Theoretical Analysis of Heat Exchangers in an Ocean Thermal Energy Conversion System

by

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A Thesis Submitted to the Faculty of the
College of Engineering
In Partial Fulfillment of the Requirements for the Degree of
Master of Science in Engineering

Florida Atlantic University
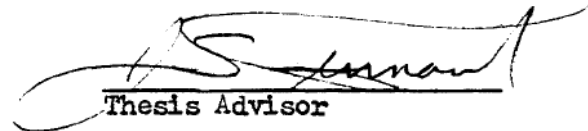**Boca Raton, Florida**
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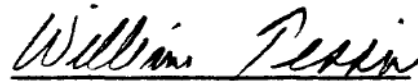**Updated October 2017**

Theoretical Analysis of Heat Exchangers in an
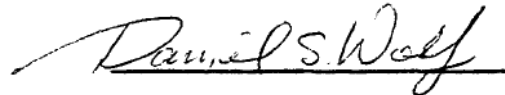# Ocean Thermal Energy Conversion System

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This was prepared under the direction of the candidate's thesis advisor, Dr. Jeffrey s. Tennant, Department of Ocean Engineering, and has been approved by the members of his supervisory committee. It was submitted to the faculty of the College of Engineering and was accepted in partial fulfillment of the requirements for the degree of Master of Science in Engineering.
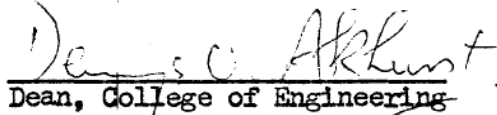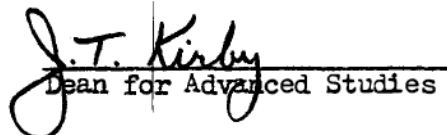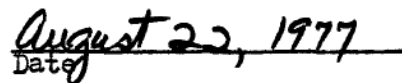
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Preface

Forty years have elapsed since I submitted this work to the graduate school. The analysis, derivations, equations, and data are still relevant and useful, but so much has happened since the invention of microcomputers. The areas of this study most greatly impacted are the graphics, data reductions, and document formatting, especially the equations. It is for these reasons that I have updated this document in hopes that the information contained herein may be more useful to future researchers. I have changed as little as possible in order to retain the original content. I have updated the thermodynamic properties of Ethane, as there may have been one or more typographical errors in the coefficients.

Acknowledgements

Abstract

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A theoretical study was conducted to determine the relationships between the major controlling factors in the heat exchangers in an Ocean Thermal Energy Conversion system. A digital computer model was developed to simulate and analyze the system. Variations in the thermophysical properties of the seawater and working fluid were considered in the analysis. Effects of variation in excess temperature differential on phase change heat transfer rates were also considered. The net power output of the system was determined from a Rankine cycle analysis. The results of this investigation show the necessity of a Rankine cycle analysis and. the inclusion of fluid property variations. A significant difference can be i seen in the net power output of such a system per dollar invested in the heat exchangers projected by this analysis and the analysis of other investigators that have not considered these factors.

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Chapter 1. Introduction

The utilization of the thermal gradient in the oceans as a replenishable natural energy source may soon be economically feasible. The advances in technology, especially in the areas of fabrication and implementation, brought about by the recent thrust of development in nuclear power and offshore petroleum exploration have made production of a full scale system possible. In fact, several preliminary designs have already been presented (1)[*].

Due to the hostile environment of the oceans, it is recognized that the most costly - and perhaps critical in design - components in an Ocean Thermal Energy Conversion (OTEC) system would be the heat exchangers. Constant exposure to corrosive elements and bio-degraders demand special design considerations. Unfortunately corrosion and biofouling present the greatest problem in the warm surface waters that are so necessary to the economic operation of an OTEC system (2,3).

Overview

The thermal energy converter studied in this investigation is a system which will take advantage of the difference in temperature of the warm surface waters and the colder waters of the deep. In the tropical areas surrounding South Florida and the Caribbean, the temperature of the surface waters (within the first 100 feet) varies from 76 to 82F. A temperature of 78F is given by Smith (4) as a yearly average to be expected off South Florida where installation of a prototype is most likely. Although Smith gives a value of 46F for the lower temperature at a depth of 1200 feet, other investigators (1) have used from 38 to 46.5F. Primarily the value of 46F was used in this investigation; however, some analysis was carried out using both 46 and 38F for comparison.

The principle behind the operation of such a system is the ex- traction of heat from the warmer waters to evaporate a fluid. This fluid is expanded in a turbine that drives electric generators. The fluid leaving the turbine is then condensed, rejecting its heat to the colder water (which is pumped up from the deep). Finally, the fluid is compressed and then introduced again into the evaporator. A schematic representation of the system appears in Figure 1.

Objectives of the Study

Ascertaining the controlling factors in the overall heat transfer mechanism in the OTEC heat exchangers and affecting a scheme for optimization would make possible a more intelligent design of the system. The development of a mathematical model that will describe the system is the first step in effecting this optimization.

The specific objectives of the study are to develop a model of the system and to use this model to determine the extent that the con- trolling factors affect the system. This model will be used to:

1. determine the interaction between these controlling factors,
2. establish a scheme for optimization of these factors,
3. outline basic design criteria,
4. project the system's potential and limitations,
5. and demonstrate the need for further research and experimentation explicitly.

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[*] Numbers in parenthesis correspond to references listed at the end.
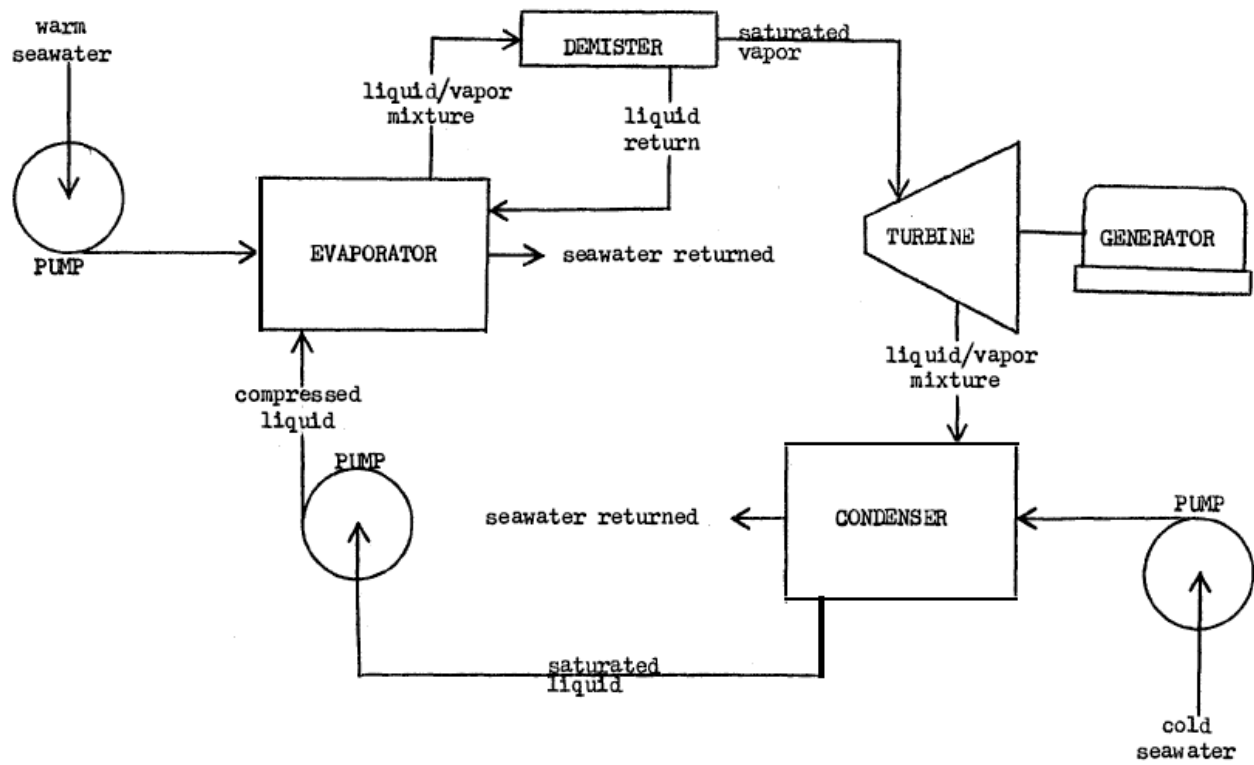
Figure 1. Schematic Representation of OTEC System

Chapter 2. Description of General System

Description of the various possible designs of an overall OTEC system would be too lengthy to include here. The intent of this investigation is to study only the heat exchangers and those subsystems which are directly related. These subsystems being the seawater pumps, recirculating pump, and the turbine. It is the description and modeling of these subsystems that the formulation of the model deals with specifically.

