most widely used in draft limited, and high speed craft with the external geometry justifying the associated ducting losses. Individual designs for high power applications are normally unique to the vessel and little standard hardware is available. Theoretically, very high power units could be built (the OTEC effluent thrust concept, for example). Thrust vector heading control must be accomplished at the nozzle and effects upon performance can be defined for specific designs only.

- Pod Thruster

Although this concept has been proposed for offshore use little data is available. The concept reduces the power transmission constraints imposed on right angle drive units by providing straight shafting from prime mover to propeller. Azimuthing units could be built; however, hydrodynamic loads for a unit rotated from the prevailing current axis would be high. The concept offers the potential to extend azimuthing thruster power capability while maintaining good thrust vector control and could be considered if power levels higher than available from conventional azimuthing units were required.

These configurations are diverse geometrically, but there are certain similarities in performance which allow developing a single model for general thruster analysis. The objective of this model will be to make approximate power, size and cost estimate for OTEC dynamic positioning analysis. Thrusters used in dynamic positioning systems normally operate at or near bollard conditions, i.e., the platform speed through the water (or water speed past a fixed platform) is approximately zero. Thruster performance at bollard pull may be described by an expression relating thrust and power:

$$T/\dot{W}_t = C (\dot{W}_t/A)^{-1/3}$$

or an equivalent relationship. In this expression, T is thrust in Newtons, $\dot{W}_t$ the input power in kilowatts, A is the propeller disc area in square meters and C has units of $(\text{kg}/\text{m}^3)^{1/3}$. The exact form of the constant (often called a Merit Coefficient) varies among authors but the basic

relationship between $T/\dot{W}_t$ and $\dot{W}_t/A$ is common. A similar treatment⁽²⁾ reports ranges of performance for various configurations resulting in

- $C = 1156 \text{ to } 1203 \text{ (kg/m}^3\text{)}^{1/3}$ for ducted propellers (Kort nozzle)
- $C = 915 \text{ to } 947 \text{ (kg/m}^3\text{)}^{1/3}$ for nonducted propellers

These values are for thrusters which are designed for a predominant thrust direction and are operating in that design direction. Based upon the consideration of thrust to power ratio only, a well-designed, ducted (Kort nozzle) thruster will always exceed the performance of an open propeller under bollard conditions. The difference in performance narrows with increasing speed of advance but the Kort nozzle unit will show some improvement for the entire range of OTEC design currents. Thruster performance reduces as speed of advance increases. Typical units, designed for bollard conditions, derate approximately 8-12 percent per meter per second of advance velocity.

The results of a survey of thrusters typical of those used in the offshore oil industry are included in Table 5-1. Figure 5-12 is a plot of thrust to power ratio versus propeller loading for the thrusters surveyed.

Although there is data spread resulting from variations in configuration the general correlation between thrust to power ratio and propeller loading ratio is clear. The following analysis will assume a baseline Kort nozzle, right angle drive thruster with $T/\dot{W}_t = 178 \text{ N/kW}$ and $\dot{W}_t/A = 200 \text{ kW/m}^2$; however, other values or configurations consistent with the general data could be used by the reader. The baseline model is clearly within the state-of-the-art. The configuration chosen could be applied to any platform. The lower than average propeller loading reflects relaxed geometrical constraints anticipated for the very large OTEC platform. The lower than average value of $C = 1040$ reflects potential increase in shaft seal losses for increased immersion depths for the OTEC application.

Note that the relatively low thrust to power ratios of the two low speed water jets included on the plot appear to result from high disc loading and, assuming cavitation criteria were not exceeded, extension to lightly loaded, higher thrust to power ratio jets seems reasonable. The tunnel thruster is similar to the water jet in configuration and lies on a line through the water jet points.

TABLE 5-1. THRUSTER SURVEY

Manufacturer	Description ⁽¹⁾	Input Power $\dot{W}_t$ (kW)	Propeller Diameter D (m)	Propeller Speed N (RPM)	Ballard Thrust T (kN)	Cost Estimate C (k\$)	Weight W (kN)	Remarks ⁽²⁾
1a. Baylor	R, A, CP, N	2238	3.05	180	382	950	1000	(c) (d)
1b.	↓	↓	3.4		467	↓	1000	(c) (e)
2.	↓	1343	3.05	190	285	650	890	(c) (d)
3.	↓	1120	3.05	165	165	550	800	(c) (d)
4. Bird-Johnson	R, A, CP, N	1120	2.42	259	157	175	187	(a) (d)
5.	↓	1679	2.83	205	231	250	294	(a) (d)
6a.	↓	1939	3.34	—	363	325	480	(a) (d)
6b.	↓	2611	↓	180	460	325	480	(a) (d)
6c.	↓	↓	↓	↓	↓	850	—	(c) (d)
7. Schottel	R, A, FP	858	2.30	205	165	300	130	(b) (d)
8.	↓	1678	2.80	244	310	530	378	(b) (d)
9.	↓	3357	3.50	202	623	590	400	(a) (d)
10.	↓	4625	4.20	150	800	1570	390	(a) (e)
11. Bird-Johnson	T, CP	448	1.65	325	66	75	51	

TABLE 5-1. THRUSTER SURVEY (Continued)

Manufacturer	Description ⁽¹⁾	Input Power $\dot{W}_t$ (kW)	Propeller Diameter D (m)	Propeller Speed N (RPM)	Ballard Thrust T (kN)	Cost Estimate C (k\$)	Weight W (kN)	Remarks ⁽²⁾
12. Bird-Johnson	T, CP	746	2.01	295	111	100	82	
13. ↓	↓	1119	2.41	240	167	150	127	
14. ↓	↓	2238			334	300		

1. R: Right Angle Drive; Z: Z-drive; A: Azimuthing; FP: Fixed Pitch; CP: Controllable Pitch; N: Kort Nozzle; T: Tunnel
2. a) Bare drive, no motor; b) Encapsulated, no motor; c) encapsulated, complete; d) data for existing unit; e) estimate

Note: Numbers at data points refer to entries in Table 5-1.

Figure 5-12. Performance of Surveyed Thrusters at Bollard Pull

Propeller diameter for the baseline thruster model may be calculated using:

$$D = 0.08 \sqrt{\dot{W}_t}$$

where D is the diameter in meters and $\dot{W}_t$ is thruster power in kilowatts. Figure 5-13 is a plot of this baseline diameter vs. power for thrusters to 6000 kW. These values may be used as a rough guide for layout purposes. If diameters significantly different from these are used, thruster performance should be re-evaluated using Figure 5-12.

Figure 5-14 is a plot of propeller diameter versus power to (speed)³ ratio for the thrusters surveyed. The data can be used for estimating one unknown of the three variables.

Figures 5-15 through 5-17 include survey data of thruster weight, thrust, and cost plotted against thruster power. Note that the amount of ancillary equipment included with the surveyed units varies considerably (see notes in Table 5-1). The support equipment for a self-contained right angle thruster, including prime mover, pressure proof housing, auxiliary motors, hydraulics, foundations, and controls constitute a significant weight and cost contribution. The baseline thruster model is a complete unit with all of these auxiliaries. Figure 5-15 presents thruster weight per unit thrust versus power for the right angle drive units surveyed. Auxiliaries can contribute approximately 50% of the total unit weight. Figure 5-16 indicates that bare thrusters and auxiliaries share approximately equal cost per unit weight in the range up to about 2500 kW. The very limited samples indicate that this variable may increase rapidly beyond this standard industry range.

