SEAWATER PUMP STUDY

OCEAN THERMAL ENERGY CONVERSION PROGRAM

Final Report

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PREFACE

The Westinghouse Electric Corporation, Oceanic Division, has completed a twelve-month study of platform station keeping and seawater pumps for an Ocean Thermal Energy Conversion (OTEC) Plant. This study was performed for the U.S. Energy Research and Development Administration (now the Department of Energy) under contract E(11-1)-4071. Volume 1, Report No. C00-4071-1, covers the Platform Station Keeping Study and Volume 2, Report No. C00-4071-2, covers the Seawater Pump Study.

The effort reported by Volume 1 and Volume 2 was conducted under the Applied Technology Programs office, headed by Bradley L. Johnston. The study team was directed by Dr. Thomas E. Little as the project manager/principal investigator. Major contributions were made to the study reported here by Dr. Terry Wright and G. Russell Howland of Westinghouse R&D Center, Fluid Systems Laboratory, West Lafayette, Indiana and by Wilmott G. Brown and A. LeBreton of Westinghouse, Steam Turbine Division, Lester, Pennsylvania.

ABSTRACT

The pumping power required to move cold seawater and warm seawater through an Ocean Thermal Energy Conversion (OTEC) power plant is a significant portion of the plant power output; therefore, seawater pump performance, sizing, and cost information are very influential inputs into any power plant system design optimizations. The analysis and evaluation of large seawater pumping systems selected specifically for the OTEC application are provided with a view toward judging the impact of pump selection on overall OTEC power plant performance. A self-contained bulb, direct drive, axial flow pump was found to have a distinct advantage in performance and arrangement flexibility. A design of a pump operating at a net total head rise of 3.5 meters and a flow capacity of $100 \text{ m}^3/\text{s}$ is presented including pump blade geometry (profiles), pump diffuser geometry, and pump/diffuser configuration and performance. Results are presented in terms of the geometric and power requirements of several related pump designs over a range of seawater capacity from $25 \text{ m}^3/\text{s}$ to $100 \text{ m}^3/\text{s}$. Summary analysis and evaluations include pump design weights and cost estimates.

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SECTION 1

INTRODUCTION

1.1 General Statement of the Problem

An initial investigation into the general characteristics required of seawater pumps for the OTEC application was reported by Westinghouse Oceanic Division FY76 study (Reference 1). The accumulated data of the reference report indicated that the head and capacity requirements of the OTEC power plant seawater pumps lead directly to the selection of axial flow pump designs. Typical power system operational requirements obtained from a survey of previous OTEC studies (References 2 through 10) indicate that the seawater pumps must operate at low pressure heads and very high rates of flow; typical values are $H = 3.5\text{m}$ for the pump head required and $Q = 25\text{ m}^3/\text{s}$ up to $Q = 100\text{ m}^3/\text{s}$ for pump volumetric flow rate. Figure 1-1 shows four seawater pump configurations that were considered as candidate concepts for the OTEC application.

A self-contained bulb, direct drive, axial flow pump was found in past studies to have a distinct advantage in performance and arrangement flexibility. The diffusing sections of the pump bulb unit, located downstream of the stator blade section, must be efficient to achieve minimum required pumping power. The diffusing sections referred to here consist of an annular diffuser downstream of the rotor-stator blade section followed by a conical (or parabolic shape) diffuser. This diffusion is needed to reduce the relatively high fluid velocity discharged from the pump throat to a more usable fluid velocity of approximately two meters per second with the minimum loss of available energy.

The self-contained bulb arrangement provides the compactness and shape that allows maximum freedom in shaping a high efficiency pump-diffuser unit; the flow is straight through with minimum turns and obstructions such that viscous dissipation losses can be held to a minimum. Also, the bulb configuration allows the entire integrated (matched) pump-diffuser unit to be produced as a "plug-in" unit requiring no field drive train assembly or alignment.

Axial Pump Configurations

Self-Contained Bulb Pump

External Drive Pump

External Drive Pump

Vertical Pit Pump

Figure 1-1. Seawater Pump Configurations Considered for OTEC Application

It is suggested that the best seawater pump design, in terms of overall pumping efficiency and operational flexibility, will not be an off-the-shelf unit but will be a special design having features unique to the OTEC plant seawater flow path requirements. To insure that the OTEC seawater pumps are developed in an orderly and timely fashion it is necessary to establish axial pump parametric designs sufficient to accomplish hydrodynamic tradeoff analyses leading to a selection of a particular preliminary design configuration. This particular design configuration can then be evaluated and compared with future alternatives as to drive motor selected, overall performance, mechanical considerations, maintainability, and overall system cost.

1.2 Study Objectives

The efforts of this study were directed to selecting a preliminary design configuration of an OTEC seawater pump that can be used to establish performance, size, and cost baseline information. To accomplish this, the following specific study objectives emerge as most descriptive of the major areas of concentration:

- Select seawater pump operational requirements such as head, capacity range, envelope restrictions, etc., that are representative of a typical OTEC power plant.
- Establish a logical systems approach leading to the identification and description of issues and evaluation criteria needed to select candidate seawater pump (warm water and/or cold water) design configurations.
- Layout and describe a specific seawater pump design.
- Provide a summary analysis and evaluation of the established pump design configuration that includes weight, cost, and overall impact on the OTEC power plant.

The work flow logic, Figure 1-2, shows schematically the pump design process, consistent with the above stated objectives, that was used during this study. As indicated here, a pump with a volumetric flow rate of $100 \text{ m}^3/\text{s}$, designated as the "baseline" design, was used as the model for the bulk of the direct analysis. The $25 \text{ m}^3/\text{s}$ capacity pump design was carried through the study to show the relative influence of pump capacity to size, design speed, weight, and cost.

Work Flow Logic — Seawater Pump Analysis
H = 3.5 m Total Head Rise

Figure 1-2. Logic Schematic for the Seawater Pump Study Effort

1.3 Study Requirements and Constraints

From a consideration of the operating conditions experienced by an OTEC plant, the following seawater pump design guidelines were used:

- Seawater volumetric flow rate, $Q = 100 \text{ m}^3/\text{s}$ for the "baseline design" and $Q = 25 \text{ m}^3/\text{s}$ for the secondary design. Previous OTEC studies have indicated that the cold seawater volumetric flow rate required will be approximately $4 \text{ m}^3/\text{s}$ per megawatt of plant net power and that the warm seawater flow required will be approximately the same. Therefore, a 100-MWe plant (net power) would require 8 large pumps ($Q = 100 \text{ m}^3/\text{s}$) or 32 small pumps ($Q = 25 \text{ m}^3/\text{s}$) to handle combined cold and warm seawater flows.
- A baseline pump pressure head of 3.5 meters and a head range of 2 to 5 meters of seawater was selected as relevant to the cold water and/or the warm water pumping requirement for a typical OTEC plant.
- A seawater density of $1027 \text{ kg}/\text{m}^3$ was selected for the cold seawater and a density of $1024 \text{ kg}/\text{m}^3$ was selected for the warm seawater.
- Structurally all pump designs must be capable of continuously operating under sea state nine conditions with a maximum of 0.5g acceleration due to a platform motion. Also, all designs must operate at 100-meter sea depths, therefore the pump motor and drive system components must be isolated from seawater contact by using seals and/or pressure compensated pod packaging that is minimally affected by submerged location.
- Slow startup/shutdown capability is desired to minimize pressure surges in the seawater piping and heat exchangers.

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SECTION 2

EXECUTIVE SUMMARY

This report provides the results of a twelve-month study concerned with the selection and preliminary design analysis of seawater pumping systems for Ocean Thermal Power Plants. The seawater pumping system referred to here includes a hydraulic pump (rotor-stator and mechanical drive), a drive motor, pump inlet section, and diffuser sections necessary to provide the desired fluid exit velocity and net head rise.

A pump net head rise (total) of 3.5 meters of seawater and pump volumetric flow rate of $100 \text{ m}^3/\text{s}$ and $25 \text{ m}^3/\text{s}$ were selected as the operation requirements for the design effort reported here.

As an outgrowth of this effort a self-contained bulb, direct drive axial flow pump, shown by Figure 2-1, was found to have a distinct advantage in performance and arrangement flexibility. The diffuser sections of the pump bulb unit, located downstream of the stator blade section, must be efficient to achieve minimum required pumping power. The diffuser sections referred to here consist of an annular diffuser downstream of the rotor-stator blade section followed by a conical (or parabolic shape) diffuser. These diffuser sections are needed to reduce the relatively high fluid velocity discharged from the pump throat to a velocity that is more compatible with the power plant seawater flow path (approx. 2 m/s). It is desirable to achieve this velocity reduction with the minimum loss of available energy.