Geometric Considerations

In this investigation only three basic heat exchanger configuration were considered& a shell and tube (in which the seawater flows inside the tubes and the working fluid surrounds the tubes in a 'pool' situation) , a shell and tube (in which the seawater flows inside the tubes and the working fluid is sprayed from above and falls over the tubes) , and an extruded plastic honeycomb (where both the seawater and working fluid flow in closed passages).

The geometric variations of the shell and tube heat exchangers considered in the analysis were length and diameter of the tubes. Only one geometric configuration of the honeycomb heat exchangers was analyzed. This configuration appears in Reference 1.

The external parameters considered in the analysis are the depth from which the cold water must be raised, heat exchanger material, and required tube wall thickness. A depth of 1200 feet was chosen, as it is representative of the values given in Reference 1 as well as the figure for the depth at which the average temperature is 46F in Reference 4. The heat exchanger materials investigated were titanium, aluminum, and the plastic of the honeycomb heat exchangers. The required wall thickness in the case of the shell and tube heat exchangers was determined from the maximum pressure that would be experienced in the system and the corrosion rate. The method used is given in Appendix VTII.

Hydrodynamic Considerations

The hydrodynamic considerations of the analysis were& effects of geometry on flow characteristics, velocity of the seawater in the heat exchangers, Reynolds number, heat transfer coefficient, and pumping work. It is the aim of this investigation to determine the relationships between these hydrodynamic parameters and affect a maximization scheme that will trade off desired and undesired performance effects. This maximization is achieved. in the computer program through the economic return factor net power output per unit cost ('WPD').

The pumping work is computed by considering the following entrance losses, friction losses, dumping losses, and the power required to lift the colder, denser water (in the case of the condenser). The quantities of seawater required by OTEC would most probably be supplied by a propeller type device. After considering the values given by the investigators in Reference 1a conservative estimate of 76% was selected for the efficiency of the seawater pumps.

The flow characteristics of the seawater were determined from the Reynolds number (using the hydraulic diameter) and the shape of the ducts. The correlations used to determine the hydrodynamic parameters were taken from Reference 5 and 6. These correlations appear in Appendix

VIII.

Biofouling is one of the most difficult problems to overcome in the effective design of OTEC system. M. A. Wood (7) has shown that it is possible to discourage the attachment of biofoulers

to the heat exchanger surfaces if 'unfavorable' hydrodynamic conditions can be maintained. Turbulence and wall shear stress in excess of 4×10^{-5} psi have been shown to be two effective deterrents. These become flow criteria in the computer program and are discussed in Appendices V and VIII.

Thermodynamic Considerations

The choice of working fluids to be used in the OTEC system must involve an assessment of the following characteristics:

1. The working fluid must be chemically compatible with the heat exchanger material.

2. The fluid should have a relatively high latent heat and enthalpy (thus requiring less equipment and power to circulate).

3. The fluid should have a high thermal conductance in phase change heat transfer within the temperature range expected in OTEC. There are several other fluid characteristics that must be considered in the choice; however, they are not directly related to the heat exchangers. The three fluids that are most prevalent in the current literature area ammonia, propane, and ethane. All three are analyzed separately in this investigation.

The following assumptions are made to facilitate analysis of the heat exchangers:

1. In the 'pool' type shell and tube evaporator, it is assumed that the compressed liquid enters the evaporator at the conditions of the exit of the recirculating pump. It is also assumed that the temperature of the fluid increases linearly until at the top, only vapor remains.

2. In the case of the 'spray' type shell and tube evaporator, it is assumed that the compressed liquid enters the evaporator in the same state as the 'pool' type with the exception being, that it is sprayed over the tubes. It is also assumed that the liquid falls from one tube to the next, decreasing in amount until only vapor remains at the bottom.

3. It is assumed that the process taking place in the honeycomb heat exchangers can be described by the correlation of Kleis. This correlation appears in Reference 8 and is described in Appendix III.

Chapter 3. Method of Analysis

The analysis of the system is divided into six steps as follows: 1) election of the geometric configuration of the heat exchangers, 2) determination of the Rankine cycle, 3) selection of the velocities of the seawater in the heat exchangers, 4) estimation of the effective average heat flux in the evaporator and condenser, 5) adjustment of the exchangers, 6) maximization through the economic return factor. This economic return factor is the net output per unit investment in the heat exchangers; and is used as an indicator by the computer model to determine the optimum selection of the geometric configuration, Rankine cycle, and velocities of the seawater.

Selection of Geometric Configuration

In the case of the shell and tube heat exchangers, the geometric variations considered were the length and diameter of the tubes. Primary analysis was conducted for tubes ranging in diameter from 3/8 inch to 3 inches and from 10 to 120 feet in length. It was determined from this primary analysis that the geometric configurations yielding the highest economic return factor were diameters ranging from 1 to 2½ inches with length/diameter ratios between 200 and 800. The selection of the tube diameter spans these values inclusively in t inch increments. The length of the tubes is selected at equal spacing between the upper and lower figure determined from the diameter and length/diameter ratio. In all instances the evaporator and condenser were assumed to have been constructed using the same length and diameter of tubes, however not necessarily the same number of tubes. Only one configuration of plastic honeycomb heat exchangers was investigated.

Determination of the Rankine Cycle

The Rankine cycle was defined by the selection of two temperatures. The definition of the cycle as well as the method used to select these temperatures appears in Appendix III. The Rankine cycle prescribed the thermodynamic properties of the working fluid at the entrance and exit of the heat exchangers and the gross power output. Also prescribed by the Rankine cycle was the total heat that must be transferred by the evaporator and condenser.

Velocities of the Seawater

The velocities of the seawater in the heat exchangers were determined from the wall shear criterion of M. A. Wood (7). Fluid properties and hydraulic diameter of the passages were also used in the determination. The correlation used was taken from Reference 6 and appears in Appendix VIII. The velocities of the seawater were then selected starting with this minimum value and incrementing upward. The upper limit of the velocities was determined by the realization of a local maximum in the economic return factor, 10 feet per second, or a Reynolds number above 106 (which ever came first). The last two criteria were the result of preliminary analysis. The hierarchy of this selection and maximization process is given in Appendix VIII. It was assumed that the velocity of the seawater in the evaporator was not necessarily equal to that in the condenser.

Effective Average Heat Flux

In the analysis, the heat exchangers were assumed to be approximately the size required to have a net output of 100 MWe. This size is typical of those in Reference 1. A value for the effective average h eat flux in the evaporator was arrived at by the method discussed in Appendix V. The h eat flux in the condenser was determined from the correlations given in

Appendix IV. The effective average heat flux is the quantity of heat transferred per unit time per unit area, and is a measure of the ability of the heat exchangers to transfer heat under the conditions assumed.

Adjustment of Heat Exchanger Size

The sizes of the heat exchangers were determined from the total heat transferred and the effective average heat flux. The total h eat transferred by the heat exchangers was determined from the detailed Rankine cycle description and the flow rate of the working fluid. The flow rate of the working fluid was assumed to be unity as it was later to be cancelled in calculation of the economic return factor. The quotient of these two quantities then yielded a value of required area. The size of each heat exchanger was determined by the number of tubes (of the specified length and diameter) required to supply this area. The total power required to pump the seawater and the approximate cost of the heat exchanger elements were determined from this adjusted size.