Figure 5-17 shows that cost per unit thrust is nearly constant for a given configuration. As with weight, the inclusion of auxiliaries increases the base thruster price by a factor of more than two. A fixed cost of 2 \$/N is selected as the baseline. A cost near the bottom of the complete unit range is used to reflect the advantage of building a reasonable quantity of standard OTEC units. The cost of thrusters for a single demonstration plant could be higher, but the baseline value should not be exceeded by more than 25%.

Figure 5-13. Thruster Propeller Diameter vs. Power for Baseline Model

Figure 5-14: Thruster Diameter vs. Power/RPM³ Survey Data

Figure 5-15. Thruster Survey Weight Per Unit Thrust

Figure 5-16. Thruster Survey Cost Per Unit Weight

Figure 5-17. Cost of Right Angle Drive Thrusters

5.5 Deep Water Mooring

This section identifies requirements and constraints for OTEC moorings and describes hardware capable of meeting these needs. The hardware descriptions include performance and cost data, independent of a specific OTEC configuration. In Section 6, these data are combined with OTEC drag data to predict OTEC station keeping system performance and cost.

One of the most demanding requirement for an OTEC mooring is water depth. The FY76 Westinghouse mooring study⁽¹⁾ considered site water depths to 6000 meters. Site A water depth is greater than 3050 meters. The most current ERDA environmental report⁽³⁾ specifies water depths for prime site candidates as:

● Keahole Point, Hawaii	Z = 960 m
● Punta Tuno, Puerto Rico	Z = 1200 m
● New Orleans (Gulf Site)	Z = 1200 m
● Key West, Florida	Z = 1476 m

In light of this latest data, 1500 meters is used as a baseline design depth for an OTEC mooring.

Although this section will not address specific OTEC configurations, the range of horizontal forces to be reacted by the mooring must be considered. The upper limit of drag on the largest OTEC platform in the most severe environment is approximately 51 MN. This value will be used as an upper bound for initial discussions.

OTEC plants delivering electrical power to an on-shore grid will require an electrical riser cable connecting the plant to a mid-water column or bottom distribution point. This riser is currently under study by others and no data are available to relate riser mechanical constraints and platform station keeping requirements. ERDA has suggested a watch circle design goal of 5% of water depth for current study purposes. Watch circle is defined as the horizontal platform excursion from a neutral (no load) position under the design environmental load, i.e., the platform horizontal excursion is limited to a circle, centered at the neutral position, with a radius

equal to 5% of the water depth. This is the requirement commonly used by the offshore oil industry for drill rig positioning during drilling operations in normal weather conditions. It is found practical for the hardware used in that industry; however, drilling operations are normally stopped during severe storms and the station keeping requirements relaxed to a larger, survival watch circle. Assuming that the OTEC watch circle requirement holds for storm conditions, this OTEC requirement is significantly more stringent than that of the oil industry. Data will be presented for watch circles from 5 to 30% to allow evaluation of the consequences of this requirement.

In summary, the baseline OTEC mooring system requirements are:

- Design depth 1500 meters
- Horizontal force to 51 MN
- Watch circle radius 5% of water depth

The FY76 Westinghouse study⁽¹⁾ evaluated state-of-the-art mooring lines and an advanced line concept, hollow cylindrical links (HCL). Tables 5-2 and 5-3, taken from that study, summarize characteristics of these lines. Figure 5-18 is a sketch of the HCL concept. The earlier study concluded that the most promising mooring line candidates for OTEC moorings are a two-component nylon/chain line and the advanced HCL design. The advanced line is the only candidate which satisfies the entire OTEC drag range requirement. Both will be analyzed in this study.

A study of deep water anchors was conducted by the Civil Engineering Laboratory (CEL) concurrently with the Westinghouse studies.⁽⁴⁾ This study does not include anchor analysis. Data reflects mooring line performance only. Vertical forces at the anchor have been included to provide inputs for a complete mooring system analysis.

Two computer routines were used to generate the mooring data presented in this section. The first is an iterative solution of two component mooring line catenary equations based upon Appendix C of Reference (5). This is a static solution considering only end point line loads and line weight. The slight improvement in accuracy gained from considering line drag forces is

TABLE 5-2. MOORING LINE PHYSICAL PROPERTIES

Material	Configuration	Size		Static Breaking Strength T_{max} (MN)	Weight In Air Wt (N/m)	Weight In Water [Wt- Δ] [*] (N/m)	Data Source
Steel	Chain (ORQ)	15.2 cm dia		16	5251	4570	Ramnas Bruk
Steel (alloy)	Chain (Super)	12.7 cm dia		15	3682	3200	Ramnas Bruk
Steel	Wire Rope	6.4 cm dia		1.7	126	110	U.S. Steel
Synthetic Fiber	Nylon Braid	76.2 cm circ		10	389	48	Sampson
Steel (HY80)	Design Depth	<u>Diameter</u>	<u>Wall Thickness</u>				
Hollow Cylindrical	(m)	(m)	(cm)				
Links, L/D = 10	1500	0.520	1.72	15	2595	714	Westinghouse
	3000	0.466	1.93			908	Conceptual
	6000	0.337	2.80			1297	Design
	1500	1.62	5.35	146	25,261	6947	
	3000	1.46	6.06			8841	
	6000	1.05	8.71			12630	
	1500	2.84	9.37	445	76,996	21174	
	3000	2.54	10.54			26949	
	6000	1.84	15.27			38498	

^{*} Δ = seawater displacement (buoyant force)

TABLE 5-3. MOORING LINE COST DATA, BASIS FOR COST ESTIMATE
(MID-1976 DOLLARS)

Line Type	Size	Static Breaking Strength $T_{\max}$ (MN)	Line Cost	
			$\left(\frac{\$}{\text{m}}\right)$	$\left(\frac{\$}{\text{metric ton}}\right)$
Chain (Super)	12.7 cm dia	15	512 ^(a)	1360
Nylon Braid	76.2 cm circ	10	350 ^(b)	8820
Hollow Cylindrical Links (HY 80 Steel)	L/D = 10 ↓	15 146 445		3400 ^(c) ↓

(a) Swedish Chain Sales Corp., W. Caldwell, N. J.

(b) Sampson Ocean Systems, Inc., Boston, Mass.

(c) Material Cost + Construction Cost (Westinghouse estimate)

Material: HY-80 Steel

Figure 5-18. Hollow Cylindrical Link (HCL) Concept

not warranted in a general study of this nature. General results will be valid and the more complex analysis should be used only for detailed line design. The two-component solution is used for both the nylon/chain and the HCL lines.

The second analysis solves the general equations for a multi-point mooring with an arbitrary number of legs. The solution uses catenary data generated by the first routine and is iterated to obtain a solution for the desired watch circle.

