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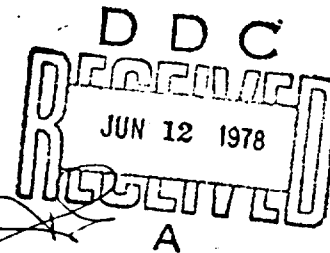
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DIMENSIONAL ANALYSIS OF OCEAN THERMAL ENERGY CONVERSION  
HEAT EXCHANGERS.

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| This paper points out certain historical highlights and problems connected<br>with development of electrical energy from deep-ocean thermal differences.<br>Natural and economic factors which have focused attention on this type of<br>energy development are mentioned, as well as areas of support by the National<br>Science Foundation, the Energy Research and Development Administration and the<br>U.S. Navy.<br><br>Dimensional analysis is used to develop a list of dimensionless groups (over) |                       |                                                                                                                        |

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of factors having significance in OTEC (Ocean Thermal Energy Conversion) heat exchangers. Certain of these groups are then evaluated for a model and prototype OTEC-Type heat exchanger using the same working fluid and experiencing the same working fluid flow rate per unit area. A discussion of the evaluation and conclusions complete the report.

ABSTRACT

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## SUMMARY

Chapter I, the Introduction, points out certain historical highlights and problems connected with development of electrical energy from deep-ocean temperature differences. Included also are certain natural and economic factors which more recently have focused national attention on this method of energy development. Areas of support by the National Science Foundation, the Energy Research and Development Administration, and the U. S. Navy are brought out.

Chapter II investigates the scaling problems of heat exchangers in OTEC (Ocean Thermal Energy Conversion) power plants. Dimensional analysis is used to develop a list of dimensionless groups of factors affecting heat transfer on both the sea water and working fluid sides of the heat exchangers.

Chapter III evaluates certain of these dimensionless groups for a prototype and model OTEC-type condenser. This evaluation is based on the same working fluid and the same working fluid flow rate per unit area in the model and the prototype.

Chapter IV lists the results of the evaluation and states pertinent conclusions drawn therefrom.

## Chapter I. INTRODUCTION

### 1. Background and Early Problems

One of the earliest observations of the possibilities for extracting useful energy from the temperature difference existing between warm upper layers and the considerably colder deep layers of some ocean areas is attributed to D'Arsonval in 1881.<sup>1</sup> Later, a student of D'Arsonval's named Georges Claude had both the vision and the courage necessary to fabricate, erect and operate an open cycle vapor power plant capable of generating 22 kW power from approximately 200 liters per second of warm sea water and the same flow of sea water some 14 degrees Centigrade colder.<sup>2</sup>

Even though Claude's efforts were successful in proving the technical correctness of the theory, they were anything but a success from an economic standpoint. His use of (sea) water as the cycle energy vehicle, his selection of a land-based plant, the absence of certain necessary pieces of oceanographic information and the nature of his attempt as a first-of-its kind all combined to overshadow the fact that his main objective had been satisfied.

Encouraged by the operational success of his venture against some almost insurmountable odds Claude predicted that, with continued effort in component development and refinement of design, large plants of this nature would soon be running year-round oblivious to both the natural and economic

<sup>1</sup>D'Arsonval, Dr. A., "Utilisation des Forces Naturelles," Avenir de l'électricité, Revue Scientifique, pp 370-372 (Sept. 17, 1881).

<sup>2</sup>Claude, Georges, "Power From the Tropical Seas," Mechanical Engineering, Vol. 52:12, pp 1039-1044 (Dec 1930).

factors which sometimes plagued the fossil fueled and hydro-electric plants of that era.

## 2. More Recent Studies

In spite of the optimistic predictions and high expectations expressed by Claude in 1929, not much more was heard concerning the tropical sea power plant during the economically difficult 1940 and 1950 decades, but certain other forces came into action to focus attention on the oceans of the world during the 1960 decade. Experience gained in deep ocean drilling for oil, the "Mohole Project," and many other surface and deep-ocean projects served to increase the confidence with which such projects could be approached. During this period of time a number of papers and articles were published by J.H. Anderson and J.H. Anderson, Jr.<sup>3</sup> which carried fairly detailed design calculations on the feasibility of large sea thermal power plants proposed for the Caribbean Sea.

## 3. NSF and ERDA Support

For economic and other reasons, emphasis on oceanographic research seemed to wane in the latter part of the 1960 decade and concern for energy resources began to rise, not only in the United States, but in other areas of the world as well.

As the effort increased to identify all possible means for

<sup>3</sup> Anderson, J. Hilbert and James H. Anderson, Jr. - "Thermal Power From Sea Water," Mechanical Engineering, Vol. 88:4, pp 41-46(April 1966).

reaching energy independence, the National Science Foundation, through its Research Applied to National Needs (RANN) program, began funding a group of projects in the solar energy area which became known as the Ocean Thermal Energy Conversion (OTEC) program. Starting with a modest \$84,100 program in Fiscal Year 1972 the funding increased to \$730,700 in FY 1974 and to approximately \$3 million in FY 1975.<sup>4</sup> Budget Authority for FY 1976 was set at \$8.1 million and Budget Obligation at \$6 million.<sup>5</sup>

An important segment of the FY'75 NSF/RANN program was made up of two industrial team studies: one by a Lockheed, Bechtel, T.Y. Lin International consortium, and the other by the Ocean and Energy Systems group of TRW, Inc. along with Global Marine Development, Inc. and United Engineers & Constructors.<sup>6,7</sup> These studies were essentially engineering evaluations of a basic design for an OTEC plant and concluded generally, that, while problems remained to be solved, plants of 100 megawatts capacity were feasible within the range of current technology. At the same time these rather

<sup>4</sup>Cohen, Robert "OTEC Program Overview," Proceedings, Third Workshop on Ocean Thermal Energy Conversion, APL/JHU SR75-2, p 47 (Aug 1975)

<sup>5</sup>ERDA 76-1 Volume 2: Program Implementation, "A National Plan for Energy Research, Development & Demonstration: Creating Energy Choices for the Future" p 110 (1976)

<sup>6</sup>Trimble, Lloyd C., B.L. Messinger, H.G. Ulbrick, Geoffrey Smith and T.Y. Lin, "Ocean Thermal Energy Conversion System Study," Proceedings, Third Workshop on Ocean Thermal Energy Conversion, APL/JHU SR 75-2, pp 3-21

<sup>7</sup>Douglass, Robert H., "Ocean Thermal Energy Conversion: An Engineering Evaluation," Proceedings, Third Workshop on Ocean Thermal Energy Conversion, APL/JHU SR 75-2, pp 22-38 (Aug 1975).

large studies were in progress, a number of smaller efforts on component configurations, component materials and design choices, site selection criteria and many others were being carried on. On January 19, 1975 the Energy Research and Development Administration was formed and its Division of Solar Energy was given responsibility for certain aspects of the nation's energy program, among which were the developmental aspects of the OTEC program. OTEC Workshops were held in 1973, 1974, 1975, and 1977 for the purpose of reviewing the status of the program and assessing the direction of future efforts. They also afforded newcomers to the program opportunities to become acquainted with what had already been accomplished and to discuss current problems with people already involved in the effort.