The self-contained bulb arrangement provides the compactness and shape that allow maximum freedom in shaping a high efficiency pump/diffuser unit; the flow is straight through with minimum turns and obstructions such that viscous dissipation losses can be held to a minimum. Also the bulb configuration allows the entire integrated (matched) pump/diffuser unit to be produced as a "plug-in" unit requiring no field drive train assembly or alignment. The rotating blades are driven directly from the motor (no transmission, belts, or chains are required).

Axial Flow Seawater Pump/Diffuser Configuration

Figure 2-1. Axial Flow Seawater Pump/Diffuser Configuration

As a result of this study it is suggested that the best seawater pump design, in terms of overall pumping efficiency and operational flexibility, will not be an off-the-shelf unit but will be a special design having features unique to the OTEC plant seawater flow path requirements. To insure that the OTEC seawater pumps are developed in an orderly and timely fashion, it is necessary to establish axial pump designs in sufficient depth to accomplish hydrodynamic tradeoff analysis that in turn can be used to support the selection of beneficial design configurations. These select configurations can then be evaluated and compared with future alternatives. Issues of special interest are drive motor selection, overall performance, mechanical considerations, maintainability, and overall system cost.

As indicated by the discussion above, a direct drive, axial flow pump, driven by an electric motor, is a strong candidate for an OTEC seawater pump. There are several types of electric motors that can be considered for the seawater pump prime mover:

- AC synchronous machines
- DC machine (conventional)
- DC homopolar machines (with segmented magnetic circuit to give higher voltage operation)

AC, DC, and DC homopolar machines were considered because it was felt that each type would be able to offer some distinct advantages for the bulb-mounted pump applications.

At this point in the pump design process, an AC synchronous drive motor with a predicted efficiency of 93% to 94% is selected as the most flexible approach. A key point in the selection of the pump motor concerns the very low speed required for the impeller. A direct drive shaft at 55 rpm requires unusually low-speed motors. To reach this speed in a bulb unit with an AC motor requires a low frequency excitation at 15 Hz and a motor that has a high length/diameter ratio. The synchronous motor fed by a low frequency (12 to 15 Hz) supply compares favorably with respect to motor weight, cost, and maintenance.

The pump/diffuser design configuration selected for 100 m³/s capacity and 3.5 meter net head rise (total) was determined to be 26.3 meters in overall length with a rotor tip diameter of 4.9 meters, a diffuser exit diameter of 8 meters and a efficiency, $\eta' = 76\%$ at design point. The efficiency, η' , used here is the dimensionless product of capacity, Q and net total head rise, H (between pump inlet and diffuser outlet) divided by the input shaft power, P.

$$\eta' = \rho QgH/P.$$

The cost of the 100 m³/s pump unit, excluding the outer shell and diffuser shell, totals to \$1.74 x 10⁶ dollars or approximately \$139 per kilowatt of plant net power. The 100 m³/s pump AC motor, rated at 4662 kw, costs approximately 61% of the above total cost.

To obtain a pump with a 25-m³/s capacity and a 3.5-meter net head rise, it is necessary to halve the rotor tip diameter to 2.45 meters and double the pump rotational speed (to 110 rpm). Although the blade cord Reynolds number of the 25-m³/s pump is halved, no further correction is necessary on the efficiency ($\eta' = 76\%$) since the Reynolds number effects are very small at the large Reynolds numbers of the pumps considered here. Therefore, the input power of the 25-m³/s pump is one-fourth the input power of the 100-m³/s pump.

The cost of a 25-m³/s pump unit, excluding the outer pump and diffuser shell, totals to \$517,100 dollars or approximately \$165 per kilowatt of plant net power. The AC motor, rated at 1167 kw, costs about 45% of the total cost for a 25-m³/s capacity pumping unit.

SECTION 3

RESULTS AND DISCUSSION

3.1 Survey of Comparable Equipment

A survey of comparable equipment, reported by Westinghouse Oceanic Division report (Reference 1) and summarized by Table 3-1, indicates the general availability of pumps in the OTEC performance range. The low-head, low-speed OTEC operating requirements were most difficult to match. High-head, high-speed units were more commonplace, but these were of the centrifugal and mixed flow types and were not comparable to the required axial flow design. During this study no production units of the self-contained bulb configuration were located that approximated the design requirements. Most available pumps were of the side discharge, wet pit configuration that use external prime movers and long driven shaft.

Even though the operational characteristics of the pumps summarized by Table 3-1 were not exactly parallel to the calculated OTEC requirements, it is possible to derive considerable information from the data by plotting and establishing trend lines as shown by Figures 3-1 through 3-3. Figures 3-1 and 3-2 show the trend of pump size and speed with pump capacity. It is instructive to note that the larger pump sizes surveyed have a hub radius of approximately 3 meters with speeds of 35-70 rpm; also, pump speed decreases as pump size increases. This trend will be supported by analytical results in later discussions; however, at this point it can be stated that for large capacity pumps ($r_o > 3\text{m}$) the pump drive/transmission unit should be capable of speeds less than 50 rpm.

Figure 3-3 provides a review of typical pump efficiencies for existing large capacity pumps as related to pump specific speed. The solid line represents an accumulation of data for existing pump design (Reference 2) and the points are from the survey summarized by Table 3-1. A maximum pump efficiency of 86 to 88% seems to be indicated by the accumulation of data so far; however, the OTEC pump requirements of low head and high capacity cannot be considered typical nor can we anticipate the resulting hardware to be typical. Based on the initial

TABLE 3-1. SURVEY OF PUMP MANUFACTURERS PUMP CHARACTERISTICS

Manufacturer	Type	Flow, Q (m ³ /s)	Head, H (m)	Radius, r _o (m)	Speed, N (rpm)	Specific Speed, N _s [rpm(gpm) ^{1/2} /ft ^{3/4}]	Pump Efficiency, η (%)	Pump Weight, Wt. (kN)
① Worthington	Vertical Pit	39.74	6.10					458
② Worthington	Vertical Pit	12.6-16.0	9.10	1.8	200-225	7000-8000	80-87	382(a)
③ Allis-Chalmers	Vertical Pit	31.5	6.10	2.13 (bell)			80-90	441(b)
④ James Leffel	Blades Only	113		3.05				445(e)
⑤ Mitsubishi	Bulb	113	2.72	3.05	54	14,000		1530(c)
⑥ Mitsubishi	Bulb	85	1.21	3.25	35	14,500		1430
⑦ Ingersoll Rand	Vertical Pit	14.5	7.62	1.05	235	10,000	86.5	177(a)
⑧ Ingersoll Rand	Mixed Flow	22.1	10.7-16.8	1.37	270	11,100	87	688(a) 220(d)
⑨ Byron Jackson	Axial Flow Control	22.2	1.5-1.8	1.60	90	15,900	80	538(a)
⑩ Bofors-Nohab	Turbine Generator		5.3	3.2	68.2			
⑪ Mitsubishi	Centrifugal Pump Turbine	76.7	400		300			
⑫ Voith	Turbine Generator		42	3.68				
⑬ Escher-Wyss	Pump Turbine (Bulb)		16					
⑭ Ateliers de Charmille	Pump Turbine (Bulb)	227	4.57	2.67	94		90	
⑮ Stork	Axial Flood Control	31.5		2.44				

(a) Pump less drive train (b) Pump less drive train and housing (c) Pump and Motor, less housing (d) Motor drive (e) Blades only

C.W. PUMP SURVEY OUTER RADIUS VS CAPACITY

76075A022

Figure 3-1. Pump Outer Casing Radius Versus Flow Capacity

C.W.PUMP SURVEY SPEED VS OUTER RADIUS

78075A021

Figure 3-2. Pump Rotational Speed Versus Pump Outer Casing Radius

Typical Pump Efficiencies For Existing Large Capacity Pumps

	Q (m ³ /s)	H (m)	N (rpm)	η (%)
②	12.6	9.1	200	80-87
⑦	14.5	7.6	235	86.5
⑧	22.1	10.7	270	87
⑨	22.1	1.8	90	80

Figure 3-3. Typical Pump Efficiencies Versus Specific Speed

industrial pump manufacturer's survey, it can be stated that the optimum pump, in terms of efficiency, maintainability, and operational flexibility, will not be an off-the-shelf item but will be a special design having features unique to the OTEC plant seawater flow path requirements.

In summary, the operational requirements for an OTEC pump were utilized as the first step in the selection of a pump type. The next step involved a state-of-the-art survey and examination of existing large capacity pumps that approximated the OTEC pump requirements. At this point an analysis based on ideal hydraulic assumptions and a representative (but not optimum) hydrofoil blade section was utilized to obtain additional information about large axial flow pumps.