Maximization through the Economic Return Factor

The gross power output of the turbine per un1t flow rate of the working fluid was found from the Rankine cycle analysis. The gross power output of the system was then determined from the product of the gross power output of the turbine per un1t flow rate of the working fluid, the flow rate of the working fluid, and the efficiency of the generator. The net power output of the system was subsequently determined by reducing the gross power output by the total pumping power.

The approximate cost of the heat exchanger elements was divided into two parts: the cost of material and the cost of fabrication. The cost of the material was calculated from the present cost of the material per un1 t weight and the total weight of the elements. This price was then increased by a representative factor for shell/tube material weight ratio given in Reference 10. The approximation was made that the fabrication cost of the heat exchangers could be estimated on a flat rate per tube basis. It is obvious that there are many other factors entering into the cost of fabrication, however estimation of these factors would be speculation at this time. A more accurate determination would require a tentative pricing schedule. The actual values used are given in Appendix VIII.

The economic return factor is the ratio of the net power output to the estimated cost of the heat exchangers. It is this value that was used throughout the analysis to determine the optimum selection of the length and diameter of the tubes, Rankine cycle, and the velocities of the seawater in the heat exchangers. The order in which these parameters were maximized may be seen in the flowchart of the main computer program, Appendix VIII.

Maximization Schemes

The two maximization schemes which were used to determine the Rankine cycle may be characterized by: l) symmetric and 2) asymmetric temperature distributions. The first maximization scheme operated under the assumption that the total temperature difference available to the system was symmetrically distributed between the Rankine cycle and the thermal circuit. The temperature differential available to the condenser was assumed to be equal to that available to the evaporator. The ratio of the temperature difference available to the Rankine cycle, to that available to the evaporator and condenser was initially assumed to be unity. This ratio was subsequently adjusted by the computer model until a maximum value of the economic return factor was achieved. The second maximization scheme did not assume symmetry in the

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temperature distribution. However, the values of the temperature differentials arrived at by the first scheme became the initial values used in the second. The temperature difference available to the Rankine cycle was unchanged. However, the temperature difference available to the evaporator and condenser were adjusted in an attempt to converge the sizes (thus cost) of the two heat exchangers.

Chapter 4. Fluid Properties

Variation of the fluid properties is one of the major factors considered in this investigation that seems to be absent in the investigation of others. It is these variations of the fluid properties with the phase change heat transfer correlations that transform the analysis of OTEC into a highly nonlinear problem. It is the intent s investigation to demonstrate the necessity of considering these variations in an analysis of the system.

Magnitude of Property Variations Expected

The ma.gn1tude of variations in the fluid properties of the seawater and the three working fluids examined can be seen directly in Appendix VI and from the correlations in Appendix VII. The more significant changes are summarized here, taking into account the variation over the temperature range from 46 to 78F.

Seawater

The fluid property of seawater subject to the largest variation is the Prandtl number, which ranges from 10.44 to 6.28 (some 66%). The kinetic viscosity also experiences a large variation (.58%). The other seawater are fairly constant over the range of temperatures.

Ammonia

Most of the fluid properties of ammonia do not experience any great variation over this range of temperatures. The largest variations are seen in the saturation pressure (79.5%), the density of the saturated liquid (75.9%), the enthalpy of the saturated liquid (38.8%), and the surface tension (20.6%).

Propane

Although no one property of propane changes as significantly over the range of temperatures as do the properties of ammonia, the individual properties have a larger average variation. The larger variations being in the density of the saturated vapor (61.6%), the saturation pressure (61.6%), the enthalpy of the saturated liquid (40.4%), the enthalpy of the saturated liquid (35.5%), and the surface tension (31.4%).

Ethane

The critical temperature of ethane is 90.1F. As a result, the properties of this fluid vary rapidly with even small changes in temperature near the critical point. These variations include: density of the saturated vapor (88.8%), specific heat of the saturated vapor (52. 9%), saturation pressure (48.4%), dynamic viscosity of the saturated liquid (30.3%), conductivity of the saturated liquid (23%), enthalpy of the saturated liquid (12.3%), and the largest variation, surface tension (386%).

It can easily be seen from the magnitude of some of the variations that failure to take such variations into account in an analysis could result in erroneous conclusions.

The Effect of Property Variations

The effect of the property variations on the system can be predicted using the correlations for the heat transfer coefficients. The accuracy 1n predicting the effects on the system is limited by the accuracy of the correlations used, since great effort was taken to insure the accuracy of the

fluid properties. The individual con-elations are given (with some justification) in Appendices IV and VIII.

The effect of the change in the Prandtl number and kinematic viscosity on the thermal conductance of the flowing seawater is quite a b1t less than the variations of the individual properties. This is a result of the mutually effect in the correlation used. In fact, the variation over the entire 32 degrees (although the seawater varies no more than 5 degrees in any one tube) is only 11.9% significantly smaller than the 66 and 58% property variations.

The effects of the property variations of the working fluid show up in the comparison of the heat transfer coefficients of the phase change in the evaporator and condenser. The greatest effects can be seen in the case of ethane. This is a result of the previously mentioned proximity of the temperatures experienced in the system to the critical temperature of the fluid.

An example of the error that might be introduced into the analysis of the system by not considering fluid property variations is given here. In analyzing the Rankine cycle for ammonia, it was determined that a satisfactory operating condition for the evaporator would be defined by a temperature of 50F for the subcooled liquid entering the bottom and 66F for the saturated vapor leaving the top. The effective average heat flux assuming fluid property variations was determined to be 3850 BTU/hr/ft². This value was found from the numerical solution to the appropriate differential equations and 20 point Gaussian quadrature. This method is discussed in Appendix V. If however, it is assumed that the fluid properties remain constant, a misleading figure of 12,160 is computed! The values of the fluid properties used in this second determination were the standard values given by the Chemical Rubber Co. (Reference 19).

Chapter 5. Results

The detailed results of the investigation are manifested in the outputs of the computer program. A selection of these outputs appears in Appendix II. Table 1 lists the seventeen different runs and the variations considered in each. Figures 2 through 5 are plots of the economic return factor watts per dollar ('WPD') as a function of length/diameter ('L/D') with diameter of the tubes as a parameter. Figures 2 through 4 deal with the 'pool' type evaporator, whereas Figure 5 deals with the ' spray' type. The values plotted are the result of four levels of maximization (described in Appendix VIII) and are summary data points. The largest values attained by both maximization schemes are used. The similarity that can be seen in the plots of the economic return factor is also present in the plots of data not directly represented here.