#### 4. Navy OTEC Involvement

Supplies and cost of fossil fuels, particularly oil, are of paramount interest to the U.S. Navy. From both operational and economic standpoints, it is imperative that the Navy have first-hand knowledge not only of near-term energy supplies and costs, but also of long-range plans and projections. In this regard it is also to the Navy's advantage to be aware of alternate energy sources and any impact they may have on the national (and even international) energy supply. For these reasons and the fact that the Navy has

perhaps the largest store of information and expertise available regarding operations on, under, and above the oceans, the Navy has become involved with and given administrative support to certain facets of OTEC development.

The Naval Research Laboratory in Washington, D. C. has been actively engaged in studying the environmental effects of placing an OTEC plant in ocean waters. Because of the huge quantities of both warm and cold waters used by a large capacity plant, opportunities exist for both beneficial and deleterious disturbances to the various eco-systems involved. Additional information is needed and is actively being sought. Wake effects from such a plant, whether anchored or moving, are another area of concern being studied by the Naval Research Laboratory.

The Naval Underseas Center, New London, Connecticut, has been assisting with certain aspects of the plant's structural analysis. Both the supporting framework and the cold water pipe are structures of unusual dimensions and present unique problems.

The Naval Postgraduate School is investigating problems of plant platform dynamics. Similitude relationships are being applied to check for circumstances of unusual or critical action.

David Taylor Naval Ship Research Development Center, Annapolis, is working on mechanical cleaning of OTEC heat exchangers.

The Naval Facilities Engineering Command is involved with assisting ERDA on ocean engineering problems and with the administration of some of the many contracts underway on the various aspects of OTEC development. The Civil Engineering Laboratory is working on OTEC anchor development and marine concrete applications.

Still another area of Navy involvement with OTEC is through three of its midshipmen. Tom Frey, a 1974-1975 Naval Academy Trident Scholar, and graduate of the Class of 1975, did much of the fabrication, modification and testing of the operating OTEC model which was eventually demonstrated at the Third Workshop on Ocean Energy Conversion in Houston, Texas on 8 May 1975. Two additional midshipmen, Bruce Montgomery and Gary Hall, U.S. Naval Academy Class of 1976, performed a literature search and did preliminary development work on the parameters involved in the dimensionless ratios used in this simulation study. Their work was part of a senior research elective course in Marine Engineering.

##### 5. Literature Search and Review

Since 1970 a considerable fund of knowledge concerning the many facets of OTEC plants has been developed and published. Much of this activity came in response to NSF (RANN) requests for proposals on particular problems, but some was generated in the private sector as the total OTEC

picture began to take shape. When information from the literature search\* for this project was evaluated, one of the questions most frequently left unanswered was the performance of the heat exchangers. While most

observers agreed that the necessary heat exchangers could be built within the framework of present technology, uncertainty existed regarding performance characteristics of the exchangers in the environment to which they would be subjected. Because economic studies invariably showed that heat exchanger costs would constitute roughly 50%<sup>6</sup> of the total plant cost, and since these heat exchangers are roughly double the size of the largest current heat exchanger of similar style used in power plant work, interested individuals felt that pilot plant tests would have to be made before any real credibility in performance could be achieved. In order to investigate the dimensionless parameters pertinent to this type of heat exchanger simulation, this study was undertaken.

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\*See Reference List

<sup>6</sup>Ibid, page 8

## Chapter II. DIMENSIONAL ANALYSIS

### 1. Techniques of Modeling Through Dimensional Analysis

Dimensional analysis is a mathematical technique involving the formation of dimensionless groups of related physical properties. This method is useful in simplifying a problem by combining into dimensionless ratios or groups the applicable variables. These groups can then serve as a good basis for construction of a model.

The technique is based on the fact that any physical property or quantity can be expressed in a small number of fundamental dimensions. For the purpose of this study, four fundamental dimensions are used, which are as follows:

1. Length (L)
2. Time (T)
3. Mass (M)
4. Temperature ( $\theta$ )

The larger the number of variables involved in the system in relation to the number of dimensions used, the greater will be the number of dimensionless groups formed. It must be fully realized that the correct formation of a dimensional group in no way insures the physical correctness of the parameters chosen to construct the dimensionless groups, nor does it insure that all necessary parameters describing the system have been included. Finally, the

groups which are eventually chosen must be verified by experimental evidence and investigative experience.

The theoretical basis which has been chosen to develop dimensionless groups from a list of parameters is Buckingham's Pi theorem. Because of its overall importance, it is stated below.

Buckingham's Pi Theorem: If a physical equation exists among "n" parameters, it may be equivalently expressed as an equation among (n-K) dimensionless groups of these parameters, where K= number of fundamental dimensions involved in the "n" parameters. If Q denotes the physical parameters and  $\pi$  denotes the dimensionless combinations of some of the Q's, the theorem states that a functional relation of the form,

$$f(Q_1, Q_2, Q_3, Q_4, \dots, Q_n) = 0$$

may be expressed as the following function of dimensionless groups:

$$g(\pi_1, \pi_2, \pi_3, \pi_4, \dots, \pi_{n-K}) = 0.$$

Each term  $\pi$  is of the form  $\pi = Q_1^a Q_2^b Q_3^c \dots Q_n^z$ , where the exponents a, b, c, ..., z are such that  $\pi$  has no dimensions, some exponents being zero.

The group of  $\pi$  terms must form a complete set.

In forming a set of dimensionless groups certain rules must be followed. If the system involves "n" parameters and "K" fundamental dimensions, then the following conditions must be met by the fundamental quantities, of which there are

the same number as the fundamental dimensions:

- (1) product of the "K" fundamental quantities must not be dimensionless;
- (2) the "K" fundamental quantities contain all involved dimensions;
- (3) no two of the "K" fundamental quantities can have identical dimensions.

With these rules obeyed, a total of " $n-K$ "  $\pi$  terms can be formed, each  $\pi$  term consisting of a product of " $K+1$ " parameters, " $K$ " of which are the fundamental quantities. Each parameter is raised to an unknown power. Therefore, each  $\pi$  term has  $K$  unknowns, namely the exponents to the " $K$ " fundamental quantities. Since " $K$ " fundamental dimensions are involved in the product, " $K$ " simultaneous equations can be formed with each equation expressed in exponents of only one dimension. With " $K$ " equations and " $K$ " unknowns, the desired exponents that will produce a dimensionless group can be found.