3.2 Candidate Pump Configuration Analysis

a. Methods of Analysis

A design analysis computer program has been developed by Westinghouse R&D Center, Fluid System Laboratory, which provides a complete geometric description of an axial flow turbo machine based on the input requirements of head rise, volume flow rate, rotational speed, and diameter. The solution or design procedure is based on three basic elements — an inviscid, incompressible streamline solution; a semi-empirical cascade life-prediction model; and an empirical loss and loading-limit calculation.

The inviscid solution for the flow through a blade or vane row is governed by the Euler equation with conservation of mass and determination of energy addition associated with total head rise through the machine. This flow is constrained to satisfy "simple" radial equilibrium; radial velocities, as such, are not considered (see, for example, References 3 and 4). The output of this calculation is in the form of normalized velocity vector distributions to be used in the next calculation for determining two-dimensional blade cascade parameters.

Blading parameters are chosen at each radial station of the turbomachine which will conform to the velocity vector distribution requirements of the inviscid solution and which will, in fact, generate those flow patterns. To provide the method of choosing these cascade sections, a subroutine relates blade or vane solidity, camber, thickness, and stagger angle so that the required flow patterns are generated. The method is based on the theoretical work of Mellor (Reference 5) whose approach consists of determining the distribution of vortex strength and source-sink strength along a blade mean camber line which will satisfy normal flow boundary conditions and the Kutta condition at the blade trailing edge. Mellor's solution has been empirically modified for this work based on correlations with extensive NACA 65-series blower blade data (see Reference 6).

To augment and control the use of the cascade calculation, a method was developed to estimate the losses associated with a given blade or vane in a two-dimensional cascade and a correlation was developed to provide a conservative estimate of the maximum load a blade can support without suffering severe energy losses. Loading limit is provided by a correlation of the

permissible range of blade angle of attack above the design or minimum loss value and relates this range to the cascade parameters. Both correlations yield conservative results and serve primarily to eliminate poor design choices and to limit ranges of investigations.

Solution and design of a turbomachine is initiated as a simple free-vortex form. An initial inviscid solution is generated and a check is made to determine whether the design may be constructed within reasonable blading load limits. If this check fails, the design is termed unworkable since the blading will stall, so hub size is increased from a minimum value toward a specified maximum. If a load check still fails, hub size is again increased and so forth until maximum hub size is exceeded (or a solution is reached). When hub size is exceeded, swirl flow is perturbed from free vortex, minimum hub size is reset, and the search continues subject to a size constraint in the ratio of discharge maximum and minimum axial velocity values. If under these conditions hub size is again exceeded, no solution is generated and warnings are printed. Several levels of control of design details are available as options depending on user expertise.

For calculation of a complete characteristic or pressure-volume curve, it is required to know the lift and drag for values of blade incidence which are both higher and lower than design point values. Selected cascade configurations from NACA data (Reference 6) were used to create a correlation of off-design loss values based on blade incidence, loss at design point, design point incidence, and blade stall incidence. In the model it is assumed that the loss at design point and both positive and negative stall are connected by a parabola; beyond positive and negative stall, loss values are governed by linear extensions from the parabola. The same set of data was used to examine blade lift behavior for incidence beyond positive stall. In general, there is a leveling-off of cascade lift with increasing incidence as the stall value is approached and exceeded, and this behavior is not predicted by the method of Reference 5. A simple means of placing an upper limit on lift is to join the theoretical lift curve to a horizontal line whose lift value is equal to the theoretical value at stall incidence. A slight refinement is added to this concept by adding a small linear derate to the theoretical curve with horizontal extension to effect a mild decline in lift force with very high incidence. In addition to the off-design lift and loss models, the influence of end-wall boundary layer interference on blade loss coefficients has been included in an approximate correlation developed by Howell (Reference 7). The final correction to the lift and loss calculations consists of using an empirical determination of the

additional blade drag, pressure loss, and flow recirculation associated with the running clearance between blade tips and housing or running ring. This calculation is based on the work of Reference 8, and for small clearance has a minor effect on the performance of the turbomachine.

Use of these extended methods is carried out with a logically inverted form of the design analysis program, discussed above, to provide radial distributions of force vectors on the blading so that torque may be calculated by integration of tangential force components acting on the blades. Determination by the program of the accompanying head rise at volume flow rate is used with the calculated torque or shaft power to determine the efficiency as the ratio of output power (product of flow and head rise) to input power.

It should be noted that the experimental data on which the correlations for the above analyses are based were for the most part acquired at blade chord Reynolds numbers around one-half million. Since the pumps considered in this work are characterized by blade chord Reynolds numbers between ten and twenty million, some reduction in loss or increase in efficiency is to be expected. The quantitative dependence on Reynolds numbers is considered at length in References 9 and 10, and in this work it is stated that the ratio of losses (i.e., 1.0 minus efficiency) is equal to the inverse ratio of Reynolds numbers raised to the power, n . " n " is assigned values in the literature ranging from 0.167 to 0.45. For purposes of conservatism, a value of 0.167 is used in this work. Furthermore, full-scale Reynolds number effects are assumed to virtually vanish beyond four million (as suggested by Pampreen, in Reference 10). For the pumps considered in this work, such a calculation results in an increase of efficiency of about three percentage points, which is interpreted as a reasonable expectation of favorable scale effect.

The methods used to analyze candidate configurations for the diffusers for the various pumps considered are based on empirical results of the work of Sovran and Klomp, Reference 11, and Idel'Chik, Reference 12, for axisymmetric annular diffusers, and on the work of McDonald and Fox, Reference 13, and Idel'Chik for axisymmetric conical and exponential diffusers. As is strongly emphasized in these references and in References 14 and 15, adequate diffuser design must include a determination of the stability of the flow in the diffusing element.

If diffusion is too rapid, the flow will be characterized by very high losses, or by intermittent stall or pressure-flow oscillations of possibly disastrous magnitude. Much of the work of References 11, 13, 14, and 15 have been devoted to establishing boundaries, in terms of inlet-to-outlet-area ratio versus length-to-inlet-height ratio below which such unstable flows will be avoided. The diffusers used in this work have all been selected to lie in the stable flow range both in order to achieve efficient and to achieve steady performance. This approach should avoid pump and overall system problems and should provide a degree of performance margin when various elementary diffusers are used in series in a single configuration.

In matching pump designs to diffuser configurations, it was necessary to determine, at least roughly, the physical dimensions of the motor to decide if a flared motor pod was required or if the motor could nestle behind the hub of the pump. A simple scaling rule was used to establish motor size by assuming that motor power density (watts/meter³) was invariant over the range of consideration for this work. A baseline motor was used which was characterized by a motor housing (pod) diameter requirement of approximately 2.7 meters, a housing length of approximately 3.2 meters for a rated output of 3360 kw. Motor length to diameter ratio was held constant for all scaling calculations.

b. Pump/Diffuser Parametric Description

The methods described in the preceding section were used to determine configuration and typical performance of a series of seawater pumps that could be used for the OTEC application. A self-contained bulb, direct drive, axial flow pump arrangement, as shown by Figure 3-4, was found to have a distinct advantage in overall performance and arrangement flexibility. Figure 3-5 and Figure 3-6 show matching pump/diffuser assemblies that cover a range of motor pod size and pump capacity. Each pump shown has 6 rotor blades and 11 stator blades. The diffuser sections, located downstream of the stator blades, provide a recovery of kinetic energy developed by the rotor blades into useful static pressure at the diffuser discharge. A diffuser discharge velocity of 2 m/s was used for all the configurations shown. Both conical (dash line) and parabolic (solid line) diffusers are shown with each pump design for completeness. It is felt that the reduced lengths of the parabolic diffusers should preclude use of the conical elements but it must be noted that the performance of the conical elements can be predicted with greater confidence, particularly in the area of flow stability, and that the cone is an inherently simple shape to construct.

Figure 3-4. Axial Flow Seawater Pump/Diffuser Configuration

Pump-Diffuser Assemblies

$Q = 100 \text{ m}^3/\text{s}$ And $H = 3.5 \text{ m}$

$\eta' = 0.76$

- (a) 4.9 m Tip Dia. 55rpm
2.9 m Hub Dia.

$\eta' = 0.74$

- (b) 5.5 m Tip Dia. 50rpm
3.5 m Hub Dia.

$\eta' = 0.72$

- (c) 6.1 m Tip Dia. 45rpm
4.1 m Hub Dia.

$$\eta' = \frac{\rho Q g H}{P}$$

Figure 3-5. Pump/Diffuser Configuration And Efficiency For $100 \text{ m}^3/\text{s}$ Flow Capacity at 3.5 m Head

Pump-Diffuser Assemblies

$Q = 25 \text{ m}^3/\text{s}$ And $H = 3.5 \text{ m}$

(a) 2.45 m Tip Dia. 110 rpm
1.46 m Hub Dia.

(b) 2.75 m Tip Dia. 100 rpm
1.77 m Hub Dia.

(c) 3.05 m Tip Dia. 90 rpm
2.04 m Hub Dia.