Table 1. Configuration of Computer Runs

| run number	working fluid	heat exchanger material	heat exchanger type	method of introduction	maximization scheme	temperature	
						hot	cold
1	ammonia	titanium	shell & tube	pool	1	78	46
2	ammonia	titanium	shell & tube	pool	2	78	46
3	ammonia	titanium	shell & tube	pool	1	78	38
4	ammonia	titanium	shell & tube	pool	2	78	38
5	propane	titanium	shell & tube	pool	1	78	46
6	propane	titanium	shell & tube	pool	2	78	46
7	propane	titanium	shell & tube	pool	1	78	38
8	propane	titanium	shell & tube	pool	2	78	38
9	ethane	titanium	shell & tube	pool	1	78	46
10	ethane	titanium	shell & tube	pool	2	78	46
11	ethane	titanium	shell & tube	pool	1	78	38
12	ethane	titanium	shell & tube	pool	2	78	38
13	ammonia	titanium	shell & tube	spray	1	78	46
14	ammonia	titanium	shell & tube	spray	2	78	46
15	ammonia	plastic	honeycomb	spray	1	78	46
16	ammonia	plastic	honeycomb	spray	2	78	46
17	iso-butane	aluminum	shell & tube	pool	1	78	46
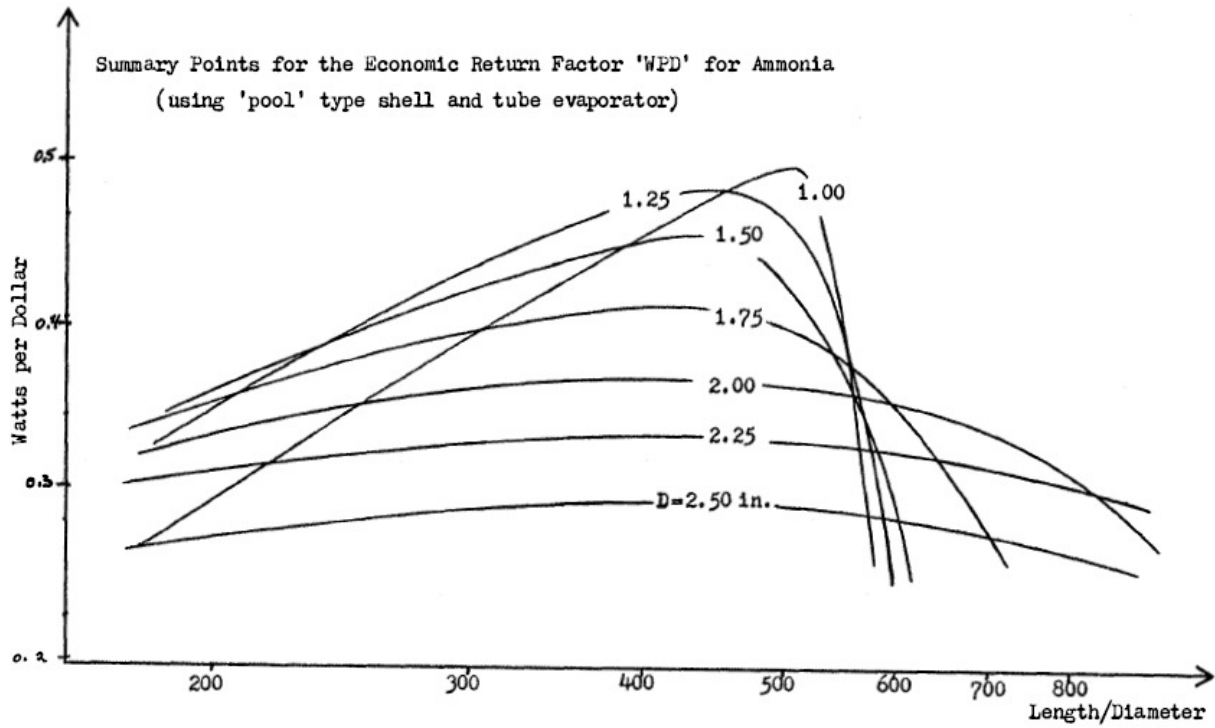
Figure 2. Plot of 'WPD' vs. 'L/D' at Various Diameters (for Ammonia)
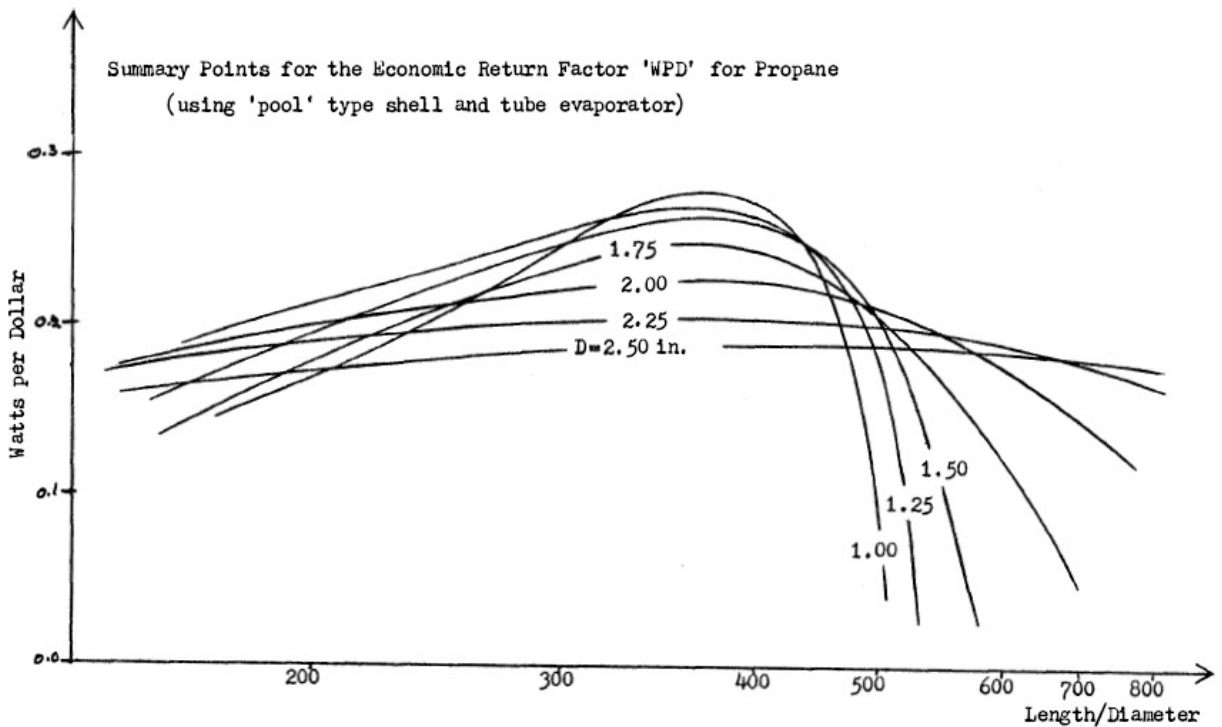

Figure 3. Plot of 'WPD' vs. 'L/D' at Various Diameters (for Propane)