The following page illustrates this technique for an arbitrary system consisting of 7 parameters and 3 fundamental dimensions:

EXAMPLE: METHOD OF SYSTEMATIC CALCULATION FOR DIMENSIONAL ANALYSIS

$$\text{System: } f(Q_1, Q_2, Q_3, \dots, Q_7) = 0$$

$$n = 3 + 4 \quad K = 3 \quad n-K = 4$$

$$\pi_1^1 = Q_1^{a_1} Q_2^{b_1} Q_3^{c_1} Q_4^{d_1}$$

$$\pi_2^1 = Q_1^{a_2} Q_2^{b_2} Q_3^{c_2} Q_5^{d_2}$$

$$\pi_3^1 = Q_1^{a_3} Q_2^{b_3} Q_3^{c_3} Q_6^{d_3}$$

$$\pi_4^1 = Q_1^{a_4} Q_2^{b_4} Q_3^{c_4} Q_7^{d_4}$$

We can choose  $d_1 = d_2 = d_3 = d_4 = 1$ , since the root of a dimensionless quantity is dimensionless.

$$\pi_1 = Q_1^{a_1} Q_2^{b_1} Q_3^{c_1} Q_4$$

$$\pi_2 = Q_1^{a_2} Q_2^{b_2} Q_3^{c_2} Q_5$$

$$\pi_3 = Q_1^{a_3} Q_2^{b_3} Q_3^{c_3} Q_6$$

$$\pi_4 = Q_1^{a_4} Q_2^{b_4} Q_3^{c_4} Q_7$$

For each dimensionless term  $\pi$ , the product involves 3 dimensions which must all cancel out. Therefore, for each  $\pi$ , three simultaneous equations, one for each dimension, exist; along with three unknowns;  $a_i, b_i, c_i$ ;

$i = 1, 2, 3, 4$ . Hence, the various  $a$ 's,  $b$ 's,  $c$ 's can be found and the dimensionless group is formed.

## 2. Determination of Significant Parameters

The list of significant parameters has been obtained by considering the energy equation that would be applicable to a condenser or evaporator. From these considerations the significant parameters have been separated into two groups: Those parameters applicable to the working fluid side of the condenser and evaporator (Table 2.1); and those important on the sea water side of the condenser and evaporator (Table 2.2). In constructing Tables 2.1 and 2.2, it was assumed that the heat exchangers (evaporator and condenser) would be of the shell and tube type. Fins on the heat exchanger tubes were not considered.

TABLE 2.1

Significant Parameters for Working  
Fluid Side of Condenser or Evaporator

| <u>Parameter</u>                                                 | <u>Symbol</u>     | <u>Dimensions</u>      |
|------------------------------------------------------------------|-------------------|------------------------|
| 1. Shell diameter                                                | $d_s$             | L                      |
| 2. Hydraulic diameter (working fluid side)                       | $d_o$             | L                      |
| 3. Tube length                                                   | $z$               | L                      |
| 4. Tube roughness                                                | $f$               | L                      |
| 5. Tube thermal conductivity                                     | $K_1$             | $MLT^{-3}\theta^{-1}$  |
| 6. Velocity of working fluid                                     | $V_{WF}$          | $LT^{-1}$              |
| 7. Density of working fluid                                      | $\rho_{WF}$       | $ML^{-3}$              |
| 8. Change in working fluid density across heat exchanger         | $\Delta\rho_{WF}$ | $ML^{-3}$              |
| 9. Viscosity of working fluid                                    | $\mu_{WF}$        | $MT^{-1}L^{-1}$        |
| 10. Pressure drop through working fluid side of heat exchanger   | $\Delta P_{WF}$   | $MT^{-2}L^{-1}$        |
| 11. Convective heat transfer coefficient of working fluid        | $h_{WF}$          | $MT^{-3}\theta^{-1}$   |
| 12. Specific heat of working fluid                               | $C_{p\ WF}$       | $L^2T^{-2}\theta^{-1}$ |
| 13. Thermal conductivity of working fluid                        | $K_{WF}$          | $MLT^{-3}\theta^{-1}$  |
| 14. Change in temperature of working fluid across heat exchanger | $\Delta T_{WF}$   | $\theta$               |
| 15. Acceleration of gravity                                      | $g$               | $LT^{-2}$              |
| 16. Heat of vaporization of working fluid                        | $h_{fg}$          | $L^2T^{-2}$            |
| 17. Condenser pressure, working fluid                            | $P_{c,WF}$        | $MT^{-2}L^{-1}$        |

TABLE 2.1 (cont'd)

| <u>Parameter</u>                                      | <u>Symbol</u>     | <u>Dimensions</u> |
|-------------------------------------------------------|-------------------|-------------------|
| 18. Density of saturated liquid, working fluid        | $\rho_{l,WF}$     | $ML^{-3}$         |
| 19. Density of saturated vapor, working fluid         | $\rho_{g,WF}$     | $ML^{-3}$         |
| 20. Surface tension of working fluid                  | $\sigma_{WF}$     | $MT^{-2}$         |
| 21. Log mean temperature difference in heat exchanger | $\Delta T_{LMTD}$ | $\theta$          |

TABLE 2.2

Significant Parameters for Sea Water  
Side of Condenser or Evaporator

| <u>Parameter</u>                                                     | <u>Symbol</u>     | <u>Dimensions</u>      |
|----------------------------------------------------------------------|-------------------|------------------------|
| 1. Shell diameter                                                    | $d_s$             | L                      |
| 2. Tube inside diameter                                              | $d_i$             | L                      |
| 3. Tube length                                                       | $l$               | L                      |
| 4. Tube roughness                                                    | $f$               | L                      |
| 5. Tube thermal conductivity                                         | $K_i$             | $MLT^{-3}\theta^{-1}$  |
| 6. Fouling on sea water side of heat exchanger                       | $F$               | L                      |
| 7. Velocity of sea water                                             | $V_{SW}$          | $LT^{-1}$              |
| 8. Ocean current velocity                                            | $V_c$             | $LT^{-1}$              |
| 9. Density of sea water                                              | $\rho_{SW}$       | $ML^{-3}$              |
| 10. Change in density of sea water across heat exchanger             | $\Delta\rho_{SW}$ | $ML^{-3}$              |
| 11. Viscosity of sea water                                           | $\mu_{SW}$        | $MT^{-1}L^{-1}$        |
| 12. Sea water pressure drop through heat exchanger                   | $\Delta P_{SW}$   | $MT^{-2}L^{-1}$        |
| 13. Sea water convective heat transfer coefficient                   | $h_{SW}$          | $MT^{-3}\theta^{-1}$   |
| 14. Specific heat of sea water                                       | $C_{PSW}$         | $L^2T^{-2}\theta^{-1}$ |
| 15. Thermal conductivity of sea water                                | $K_{SW}$          | $MLT^{-3}\theta^{-1}$  |
| 16. Change in average temperature of sea water across heat exchanger | $\Delta T_{SW}$   | $\theta$               |