Figure 3-6. Pump/Diffuser Configurations And Efficiency For $25 \text{ m}^3/\text{s}$ Flow Capacity At 3.5 m Head

The pump-diffuser assembly dimensions and performance at design point conditions are tabulated in Table 3-2. The efficiency, η' , is the overall efficiency of the assembly, calculated as the product of volume flow rate and net total pump head rise (between pump inlet and diffuser outlet) divided by the input shaft power. The input power is based on the hydrodynamics of the rotor blades and does not include bearing torques, motor windage, or electrical losses.

A review of the configurations listed in Table 3-2 indicate that configuration no. 1 for a 100-m³/s capacity pump and configuration no. 4 for a 25-m³/s capacity pump are superior from the standpoint of minimum input power required. Also, as evident from Figure 3-5 and Figure 3-6, the general trend is an increase in overall length with decreasing pump size, so that a cost and/or arrangement tradeoff exists. Since the cost (per pound) of rotating equipment including the motor) is generally several times the cost for stationary elements, the longer more efficient pump/diffuser configurations (no. 1 and no. 4) will probably represent minimum first cost seawater pumping units. Since, in addition, the required power for the smaller pumps is lower, long-term evaluated cost should yield the same preference.

The specific speed presented in Table 3-2 for each pump design is a useful parameter that is the same for all pumps that are geometrically similar. In its most basic form, specific speed is a dimensionless product of the pump discharge Q , the head H , and the speed ω , such that

$$n_s = (\omega Q^{1/2}) / (gH)^{3/4}. \quad (1)$$

The pump specific speed defined above has a definite value for a machine of given design operating at maximum efficiency (see Figure 3-3). It is a common industrial practice in pump design to express pump speed in revolutions per minute (rpm), the discharge in gallons per minute (gpm), and the head in feet. From these relations a widespread and commonly used dimensional form of specific speed can be defined as:

$$N_s = (N Q^{1/2}) / (H^{3/4}) \text{ [rpm (gpm)}^{1/2} \text{ / ft}^{3/4}] \quad (2)$$

TABLE 3-2. SUMMARY OF CANDIDATE PUMP/DIFFUSER ASSEMBLIES,
CONFIGURATIONS AND PERFORMANCE

(a) No.	Flow, Q (m ³ /s)	Head, H (m)	Outer Dia., D (m)	Speed, N (rpm)	Total (b) Efficiency, η' (%)	Power, P (kW)	Hub To Tip Ratio, d/D (d'less)	Specific Speed n_s (d'less)	Flow Coeff. ϕ (d'less)
1	100	3.5	4.90	55	76	4650	0.59	3.79	0.58
2	100	3.5	5.50	50	74	4740	0.64	3.50	0.50
3	100	3.5	6.10	45	72	4870	0.67	3.19	0.43
4	25	3.5	2.45	110	76	1160	0.59	3.79	0.58
5	25	3.5	2.75	100	74	1190	0.64	3.50	0.50
6	25	3.5	3.05	90	72	1220	0.67	3.19	0.43
(a) Each pump has 6 rotor blades and 11 stator blades									
(b) $\eta' = \rho QgH/P$									

where the units of N are rpm, Q is in gpm, and H is in feet. The unit conversion

$$N_s = 2733 n_s [\text{rpm (gpm)}^{1/2} / \text{ft}^{3/4}] \quad (3)$$

can be used to convert specific speed into the familiar units shown from the dimensionless form, n_s .

For an axial flow pump the flow coefficient, ϕ , presented in Table 3-2 for each pump design is defined as

$$\phi = 2V_1 / \omega D, \quad (4)$$

where V_1 is the pump inlet velocity just upstream the rotor blade section. The expression above for the flow coefficient, the mass continuity equation, and the specific speed equation can be combined to give the very useful pump diameter and pump speed scaling equations as follows:

$$D = (2Q^{1/2}) / [\pi^{1/3} (n_s \phi x)^{1/3} (gH)^{1/4}] \quad (5)$$

and

$$\omega = [2n_s^{2/3} (gH)^{1/2}] / [\pi^{1/3} (\phi x)^{1/3} D] \quad (6)$$

where x is a dimensionless function of the pump hub-to-tip ratio according to the equation

$$x = 1 - (d/D)^2. \quad (7)$$

The unit conversion

$$N = 9.549 \omega [\text{rpm}] \quad (8)$$

can be used to convert pump speed to the units shown, given the rotational speed ω in radians per second.

c. Pump Inlet Section and Diffuser Section Sizing

Pump inlet configurations are quite simple and are all designed to be similar. The inlet section consists of an inner nose piece for the pump which is half of a prolate spheroid with a 45-degree elliptical profile (eccentricity of 0.500). The outer wall is a one-quarter elliptical torus of the same eccentricity. The resulting overall inlet area is nearly equal to the diffuser outlet area, yielding velocities in the range of 2 m/s for minimal inlet loss. An example is given in Figure 3-7 for the 4.9 meter, 100-m³/s capacity pump.

The pump configurations for a 100-m³/s flow rate described in the preceding subsection, included a range of sizes from a diameter of 4.9 meters to 6.1 meters with corresponding axial annulus velocities of 8.2 m/s to 6.2 m/s. The requirement placed on the diffuser is to reduce the velocity to 2 m/s, implying the need for outlet-to-inlet velocity ratios of 4.1 for the 4.9 meter pump to 3.1 for the 6.1 meter pump, with the total pressure loss in the diffuser constrained to be no more than approximately 10% of the velocity pressure in the pump annulus.

As discussed in 3.2, the length-to-inlet-height ratio of a given diffuser must be held to a minimum value in order to preclude the intermittent stall and unsteady flow in the diffuser and pump. This minimum length ratio is a strong function of the required area ratio. If, for example, pure annular elements were used from the motor to the outlet, the length of a conical diffuser for the 4.9 meter pump would be a formidable 44 meters. To escape these excessive length requirements, the configuration of Figure 3-8 is proposed, consisting of two basic elements: an annular diffuser with a cylindrical outer wall and a tapering center body to expand the flow to the full pump disc area; and an axisymmetric (non-annular) second-stage diffuser. If the annular element is designed conservatively, a fairly rapid pressure recovery can be taken in the axisymmetric element. For simplicity, a conical second element may be used, or, for reduced length, a parabolic (constant pressure gradient) second element may be used, such that both conical and parabolic design have the same loss and stability margin.

Using the methods previously outlined, element lengths and total diffuser lengths have been calculated for three pumps with diameters of 4.9 meters, 5.5 meters, and 6.1 meters. These lengths and diameters are given in Table 3-3 along with the power requirements. The nomenclature is illustrated in Figure 3-8. These diffusers satisfy the stability, loss, and outlet

Figure 3-7. Pump Inlet Configuration Shown With Pump Stage

Figure 3-8. Conceptual Illustration of Diffuser Configuration Showing Dimensional Nomenclature

velocity requirements and are subject to the constraint that the motor housing diameters are less than or (nearly) equal to their pump hub diameters to avoid the need for an expanding-contracting motor pod or bulb. Actually, for the 4.9 meter pump, hub diameter is 2.9 meters while required motor diameter is 3.06 meters. Extra length in the annular section (0.56 meters) was allowed to accommodate the small additional diffusion requirement.

The two-element diffusers for these pumps are illustrated in Figure 3-9 in order to show the dependence of increasing diffuser size with decreasing pump size. It is emphasized that a variation of 300 hp (0.223 MWe) must be evaluated against the relative costs of the small pump-long diffuser (4.9 meter pump) or large pump-short diffuser (6.1 meter pump) configurations. As seen in Figure 3-9, use of the more complex parabolic element with the 4.9 meter pump yields a substantial savings (10.0 meters) in overall length. For the 6.1 meter pump, the parabolic configuration appears to yield a relatively unimpressive length reduction (3.4 meters). While all of these configurations may appear to be somewhat long, significant reductions in length will degrade the pressure performance (require more pumping power) and will lead to problems of unsteadiness in pump performance.

The diffuser for the $100\text{-m}^3/\text{s}$ capacity pumps may be scaled directly (one-half size) for use with the 3.05 meter pump designed for a head of 3.5 meters and $25\text{ m}^3/\text{s}$ volume flow rate, since the scaled motor pod diameter requirement and the scaled hub diameter is approximately 2.04 meters (see Table 3-4). Scaling down the 4.9 meter and 5.5 meter pumps results in motor pod diameters considerably larger than the hub diameters. Thus, coupling of these pumps to their diffusers will require a flared motor pod and outer body, as shown in Figure 3-8.

Figure 3-10 shows the diffusers for the three pumps designed for $25\text{ m}^3/\text{s}$ volume flow rate. For the smallest pump (2.45 meters), the transition section to bring the flow on to the motor pod causes a significant increase in overall length. For the 2.75 meter and the 3.05 meter pumps, the results are very similar to the diffusers associated with the full size pumps, shown in Figure 3-9.