Figure 5-19 includes the major variables used to define mooring configurations. For a single point mooring, excursion is measured from the anchor. Excursion is measured from the platform position with zero horizontal force for the multi-point configuration. The watch circle requirement, used only in the multi-point case, is applied to the most compliant axis, i.e. along the bisector of the angle included between two legs. This constitutes the most severe watch circle definition. Minimum horizontal load capability of a multi-point system occurs for a net force in line with one leg. This is the direction assumed for load data.

Scope is defined as the dimensionless ratio of total line length, ℓ to the depth, Z . Horizontal excursion, X , is given for all configurations and the dimensionless ratio of excursion to depth, X/Z is given for the multi-point configurations. This variable corresponds to watch circle radius as a fraction of depth.

Figure 5-20 shows the correlation between the mooring performance data presentation in this study and physical configurations of the mooring system. A brief discussion of the loading sequence as found in the figure follows.

① With no horizontal force on the platform, the maximum line tension is equal to the suspended line weight in water. For all lines which do not float, the maximum tension occurs at the platform connection. Horizontal force on the platform increases from ① to ④ or ⑤. Vertical force on the anchor is zero until the line angle at the anchor exceeds zero at ② and increases with increasing horizontal force, until either breaking strength is exceeded ④ or the anchor drags ⑤.

● Multi-Point

Figure 5-19. Characteristics of Passive Moorings

Figure 5-20. Mooring Performance

One of the most significant issues in mooring design is the choice of design criteria. Although most practical design criteria are simple in form, they are based upon many factors including environmental loads, confidence in line mechanical characteristics and their degradation with time, operational constraints, consequences of a failure, and cost considerations. Current offshore oil rig practice is to design for a maximum ratio of line tension to breaking strength, $T/T_{\max}$, of $1/3$ for normal operating conditions and to limit $T/T_{\max}$ to $1/2$ for storm conditions. These criteria will be used for this study. Note that oil rig moorings are relatively temporary systems. The long term OTEC application could lead to greater margins; however, a choice based upon actual practice is a reasonable first assumption. A more detailed analysis will be justified when a specific hardware choice is made.

The analysis in this section will concentrate upon establishing upper limits of capability for the two line candidates. Specific OTEC applications will be given in Section 6.

Table 5-4 and Figure 5-21 include performance data for a 2-component nylon/chain mooring line using the 76.2-cm nylon rope and 12.7-cm chain listed in Table 5-2. Data are given for a water depth, of 1500 m. The scope of the nylon is 0.95 i.e., the length is 95% of the depth, to prevent bottom abrasion with a slack system. The total scope is 1.5, a value chosen to give a reasonable compromise between horizontal force capability, F_{HP} , and cost. This configuration leads to $F_{HP} = 3.4$ MN at a line tension ratio of 0.50, the design load for storm conditions. The vertical force on the anchor is zero under these conditions. The horizontal excursion includes line stretch, a significant factor for nylon line.

It is evident from Figure 5-21 that the performance of the nylon/chain combination is approximately the same for the scope of 1.5 used here and the scope of 2.2 used in the FY76 report.⁽¹⁾ The cost advantage of the lower scope is significant. The total line cost is 0.92 M\$ for a scope of 1.5 and 1.46 M\$ for a scope of 2.2.

Multi-point moorings may be divided into two major categories, namely, passive and active systems. A passive system uses mooring lines of fixed length, pretensioned at deployment to meet the design watch circle requirements. An active system incorporates mooring line winches, capable of adjusting line tension during operation. The active system provides the

TABLE 5-4. PERFORMANCE SUMMARY OF NYLON/CHAIN MOORING LINE

- Single Leg, 2-Component Line
- 76-cm Nylon
- 12.7-cm Super Chain

Depth Z (m)	Design Scope L/Z (d'less) ^a	Horizontal Excursion X (m)	Horiz Force On Platform F_{HP} (MN)	Vert Force On Platform F_{VP} (MN)	Line Tension Ratio T/T_{Max} (d'less)	Line Angle At Anchor θ_1 (deg)	Vert Force On Anchor F_{VA} (MN)
1500	1.5	1200	0.12	0.55	0.06	0	0
↓	↓	1425	0.84	1.51	0.17	↓	↓
		1852	2.15	2.71	0.35		
		1950	3.40	3.70	0.50		
		2055	7.14	7.00	1.00	31	0.06

a. Dimensionless

Figure 5-21. Performance of Single Leg Nylon/Chain Line

highest horizontal force capability. The windward legs of a passive system must react both the environmental loads and the pretension of the leeward legs. The leeward legs of an active system may be relaxed, with windward legs reacting only the environmental loads. The price paid for this increased performance is the cost of winches, extra mooring line, outboard sheaves, line stowage volume, power and control hardware. These additional costs can exceed the line cost for water depths in the range considered.

Figure 5-22 demonstrates the improved performance of an active system using the nylon/chain combination. The improvement is quite sensitive to platform excursion constraints as seen in the figure. This pronounced effect for lines including nylon results from the high elasticity of that material. Stiffer lines show less sensitivity. The ratio of line cost to horizontal force capability is shown for several points on the passive system curves and the ratio of line plus winch cost to force is shown for the active system. A winch cost of 0.5 M\$* per leg is assumed for the 76-cm nylon as well as a 25% increase in length for the active system lines. Although the total cost of the active system will include additional factors not considered here, the advantage of the active system for elastic lines is clear.

The multi-point mooring solution used assumes that lines are equally spaced in azimuth. Under these conditions, and assuming that an 8-leg system is a reasonable limit, the maximum capability of the 1500-meter depth, 2-component nylon/chain mooring is approximately 8 MN assuming a line tension ratio of 0.5 at that horizontal force. An increased number of equally spaced legs or multiple strand legs could be considered; however, the increased system complexity should be weighed carefully in these cases.

Table 5-5 and Figure 5-23 include the performance of a single, 146-MN breaking strength hollow cylindrical link (HCL) line constrained to give a zero line angle at the anchor at $T/T_{\text{Max}} = 0.33$. A scope of 2.95 is required to meet this constraint which results in no vertical force at the anchor under most operating conditions. Only a small vertical component (relative to the horizontal force capability) occurs at the storm load. Table 5-6 and Figure 5-24 include similar data for a line with a scope of 1.75. This scope was chosen as a reasonable compromise

* Samson Ocean Systems, Inc., Boston Mass.