TABLE 2.2 (cont'd)

| <u>Parameters</u>                                     | <u>Symbol</u>     | <u>Dimensions</u> |
|-------------------------------------------------------|-------------------|-------------------|
| 17. Acceleration of gravity                           | $g$               | $LT^{-2}$         |
| 18. Sea water inlet pressure                          | $P_{i,SW}$        | $MT^{-2}L^{-1}$   |
| 19. Inlet temperature of sea water                    | $T_i$             | $\theta$          |
| 20. Log mean temperature difference in heat exchanger | $\Delta T_{LMTD}$ | $\theta$          |

### 3. Formation of Dimensionless Groups

It is here where the concept of dimensional analysis becomes invaluable. Through the use of Buckingham's Pi Theorem, the twenty-one parameters in Table 2.1 can be combined into seventeen dimensionless groups, and the twenty parameters in Table 2.2 can be arranged into sixteen groups.

The dimensionless groups formed depend on the fundamental quantities chosen. Any fundamental quantities can be picked so long as they obey the three rules stated in Section 1 of Chapter II. It is also allowable to pick different sets of fundamental quantities. Thus, in principle, one could form nearly an infinite set of dimensionless groups from the parameters in Tables 2.1 and 2.2. However, most of the groups formed would have an unimportant physical significance. Thus, the fundamental quantities finally chosen were those leading to commonly used dimensionless groups, which have a known physical significance.

The fundamental quantities chosen were:

|                                               | dimensions            |
|-----------------------------------------------|-----------------------|
| (1) Characteristic diameter of heat exchanger | $L$                   |
| (2) Fluid velocity                            | $LT^{-1}$             |
| (3) Fluid density                             | $ML^{-3}$             |
| (4) Fluid thermal conductivity                | $MLT^{-3}\theta^{-1}$ |

These fundamental quantities satisfy the criteria:

- (1) their product is not dimensionless (product has dimensions of  $M^2T^{-4}\theta^{-1}$ )

- (2) they contain all 4 fundamental dimensions (M,L,T,θ)
- (3) no two of the fundamental quantities have identical dimensions.

A sample calculation of the formation of the Reynolds number from the four fundamental quantities and the viscosity parameter is shown below to illustrate the dimensional analysis technique.

Applying Buckingham's  $\Pi$  theorem to  $\mu_{WF}$  (parameter #9 in Table 2.1) gives

$$\Pi_9 = d_o^{a_1} V_{WF}^{b_1} \rho_{WF}^{c_1} K_{WF}^{d_1} \mu_{WF}$$

where  $a_1, b_1, c_1, d_1$ , are unknown exponents.

However, since the group is dimensionless, each of the fundamental dimensions (length, mass, time and temperature) must have its exponents sum to zero. Thus, the following equation can be written for length:

$$a_1 + b_1 - 3c_1 + d_1 - 1 = 0 \quad (2.1)$$

The coefficients "1" in front of  $a_1, b_1$ , and  $d_1$  represent the fact that length, velocity, and thermal conductivity are proportional to length. The coefficient "-3" before  $c_1$  represents the fact that density is inversely proportional

to volume, which is length cubed. Finally, viscosity has units of inverse length, so its exponent is -1.

Similarly, the equation for mass can be written as

$$c_1 + d_1 + 1 = 0 \quad (2.2)$$

The equation for time is

$$-b_1 - 3d_1 - 1 = 0 \quad (2.3)$$

Finally, the equation for temperature gives

$$-d_1 = 0 \quad (2.4)$$

Solving back for the unknown coefficients gives

$$d_1 = 0, \quad b_1 = -1, \quad c_1 = -1, \quad \text{and } a_1 = -1 \quad (2.5)$$

$$\text{thus } \Pi_9 = \frac{\mu_{WF}}{d_o V_{WF} \rho_{WF}} \quad (2.6)$$

Since dimensionless ratios can be inverted,  $\Pi_9$  finally becomes

$$\Pi_9 = \frac{d_o V_{WF} \rho_{WF}}{\mu_{WF}} \quad (2.7)$$

which is recognized as the Reynolds number.

Table 2.3 lists the dimensionless groups applicable to the working fluid side of the condenser or evaporator that resulted from the parameter list of Table 2.1. The fundamental quantities chosen were the working fluid heat exchanger's hydraulic diameter and the working fluid's velocity, density, and thermal conductivity.  $\Pi_1$  and  $\Pi_3$  are basic geometry scale parameters.  $\Pi_4$  is the scale factor which accounts for tube roughness.  $\Pi_5$  is the ratio

of the tube thermal conductivity to the working fluid conductivity.  $\Pi_8$ ,  $\Pi_{18}$ ,  $\Pi_{19}$  are density scale parameters.  $\Pi_9$  is the Reynolds number of the working fluid, which is the ratio of the inertial to viscous forces in the working fluid.  $\Pi_{10}$  is the Euler number, which is the ratio of the pressure forces to twice the kinetic energy of the working fluid.  $\Pi_{11}$  is the Nusselt number.  $\Pi_{12}$  is the Peclet number, which is the product of the Reynolds and Prandtl numbers. However, it has been placed in parenthesis since the Reynolds number is already a dimensionless ratio. Thus, the only ratio which needs to be scaled is the Prandtl number of the working fluid. This is the ratio used in the remainder of the report.

$\Pi_{14}$  is the Clausius number which is the Brinkman number divided by the Reynolds number. As mentioned above, since the Reynolds number is already a dimensionless ratio, the only independent group in  $\Pi_{14}$  is the Brinkman number. The Brinkman number is a ratio of viscous heating divided by the heat transported by conduction.  $\Pi_{15}$  is the Froude number which represents the ratio of inertial to gravitational forces.  $\Pi_{16}$  is a ratio of the energy for phase change to the kinetic energy of the working fluid.  $\Pi_{17}$  and  $\Pi_{21}$  are the Euler and Brinkman numbers again. Finally,  $\Pi_{20}$  is the Weber number, which is the ratio of the surface tension forces to the inertial forces.

Table 2.4 gives the dimensionless groups for the sea water side of the condenser or evaporator based on the parameter list of Table 2.2. Basically, the same dimensionless ratios arise as in Table 2.3 except these ratios are based on the sea water properties. This is because the fundamental quantities chosen to construct this table were, in addition to the tube inside diameter, the sea water's velocity, density, and thermal conductivity. Table 2.4 has been constructed by reducing those terms which contain the Reynolds number.