0 4 8 12 16 20 24 28 32 36 (m)

(A) 4.9 m PUMP OD

(B) 5.5 m PUMP OD

(C) 6.1 m PUMP OD

*Figure 3-9. Diffuser Configurations For the Full-Size Pumps
Showing Optional Conical and Parabolic Elements*

*Figure 3-10. Diffuser Configurations for the Half-Size Pumps
Showing Optional Parabolic and Conical Elements*

TABLE 3-3. DIFFUSER DIMENSIONS FOR $Q = 100 \text{ m}^3/\text{s}$ PUMPS

D (m)	d (m)	D _p (m)	l _t (m)	l _m (m)	l _a (m)	l _p (m)	l _c (m)
4.90	2.92	2.92	0	3.51	4.88	17.70	24.50
5.50	3.54	3.54	0	3.53	6.47	11.00	16.56
6.10	4.08	4.08	0	3.57	7.98	7.60	10.98

TABLE 3-4. DIFFUSER DIMENSIONS FOR $Q = 25 \text{ m}^3/\text{s}$ PUMPS

D (m)	d (m)	D _p (m)	l _t (m)	l _m (m)	l _a (m)	l _p (m)	l _c (m)
2.45	1.46	1.92	1.5	2.24	3.94	5.38	8.25
2.75	1.77	1.94	0.5	2.25	3.31	4.85	7.44
3.05	2.04	2.04	0	2.26	4.00	3.80	5.80

3.3 Drive Motor Considerations

a. Pump Motor Configuration Analysis

As indicated by the discussion of the preceding section, the overall power system constraints and power plant efficiency demands lead to a self-contained bulb, direct drive, axial flow pump, driven by an electric motor, as a strong candidate for an OTEC seawater pump. There are several types of electric motors that can be considered for the seawater pump prime mover:

- ④ AC synchronous machines
- ④ DC machine (conventional)
- ④ DC homopolar machines (with segmented magnetic circuit to give higher voltage operation).

AC, DC, and DC, homopolar machines were considered because it was felt that each type would be able to offer some distinct advantages for the bulb mounted pump application.

Factors in addition to size and efficiency that enter into the choice of motor type are:

- ④ Maintenance - particularly with respect to brush life; DC machines are at a disadvantage here
- ④ Ease of speed control for the slow start up required
- ④ Reasonable voltage and current levels
- ④ Efficiency
- ④ Cost (development and production)

To establish a base for the pump motor design, an effort was made to identify machines of similar power, speed, and shape that might be commercially available or had been built in the past. The first observation is that machines at this power level are not "off the shelf"; each unit is designed for a specific application. Secondly, the very low speed required and small diameter permissible puts this motor design well outside the design parameters of normal commercial

low-speed synchronous or DC motors. Applications for large hydraulic turbine generators approach the OTEC seawater pump requirements closely, however, design speeds tend to be higher and the pods are relatively greater in diameter.

Due to an absence of available designs a preliminary design and cost estimate for both AC and DC machines were developed to satisfy the OTEC pump motor requirements. Table 3-5 presents a summary of the motors considered for the self-contained bulb units. Two basic sizes of machines to fit the 25 m³/s pump and the 100 m³/s pump were considered. The bulb diameters, motor power required, and motor speed corresponding to the most favorable hydrodynamic efficiency (see Table 3-2) were used as motor design constraints.

Design comparisons for tradeoff purposes were determined for the AC synchronous and conventional DC machines. These comparisons are shown by Figure 3-11 and Figure 3-12.

The AC synchronous machine offers long maintenance-free service and generally lower weight and construction cost and higher efficiency than a DC machine. The conventional DC machine offers ease of power control and independence from any particular operating frequency of supply. The segmented rotor DC homopolar motor, while still developmental by comparison, was felt to offer the possibility of a more compact DC machine with the power and speed required for the smaller pod size (configuration no. 1). A closer look at the segmented rotor homopolar machine, however, presents it in an unfavorable light for this application. In a homopolar machine, current passes axially down a cylindrical rotor surface. The induced voltage (back EMF) due to the cylindrical rotor turning in the radial magnetic field is low. Increased voltage operation is obtained by segmenting the rotor and arranging for the current to make several passes through the magnetic field. For each segmentation of the rotor an extra pair of brushes and slip rings is required; while the total number of brushes goes up, the current per brush goes down due to the higher voltage and lower current operating condition.

For example, in this application, by using an 80-inch core length, at 50 rpm, and rotor diameter consistent with pod diameter, the voltage per current pass would be 19 volts. With 36 such passes connected in series, the motor terminal voltage would still be only 680 volts, but the current must now be collected 72 times instead of twice as in a conventional DC machine. It

TABLE 3-5. SUMMARY OF PUMP MOTOR CHARACTERISTICS

PUMP CAPACITY, Q	100 m ³ /s	25 m ³ /s
<ul style="list-style-type: none"> GENERAL <ul style="list-style-type: none"> Motor Pod Diameter Motor Power Required Motor Speed SYNCHRONOUS AC MOTOR <ul style="list-style-type: none"> Voltage Frequency Efficiency Total Weight Cost Estimate CONVENTIONAL DC MOTOR <ul style="list-style-type: none"> Efficiency Total Weight Cost Estimate SEGMENTED MAGNET DC MOTOR (SEGMAG) <ul style="list-style-type: none"> Efficiency Total Weight Cost Estimate 	<ul style="list-style-type: none"> 2.91 m 4662 kW 55 rpm 4160 V 13.7 Hz 94% 622.7 kN \$1,065,000 89.6% 1,236.5 kN \$1,935,000 92.9% 956.3 kN \$1,420,000 	<ul style="list-style-type: none"> 1.92 m 1167 kW 110 rpm 2400 V 15 Hz 92.5% 151.2 kN \$235,000 89.4% 253.5 kN \$495,000 90.5% 302.5 kN \$670,000

4.448 kN = 1000 lb_f

Figure 3-11. Seawater Pump Motor Size

Figure 3-12. Seawater Pump Motor Efficiency And Cost

is felt this would be prohibitive in cost and brush maintenance. The use of the homopolar machine could only be justified provided a conventional machine could not be designed for the application.

b. Power Plant Considerations

The choice of the pump motor itself cannot be made without considering the impact of the pump configuration selected on the overall power system. These considerations should include requirements for slow startup or continuous speed control and conversion from AC to DC, or AC to AC of a different frequency.

A factor in the selection of the drive train for the pump motor concerns the very low speed required for the impeller. A direct drive shaft at 45-55 rpm requires unusually low-speed motors. To reach this speed in a bulb unit with an AC motor requires a low frequency excitation at 15 Hz and a motor that has a high length/diameter ratio. Low frequency excitation for an AC machine can be provided via a cycloconverter, assuming main power to be generated at 60 Hz, or by a separate generator.

A key parameter in determining the final system will be the period over which the slow startup of the impeller must be extended. For relatively short startup periods, for example, a dissipative speed control device (such as a clutch or series resistor) could be used. For very long periods or where continuous speed control is required a more sophisticated approach is needed.

After removing technically invalid alternatives, several possible configurations remain for delivering power to the pump impellers with the required control, with varying degrees of efficiency and cost. Figures 3-13 through Figure 3-15 show block diagrams of several possible basic approaches used in conjunction with the overall plant power system.

Figure 3-13 shows the simplest electrical starting mechanism, a resistance step start controlling DC machines and the simplest continuous speed control mechanism uses thyristor half-bridge converters to provide variable voltage DC to control DC machines. These methods

Figure 3-13. Candidate DC Or DC Homopolar Motor Power Supply And Control Schemes

Figure 3-14. Candidate AC Synchronous Motor Power Supply And Control Schemes

Figure 3-15. Candidate AC Synchronous Motor Power Supply And Control Schemes

provide for operation only in the 1st quadrant of the motor operating region, i.e., it cannot provide dynamic braking or drive the motor in reverse. However, for a pump/propeller type load these features are not usually required.

Figure 3-14 represent two techniques for providing speed control for AC motors. Both approaches provide variable frequency, variable amplitude, 3-phase AC to the motor. The converter/inverter approach first rectifies the incoming AC and then converts it to variable frequency and amplitude AC. It is best suited to driving synchronous machines since a leading power factor can be forced by the machine and self-commutation of the converter can take place. The cycloconverter approach provides high quality AC sine wave output for driving either synchronous or induction machines at frequencies below 25% of the input frequency (60 Hz). It uses more complex control circuitry than the converter inverter approach, but at the power levels considered here full use can be made of semiconductor KVA rating.