Figure 5-22. Performance of Nylon/Chain Multi-Point Mooring

TABLE 5-5. PERFORMANCE OF HOLLOW CYLINDRICAL LINK LIMIT CONCEPT

- HCL
- $T_{Max} = 146 \text{ MN}$
- $\theta_1 = 0 @ T/T_{Max} = 0.33$
- Line Cost = 39 M\$
- Line Cost Ratio = 0.62 \$/N
at $T/T_{Max} = 0.50$

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Depth Z (m)	Design Scope (l/Z) Design (d'less)	Horizontal Displacement X (m)	Horiz Force On Platform F_{HP} (MN)	Vert Force On Platform F_{VP} (MN)	Line Tension Ratio T/T_{Max} (d'less)	Line Angle At Anchor θ_1 (deg)	Vert Force On Anchor F_{VA} (MN)
1500	2.95	3429	2.2	11	0.08	0	0
		3900	15	19	0.17		
		4020	27	24	0.25		
		4080	39	29	0.33		
		4130	63	37	0.50	7.5	7.1
		4155	132	62	1.00	14	28

Figure 5-23. Performance of Hollow Cylindrical Link, Limit Concept

TABLE 5-6. PERFORMANCE OF HOLLOW CYLINDRICAL LINK
REDUCED COST CONCEPT

- HCL
- $T_{Max} = 146 \text{ MN}$
- Vertical Force at Anchor = 26 MN
@ $T/T_{Max} = 0.50$
- Line Cost = 23 M\$
- Line Cost Ratio = 0.41 \$/N
@ $T/T_{Max} = 0.50$

Depth Z (m)	Design Scope (l/Z) Design (d'less)	Horizontal Displacement X (m)	Horiz Force On Platform F_{HP} (MN)	Vert Force On Platform F_{VP} (MN)	Line Tension Ratio T/T_{Max} (d'less)	Line Angle At Anchor θ_1 (deg)	Vert Force On Anchor F_{VA} (MN)
1500 ↓	1.75 ↓	2036	11	17	0.14	2.5	0.4
		2098	17	21	0.18	13	3.4
		2127	24	26	0.24	20	7.8
		2141	36	33	0.33	25	14
		2149	56	47	0.50	29	26
		2153	116	89	1.00	32	62

Figure 5-24. Performance of Hollow Cylindrical Link Reduced Cost Concept

between cost and vertical force on the anchor. The line cost ratio, C/F_{HP} at $T/T_{Max} = 0.50$, is 0.62 \$/N for the 2.95 scope and 0.41 \$/N for the 1.75 scope. Figure 5-25 summarizes the dependence of F_{HP} , cost, and F_{VA} upon scope. Given an anchor vertical force limit for example, a cost effective 146-MN HCL configuration can be selected from these data.

The 146-MN HCL configuration results in horizontal force capability consistent with the upper limit of the required OTEC drag load range. The dimensions of these links (see Table 5-2) are reasonable from the standpoint of current fabrication practices. The 146-MN HCL will be chosen as a limit mooring line design and will be used in the following discussion to generate a limit configuration for an OTEC multi-point mooring.

Table 5-7 and Figure 5-26 include performance data for a 3-point mooring in 1500-meter water using the 146-MN HCL. The cost effective scope of 1.75 is used. The 16-MN reduction in horizontal force capability at storm loads (as compared with a single leg mooring) results from line pretension which is 20% of T_{Max} . An inelastic line is assumed with the catenary providing mooring compliance. Note the significant improvement in performance of this passive configuration compared with the passive nylon line. The increased cost ratio of the 3-point configuration (1.72 \$/N) versus the single leg (0.41 \$/N) results from the added number of lines and the decrease in performance due to pretension.

Table 5-8 and Figure 5-27 show the dependence of multi-point HCL mooring performance and cost as a function of the number of legs. The 3-leg configuration is the most cost effective. Increasing the allowable horizontal force for a 146-MN HCL mooring from 40 MN to 80 MN requires an increase from 3 to 10 legs with a resultant cost ratio increase from 1.72 \$/N to 2.80 \$/N. Assuming that the 40-MN horizontal force capability were retained, a 73-MN HCL could be used with a 10-leg system but the line cost would be 62% higher than with the 3-leg system. The advantage of the 10-leg system would be a 50% reduction in vertical force at the anchor.

Figure 5-25. Hollow Cylindrical Link — Performance vs Scope

TABLE 5-7. PERFORMANCE OF MULTI-POINT MOORING LIMIT CONCEPT

- 3-Point Moor
- HCL $T_{Max} = 146$ MN
- Design Excursion: 5% @ $T/T_{Max} = 0.50$
- Line Cost = 69 M\$
- Cost Ratio = 1.72 \$/N @ $T/T_{Max} = 0.50$

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Depth Z (m)	Design Scope (L/Z) Design (d'less)	Horizontal Excursion X_{Max} (m)	X/Z (d'less)	Horiz Force On Platform F_{HP} (MN)	Total Vert Force On Platform F_{VP} (MN)	Line Tension Ratio T/T_{Max} (d'less)	Line Angle At Anchor θ_1 (deg)	Vert Force On Anchor F_{VA} (MN)
1500 ↓	1.75 ↓	0	0.0	0	66	0.20	16	5.0
		33	0.022	6.8	73	0.24	20	7.8
		60	0.040	19	85	0.33	25	14
		75	0.050	40	111	0.50	29	26
		82	0.055	100	194	1.00	32	62

Figure 5-26. Performance of Multi-Point Mooring Limit Concept

**TABLE 5-8. MULTI-POINT MOORING - PERFORMANCE VS NUMBER OF LEGS
(146 MN HCL)**

Depth Z (m)	Scope L/Z (d'less)	No. of Legs N (d'less)	Excursion (X/Z) $T/T_{Max} = 0.5$ (d'less)	Horiz Force On Platform F_{HP} (MN)	Line Tension Ratio T/T_{Max} (d'less)	Line Cost (M\$)	Line Cost Ratio (\$/N)
1500 ↓	1.75 ↓	3 4 6 8 12	0.05 ↓	40 45 54 68 96	0.50 ↓	69 92 138 184 276	1.72 2.04 2.56 2.70 2.88

Figure 5-27. Performance of Multi-Point Mooring vs Number of Legs

Section 5 — References

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SECTION 6

ANALYSIS AND EVALUATION

6.1 Performance and Cost of OTEC Station Keeping Candidates

Section 5 of this report develops general data for station keeping analysis using effluent thrust, auxiliary thrusters and deep water moorings.

Effluent thrust is a concept peculiar to the OTEC application. Thrusters and moorings are applicable to any deep ocean platform. These concepts may be used individually, or in combination to form station keeping systems. Three generic station keeping concepts were evaluated for the OTEC application, namely:

- Dynamic Positioning
- Mooring
- Combined Dynamic Positioning and Mooring.

The first candidate results in a free platform from the standpoint of the station keeping system (see Figure 6-1). The second and third candidates include permanent bottom connections and are designated fixed platform concepts. The first concept could be employed with grazing operation of the OTEC plant or with a bottom connected riser cable. Some degree of grazing could be incorporated into a combined station keeping concept. A grazing circle diameter of 10,000 meters (about 6 statute miles) is reasonable, using a single leg mooring. This division into station keeping concepts does not, then, force a certain OTEC power utilization concept and is generally applicable to the OTEC plant.

In this section, three OTEC station keeping system candidates are developed, each corresponding to one of the three major concepts. The candidates are evaluated in terms of their ability to meet OTEC station keeping force and watch circle requirements. The power penalty of the dynamic positioning system is determined for storm conditions and the capital cost of the systems is estimated.