TABLE 2.3

Dimensionless Groups Applicable to Working  
Fluid Side of Condenser or Evaporator

| <u>Fundamental Quantities</u>                              | <u>Symbol</u> | <u>Dimensions</u>     |
|------------------------------------------------------------|---------------|-----------------------|
| $Q_2$ = Hydraulic diameter of working fluid heat exchanger | $d_o$         | L                     |
| $Q_6$ = Velocity of working fluid                          | $V_{WF}$      | $LT^{-1}$             |
| $Q_7$ = Density of working fluid                           | $\rho_{WF}$   | $ML^{-3}$             |
| $Q_{13}$ = Thermal conductivity of working fluid           | $K_{WF}$      | $MLT^{-3}\theta^{-1}$ |

Dimensionless Groups

$$\left. \begin{aligned} \Pi_1 &= \frac{d_s}{d_o} \\ \Pi_3 &= \frac{l}{d_o} \\ \Pi_4 &= \frac{f}{d_o} \end{aligned} \right\}$$

Geometry Parameters

$$\Pi_5 = \frac{K_1}{K_{WF}}$$

Ratio of Conductivities

$$\Pi_8 = \frac{\Delta\rho_{WF}}{\rho_{WF}}$$

Density Ratio

$$\Pi_9 = \frac{d_o V_{WF} \rho_{WF}}{\mu_{WF}}$$

Reynolds Number of Working Fluid

TABLE 2.3 (cont'd)

Dimensionless Groups

$$\Pi_{10} = \frac{\Delta P_{WF}}{\rho_{WF} V_{WF}^2} \quad \text{Euler Number}$$

$$\Pi_{11} = \frac{h_{WF} d_o}{K_{WF}} \quad \text{Nusselt Number}$$

$$\Pi_{12} = \frac{d_o V_{WF} \rho_{WF} C_{pWF}}{K_{WF}} \quad \text{Peclet Number which reduces to}$$

$$\Pi_{12} = \frac{C_{pWF} \mu_{WF}}{K_{WF}} \quad \text{Prandtl Number}$$

$$\Pi_{14} = \frac{K_{WF} \Delta T_{WF}}{V_{WF} \rho_{WF} d_o} \quad \text{Clausius Number which reduces to}$$

$$\Pi_{14} = \frac{\mu_{WF} V_{WF}^2}{K_{WF} \Delta T} \quad \text{Brinkman Number}$$

$$\Pi_{15} = \frac{V_{WF}^2}{g d_o} \quad \text{Froude Number}$$

$$\Pi_{16} = \frac{h_{fg}}{V_{WF}^2} \quad \frac{\text{Energy for Phase Change}}{\text{Kinetic Energy}}$$

$$\Pi_{17} = \frac{P_{c, WF}}{\rho_{WF} V_{WF}^2} \quad \text{Euler Number}$$

TABLE 2.3 (cont'd)

Dimensionless Groups

$$\Pi_{18} = \frac{\rho_{l,WF}}{\rho_{WF}}$$

$$\Pi_{19} = \frac{\rho_{g,WF}}{\rho_{WF}}$$

Density Ratios

$$\Pi_{20} = \frac{\sigma}{\rho_{WF} V_{WF}^2 d_o}$$

Weber Number

$$\Pi_{21} = \frac{K_{WF} \Delta T_{LMTD}}{V_{WF}^3 \rho_{WF} d_o}$$

Clausius Number which reduces to

$$\Pi_{21} = \frac{\mu_{WF} V_{WF}^2}{K_{WF} \Delta T_{LMTD}}$$

Brinkman Number

TABLE 2.4

Dimensionless Groups Applicable to Sea Water  
Side of Condenser or Evaporator

| <u>Fundamental Quantities</u>                | <u>Symbol</u> | <u>Dimensions</u>     |
|----------------------------------------------|---------------|-----------------------|
| $Q_2$ = Tube inside diameter                 | $d_i$         | L                     |
| $Q_7$ = Velocity of sea water                | $V_{SW}$      | $LT^{-1}$             |
| $Q_9$ = Density of sea water                 | $\rho_{SW}$   | $ML^{-3}$             |
| $Q_{15}$ = Thermal conductivity of sea water | $K_{SW}$      | $MLT^{-3}\theta^{-1}$ |

Dimensionless Groups

$$\left. \begin{aligned} \Pi_1 &= \frac{d_s}{d_i} \\ \Pi_3 &= \frac{l}{d_i} \\ \Pi_4 &= \frac{f}{d_i} \end{aligned} \right\}$$

Geometry Parameters

$$\Pi_5 = \frac{K_1}{K_{SW}}$$

Ratio of Conductivities

$$\Pi_6 = \frac{F}{d_i}$$

Fouling Factor

$$\Pi_8 = \frac{V_c}{V_{SW}}$$

Velocity Ratio

TABLE 2.4 (cont'd)

Dimensionless Group

$$\Pi_{10} = \frac{\Delta \rho_{SW}}{\rho_{SW}} \quad \text{Density Ratio}$$

$$\Pi_{11} = \frac{d_i V_{SW} \rho_{SW}}{\mu_{SW}} \quad \text{Reynolds Number}$$

$$\Pi_{12} = \frac{\Delta P_{SW}}{\rho_{SW} V_{SW}^2} \quad \text{Euler Number}$$

$$\Pi_{13} = \frac{h_{SW} d_i}{K_{SW}} \quad \text{Nusselt Number}$$

$$\Pi_{14} = \frac{C_{PSW} \mu_{SW}}{K_{SW}} \quad \text{Prandtl Number}$$

$$\Pi_{16} = \frac{\mu_{SW} V_{SW}^2}{K_{SW} \Delta T_{SW}} \quad \text{Brinkman Number}$$

$$\Pi_{17} = \frac{V_{SW}^2}{g d_i} \quad \text{Froude Number}$$

$$\Pi_{18} = \frac{P_{i,SW}}{\rho_{SW} V_{SW}^2} \quad \text{Euler Number}$$

$$\Pi_{19} = \frac{\mu_{SW} V_{SW}^2}{K_{SW} T_i} \quad \text{Brinkman Number}$$

$$\Pi_{20} = \frac{\mu_{SW} V_{SW}^2}{K_{SW} \Delta T_{LMTD}} \quad \text{Brinkman Number}$$

## 2.4 Reduction of Dimensionless Groups

Tables 2.3 and 2.4 suggest many dimensionless ratios to be scaled in an OTEC model plant. However, certain groups are immediately scaled by using similar materials and fluids in an OTEC model and prototype. Also, some groups will have negligible effect on the heat exchanger's performance.