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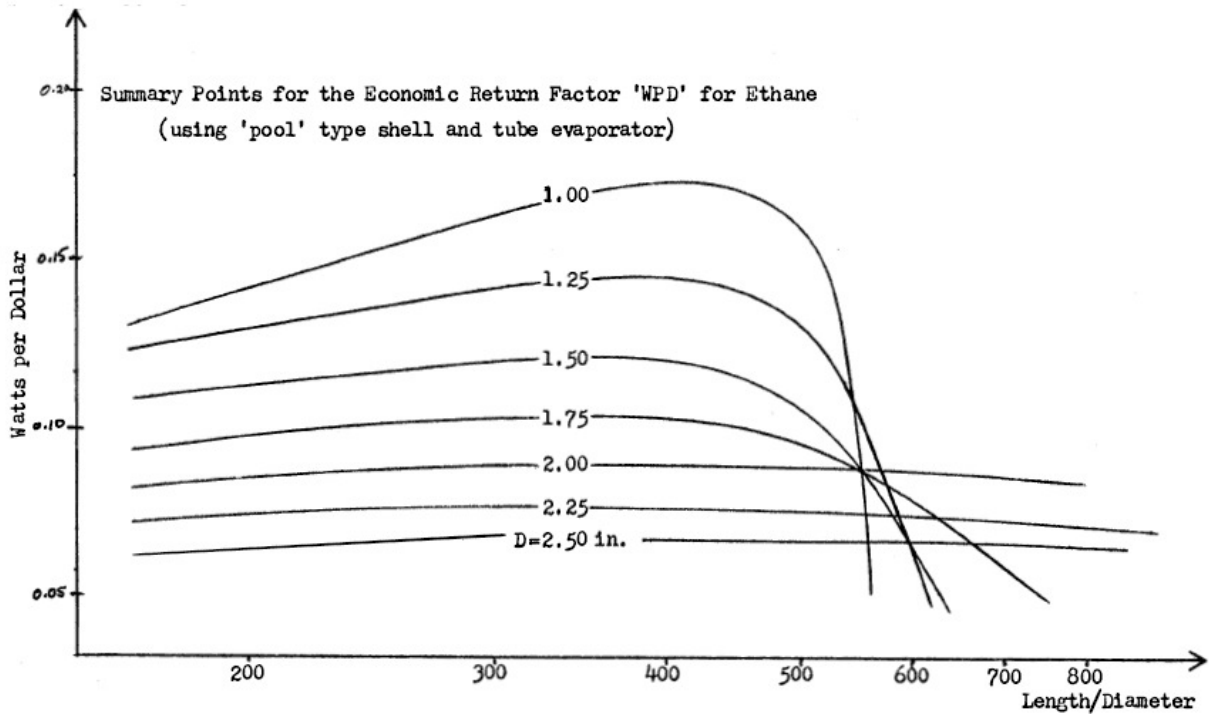
Figure 4. Plot of 'WPD' vs. 'L/D' at Various Diameters (for Ethane)
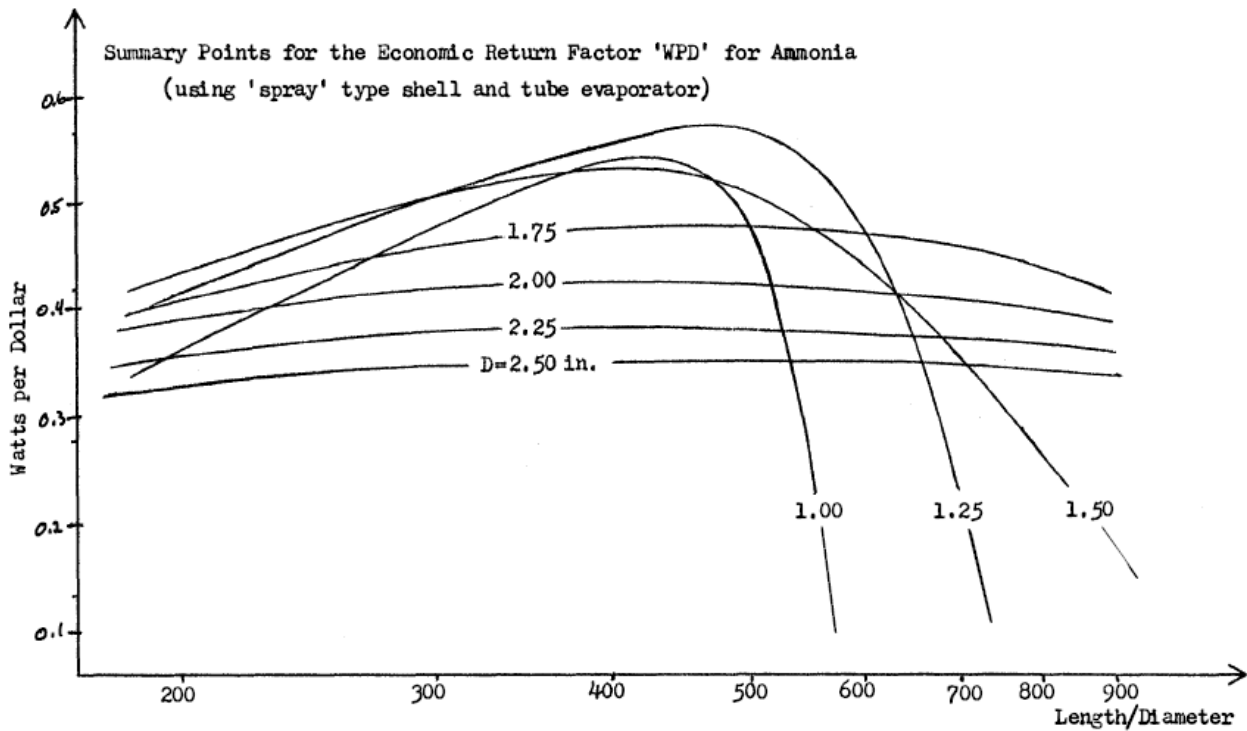
Figure 5. Plot of 'WPD' vs. 'L/D' at Various Diameters (for Ammonia)

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The characteristic curves o£ the economic return factor 'WPD ' corresponding to diameters greater than 2 inches are representative of the solutions obtained by using a more linear analysis. However, the characteristic curves corresponding to tubes of smaller diameter demonstrate the nonlinearity o£ the problem when variation of fluid properties are considered. It should be noticed that the increase and rapid decrease of the economic return £actor generally becomes more pronounced as the diameter is reduced.

This solution behavior is the result of two controlling factors: 1) the necessity of maintaining the minimum tolerable wall shear to discourage attachment of biofoulers and 2) the dependence of the thermal conductance in phase change heat transfer on the excess temperature differential. This first controlling factor is a design criterion based on the investigation of M. A. Wood (7) and is discussed in Appendix IV. The second controlling factor is evidenced in the phase change heat transfer correlations described in detail in Appendix IV.

This rapid decrease in the economic return factor experienced with increasing tube length is a result of the following: As the length o£ the tubes is increased, the required pumping power is increased. If the pumping power is intolerably high, the flow rate of the seawater is reduced by the computer model. When the flow rate of the seawater is reduced, the change in temperature of the seawater as it flows through the tubes is increased. The thermal circuit as described in detail in Appendix V is determined from this temperature difference and the definition points of the Rankine cycle. As more of the available temperature difference in the thermal circuit is taken up by this change in temperature of the seawater, the temperature difference remaining for the phase change heat transfer is reduced. Since the thermal conductance in phase change heat transfer is dependent on this temperature difference, as the difference is reduced, the conductance is reduced exponentially. This reduction in the thermal conductance of the phase change process results in a reduction of the total heat transferred. This reduction in the total heat transferred reduces the gross power output. Due to this reduction in the gross power output, the pumping power becomes even more intolerable. The final end of this process is the inability of the system. to maintain the minimum. tolerable wall shear with net power output.

Not represented were the graphs of the economic return factor for the aluminum. heat exchangers. Although the solution was close in form to that of the titanium heat exchangers of the same configuration, the unit cost was almost 5 times that of the titanium.. This is a result of the fact that the subroutine of the computer program used to determine the required wall thickness of the tubes had as criteria the corrosion rate and pitting factor (values taken from Reference 2 and 3). The pitting factor is the ratio of the deepest penetration to average metal penetration as determined by the weight loss of the specimen. A pitting factor of unity would represent uniform attack. Since the corrosion rate of aluminum is at least 20 times that of titanium and the pitting factor is almost 3 times, the additional required wall thickness more than offset the lower price of the aluminum.. The result of an assumed life of fifty years was an impractical situation for aluminum without electro-potential protection (which is not possible inside tubes unless individual wires are run through each tube).

The plastic heat exchangers configurations were also analyzed in the form proposed (l). The result was an economic return factor quite bit below that of the shell and tube heat exchangers. This was a result of three factors: l) the small passages through which the seawater must be pumped (t inch by t inch, 24 feet long) created such frictional losses that pumping power was significantly higher than that experienced in the shell and tube heat exchanger, 2) the working fluid ammonia in this case) was to be sprayed over the top of a series by 0.32 inch rectangular

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passes, 52 feet in depth; assuming that e evaporation process may be described by the correlation of Kleis Appendix IV), this process yields lower values than are possible shell and tube configurations, 3) the plastic heat exchangers have an individual life expectancy of twelve years, thus requiring co replacement four times over the fifty year total life span of the system.

Selection of Heat Exchanger Configuration

If the heat exchanger configuration were to be selected on the basis of economy alone, and judging from the results of this analysis, there would be little doubt that the shell and tube, 'spray' type evaporator using ammonia would be the optimum choice. Injecting the subcooled liquid into the evaporator such that it falls over a bank of tubes should, as the analysis verifies, yield a higher average heat flux thus require a smaller evaporator. Ii' the same accuracy may be assumed for the correlations in each case, an increase in the economic return factor of 3:3% may be expected in the use of the 'spray' type over the 'pool' type evaporator (the condenser in both cases is assumed to be the same).

Effects of Fluid Variations

The areas considered in this investigation, upon which the fluid property variations have the most significant effect, are the Rankine cycle and the effective average heat flux in the evaporator and condenser. Analysis of the Rankine cycle determines the gross power output of the system, while analysis of the heat exchangers determines the required size of the system. The net power output is found by reducing the gross power h'-J the pumping power. The cost of the heat exchangers is determined from the required size through the method discussed in Chapter III. Since the economic return factor is the quotient of these two quantities, it is of equal importance to accurately determine both the net power output and projected cost.