Figure 6-1.. Station Keeping Concepts

Dynamic Positioning

Dynamic positioning of the OTEC platform can be implemented using effluent thrust, thrusters, or a combination of the two. These concepts are discussed in detail in Sections 5.3 and 5.4

Generation of effluent thrust equal to plant drag load can require significant increases in seawater pump power. Overall plant efficiency is sensitive to seawater pump efficiency and variation in pump power by factors of 2 to 5 to meet varying effluent thrust requirements would result in an unacceptable pump design range and lower plant efficiency under all conditions. Effluent thrust capability of the seawater transport system could be increased by adding new pumps in the seawater ducting to increase the head at a converging exhaust nozzle. In effect, the seawater transport system would incorporate a water jet thruster system in series with the primary pump. The apparent advantage of such a system for some OTEC plant sizes results from utilization of momentum increase due to the primary pumps. The power required for this momentum increase has already been charged to the seawater system and increases the apparent efficiency of the effluent thrust system. This effect is most pronounced when the additional momentum increase is small. The inclusion of an additional internal pump is merely a variation in configuration of a combined system using effluent thrust from the baseline seawater transport system and auxiliary thrusters. The performance of the added thrusters will follow the performance trend of Figure 5-12. Assuming a favorable impeller loading for the internal configuration, the performance of this system could approach that of external thrusters; however, the penalties for this arrangement are perturbation of the critical seawater path geometry and nonstandard thruster design. In summary, internal augmentation of effluent thrust offers no apparent advantage at this time.

Utilization of effluent thrust from the baseline seawater transport system is attractive for the larger OTEC plant sizes. Assume that the heat exchanger exit velocity of 2 m/s were maintained at the discharge i.e., the diffuser is not used. Figure 6-2 is a plot of the thrust contribution of this configuration as a fraction of the total thrust required to equal plant drag. The data are based upon use of only cold water effluent. The figure shows that the baseline cold water system can provide all of the thrust required under storm conditions for the spar configuration in Site A for plants above 600 MWe. Cold water effluent thrust meets 45% of the

Figure 6-2. Effluent Thrust Contribution Versus Plant Net Power

requirement for a 1000-MWe semisubmersible plant in Site A. Clearly, effluent thrust should be considered for the larger plant sizes. Note that no significant power penalty is attributed to thrust generation for this configuration.

Although the data in Section 5.3 indicate that effluent thrust could be doubled using the warm water exhaust, note that this result is based upon the assumption of zero current velocity into the intake duct. This is a good assumption for the cold water system; however, current velocities near the 2 m/s effluent velocity may be experienced at the surface. Inlet velocities of this magnitude would significantly reduce the net momentum change for the warm water system. The effluent thrust estimates made in Section 5.3 are based upon very simple assumptions and a detailed analysis of a specific configuration is required to obtain accurate performance estimates for low velocity effluent thrust.

The cost of an effluent thrust system can be determined only through development of a specific configuration for a specific plant design. No estimates will be made in this study.

Performance and cost estimates for a dynamic positioning system using auxiliary thrusters may be made using the thruster model developed in Section 5.4. The two major characteristics required for this analysis are:

- Thrust/Power, $T/\dot{W}_t = 178 \text{ N/kW}$
- Cost/Thrust, $C/T = 2 \text{ \$/N}$

Combining the thrust-to-power ratio with the drag data presented in Figure 5-5 yields the thruster power, $\dot{W}_t$, versus plant net power relationships shown in Figure 6-3. As in Figure 5-5, the heavy vertical lines represent the 20 and 100-MWe plants in Sites B and A, respectively, and include the range of plant configurations. The vertical axis on the right indicates the number of 2240-kW (3000 HP) thrusters required to generate the storm condition thrust. Use of 8 to 10 thrusters would be consistent with current dynamic positioning designs. A significant increase from this number of thrusters seems unreasonable from both control and flow interference standpoints. It would be more reasonable to use large azimuthing thrusters of advanced design or fixed screw propulsion for the higher power levels.

Figure 6-3. Total Thruster Power Required Vs. Plant Net Power

Figure 6-4 is a more general presentation of the same data. A displacement scale has been added to allow inclusion of data from existing dynamically positioned vessels. Data points for several dynamically positioned drill ships and semisubmersible drills rigs are included. The line, $\dot{W}_t \propto \Delta^{2/3}$, is a best estimate extrapolation of the semisubmersible data points into the OTEC displacement range (see Section 5.1 for discussion of scaling techniques). This line corresponds reasonably with the data for the semisubmersible OTEC configuration in the baseline environment.

The line, $\dot{W}_t = \dot{W}_{net}$, corresponds to utilization of the entire plant output for propulsion. The portion of the OTEC range above this line (crosshatched) could not be satisfied unless additional power were added to the OTEC plant output. The $\dot{W}_t = \dot{W}_{net}$ line assumes that full plant output is available during the worst case (storm) conditions. In fact, this line will lower due to reduced temperature differential in storm conditions. Note that this plot is for a system using auxiliary thrusters, only and does not include effluent thrust.

Figure 6-3 indicates that dynamic positioning of a small OTEC plant using thrusters is reasonable for the milder environment while Figure 6-2 demonstrates the usefulness of effluent thrust for large plants. Figure 6-5 shows the thruster power required for a combined system using both thrusters and baseline effluent thrust. The contribution of effluent thrust for plants with net power output below 100 MWe is small and could be neglected. The thruster requirement for plants above 500 MWe in the mild environment is small. Assuming that the OTEC plant had the most favorable drag characteristics assumed for this study and that the more benign site were chosen, the thruster power required is nearly constant for plants from 20 to 300 MWe and drops to zero at 600 MWe. The thruster power required for a 500-MWe OTEC plant meeting these favorable conditions would be less than that for a semisubmersible drill rig with 1/30 the displacement of the OTEC plant.

Figure 6-6 is a plot of unit cost (\$/kWe) for thrusters with total thrust equal to the drag loads given in Figure 5-3. The data assume a cost to thrust ratio of 2.0 \$/N. Effluent thrust is not used. Figure 6-6 presents similar data for a combined system using thruster and baseline effluent thrust. The cost of effluent thrust is assumed to be negligible.

Figure 6-4. Thruster Power for Dynamic Positioning Without Effluent Thrust

Figure 6-5. Thruster Power for Dynamic Positioning With Effluent Thrust

Figure 6-6. Unit Cost of Thrusters Without Effluent Thrust

Figure 6-7. Unit Cost of Thrusters With Effluent Thrust

The data presented in Figures 6-5 and 6-7 emphasize the significance of environment and plant configuration in determining feasibility of dynamic positioning for the OTEC platform. Dynamic positioning is feasible from both performance and cost standpoints for all plant sizes considered at Site A. Dynamic positioning is clearly not feasible for small plants (20 MWe) operating in Site B. There is insufficient onboard power to generate the required storm condition thrust and the cost would be prohibitive.

Mooring

Section 5.5 developed performance data for moorings using nylon/chain and hollow cylindrical link (HCL) lines. Reasonable limits of capability for these configurations were developed for the baseline OTEC site depth. Figure 6-8 combines these performance limits with the OTEC drag data from Section 5.2. The horizontal lines indicate limits of performance for the respective configurations. Only the HCL meets the performance requirements of the entire OTEC drag range. The nylon/chain mooring is limited to mild environments.