The dimensionless groups which depend only on physical properties of the working fluid, sea water, or heat exchanger materials are  $\Pi_5$ ,  $\Pi_{12}$ ,  $\Pi_{18}$ ,  $\Pi_{19}$  in Table 2.3 and  $\Pi_5$  and  $\Pi_{14}$  in Table 2.4.

Further, if the tubes are very smooth, then  $\Pi_4$  in Tables 2.3 and 2.4 will have a negligible effect, since  $\Pi_4$  is a measure of tube roughness.

$\Pi_6$  in Table 2.4 is a measure of the fouling in an OTEC plant. It is essential for the success of these plants, that fouling not be significant. Probably chemical or mechanical cleaning will be used to minimize this problem. In any case, if the tubes have not undergone significant fouling then  $\Pi_6$  in Table 2.4 will not be a significant scaling factor.

$\Pi_8$  in Table 2.3 is a measure of the density change in the working fluid. This is primarily controlled by the pressure change across the heat exchanger. The pressure drop is scaled in  $\Pi_{10}$ . Thus, if  $\Pi_{10}$  is scaled, then  $\Pi_8$  will be scaled. Therefore,  $\Pi_8$  will not be further considered. Similarly, in Table 2.4,  $\Pi_{10}$  and  $\Pi_{18}$  are related to  $\Pi_{12}$ .

The Brinkman number appears as  $\Pi_{14}$  and  $\Pi_{21}$  in Table 2.3 and  $\Pi_{16}$ ,  $\Pi_{19}$ , and  $\Pi_{20}$  in Table 2.4. Since the Brinkman number is a measure of the viscous heating in a fluid, it is usually negligible and thus these parameters should have a small effect on the heat exchanger performance.

The Froude number appears as  $\Pi_{15}$  in Table 2.3 and  $\Pi_{17}$  in Table 2.4. However, the Froude number is important only when gravity forces are important. Gravity forces are important if the density changes are large, which is measured by the  $\Pi_8$  parameter in Table 2.3 and  $\Pi_{10}$  in Table 2.4. As mentioned above, these two parameters are effectively scaled if the Euler number is scaled. Thus the Froude number will not be further considered.

The remaining important dimensionless groups needed to scale an OTEC plant are listed in Tables 2.5 and 2.6.

TABLE 2.5

Significant Dimensionless Groups Applicable to Working Fluid Side, Condenser or Evaporators, when similar materials and fluids are used in prototype and model.

Dimensionless Groups

$$\pi_1 = \frac{d_s}{d_o}$$

$$\pi_3 = \frac{\ell}{d_o}$$

$$\pi_9 = \frac{d_o V_{WF} \rho_{WF}}{\mu_{WF}}$$

$$\pi_{10} = \frac{\Delta P_{WF}}{\rho_{WF} V_{WF}^2}$$

$$\pi_{11} = \frac{h_{WF} d_o}{K_{WF}}$$

$$\pi_{16} = \frac{h_{fg}}{V_{WF}^2}$$

$$\pi_{20} = \frac{\sigma}{\rho_{WF} V_{WF}^2 d_o}$$

TABLE 2.6

Significant Dimensionless Groups Applicable to  
Sea Water Side, Condenser or Evaporator, when  
similar materials are used in prototype and model.

Dimensionless Groups

$$\Pi_1 = \frac{d_s}{d_i}$$

$$\Pi_3 = \frac{\ell}{d_i}$$

$$\Pi_8 = \frac{V_c}{V_{SW}}$$

$$\Pi_{11} = \frac{d_i V_{SW} \rho_{SW}}{\mu_{SW}}$$

$$\Pi_{12} = \frac{\Delta P_{SW}}{\rho_{SW} V_{SW}^2}$$

$$\Pi_{13} = \frac{h_{SW} d_i}{K_{SW}}$$

### Chapter III. ANALYSIS OF DIMENSIONLESS GROUPS

#### 1. Simulation Compatibility

In Chapter II the important dimensionless groups governing the thermodynamic performance of an OTEC plant have been developed. These dimensionless groups, which are applicable to the working fluid and sea water of an OTEC condenser or evaporator, are presented in Tables 2.3 and 2.4. These groups can be used to show the effect of scaling parameters between a proposed pilot plant test model and a full scale prototype.

The potential parameters that could be scaled in an OTEC heat exchanger are given in Tables 2.1 and 2.2. Obviously there are an infinite number of combinations of parameters that could be scaled. Investigating all these potential combinations of scaling is beyond the scope of this report. However, an example using a hypothetical test model would illustrate how an investigator could apply these dimensionless groups and thus evaluate the usefulness of such a test model.

#### 2. Example of Dimensionless-Group Compatibility

As an example, consider the OTEC condenser proposed in the Lockheed study.<sup>6</sup> The significant parameters for the proposed prototype plant are given in the first column of Table 3.1, while the second column gives a set of proposed test model parameters. Essentially such a model would be a

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<sup>6</sup>Ibid. page 8

one-tenth ( $\frac{1}{10}$ ) physical scale model of the prototype plant. However, both model and prototype would use ammonia as the working fluid.

To simplify evaluation of the model and prototype condensers several assumptions have been made. They are as follows:

- (1) Condenser is single pass shell and tube type.
- (2) Working fluid, ammonia, flows vertically through condenser while the sea water flows horizontally through condenser tubes as shown in Figure 3.1
- (3) Gravity and momentum changes of working fluid have negligible effect on working fluid pressure drop.
- (4) Ammonia remains in the vapor phase throughout the condenser.

The last assumption has been made to simplify the computation of the heat transfer coefficient and the pressure drop on the working fluid side of the condenser.

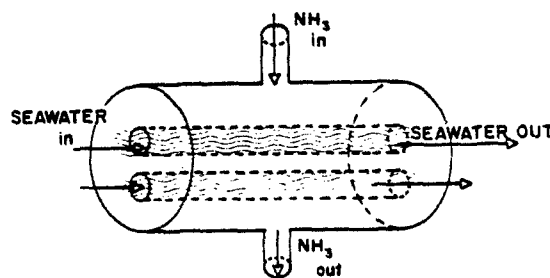


Figure 3.1 Schematic Diagram of Hypothetical OTEC Shell and Tube Condenser.