If some constant values were assumed for the fluid properties, and some constant value for the heat transfer coefficients, the resulting system (except for pumping work) would be linear. This would give the false impression of being able to transfer heat w1.th small temperature differentials, far in excess of the actual capability. The correlations used in this analysis to determine the thermal conductance of the phase change are very sensitive to temperature differential. Since the temperature differences experienced in OTEC are quite small in comparison to many conventional power systems, even greater effort must be made to determine these differentials accurately. Inclusion of fluid property variations in the analysis makes possible a more accurate description of the actual processes which will be taking place in the system. It is therefore imperative that these variations be considered.

Chapter 6. Conclusions and Recommendations

In conclusion, it could be said that the two main points brought out by this investigation are the necessity of a Rankine cycle analysis and the inclusion of fluid property variations in the model. It can be seen from comparison of the characteristic curves of the economic return factor, to the linear relationships quoted by some investigators which have neglected these considerations, that certainly an unrealistic picture is given of the system if these points are not taken.

The correlations used in the heat transfer coefficients have been discussed and appear in Appendices IV and VITI. These correlations were selected in an effort to best describe the system. Since nothing of this type or size has been built, the applicability of these correlations and others must be verified by experiment.

Three working fluids were investigated: ammonia, propane, and ethane. Many other fluids should be considered in an effort to find the one best suited to the situation. Working fluids should be considered which have a high enthalpy and other thermophysical properties that would result in the highest possible phase change heat transfer coefficients under the conditions present in OTEC. It is important that all three factors (enthalpy, thermal conductance in evaporation, and condensation) be as high as possible. If any one of these factors is low, that is low in comparison to those obtained u sing other fluids, the result will be a low economic return factor.

The computer model was used to determine the Rankine cycle corresponding to maximum economic return factor for the 'pool' type shell and tube heat exchangers; and the resulting effective average heat transfer coefficients were compared for the three working fluids examined. The effective average thermal conductance in the evaporator using ammonia was approximately 475 BTU/hr/ft²/°F, whereas in the condenser it was close to 1150 BTU/hr/ft²/°F or propane these values were around 450 and 250 respectively; and for ethane, 1200 and 200. From these values it can be seen that ethane has the highest evaporative thermal conductance but unfortunately the lowest thermal conductance in condensation. On the other hand, ammonia has the highest thermal conductivity in condensation and about the same evaporative thermal conductance as propane. Ammonia is the most promising of these three fluids, a conclusion which is supported by other investigators (1).

Of the independent variables (that is those which could be changed after assembling an OTEC system), perhaps the two affecting the economic performance of the system most are the definition temperature s of the Rankine cycle and the velocities of the seawater in the heat exchangers. The selection of these independent variables has be en optimized by the computer model. This limits the economic return factor that can be achieved after construction to within the value obtained by the computer model for the specific length and diameter of the tubes selected in the design.

It can be seen from. the plots of the economic return factor that while the selection of the working fluid, material, and mode of introducing the subcooled liquid into the evaporator ('spray' or 'pool') may determine the maximum economic return factor, the allowable error in the selection of the tube length is dependent on the selection of the diameter. The larger diameter tubes allow a wider range of acceptable tube length; however the smaller diameters are more economical. A very thorough analysis must be done before the additional risk of using a smaller diameter tube would be tolerable. This risk may be an absolute necessity if the cost of the heat exchangers is as reflective of the cost of the total system as some investigators have pointed out.

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It can be seen, for example, in the case of ammonia (Figure 2), that the use of 1 inch diameter tubes rather than 2 inch tubes might prove to reduce the cost of the system by the 29% indicated.

As mentioned, the problem of biofouling has not been sufficiently studied. The results of M. A. Wood (Reference 7) show real promise in controlling this problem by maintaining 'unfavorable' hydrodynamic conditions, a solution more acceptable by all who are concerned for the environment than some that have been proposed. Investigation in this area by engineers is disappointingly incomplete. Wood's study was concerned primarily with macrofoulers, leaving the problem of microfoulers to be dealt with. Much experimentation over extended periods of time is needed to be able to predict the total effect of this problem on OTEC.

It is clear that a great deal of research and experimentation is needed to determine and verify many assumptions which must be made in the design and analysis of an OTEC system. It should be obvious from the characteristic curves of the economic return factor, that this research and experimentation should be performed before any full scale is constructed. It is the hope of this investigator that in pointing out these details it may be possible to make the need for and experimentation of this type known.

Appendix I. Definition of Symbols

AC   the surface area required per unit flow rate of the working fluid in the condenser (ft²-sec/pound)

ACMAX   the 'AC' corresponding to the maximum 'WPD'

AE   the surface area required per unit flow rate of the working fluid in the evaporator (ft²-sec/pound)

AEMAX   the 'AE' corresponding to the maximum 'WPD'

CARNOT   the Carnot efficiency of the system

CL   the specific heat of the saturated liquid (BTU/pound/°F)

COST   the cost of the heat exchanger material (dollar/pound)

COSTC   estimated baseline cost of the condenser per unit flow rate of the working fluid (dollar-sec/pound)

COSTE   estimated baseline cost of the evaporator per unit flow rate of the working fluid (dollar-sec/pound)

CSTFCT   cost factor increase of material for manufacturing

CSWC   specific heat of the seawater at the inlet to the condenser (BTU/pound/°F)

CSWE   specific heat of the seawater at the inlet to the evaporator (BTU/pound/°F)

CTUBES   the number of tubes required in the condenser per unit flow rate of the working fluid (tubes-sec/pound)

D   inside diameter of the tubes (ft)

DE   differential change in enthalpy of the working fluid (BTU/pound)

DEN   density of the heat exchanger material (pound/ft³)

DH   height which the cold water must be raised to the condenser (ft)

DHPUMP   differential work done by the recirculating pump (BTU/pound)

DL   differential length of the tubes (ft)

DOLS   estimated cost of the two heat exchangers per unit flow rate of the working fluid (dollar-sec/pound)

DPP   fabrication cost factor per tube (dollar/tube)

DPTMAX   maximum differential pressure across the turbine (psi)

DRHO   the difference in density of the cold and warm seawater (slug/foot³)

DT   the difference in the temperature of the cold and warm water (°F)

DTP   rise in temperature of the working fluid due to inefficiency of the recirculating pump (°F)

DTSWC   average rise in temperature of the seawater that passes through the condenser (°F)

DTSWCX   the 'DTSWC' which corresponds to the maximum 'WPD' (°F)

DTSWE   a representative change in temperature of the seawater in the evaporator (°F)

DTSWEX   the 'DTSWE' which corresponds to the maximum 'WPD' (°F)