The two mooring line candidates may be compared for plant sizes within the range of capability of both candidates. Consider nylon/chain and HCL moorings for a 100-MWe plant in Site A. Assume a nominal storm drag load of 4 MN. Table 6-1 summarizes the characteristics of the two configurations. Although the horizontal load capabilities of the two configurations are the same, there is a difference in vertical loads on the platform and anchor. The vertical force on the platform varies by a factor of 1.5 between configurations; however, the 11-MN value for the HCL mooring is only 1.2% of the 100-MWe plant displacement. This force is much less than the reserve buoyancy for a reasonable platform design and presents no problem of plant pull down.

The difference in vertical force at the anchor could be a significant issue if drag embedment anchors were considered feasible for this application; however, the OTEC anchor study⁽¹⁾ indicates that deadweight or pile anchors are the most likely candidates for OTEC application. Either form of anchor could be designed to resist the 2.6-MN vertical force. An anchor cost tradeoff is required to test the significance of this performance difference.

Figure 6-8. Relative Performance of Moorings

TABLE 6-1. NYLON/CHAIN AND HCL MOORINGS FOR 100-MWe PLANT

	NYLON/CHAIN (Active)	HCL (Passive)
Water Depth, Z	1500 m	1500 m
Storm drag load, F_{HP} @ $T/T_{max} = 0.050$	4 MN	4 MN
Line Strength, T_{max}	10 MN	14.6 MN
Number of legs	4	3
Scope L/Z	1.5	1.75
Design Excursion X/Z	0.05	0.05
Vertical Force on Platform at $T/T_{max} = 0.5$	7.4 MN	11 MN
Vertical Force on Anchor of $T/T_{max} = 0.50$	0	2.6 MN
Line Cost, C	5.7 M\$ ^(a)	6.9 M\$
Cost Ratio, C/F_{HP}	1.42 \$/N	1.72 \$/N
Unit Cost, $C/\dot{W}_{net}$	57 \$/kWe	69 \$/kWe

a. Includes 4 winches at 0.5 M\$ per winch.

The cost data for the two candidates indicate that the HCL line cost is 21% greater than that of the nylon/chain line and winches. This difference is insignificant in light of the general nature of the cost estimates. Neither estimate includes anchor cost. The nylon line cost is an estimate for a line size which has been built only in very short lengths to verify manufacturing capability. The nylon/chain estimate does not include costs of auxiliary sheaves, rope stowage, controls, or winch power consumption.

Figure 6-9 is a plot of unit cost of mooring line (including winches for nylon) versus plant net power. The HCL data cover the entire range of OTEC drag loads while the nylon/chain data are limited to Site A. HCL data are based upon a 3 point mooring. Nylon/chain data are based upon the most cost effective design assuming a 10 MN line limit.

One of the most significant variables to be considered in comparing the two configurations is line life. Nylon moorings have typical life expectancies of from 1 year in some North Sea single point mooring (SPM) applications to 5 or more years in milder environments. Ten years is an optimistic upper limit. An HCL mooring line should have a life expectancy of 10 to 40 years. The total capital cost of line over the life of an OTEC plant could be many times the initial investment.

Combined Systems

In the foregoing discussions, dynamic positioning and mooring candidates have been characterized by several general ratios. The baseline thruster model is defined by Thrust/Power, $T/\dot{W}_t = 178 \text{ N/kW}$ and Cost/Thrust, $C/T = 2 \text{ \$/N}$.

The mooring candidates examined can be characterized by the ratio, Cost/Horizontal Force, $C/F_{Hp} = 1.34 \text{ to } 1.72 \text{ \$/N}$ for a water depth of 1500 meters. The cost-to-thrust ratio for thrusters is analogous to the cost-to-horizontal force ratio for moorings and the moorings are found to be somewhat more economical in this analysis. In this section the unit price ratio, (cost to plant net power) is applied to both dynamic positioning and moorings. Since this ratio is a derivative of the cost-to-thrust (or horizontal force) ratio, moorings again show some advantage over thrusters; however, the addition of effluent thrust gives dynamic positioning an advantage for large plants (see Figures 6-7 and 6-9). Plant size will influence the choice of a station

Figure 6-9. Unit Cost of Mooring Lines

keeping system; however, in the plant size range anticipated for the demonstration plant, neither candidate is clearly superior. The final choice for the demonstration plant may be based upon more complex considerations.

A third, hybrid concept warrants examination. Consider a combined dynamic positioning/mooring system for a 100-MWe demonstration platform. Table 6-2 and Figure 6-10 include performance and costs for such a system. The performance curves in the figure are based upon use of the mooring system, alone until the line tension ratio, $T/T_{\max}$, reaches 0.33. At this point, thrust is added, maintaining this tension ratio until full thrust capability is developed. Further increase in drag force increases the tension ratio which reaches $T/T_{\max} = 0.50$ at a drag force of about 6 MN. This system is adequate for the upper limit of predicted drag forces in Site A and could be scaled down for lower storm drag loads. The configuration is included to demonstrate the effect of the combination, not an optimum design point.

This configuration offers several advantages over the pure dynamic positioning or mooring configurations. The use of thrusters allows self propulsion of the platform for anchor setting or transit. The mooring provides station keeping capability with no power penalty during normal operating conditions, while the thrusters decrease vertical force on the anchor by an amount proportional to their contribution to the total horizontal force. During storm conditions, the thrusters can reduce dynamic loads on the mooring, reducing fatigue effects. The thrusters could provide redundancy, compensating for a mooring leg failure, or could be used in lieu of a mooring leg during mooring repair.

6.2 Technical Evaluation

This study expands the OTEC station keeping data base established in the FY76 Westinghouse study. The presentation combines parametric data and analysis of specific hardware configurations to provide flexible models of station keeping concepts. Competitive candidates are reduced to common terms of cost per unit horizontal force (\$/N) and cost per unit of plant net power (\$/kWe) to facilitate comparison and to compare station keeping cost with that of other plant hardware.

TABLE 6-2. COMBINED MOORING/THRUSTERS – 100 MWe CONCEPT

- 3 Point Moor, 14.6 MN HCL Line
- 5-3000-HP Thrusters
- Platform Excursion Limited To 5% Of Depth @ $T/T_{\max} = 0.50$
- Line Cost = 6.9 M\$
- Thruster Cost = 4.0 M\$

Depth, Z (m)	Design Scope, $(\frac{L}{Z})$	Horizontal Excursion, X (m)	X/Z	Horiz Force on Platform, F_{Hp} (MN)	Total Vert Force on Plat, F_{VP} (MN)	Line Tension Ratio, $T/T_{\max}$	Thrust Ratio, $F/F_{\max}$	Line Angle at Anchor, θ (deg)	Vert. Force on Anchor, F_{Va} (MN)
1500	1.75	0	0	0	6.6	0.20	0	16	0.5
1500	1.75	33	0.022	0.7	7.3	0.24	0	20	0.8
1500	1.75	60	0.040	1.9	8.5	0.33	0	25	1.4
1500	1.75	60	0.040	2.9	8.5	0.33	0.50	25	1.4
1500	1.75	60	0.040	3.9	8.5	0.33	1.00	25	1.4
1500	1.75	75	0.050	6.0	11	0.50	1.00	29	2.6
1500	1.75	82	0.055	12.0	19	1.00	1.00	32	6.2

Design $\dot{W}_{NET} = 100 \text{ MW}_e$

Figure 6-10. Performance of Combined System

The data presented are intended to establish a framework within which others may make tradeoffs and select final configurations. Although comparisons of candidate systems are made in this report, they are not offered as firm decisions but as examples of the use of the general data. The detailed selection of station keeping hardware is contingent upon specific site and plant configuration and must be conducted at the overall OTEC system level.