TABLE 3.1

## Condenser Characteristics

| Parameter                         | Lockheed<br>Prototype | Proposed Model     |
|-----------------------------------|-----------------------|--------------------|
| Shell diameter, ft                | 72.7                  | 7.27               |
| Condenser horizontal length, ft   | 56                    | 56                 |
| Tube diameter, inches             | 2                     | 2                  |
| Number of tubes                   | 120,000               | 1200               |
| Tube surface area, sq.ft.         | $3.47 \times 10^6$    | $3.47 \times 10^4$ |
| NH <sub>3</sub> flow rate, lb/sec | 3700                  | 370                |
| Working fluid                     | NH <sub>3</sub>       | NH <sub>3</sub>    |

Obviously, including condensation would greatly increase the heat transfer coefficient as well as significantly change the pressure drop. Thus the numerical values calculated for these quantities should not be taken to reflect a typical OTEC plant. However, the object of this report is to compare dimensionless groups between a model and prototype. Thus the relative values of these quantities are important, not the magnitudes of the numbers. Also, the calculation of a two-phase pressure drop and heat transfer coefficient is dependent on many variables. For example, the flow regime, the variation of quality with distance through the heat exchanger, the tube configuration, etc. Thus two-phase quantities calculated would not be generally applicable to all OTEC plants.

Based on a shell diameter,  $d_s$ , of 72.7 ft. and 120,000 tubes for the prototype plant and 7.27 ft. shell diameter with 1200 tubes for the model, the quantities given in Table 3.2 were calculated.

TABLE 3.2

| Parameter                                                                          | Lockheed Prototype  | Proposed Model      |
|------------------------------------------------------------------------------------|---------------------|---------------------|
| Cross sectional area of sea water tubes in condenser, $\text{ft}^2$                | $2.62 \times 10^3$  | $2.62 \times 10^1$  |
| Volume of sea water tubes in condenser (assuming 56 ft tube length), $\text{ft}^3$ | $1.467 \times 10^5$ | $1.467 \times 10^3$ |
| Volume of condenser, $\text{ft}^3$                                                 | $2.325 \times 10^5$ | $2.325 \times 10^3$ |
| Volume occupied by working fluid in condenser, $\text{ft}^3$                       | $8.58 \times 10^4$  | $8.58 \times 10^2$  |

It was further assumed that the volume of the working fluid was that of a rectangle with horizontal length of 56 feet and equal height and width. Such a configuration is shown in Figure 3.2.

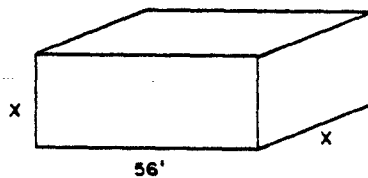


Figure 3.2 Hypothetical Rectangular Volume Occupied by Working Fluid.

Therefore for both the prototype and model

$$\begin{array}{l} \text{Volume occupied by working} \\ \text{fluid in condenser} \end{array} = \chi^2 (56 \text{ ft}) \quad (3.1)$$

Solving for  $\chi$  gives the following results, which are listed in

Table 3.3:

TABLE 3.3

| Parameter                                                                           | Lockheed<br>Prototype | Proposed Model |
|-------------------------------------------------------------------------------------|-----------------------|----------------|
| Working fluid vertical length<br>in condenser ( $\chi$ ), ft                        | 39.1                  | 3.91           |
| Working fluid flow area<br>in condenser ( $\chi \times 56'$ ), $\text{ft}^2$        | 2190                  | 219.0          |
| Working fluid mass flow<br>rate per unit area in<br>condenser, $\text{lb/sec-ft}^2$ | 1.69                  | 1.69           |

The working fluid mass flow rate is found by dividing the ammonia flow rate given in Table 3.1 by the working fluid flow area in the condenser. As can be seen, both the prototype and model have the same mass flow rate per unit area in the condenser,  $1.69 \text{ lb/sec-ft}^2$ .

The Reynolds number is found by using the relationship

$$R_e = \frac{d_o G'}{\mu_{WF}} \quad (3.2)$$

where

$d_o$  = hydraulic diameter, feet

$\mu_{WF}$  = working fluid viscosity  $\text{lb}_m/\text{ft-sec}$ .

$G'$  = mass flow rate per unit area,  $\text{lb}_m/\text{sec-ft}^2$

The hydraulic diameter is based on the condenser tube geometry shown in Figure 3.3 and can be calculated as follows:

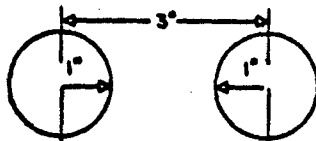


Figure 3.3 Condenser Tube Geometry

$$d_o = \frac{4A}{P} \quad (3.3)$$

Where

A = working fluid cross sectional flow area,  $\text{ft}^2$

P = working fluid wetted perimeter, ft

For the condenser shown in Figure 3.3, the cross sectional flow area and wetted perimeter are

$$A = \frac{(3 \text{ in}) (2 \text{ in}) - \pi (1 \text{ in})^2}{144 \frac{\text{in}^2}{\text{ft}^2}} = 0.0198 \text{ ft}^2$$

$$P = \frac{\pi (2 \text{ in})}{12 \frac{\text{in}}{\text{ft}}} = 0.523 \text{ ft}$$

Thus the hydraulic diameter is 1.81 inches.

If tube geometry is assumed to be the same in both prototype and model, then their respective hydraulic diameters,  $d_o$ , must be equal. Also, if the same working fluid is used in the prototype and test scale model, then the viscosity

must be the same. Therefore, since the prototype and test scale model have the same mass flow rate per unit area, their Reynolds numbers must be equal. The value for these quantities are given in Table 3.4

TABLE 3.4

| Parameter                                                   | Lockheed<br>Prototype | Proposed Model |
|-------------------------------------------------------------|-----------------------|----------------|
| Hydraulic diameter, inches                                  | 1.81                  | 1.81           |
| Working fluid viscosity, $\frac{\text{lb}_m}{\text{ft-hr}}$ | 0.0235                | 0.0235         |
| Reynolds number                                             | 39,049                | 39,049         |

Since the Reynolds number and the tube arrangement are identical in the model and prototype, the pressure drop of the working fluid is calculated as

$$\Delta p_{WF} = f \frac{\rho_{WF} V_{WF}^2 l}{2g d_o} \quad (3.4)$$

where

$f$  = friction factor, dimensionless

$\frac{\rho_{WF} V_{WF}^2}{2g}$  = dynamic head, lb/ft<sup>2</sup>

$\frac{l}{d_o}$  =  $\frac{\text{working fluid vertical length (in condenser)}}{\text{hydraulic diameter}}$

The friction factor,  $f$ , is a function of Reynolds number, so it is identical in model and prototype. Also, the dynamic head is a function of Reynolds number and tube arrangements, so it is equal in both model and prototype.

Thus

$$\frac{\Delta p_{WF} \text{ Prototype}}{\Delta p_{WF} \text{ Model}} = \frac{\frac{l}{d_o} \text{ Prototype}}{\frac{l}{d_o} \text{ Model}} \quad (3.5)$$

Since in this example the prototype has a vertical length to diameter ratio 10 times that of the model, then

$$\frac{\Delta p_{WF} \text{ Prototype}}{\Delta p_{WF} \text{ Model}} = 10 \quad (3.6)$$

The consequence of equation (3.6) is that the Euler number,  $\Pi_{10}$ , won't be scaled between the model and prototype.