EFFGEN   the efficiency of the generators

EFFPMP   the efficiency of the seawater pumps

EFFTUR   the efficiency of the turbines

EFRPMP	the efficiency of the recirculating pumps
EH	the enthalpy of the saturated vapor at 'TH' (BTU/pound)
EL	the enthalpy of the saturated liquid at 'TL' (BTU/pound)
ETUBES	the number of tubes required in the evaporator per unit flow rate of the working fluid (tubes-sec/pound)
H	enthalpy (BTU/pound)
HB	average boiling conductance (BTU/hr/ft²/°F)
HBMAX	the 'HE' which corresponds to the maximum 'WPD'
HC	the average conductance of condensation (BTU/hr/ft²/°F)
HCMAX	the 'HC' which corresponds to the maximum 'WPD'
HE	the overall conductance in the evaporator (BTU/hr/ft²/°F)
HEMAX	the 'HZ' which corresponds to the maximum 'WPD'
HK	the overall conductance in the condenser (BTU/hr/ft²/°F)
HKMAX	the 'HK' which corresponds to the maximum 'WPD '
IND	integer variable, serving as an error return indicator
KC	thermal conductivity of the seawater at the inlet to the condenser (BTU/hr/ft/°F)
KE	thermal conductivity of1the seawater at the inlet to the evaporator (BTU/hr/ft/°F)
KTW	thermal conductivity of the heat exchanger material (BTU/hr/ft/°F)
L	the length of the tubes (ft)
LD	the quotient of length and diameter (dimensionless)
LMAX	the maximum investigated length of the tubes (ft)
LMIN	the minimum investigated length of the tubes (ft)
NPASS	maximum number of re-entrant passes through the program allowed to achieve convergence
OD	outside diameter of the tubes (ft)
PAVABL	the theoretical maximum power available from the system per unit flow rate of the working fluid (BTU/pound)
PC	percent of the gross output consumed by the seawater pumps in the condenser
PE	percent of the gross output consumed by the seawater pumps in the evaporator
P	pressure (psi)
PMPTOT	total power consumed by the pumps per unit flow rate of the working fluid (BTU/pound)
PNET	the net power output of the system per unit flow rate of the working fluid (BTU/pound)
POUT	the power output of a Carnot cycle operating under the same conditions per unit flow rate of the working fluid (BTU/pound)
PPC	the power consumed by the seawater pumps in the condenser per unit flow rate of the working fluid (BTU/pound)
PPE	the power consumed by the seawater pumps in the evaporator per unit flow rate of the working fluid (BTU/pound)

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PRC	the Prandtl number of the seawater at the entrance to the condenser
PRE	the Prandtl number of the seawater at the entrance to the evaporator
REC	the Reynolds number of the flow in the tubes in the condenser
REE	the Reynolds number of the flow in the tubes in the evaporator
RENUMC	the 'REC' which corresponds to the maximum 'WPD'
RENUME	the 'REE' which corresponds to the maximum "WPD'
RHOHGH	the density of the seawater at the inlet to the condenser (slug/ft³)
RHOLOW	the density of the seawater at the inlet to the evaporator (slug/ft³)
RNUC	kinematic viscosity of the seawater at the inlet to the condenser (ft²/sec)
RNUE	kinematic viscosity of the seawater at the inlet to the condenser (ft²/sec)
RPOUT	the percent that the net output is of the Carnot output
TH	the temperature of the warm seawater (°F)
TL	the temperature of the cold seawater (°F)
TW	the thickness of the tube wall (ft)
TWFC	the temperature of the working fluid in the condenser (°F)
TWFE	the temperature of the working fluid at the exit of the evaporator (°F)
VC	the velocity of the seawater in the tubes in the condenser (ft/sec)
VCMIN	the velocity which will produce the minimum acceptable wall shear in the tubes in the condenser (ft/sec)
VE	the velocity of the seawater in the tubes in the evaporator (ft/sec)
VEMIN	the velocity which will produce the minimum acceptable wall shear in the tubes in the evaporator (ft/sec)
VSWC	the 'VC' which corresponds to the maximum 'WPD'
VSWE	the 'VE' which corresponds to the maximum 'WPD'
WLSHRC	the 'WSC' which corresponds to the maximum 'WPD'
WLSHRE	the 'WSE' which corresponds to the maximum 'WPD'
WPD	the estimated output/cost ratio for the heat exchangers (watts/dollar)
WSC	the wall shear in the tubes in the condenser (psi)
WSE	the wall shear in the tubes in the evaporator (psi)
WSMIN	the minimum acceptable wall shear (psi)
WT	the weight of the heat exchanger material required per unit flow rate of the working fluid (sec)
WTIC	the weight of heat exchanger material required to construct the condenser per unit flow rate of the working fluid (sec)
WTICMX	the 'WTIC' which corresponds to the maximum 'WPD'
WTIE	the weight of heat exchanger material required to construct the evaporator per unit flow rate of the working fluid (sec)
WTIEMX	the 'WTIE' which corresponds to the maximum 'VPD '

WTPERA   the weight of heat exchanger material required to produce one square foot of surface area (pound/ft²)

X   quality of the working fluid (pounds vapor/pounds liquid)

ACS   the gross sectional area of the inlet to the heat exchanger (ft²)

DTPERQ   the change in temperature of the seawater per unit heat transferred .per unit flow rate of the working fluid (°F/BTU)

F   friction factor

HBIO   the thermal conductance of the biofouling layer (BTU/hr/ft/°F)

HCORR   the thermal conductance of the corrosion product build-up (BTU/hr/ft/°F)

HE   the convective thermal conductivity of the seawater (BTU/hr/ft/°F)

LMTDSW   the log mean temperature difference of the seawater (°F)

Q   heat flux rate (BTU/hr/ft²)

Subscripts

C   referring to the condenser

E   referring to the evaporator

H   referring to a higher or hotter quantity

L   referring to a lower or colder quantity; referring to the saturated liquid state

S   referring to a state or quantity which is associated with an isentropic process

V   referring to the saturated vapor state

1,2,3,4,5 referring to the respective definition points of the Rankine cycle

Appendix II. Selected Output of Computer Program

The output of the computer program for those configurations dealing with propane as the working fluid appear in Tables 2 through 5· Critical information as to the assumed quantities and units of the output variables are given in the tables.

It should be noted when making an evaluation of the economic return factor 'WPD', that this figure is arrived at u sing estimated costs. This figure does not represent the total cost of the system, only the elements of the heat exchangers, the intention of this figure is not to project a cost for the system, but to allow a comparison to be made from one geometry and working fluid to another. The actual value of the economic return factor is not as important as the trends observed and the conclusions that can be drawn from its comparative values.

Table 2. Output of Computer Program for Propane (Scheme 1/46F)

MAXIMIZATION SCHEME 1 -- TEMPERATURE OF SEAWATER 78°F AND 46°F

●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●● OCEAN THERMAL ENERGY CONVERSION SYSTEM SIMULATOR ●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●●

TH=78.0DEG   TL=46.0DEG   KTW=11.56BTU/HR/FT/DEG   DH=1200.FT   POUT= 9.7BTU/#   DPTMAX=53.27PSI   COST=5.0$/#   DP=25.$/TUBE