Figure 3-15 shows a mechanical start approach using a clutch to control the rate of transfer of power from the turbine shaft to an AC generator that supplies the 15 Hz power necessary for energizing the motors. Also shown is a scheme where variable speed control can be provided for any one motor from primary power supplied by a diesel generator set or by reverse power flow from the grid during initial power plant startup and subsequently from the platform main 60 Hz generator. Each motor can be started and brought up to speed and switched in turn to the fixed 15 Hz pump motor supply. In this way the full benefit of a variable speed control for startup is achieved; although, cycloconverter electronics for only one pump unit power level is required.

Figure 3-16 shows a cycloconverter schematic illustrative of an electronic speed control for full range control of an AC machine. Typical cycloconverter cost are \$100 to \$150 per kW.

c. Selected Pump Drive Motor

At this point in the pump design process, an AC synchronous drive motor is selected as the most flexible approach as a drive unit for the OTEC seawater pump.

Figure 3-16. Cycloconverter Schematic

A key point in the selection of the pump motor concerns the very low speed required for the impeller. A direct drive shaft at 55 rpm requires unusually low-speed motors. To reach this speed in a bulb unit with an AC motor requires a low frequency excitation at approximately 15 Hz and a motor that has a high length/diameter ratio. The synchronous motor fed by a low frequency (12 to 15 Hz) supply compares favorably with respect to motor weight, cost and maintenance.

A power supply and control system similar to scheme (b) shown by Figure 3-15 can handle startup requirements from either standard diesel generator equipment or a 60 Hz power tie and can also provide speed controlled reversal of flow direction if this capability is required.

3.4 Selected Pump Design Configuration and Performance

This section is concerned with establishing a preliminary design configuration of a typical OTEC seawater pumping system. The pump configuration analysis of the previous sections identified overall pump efficiency for several motor pod sizes. It was also determined that an AC motor of acceptable performance could be designed to fit within the pod diameter and length corresponding to the most efficient pod configuration.

a. Preliminary Design Layout and Description

The pump design layout shown by Figure 3-17 was selected to verify that construction of such a pump is feasible and that sufficient volume and structural margin exist to accommodate modifications that may arise during more detailed design phases.

The selected pump design configuration consists of a rotating nose cone and 6 rotor blades followed by 11 stator blades, that are fixed to a stationery pod structure, followed by the tail cone structure. The rotor and stator profiles are based on a 10% thick blade with a NACA-65A profile. The rotor and stator blade design parameters are listed in Table 3-6. The stator blades are also attached to the outer pump shell and provide the support for the pod cylinder assembly. The pod cylinder assembly is made of the (1) pod cylinder; (2) motor shaft; (3) AC motor; (4) nose cone assembly consisting of fixed pitch rotor blades, blade support structure and nose cone; and (5) tail cone assembly.

Corrosion resistant stainless steels are used for surfaces in contact with seawater. The pod external structure is constructed of stainless steel plate, such as 316L, formed to the shapes desired and ring stiffened to provide the required strength with a minimum weight penalty. The nose and tail cone shapes (prolate ellipsoids) were selected because of hydrodynamic considerations. This structure is sized to provide a pressure boundary, with margin, for the motor cavity when submerged to sea depths up to 100 meters and also to foundation and motor stator, shaft bearing and seal supports. Access for assembly and maintenance is provided through closure plates in the nose and tail cones.

Figure 3-17. Design Layout For A 100 m³/s Flow Capacity Pump

TABLE 3-6. BLADE LAYOUT PARAMETERS FOR 100 m³/s CAPACITY PUMP

Radius (meters)	Chord (meters)	Blade Geometric Angle (degrees)	Camber (degrees)
2.92 (Hub)	1.522	57.07	41.58
3.12	1.595	53.95	36.56
3.32	1.645	51.25	33.08
3.52	1.670	48.76	30.02
3.72	1.690	46.49	27.53
3.92	1.695	44.43	25.51
4.12	1.690	42.53	23.82
4.32	1.680	40.77	22.33
4.52	1.660	39.14	21.05
4.72	1.642	37.63	19.96
4.90 (Tip)	1.620	36.22	18.92
Stator Chord — 1.671 meters Stator Blade Angle — 79.89 degrees Stator Camber — 30.70 degrees			

The motor shafts are machined of 347 stainless steel and are attached to the pump motor rotating member. The bearing support structure, fore and aft, provide for the accurate positioning of the motor shaft and the pump rotating blades with respect to the rest of the pump assembly. The bearings are sized to provide sufficient radial and thrust load carrying capability to maintain the alignment of the rotor shaft under the full range of loads and to provide a bearing life of approximately 50,000 hours.

The front bearing support foundation also locates the seal structure which houses the seal assemblies. These assemblies consist of two self-aligning, floating face seals and an inflatable seal that is used for installation/removal of the seawater pump. The space between the two face seals is connected to a pump that will discharge leakage to the ocean surface. In addition, the bearing support structure is sealed to help insure a dry atmosphere in the motor cavity if water spray/moisture should get by the second seal.

Electrical leads and tubes to the pump motor, motor case and seal cavity are provided through one or more of the eleven stator blades. The various tubes, in addition to evacuating the seal cavity and pressurizing the inflatable seal, provide a means for cooling, purging and atmosphere control of the motor and motor housing.

Preliminary stress, stability and load calculations have been made on the selected pump arrangement (Figure 3-17), considering all aspects of thrust loads, buckling loads, pressure loads, bending moments, critical speed, torque loading, stress concentration and fatigue endurance factors. This arrangement has the flexibility to be mounted in either a horizontal or vertical position and all calculations were made considering the worst case conditions. The motor shaft calculations indicate: (1) a first critical speed of 900 rpm which is far above the 55 rpm operating speed; and (2) a safety factor of 2.3 with respect to the fatigue endurance stress for fully reversed bending. All other components exhibited higher safety factors indicating that the concept is extremely rugged and has plenty of margin to allow tradeoffs among weight, life, cost and safety factors during detail design.

During the tradeoff analysis leading to the seawater pump configuration presented here many alternatives were considered and certainly should be reconsidered during any detailed design effort. In all cases, however, the selected configuration allows sufficient space for growth, alteration, and rearrangements; as may be required during detailed design. Some of the considerations that deserve further comment are listed below:

- Bearings - The concept depicts radial and thrust bearings working in combination to react the imposed loads. These bearings were selected because they were typical of off-the-shelf bearings and had an adequate rated life of 50,000 hours under load. Other styles such as angular contact or tapered roller bearings could be utilized if hardware requirements, cost, schedule or other parameters so dictate.
- The external shape of the structure is dictated by the hydrodynamic considerations discussed in previous sections. The ellipsoidal shapes provide more stiffness than straight cylindrical sections; therefore, a lighter structure is possible; however, the added complexity and cost of ellipsoid fabrication is a parameter that, during detail design, can be traded-off against hydrodynamic performance and weight to insure the optimum selection.
- Pump Construction/Material - The construction techniques and materials shown in the selected pump configuration are but one out of many variations that could be employed. Examination of the material selection and construction techniques should be conducted in conjunction with the overall plant maintenance philosophy and provisions. Weight/volume/cost constraints are influenced by the plant, the cold pipe, and the mounting interface between the seawater duct and the pump. In the pump configuration presented here, the pump interfaces with the seawater duct through the outer shell, thereby allowing the critical dimensions between the rotating group and the outer shell to be closely controlled within the pump assembly. In this manner, the mounting provisions and adjustments and any material incompatibilities have been shifted to the non-critical area outside the pump fluid flow path.

b. Rotor-Stator Configuration and Performance

The pump flow, pressure, and density requirements for the selected pump design are as follows:

Flow - $100 \text{ m}^3/\text{sec}$

Pressure - 3.5 meters of H_2O (total head rise)

Density - $1024 - 1027 \text{ kg/m}^3$

The pump sizes studied, using the methods outlined in the preceding Section, ranged from outer diameters of 4.6 meters to 6.1 meters and speeds of 35 to 70 rpm. Corrections were made to the power and efficiency for Reynolds number scaling, as previously described. A diffuser pressure loss of approximately 10% of the pump annulus velocity pressure and a rotor tip clearance of 7 mm was assumed for the analysis of this section. Some of the combinations (low speed and small diameter) could not meet the required flow and pressure conditions and, therefore, represent design limits. Although the minimum power is obtained with a diameter of 4.6 meters, the turning angles required of this pump (particularly near the hub) are approaching the maximum allowable, and therefore it is questionable if the required performance could be obtained. A good compromise between minimum power and acceptable design practice is a 4.9 meter (rotor tip diameter) axial pump running at 55 rpm. The basic dimensions of the pump are as follows:

Outer diameter	4.9 meters
Hub diameter	2.92 meters
Speed	55 rpm
Number of rotor blades	6
Number of stator blades	11

The rotor profile at the hub, mean, and tip are shown in Figure 3-18. The rotor and stator profiles are based on a 10% thick blade with a NACA-65A profile (for blade thickness distribution, see Reference 16). The blade twist is indicated by the stacked drawing in Figure 3-19. The stator profile is shown in Figure 3-20. The stator is designed uniform with no radial twist. The blade design parameters are listed in Table 3-6 and in Figure 3-21. The camber line has a circular arc profile.