The Euler number is given by

$$\text{Euler number} = \Pi_{10} = \frac{\Delta p_{WF}}{\rho_{WF} V_{WF}^2} \quad (3.7)$$

Because the density ( $\rho_{WF}$ ) and the velocity of the working fluid ( $V_{WF}$ ) are the same in model and prototype, the ratio of the Euler numbers is given by

$$\frac{\text{Euler number (Prototype)}}{\text{Euler number (Model)}} = 10 \quad (3.8)$$

The heat transfer coefficient,  $h_{WF}$ , was also found for both the model and the prototype. The results were obtained using appropriate graphs from a heat exchanger design book.<sup>8</sup> The results are given in Table 3.5.

<sup>8</sup>Fraas and Ozisik, Heat Exchanger Design; John Wiley & Sons, Inc. (1965).

TABLE 3.5

| Parameter                                                         | Lockheed<br>Prototype            | Proposed<br>Model                |
|-------------------------------------------------------------------|----------------------------------|----------------------------------|
| Heat transfer<br>coefficient (ammonia<br>working fluid $h_{WF}$ ) | 10.13 Btu/hr-ft <sup>2</sup> -°F | 11.13 Btu/hr-ft <sup>2</sup> -°F |
| $\frac{l}{d_o}$ (working fluid vertical<br>length in condenser)   | 259.2                            | 25.92                            |

Basically the difference in the heat transfer coefficients between the prototype and the model was due to the different  $\frac{l}{d_o}$  ratios. Thus

$$\frac{h_{WF} \text{ Prototype}}{h_{WF} \text{ Model}} = 0.91 \quad (3.9)$$

The consequence of equation (3.9) is that the Nusselt number  $Nu_{11}$ , won't be scaled between the model and the prototype. The Nusselt number is given by

$$\text{Nusselt number} = \frac{h_{WF} d_o}{K_{WF}} \quad (3.10)$$

The hydraulic diameter,  $d_o$ , and working fluid thermal conductivity,  $K_{WF}$ , are the same in the model and prototype. Thus the ratio of Nusselt numbers between these models is given by

$$\frac{\text{Nusselt number (Prototype)}}{\text{Nusselt number (Model)}} = \frac{h_{WF} \text{ Prototype}}{h_{WF} \text{ Model}} \quad (3.11)$$

Then using equation (3.9) the ratio of Nusselt numbers becomes

$$\frac{\text{Nusselt number (Prototype)}}{\text{Nusselt number (Model)}} = 0.91 \quad (3.12)$$

TABLE 3.6

Summary of Dimensionless Group Ratios in an OTEC Condenser from Model in Chapter 3

| Dimensionless Group Ratios from Table 2.5                   | Dimensionless Group                        | Significance                              | Ratio                           |
|-------------------------------------------------------------|--------------------------------------------|-------------------------------------------|---------------------------------|
| $\frac{\Pi_1 \text{ Prototype}}{\Pi_1 \text{ Model}}$       | $\frac{d_s}{d_o}$                          | Geometry parameter                        | 10                              |
| $\frac{\Pi_3 \text{ Prototype}}{\Pi_3 \text{ Model}}$       | $\frac{L}{d_o}$                            | Geometry parameter                        | 10                              |
| $\frac{\Pi_9 \text{ Prototype}}{\Pi_9 \text{ Model}}$       | $\frac{d_o V_{WF} \rho_{WF}}{\mu_{WF}}$    | Reynolds number                           | 1                               |
| $\frac{\Pi_{10} \text{ Prototype}}{\Pi_{10} \text{ Model}}$ | $\frac{\Delta p_{WF}}{\rho_{WF} V_{WF}^2}$ | Euler number                              | 10                              |
| $\frac{\Pi_{11} \text{ Prototype}}{\Pi_{11} \text{ Model}}$ | $\frac{h_{WF} d_o}{K_{WF}}$                | Nusselt number                            | 0.91                            |
| $\frac{\Pi_{16} \text{ Prototype}}{\Pi_{16} \text{ Model}}$ | $\frac{h_{fg}}{V_{WF}^2}$                  | Energy for phase change<br>Kinetic energy | Model does not allow evaluation |
| $\frac{\Pi_{20} \text{ Prototype}}{\Pi_{20} \text{ Model}}$ | $\frac{\sigma}{\rho_{WF} V_{WF}^2 d_o}$    | Surface tension<br>Kinetic energy         | 1                               |

Since the model does not account for change in phase,  $\Pi_{16}$  cannot be evaluated.

Finally, since the model and prototype have the same velocity and hydraulic diameter,  $\Pi_{20}$  will be scaled since  $\sigma$  and  $\rho_{WF}$  are properties of the fluid used and the fluid is assumed to be the same in the prototype and model.

Thus the important dimensionless groups, listed in Table 2.5, have been calculated in this section for a prototype and proposed model. The results are summarized in Table 3.6. The conclusions are discussed in the next chapter.

## Chapter IV CONCLUSIONS

Dimensional analysis was used to form dimensionless groups controlling heat transfer performance of an OTEC condenser or evaporator. These groups are presented in Tables 2.5 and 2.6 and are applicable to heat exchanger model and prototype comparison.

In Chapter III these dimensionless groups were analyzed on the working fluid side for a hypothetical prototype and a model OTEC condenser. The model was assumed to have one-tenth ( $\frac{1}{10}$ ) the shell diameter of the prototype, but the sea water tube diameter and condenser length were equal for both the prototype and the model. Thus,  $\Pi_1$  was scaled by a factor of 10. In addition, the working fluid flow rate was chosen such that mass flow rates per unit area between the prototype and the model were equal. Thus both the model and the prototype had the same Reynolds number ( $\Pi_9$ ).

Several important groups did not scale. The most important was the Euler number ( $\Pi_{10}$ ) which did not scale by a factor of 10. Another important group was the Nusselt number ( $\Pi_{11}$ ) which did not scale, but by less than 10%. However, the magnitude of this difference may be quite different in an actual OTEC plant, because of the assumptions used in calculating the heat transfer coefficient.

Further, if the test model had been chosen to scale the Euler number, then it could be shown that the Reynolds number would not scale (unless of course the prototype and test model were the same size). Thus it is evident that an inherent scaling problem exists between the Reynolds number and the Euler number and that there is also a scaling problem with the Nusselt number between OTEC model and prototype heat exchangers.

This analysis points up scaling difficulties only in the heat exchangers. Since scaling difficulties are shown to be present, it would probably be beneficial to perform a similar analysis on other thermal cycle components. This would lead to greater confidence in application of test results to a full scale OTEC plant.

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