Plant/Platform Description and Scaling Techniques

a. Economy of scale: The most significant scaling relationships presented in Section 5.1 are those for plant displacement and frontal area (Equations 1 and 2). Combining the two leads to the expression:

$$F_{dp} \propto (\dot{W}_{net})^{0.55}$$

where F_{dp} is the total platform drag and $\dot{W}_{net}$ is the plant net power. This relationship leads to the dramatic economy of scale which is evident throughout the data. The frontal area scaling dominates and, even if the displacement were assumed to be proportional to plant net power, i.e., $\Delta = K_{\Delta} \dot{W}_{net}$, the drag would be $F_{dp} \propto (\dot{W}_{net})^{0.67}$ and the economy of scale effect would still be significant.

b. Candidate platform characteristics: The coefficients listed in Table 5-1 provide a manageable description of a wide variety of platform configurations. The form of the model allows the reader to assess the validity of specific assumptions. Drag data are divided to show the effect of the individual parameters and results can be easily modified to reflect change of these modeling assumptions.

Environment

Environmental models for station keeping purposes can be quite complex, with specifics as varied as the number of locations selected; however, a usable environmental model is essential to any station keeping study. Section 4.2 develops such a model, presenting a variety of data from other sources for comparison. Of the three major environmental parameters

defined, surface current is the most variable among the potential OTEC sites. Current velocities from 1.5 to 3.4 m/s are included in the study (see Figure 4-3). The baseline current/depth profile (Figure 4-4) compares favorably with much detailed data for potential OTEC sites.

Wind velocities used for the Westinghouse baseline model and Sites A and B reflect storm conditions with maximum winds typical of Gulf of Mexico hurricanes (see Figure 4-2). These velocities range from 43 to 52 m/s. These values are consistent with the latest values for the Punta Tuna, New Orleans, and Key West sites and a little high for the Keahole Point site⁽²⁾.

In general, Site A is comparable to the Punta Tuna, site and Site B is comparable to the Key West site. Adjustments in the data may be made by the reader to compensate for site variations.

Platform Drag

Drag data are presented for the ranges of five candidate platforms, for a plant net power range of 20 to 1000 MWe and for three site environments. The resultant range of OTEC plant total drag is 1.7 to 51 MN (see Figure 5-4). The drag variations (maximum drag/minimum drag) considering the three variables independently, are:

<u>Variable</u>	<u>Drag Variation</u>
Plant Configuration	3.0
Environment	4.3
Plant Net Power	7.8

Even with the strong economy of scale relationship, plant size dominates these variations.

Seawater Effluent Propulsion

The OTEC seawater transport system is physically similar to a water jet propulsion system; however, the analogy should be used with care. Considering seawater transport alone, an efficient OTEC pumping system should generate the minimum thrust consistent with practical design. The exit velocities required for high thrust lead to high kinetic losses in the

pumping system. Augmentation of the baseline seawater system thrust, i.e., increasing effluent velocity above 2 m/s, results in system compromises, a decrease in overall plant efficiency and an increase in system complexity. Baseline effluent thrust with effluent velocity of 1 to 2 m/s offers substantial dynamic positioning capability for the larger plant sizes. Baseline thrust is efficient since it utilizes kinetic effects which are unavoidable in any OTEC design. Thrust obtained with the low effluent velocity is effectively free, from a power standpoint, and should be utilized if it can contribute a major proportion of the horizontal force required.

Auxiliary Thrusters

Six generic categories of thrusters are discussed qualitatively in Section 5.4 (see Figure 5-11). No attempt is made to tradeoff these configurations on a quantitative basis. The final selection would depend upon platform configuration, total thrust required, and detailed cost and operational tradeoffs. The more specialized configurations; namely tunnel, cycloidal, and water jet systems, are normally used in applications with constraints which do not apply to the OTEC application. All of these compromise performance or cost for other considerations. The right angle drive, external screw thruster with Kort nozzle is selected as a baseline configuration and characterized by a thrust-to-power ratio of 178 N/kW and a cost-to-thrust ratio of 2 \$/N. These values are consistent with data for existing hardware. Alternate schemes with performance or cost varying significantly from these values should be examined carefully, considering:

- confidence in obtaining predicted performance, i.e., relationship to state-of-the-art, and lead time
- control capability
- performance degradation due to speed of advance
- reliability
- effect upon overall plant efficiency and geometry
- maintenance cost
- off design performance, i.e., performance at reduced thrust.

Dynamic Positioning

Both effluent thrust and auxiliary thrusters should be considered for OTEC dynamic positioning. Auxiliary thrusters are most promising for the smaller plant sizes and effluent thrust for the larger. A combined system should be considered for plants in the intermediate power range (see Figures 6-5 and 6-7). Dynamic positioning is feasible for plants in moderate environments. Assuming the platform configuration with the lowest drag coefficient operating in Site A, the fraction of plant net power required for propulsion in storm conditions is:

$\dot{W}_{net}$ (MWe)	$\dot{W}_t / \dot{W}_{net}$
20	0.5
100	0.1
500	0.01

These values include the use of effluent thrust and represent a lower limit of the dynamic positioning power requirements. Dynamic positioning becomes more attractive from the standpoint of overall plant impact as plant size increases.

Mooring

a. **Mooring Lines:** Any OTEC mooring represents some extension of current practice. The combination of water depth and drag forces, alone, requires advancing the state-of-the-art. The offshore oil industry has implemented moorings in conditions close to those required for OTEC for vessels of approximately one fifth the displacement of a 100-MWe OTEC plant.

The performance of a mooring depends chiefly upon the performance of two major components; namely mooring lines and anchors. This study addresses the lines only. The most promising line candidate using industry standard materials and fabrication techniques is a 2-component line of nylon rope and chain. Assuming use of the largest nylon line and most efficient chain produced to date (see Table 5-2), single mooring legs with a horizontal force capability of approximately 3.4 MN are feasible for 1500-meter water and an operating load of one-half of line breaking strength. Although this value is more than an order of magnitude less

than the upper OTEC drag limit of 51 MN, it can be increased to approximately 8 MN in an 8-point mooring configuration and is potentially useful for large OTEC plants in the most benign environments.

With state-of-the-art hardware meeting only a portion of the OTEC drag load range, advanced concepts have been considered. The hollow cylindrical link (HCL) or buoyant chain has been suggested by several OTEC investigators and considered for application in the offshore oil industry. The concept in its most basic form is simply a metallic tension member with the strength-to-wet weight ratio improved through addition of buoyancy. The added buoyancy could be provided by auxiliary floats; however, preliminary studies have shown that buoyancy chambers integral with the tension members are more cost effective. Although reasonably high strength steels are required for efficient HCL design, the concept makes design to any strength required for OTEC application feasible. A 146-MN breaking strength HCL has been chosen as a design consistent with current fabrication techniques and capable of meeting the upper limit of the OTEC requirement.