4.9 Meter Pump Rotor Blade Profiles

Figure 3-18. Rotor Blade Profiles For $100 \text{ m}^3/\text{s}$ Flow Capacity Pump

4.9 Meter Pump Rotor Blade Stack

Figure 3-19. Rotor Blade Stack For 100 m³/s Flow Capacity Pump

4.9 METER PUMP
STATOR BLADE PROFILE

Figure 3-20. Stator Blade Profile For $100 \text{ m}^3/\text{s}$ Flow Capacity Pump

4.9 Meter Pump Rotor Blade Parameters

Figure 3-21. Rotor Blade Parameters For 100 m³/s Flow Capacity Pump

Figure 3-22 shows the distribution of lift and drag coefficients across the blade span at design point operation ($H = 3.5\text{m}$, $Q = 100\text{m}^3/\text{s}$) for the 4.9 meter pump. Lift coefficient is defined by normalizing the lift per unit span by the product of blade chord and blade relative inlet dynamic pressure. Drag coefficient is the drag force per unit span normalized in the same manner.

In order to obtain a pump with a $25\text{ m}^3/\text{s}$ capacity and a 3.5 meter net total head rise, it is necessary to halve the outer and hub diameters (to 2.45 meters and 1.46 meters respectively) and double the speed (to 110 rpm). All of the blade lengths given in Table 3-6 would be divided by two, but the angles remain the same. Although the blade chord Reynolds number of the smaller pump is halved, no further correction is necessary on the efficiency since the Reynolds number effects are very small at the large Re of the pumps considered here. Therefore, the input power of the $25\text{ m}^3/\text{s}$ pump is one-fourth the input power of the $100\text{ m}^3/\text{s}$ pump.

Once the basic rotor-stator design is known, characteristic curves for this configuration can be generated. Without a diffuser, the performance of the 4.9 meter diameter rotor-stator arrangement is provided by Figure 3-23. Pressure is defined as the total head rise of the pump only. Efficiency is also based on the pump head rise only and is not penalized by the diffuser losses. As indicated here the rotor-stator unit efficiency, η , is 86% at the selected design point; however, the pump exit velocity immediately downstream of the stator is approximately 8.2 m/s .

As shown by Figure 3-23, the design point selected for the pump occurs slightly to the right (higher flow) of the peak efficiency point; thereby a surge margin is provided that is desirable in order to prevent blade stall during periods where there are seawater flow fluctuations. At this point in the OTEC seawater pump selection process, it is not certain what magnitude of flow variations might be experienced by the pump as a result of platform heave, etc.; based on conventional turbomachinery design practices, the indicated margin is suggested at this time.

Figure 3-22. Force Coefficients For 100 m³/s Flow Capacity Pump At Design Point

4.9 Meter Pump Net Performance

Figure 3-23. Pump Net Performance Characteristic Curves

Typical off-design performance for the selected rotor-stator configuration ($Q = 100 \text{ m}^3/\text{s}$, $D = 4.9 \text{ m}$) was investigated. Figure 3-24 shows the effect of changing the blade angle by plus or minus 2 degrees. As shown here, a lower head can be obtained with larger surge margin or a higher head can be obtained at the expense of surge margin. It is instructive to note the sensitivity of the pump performance to the blade angle setting since this issue relates to a decision of fixed blade versus variable blade angle pump designs. At this point in the pump analysis process, a fixed-pitch blade was selected as the baseline configuration; however, future development efforts should be concerned with performance tradeoff comparisons between fixed-pitch and controllable pitch designs.

Figure 3-25 provides information regarding rotor-stator unit off-design characteristics as a function of pump speed. This information can be combined with power plant seawater flow path head loss curves to obtain the seawater flow versus pumping power relationships need to evaluate power plant control requirements.

c. Pump/Diffuser Assembly — Configuration and Performance

The pump configuration and rotor-stator performance reported in the previous sections describes the recommended approach to seawater pumping. Without diffuser sections the basic pump efficiency was determined to be 86% (see Figure 3-23) at design point; however, the fluid exit velocity corresponding to this arrangement is approximately 8.2 m/s . The majority of pump manufacturers use the above efficiency in published performance data describing their equipment. Unfortunately, the 8.2 m/s velocity cannot be used directly in the majority of OTEC power plant seawater flow paths; therefore, this velocity must be recovered as pressure by using diffuser sections.

The rotor-stator configurations, the diffuser configurations, and the drive motor considerations presented in the previous sections were used to select the seawater pump/diffuser preliminary design configuration shown by Figure 3-4 and described by Table 3-7. Pump/diffuser assemblies are described here that combine the basic pump with diffuser sections that are needed to reduce the seawater fluid velocity to a usable velocity of 2 m/s with maximum recovery efficiency. The recommended configurations shown by Table 3-7 consist of two basic

4.9 Meter Pump

Figure 3-24. Pump Off-Design Characteristics With Variation Of Blade Pitch Angle

4.9 Meter Pump

Figure 3-25. Pump Off-Design Characteristics With Variation Of Speed

TABLE 3-7. PUMP/DIFFUSER DESIGN CONFIGURATIONS FOR 100 m³/s AND 25 m³/s FLOW CAPACITY AT 3.5 m HEAD

Number of Rotor Blades = 6
Number of Stator Blades = 11

Capacity, Q (m ³ /s)	Head, H (m)	Speed, N (rpm)	Power, P (kW)	Efficiency, η' (%)	Casing Dia., D (m)	Hub Dia., d (m)	Length Overall, L (m)	Pod Length ℓ (m)	Exit Dia., D _e (m)
100	3.5	55	4650	76	4.90	2.92	26.3	12.3	8
25	3.5	110	1163	76	2.45	1.46	14.8	9.4	4

$$\eta' = \rho Q g H / P$$

elements; an annular diffuser with a cylindrical outer wall and a tapering center body to expand the flow to the full pump disc area followed by an axisymmetric second-stage diffuser using a parabolic (constant pressure gradient) expansion.

As indicated by Table 3-7 and Figure 3-26, the pump/diffuser assembly selected for the preliminary design configuration has an overall efficiency of 76% at design point. Here the seawater velocity at the pump/diffuser discharge is 2 m/s and the corresponding efficiency, η' , is calculated as the product of pump capacity, seawater density and net total head rise (between pump inlet and diffuser outlet) divided by the input shaft power,

$$\eta' = \rho Q g H / P.$$

Figure 3-26 provides the net performance curve of the 100 m³/s pump/diffuser configuration and the effects of adding the described diffuser section to the basic 100 m³/s rotor-stator pumping unit is obtained by comparing Figure 3-26 with Figure 3-23.

The design point selected for the pump occurs slightly to the right (higher flow) of the peak efficiency point; thereby a surge margin is provided that is desirable in order to prevent blade stall during periods where there are seawater flow fluctuations. At this point in the design process it is not certain what magnitude of flow variations might be experienced by the pump as a result of platform heave, etc.; however, based on conventional turbomachinery design practices, the indicated margin is suggested at this time.

The overall efficiency of the selected 100 m³/s flow capacity pumping unit includes a consideration of generator losses, power transmission and switching losses, motor efficiency, seal losses, and pump/diffuser efficiency. A schematic of the losses considered with an estimated overall efficiency is provided by Figure 3-27.

d. Operational Considerations

Installation is an important consideration to OTEC seawater pump maintainability. The cost and complexity of scheduled maintenance is proportional to the ease or difficulty in removing and installing the pump unit. The bulb type pump unit was selected for ease of removal and

4.9 Meter Pump And Diffuser Net Performance

Figure 3-26. Pump/Diffuser Net Performance Characteristic Curves

Overall Efficiency For 100 m³/s Pump/Diffuser

Figure 3-27. Overall Efficiency For 100 m³/s Flow Capacity Pump/Diffuser

installing. It is anticipated that maintenance will be required on a one to two year schedule to inspect and/or replace pump bearings, seals, and motor parts.

The average availability for the seawater pumps is estimated to be 94% to 96% where most of the downtime will be planned maintenance. This availability is comparable with the 10 years of service performance demonstrated by large pump-turbines now operating as part of the La Rance project (Reference 17).

The selected pump configurations were analyzed to determine cavitation characteristics as a function of pump speed; fluid velocity, and pump depth of submergence. It was found that tip cavitation would not occur if the pump was submerged to a depth greater than 2 meters.