In a single point mooring configuration for 1500-m water depth, the 146 MN HCL can react a 56 MN horizontal drag load at a working load of one-half breaking strength. The design scope (length/water depth), for this configuration is 1.75 and the resultant line cost to horizontal force ratio is 0.41 \$/N. The horizontal force capability increases to 63 MN for a line scope of 2.95 but the line-cost-ratio increases to 0.62 \$/N.

A multi-point mooring must be used to obtain the 5% watch circle specified for this study. The most economical multi-point HCL configuration is a 3-point mooring (see Table 5-6) with a scope of approximately 1.75. The watch circle criteria are met with a line pretension of 20% of breaking strength, and a horizontal force capability of 40 MN at $T/T_{max} = 0.5$ results. The vertical force on the anchor is 26 MN for these conditions. The line cost ratio is 1.72 \$/N. This is a passive configuration with no winches or other line tension control.

This configuration may be scaled to any desired horizontal force in the OTEC range. The required breaking strength will vary directly as horizontal force with the line-cost-ratio constant at approximately 1.72 \$/N. Variations in operating load and vertical force at the anchor may be examined using the performance data in Table 5-26.

b. Design Tension: As discussed in Section 5.5, normal offshore oil industry design practice for mooring lines is:

<u>Condition</u>	<u>T/T_{max}</u>
Normal Operating	0.33
Storm	0.50

where T/T_{max} is the ratio of the actual line tension to its breaking strength. These criteria have been used in this study; however, the performance data allow the reader to assess performance at any T/T_{max} desired. These criteria are reasonable for the HCL concept. The analogy between the HCL and chain, used in the offshore oil industry, is clear. Fatigue considerations for very long life design could lead to a lower design load criteria; however, the more controlled operating conditions and improved load distribution at the joints between links may reduce peak loads compared with those in chain.

Nylon mooring lines could require a marked reduction in design load. At present, no line testing equipment is capable of verifying the strength of a 76-cm line. This problem is shared by the larger line sizes now in use and the rope industry has developed reasonable arguments to justify extrapolation of rope strength data from the smaller sizes; however, confidence must decrease as the extrapolation is extended. Design loads, T/T_{max} of 0.20 and less are frequently applied to synthetic lines with $T/T_{max} = 0.33$ to 0.25 normally used if loads are well defined. The selection of design criteria can have a marked effect on overall system cost and will be an important issue as preliminary designs evolve.

Section 6 — References

1. Valent, P.J. and Atturio, J.M., "Anchor Systems" Fourth Annual Conference on Ocean Thermal Energy Conversion, Proceedings, New Orleans, Louisiana, March 1977.
2. DOE Informal Report, "OTEC Demonstration Plant Environmental Package", July 14, 1977; Point of contact: Dr. Loyd Lewis.

SECTION 7

RECOMMENDED FUTURE DEVELOPMENT NEEDS

This general study of OTEC station keeping encompasses a broad range of variables and hardware configurations. In several cases, existing hardware configurations are extended to applications beyond their current performance capabilities. In other cases, advanced hardware concepts are offered as hardware candidates. An element of risk is associated with these extensions. Historically, the OTEC program has concentrated major efforts in developmental areas, equalizing the risk of the various program elements as much as possible. This approach leads to a balanced and predictable program. This section identifies those elements of the station keeping effort requiring early attention to assure confidence in the general data trends and permit sound system analysis. Continued study of these items is required to obtain a better assessment of the true risks involved.

Platform Configuration and Environment

A very broad range of OTEC platform configuration and environments is considered in this study. It is clear from the data in Section 6 that platform drag is a critical issue in determining final station keeping system feasibility and cost. The configuration descriptions used in this study are quite general. A more detailed analysis of a specific plant configuration in a specific environment should be conducted to establish a firm data point for comparison with the general data. The analysis should include all station keeping system elements to assure compatibility of all elements and establish a comprehensive cost data point.

Effluent Thrust

The effluent thrust data in Section 5.3 are based upon simplified assumptions of thrust generation phenomena and represent an upper limit for effluent thrust performance. A detailed flow analysis for a specific plant configuration is required to define and assess the factors which degrade this ideal performance. Effluent thrust is sufficiently attractive for the larger

plants to justify establishing a more rigorous performance baseline. A minimum effort would estimate worst case performance which, taken with the ideal performance developed in this study would establish a high confidence range for the concept.

Auxiliary Thrusters

The baseline thruster model developed in Section 5.4 provides a useful estimate of thruster performance and capital cost; however, the cost of propulsion power is a crucial issue, increasing in importance with decreasing plant size. A baseline thruster power profile should be developed for one or more specific OTEC site environments to evaluate the long term cost of thruster operation.

Mooring Lines

Both mooring line candidates discussed in Section 5.5 assume some extension of current practice. Although fabrication of a short segment of 76-cm nylon line has been demonstrated, production and deployment of a deep water mooring system using this hardware has not. The performance predictions for the line are the results of a significant extrapolation from empirical data as are life predictions. The hollow cylindrical link (HCL) concept is developmental and the performance and cost estimates used in this study are based upon quite general analysis. Although no data for HCL moorings are available, even on a small scale, the extension of technology to the OTEC HCL application is analogous to that for the nylon line.

One or both of these candidate lines should be examined in more detail. Detailed analysis of a specific configuration would identify problem areas and establish a firm data point for performance and cost. If only one were pursued at this point, the HCL is the more attractive candidate. Only the HCL offers potential for satisfying the entire range of OTEC drag loads. Analytical techniques for metal structures and pressure vessels are more advanced than those for synthetic ropes. Pursuit of the HCL candidate could lead to a high confidence benchmark for OTEC mooring performance and cost analysis.

APPENDIX A
SI UNITS AND CONVERSIONS

PHYSICAL QUANTITY	SI UNIT	SI SYMBOL	CONVERSION
Length	meter	m	1m = 3.28 ft
Mass	kilogram	kg	1kg = 2.205 lb _m
	metric ton	metric ton	1 metric ton = 1000 kg 1 tonne = 2205 lb _m
Force	newton	N	1 N = 0.2248 lb _f
Pressure	pascal	Pa	1 Pa = 1 N/m ²
			1 kPa = 0.145 psi
			1 bar = 10 ⁵ Pa
			1 atm = 1.0132 bar 1 bar = 10.2 m of H ₂ O @ 4°C
Volume	cubic meter	m ³	1m ³ = 264.2 gal (U.S.) 1 m ³ = 35.3 ft ³
Volumetric Flow	cubic meter/second	m ³ /s	1 m ³ /s = 15,850 GPM
			1 m ³ /s = 35.3 ft ³ /s
Velocity	meter/second	m/s	1 m/s = 1.942 knot
			1 m/s = 3.28 ft/s
Energy	joule	J	1 J = 1 N·m
Power	watt	W	1 W = 1 J/s
	kilowatt	kW	1 kW = 1.341 hp

STANDARD UNIT MULTIPLIERS

PREFIX	SYMBOL	FACTOR
kilo	k	10 ³
mega	M	10 ⁶
Example: 1 meganewton (MN) = 10 ⁶ newton		