In addition to tip cavitation the tip clearance flow through the gap between the blade tip and the annulus wall interacts with secondary flows in the rotor tip region to form areas where cavitation might occur. It is anticipated that a tip clearance of 7mm to 10mm is reasonable for the pump designs proposed here. This tip clearance would keep the gap clearance to blade thickness ratio below 0.1 which is sufficient to prevent excessive cavitation due to the clearance between pump rotor blades and the outer shell.

Section 3 — References

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SECTION 4

SUMMARY ANALYSIS AND EVALUATION

4.1 Weight and Cost Summary

As an outgrowth of the study effort reported here, a self-contained bulb, direct drive, axial flow pump was found to have a distinct advantage in performance and arrangement flexibility. The selected configuration shown by Figure 4-1 was described in the preceding sections. Table 4-1 provides a weight breakdown of the major parts of the "baseline" 100 m³/s flow capacity pump/diffuser unit. The pumping unit total weight, including the AC motor drive but excluding the outer shell of the pump and diffuser, is approximately 1724 kn (175.8 metric tons). As indicated by Table 4-1 the pump rotating parts account for 25.5% of the total pumping unit weight; the stationary parts are 38.4% of the total and the motor is 36.1%. The stationary parts of the pump do not include the outer shell of the pump and diffuser, rather it is suggested that the outer shell (see Figure 4-1) be grouped as part of the platform weight and cost. The weight for a 25 m³/s capacity pump, scaled down from the 100 m³/s pump of Table 4-1, is approximately 580.7 kn (59.2 metric tons).

The cost estimates of the selected pumping unit configurations are provided by Table 4-2. The total cost of the 100 m³/s capacity pump is \$1,739,000 or in terms of cost per unit capacity, \$17,390 per m³/s. The total cost of the 25 m³/s capacity pump is \$517,000 or \$20,680 per m³/s. Assuming the cold water flow rate required for a typical OTEC power plant is about 4 m³/s per MWe net power and that the warm water flow required is approximately equal to the cold water flow rate then the total seawater flow required is approximately 8 m³/s per MWe net plant power; using the assumption the seawater pumping cost would be about \$139 per kWe plant net power using 100 m³/s capacity pumps or \$165 per kWe plant net power if 25 m³/s capacity pumps are used.

Seawater Pump/Diffuser Assembly **3.5-meter Total Head Rise**

	<u>$Q = 100 \text{ m}^3/\text{s}$</u>	<u>$Q = 25 \text{ m}^3/\text{s}$</u>
● Size	$L = 26.3 \text{ m}$ $D\epsilon = 8 \text{ m}$	$L = 14.8 \text{ m}$ $D\epsilon = 4 \text{ m}$
● Pump/Diffuser Efficiency, η'	76 - 82%	76 - 82%

Figure 4-1. Selected Seawater Pump/Diffuser Configurations

TABLE 4-1. WEIGHT BREAKDOWN OF 100 m³/s CAPACITY PUMP DESIGN

A. ROTATING PARTS	WEIGHT (kN)	
Blades	82.2	
Nose Cone	45.7	
Blade Support Structure	176.7	
Motor Shaft	<u>135.0</u>	439.6 kN
B. STATIONARY PARTS		
(Excluding Outer Shell of Pump/Diffuser)		
Pod Cylinder	298.7	
Sealing Structure	45.5	
Front Bearing and Structure	47.2	
Rear Bearing and Structure	30.9	
Tail Cone	94.7	
Stationary Blades	<u>144.6</u>	661.6 kN
C. AC MOTOR		
Total Motor Weight	<u>622.7</u>	<u>622.7 kN</u>
TOTAL PUMP WEIGHT (Excluding Outer Shell)		1723.9 kN

4.448 kN = 1000 lb_r

TABLE 4-2. COST ESTIMATE SUMMARY OF 100 m³/s AND 25 m³/s
FLOW CAPACITY PUMPS

<u>PUMP CAPACITY, Q</u>	100 m ³ /s	*25 m ³ /s
• General		
Motor Power Required	4662 kW	1167 kW
Pump Speed	55 rpm	110 rpm
Pump Tip Diameter	4.9 m	2.45 m
Motor Pod Diameter	2.91 m	1.92 m
• Cost Breakdown (1977 Dollars)		
AC Synchronous Motor	\$1,065,000	\$235,000
Rotating Parts (Excluding Motor)	290,000	118,900
Stationary Parts (Excluding Motor and Outer Shell)	361,000	143,000
Assembly — Ship	23,000	20,200
ESTIMATE OF TOTAL COST (1977 Dollars)	\$1,739,000	* \$517,100

* Relative cost scaled from 100 m³/s capacity pump design

4.2 Pump Design Scaling

The pump/diffuser assembly dimensions and performance at design point conditions are tabulated in Table 4-3. The pump/diffuser efficiency, η' , tabulated here, is the efficiency of the assembly, calculated as the product of volume flow rate and net total pump head rise (between pump inlet and diffuser outlet) divided by the input shaft power. The input power is based on the hydrodynamics of the rotor blades and does not include the bearing torques, motor windage, or electrical losses.

$$\eta' = \rho Q g H / P. \quad (1)$$

The specific speed presented in Table 4-3 for each pump design is a useful parameter that is the same for all pumps that are geometrically similar. In its most basic form, specific speed is a dimensionless product of the pump discharge, Q , the net head rise, H , and the rotational speed, ω , such that

$$n_s = (\omega Q^{1/2}) / (gH)^{3/4}. \quad (2)$$

For an axial flow pump the flow coefficient, ϕ , presented in Table 4-3 for each pump design is defined as

$$\phi = 2V_1 / \omega D, \quad (3)$$

where V_1 is the pump inlet velocity just upstream the rotor blade section and D is the pump rotor blade tip diameter. The expression above for the flow coefficient, the mass continuity equation, and the specific speed equation can be combined to give the very useful pump diameter and pump rotational speed scaling equations as follows

$$D = (2Q^{1/2}) / [\pi^{1/3} (n_s \phi x)^{1/3} (gH)^{1/4}] \quad (4)$$

$$\omega = [2n_s^{2/3} (gH)^{1/2}] / [\pi^{1/3} (\phi x)^{1/3} D] \quad (5)$$

TABLE 4-3. SEAWATER PUMP DESIGN SUMMARY

DESIGN CONFIGURATION

Capacity, Q (m ³ /s)	Head, H (m)	Casing Dia., D (m)	Speed, N (rpm)	Power Req., P (kw)	Hub To Tip Ratio, X = d/D (d'less)	Efficiency, η' (%)	Specific Speed, N _s (d'less)	Flow Coeff, ϕ_o (d'less)
100	3.5	4.90	55	4650	0.59	76	3.79	0.58
25	3.5	2.45	110	1163	0.59	76	3.79	0.58

TABLE 4-4.

SCALING RELATIONSHIPS

$D \propto \frac{Q^{1/2}}{H^{1/4}}$
$W \propto \frac{H^{3/4}}{Q^{1/2}}$
$P \propto H Q$

where x is a dimensionless function of the pump hub-to-tip ratio according to the equation

$$x = 1 - (d/D)^2. \quad (6)$$

It is instructive to note that the rotor tip diameter, D , rotational speed, ω , and power required, P , for geometrically similar pumps can be obtained from the following scaling relationships:

$$D \propto Q^{1/2}/H^{1/4}, \quad (7)$$

$$\omega \propto H^{3/4}/Q^{1/2}, \quad (8)$$

and

$$p \propto H Q. \quad (9)$$

The relationships above (summarized in Table 4-4), can be developed from pump similarity laws or from the information provided by Table 4-3, Equation 4, and Equation 5.

4.3 Recommended Future Development

An off-design analysis of the seawater pumps at constant speed has been made and is presented in this report. Off-design performance at other speeds is assumed to follow the normal pump scaling laws. However, during start-up at very low Reynolds numbers, scaling may not be valid. It is strongly recommended that an air model pump be fabricated and pump performance at low Reynolds numbers be evaluated to determine if a bypass or variable pitch blades are required during the start-up phase.

Compound diffusers, such as the diffuser proposed for this application, are not amenable to direct analytical approaches since the boundary layer growth in the first diffuser directly affects the performance of the second diffuser. It is recommended that more detailed computer studies be made of the diffuser to determine boundary layer growth and stability, followed by limited air model testing.

Since the seawater pumps may be operating in modules of three or four parallel pumping units, it is recommended that dynamic interactions between the pumps be studied in greater detail by means of dynamic interaction computer analysis.

APPENDIX A
SI UNITS AND CONVERSIONS

PHYSICAL QUANTITY	SI UNIT	SI SYMBOL	CONVERSION
Length	meter	m	1 m = 3.28 ft
Mass	kilogram	kg	1 kg = 2.205 lb _m
	metric ton	metric ton	1 metric ton = 1000 kg 1 metric ton = 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	1 m 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 gal/min 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		