EFFTUR=.85   EFFGEN=.95   EFFHPMP=.90   EFFPMPSW=.76   WORKING FLUID (PROPANE)   NPASS= 3

DIA (IN)	LENGTH (FT)	L/D	TWALL (IN)		TWF (F)	VSW (FPS)	RE*NUM	WALL SH (PSI)	DTSW (F)	HI*AV (BTU)	HTOT (BTU)	A*REQ (SQFT)	WTI (#/#)	COST (%)	PUMP (%)	RPOUT (%)	WPD BTU/hr/$
1.00	14.4	173.	.01271													10.23	.475
				EVAPORATOR	71.6	4.1	37915.	.66-03	1.73	347.6	199.8	.056	.0168	24.5	4.0		
				CONDENSER	52.4	2.7	14313.	.36-03	1.25	279.	167.3	.172	.0519	75.5	17.8		
1.00	25.3	303.	.01271													10.12	.707
				EVAPORATOR	71.6	4.1	37915.	.66-03	2.85	347.6	199.8	.056	.0168	24.0	3.5		
				CONDENSER	52.4	3.1	16358.	.45-03	1.86	282.	173.9	.177	.0533	76.0	14.8		
1.00	36.1	433.	.01271													8.21	.774
				EVAPORATOR	70.8	4.1	37915.	.66-03	4.05	347.1	199.7	.054	.0163	27.3	3.5		
				CONDENSER	53.2	5.0	26582.	.11-02	2.01	267.	186.2	.145	.0436	72.7	27.5		
1.25	16.1	155.	.01589													9.49	.520
				EVAPORATOR	70.8	3.8	39391.	.55-03	1.67	332.2	187.6	.058	.0217	25.8	3.6		
				CONDENSER	53.2	2.4	15833.	.27-03	1.32	263.	153.7	.166	.0626	74.2	17.3		
1.25	28.2	271.	.01589													10.41	.730
				EVAPORATOR	71.6	4.2	43767.	.67-03	2.49	344.1	197.0	.057	.0213	24.0	3.8		
				CONDENSER	52.4	3.2	21110.	.45-03	1.61	268.	167.4	.179	.0675	76.0	16.7		
1.25	40.3	387.	.01589													9.43	.844
				EVAPORATOR	70.8	4.2	43767.	.67-03	3.57	343.7	196.9	.055	.0207	24.3	3.8		
				CONDENSER	53.2	3.2	21110.	.45-03	2.41	265.	166.4	.171	.0644	75.7	13.7		
1.25	52.4	503.	.01589													3.63	.401
				EVAPORATOR	70.8	4.6	48144.	.79-03	4.22	354.3	205.5	.053	.0199	27.7	4.6		
				CONDENSER	53.2	7.5	50136.	.21-02	1.64	251.	187.0	.138	.0520	72.3	62.4		
1.50	17.7	141.	.01906													9.15	.527
				EVAPORATOR	70.8	3.9	48516.	.56-03	1.48	329.3	185.3	.058	.0264	25.7	4.0		
				CONDENSER	53.2	2.4	19501.	.27-03	1.16	253.	149.4	.169	.0765	74.3	19.7		
1.50	30.9	247.	.01906													9.75	.707
				EVAPORATOR	70.8	3.9	48516.	.56-03	2.48	329.3	185.3	.058	.0264	25.5	3.4		
				CONDENSER	53.2	2.8	22751.	.36-03	1.73	254.	155.7	.171	.0771	74.5	15.5		
1.50	44.2	354.	.01906													9.72	.792
				EVAPORATOR	70.8	4.3	53907.	.67-03	3.20	340.7	194.4	.056	.0252	24.5	4.1		
				CONDENSER	53.2	3.2	26001.	.45-03	2.15	255.	161.1	.172	.0777	75.5	15.0		
1.50	57.5	460.	.01906													7.90	.737
				EVAPORATOR	70.8	3.9	48516.	.56-03	4.18	329.2	185.3	.058	.0264	27.5	3.0		
				CONDENSER	53.2	5.3	42251.	.11-02	1.93	247.	174.0	.154	.0696	72.5	30.4		
1.75	19.1	131.	.02224													9.19	.510
				EVAPORATOR	70.8	3.5	51434.	.45-03	1.42	314.1	173.3	.062	.0329	25.5	3.2		
				CONDENSER	53.2	2.1	19381.	.20-03	1.18	250.	139.6	.183	.0963	74.5	18.5		
1.75	33.4	229.	.02224													9.52	.659
				EVAPORATOR	70.8	4.0	57863.	.56-03	2.24	326.6	183.2	.059	.0311	25.5	3.6		
				CONDENSER	53.2	2.9	27134.	.36-03	1.56	246.	151.7	.173	.0910	74.5	17.0		
1.75	47.7	327.	.02224													9.66	.726
				EVAPORATOR	70.8	4.0	57863.	.56-03	3.07	326.6	183.2	.059	.0312	25.4	3.4		
				CONDENSER	53.2	3.3	31010.	.44-03	1.94	246.	156.7	.173	.0914	74.6	16.2		
1.75	62.1	426.	.02224													9.66	.753
				EVAPORATOR	70.8	4.0	57863.	.56-03	3.82	326.6	183.2	.059	.0312	25.4	3.2		
				CONDENSER	53.2	3.7	34886.	.55-03	2.24	246.	161.0	.174	.0917	74.6	17.1		

Table 2. Continued

1.75	76.4	524.	.02224													4.71	.433
				EVAPORATOR	70.8	4.4	64292.	.68-03	4.31	337.9	192.2	.056	.0297	27.4	4.0		
				CONDENSER	53.2	7.0	65896.	.17-02	1.71	237.	174.8	.149	.07AR	72.6	54.5		
2.00	20.4	122.	.02542													9.12	.482
				EVAPORATOR	70.8	3.6	59914.	.46-03	1.30	311.8	171.5	.063	.0380	25.4	3.5		
				CONDENSER	53.2	2.1	22577.	.20-03	1.07	243.	136.7	.1A5	.1116	74.6	20.4		
2.00	35.7	214.	.02542													9.67	.604
				EVAPORATOR	70.8	4.1	67403.	.57-03	2.06	324.2	181.2	.060	.0360	24.5	3.9		
				CONDENSER	53.2	2.5	27092.	.27-03	1.58	242.	143.8	.184	.1106	75.5	15.6		
2.00	51.0	306.	.02542													9.83	.658
				EVAPORATOR	70.8	4.1	67403.	.57-03	2.83	324.2	181.2	.060	.0360	24.6	3.6		
				CONDENSER	53.2	3.0	31607.	.35-03	1.94	242.	149.4	.183	.1100	75.4	14.7		
2.00	66.3	398.	.02542													9.77	.680
				EVAPORATOR	70.8	4.1	67403.	.57-03	3.53	324.1	181.2	.060	.0360	24.7	3.4		
				CONDENSER	53.2	3.4	36122.	.45-03	2.22	242.	154.1	.182	.1096	75.3	15.4		
2.00	81.6	490.	.02542													7.54	.587
				EVAPORATOR	70.8	4.1	67403.	.57-03	4.19	324.1	181.2	.060	.0360	27.0	3.3		
				CONDENSER	53.2	5.5	5A699.	.11-02	1.90	235.	165.9	.162	.0975	73.0	33.0		
2.25	21.7	115.	.02859													8.46	.448
				EVAPORATOR	70.8	3.7	6A547.	.46-03	1.20	309.7	169.8	.064	.0432	25.3	3.7		
				CONDENSER	53.2	2.1	25830.	.20-03	.99	237.	134.2	.1A8	.1273	74.7	22.2		
2.25	37.9	202.	.02A59													9.49	.552
				EVAPORATOR	70.8	4.1	77116.	.57-03	1.90	321.8	179.4	.060	.0409	24.5	4.1		
				CONDENSER	53.2	2.6	3A996.	.27-03	1.46	237.	141.0	.185	.1257	75.5	16.A		
2.25	54.1	289.	.02859													9.67	.595
				EVAPORATOR	70.8	4.1	77116.	.57-03	2.62	321.8	179.4	.060	.0409	24.7	3.8		
				CONDENSER	53.2	3.0	36162.	.35-03	1.80	236.	146.4	.184	.1247	75.3	15.8		
2.25	70.4	375.	.02859													9.96	.611
				EVAPORATOR	70.8	4.1	77116.	.57-03	3.29	321.8	179.4	.060	.0409	23.9	3.6		
				CONDENSER	53.2	3.0	36162.	.35-03	2.24	239.	147.4	.192	.1299	76.1	13.6		
2.25	86.6	462.	.02859													8.99	.599
				EVAPORATOR	70.8	4.1	77116.	.57-03	3.91	321.8	179.4	.060	.0409	25.5	3.5		
				CONDENSER	53.2	4.3	51659.	.67-03	2.11	233.	157.0	.176	.1193	74.5	21.4		
2.50	22.8	110.	.03177													8.60	.411
				EVAPORATOR	70.8	3.7	77319.	.46-03	1.12	307.6	168.3	.064	.0484	25.3	3.9		
				CONDENSER	53.2	2.2	29135.	.20-03	.91	232.	132.0	.190	.1432	74.7	24.0		
	39.9	192.	.03177													9.44	.502
				EVAPORATOR	70.8	3.7	77319.	.46-03	1.89	307.6	168.2	.064	.0484	25.6	3.2		
				CONDENSER	53.2	2.6	34962.	.27-03	1.36	232.	13A.5	.187	.1410	74.4	18.1		
	57.1	274.	.03177													9.51	.537
				EVAPORATOR	70.8	4.2	86984.	.57-03	2.45	319.6	177.6	.061	.0459	24.7	3.9		
				CONDENSER	53.2	3.1	40789.	.36-03	1.68	231.	143.8	.185	.1396	75.3	16.9		
	74.2	356.	.03177													9.84	.552
				EVAPORATOR	70.8	4.2	86984.	.57-03	3.08	319.6	177.6	.061	.0459	24.0	3.7		
				CONDENSER	53.2	3.1	40789.	.36-03	2.10	233.	144.6	.193	.1450	76.0	14.5		
	91.3	438.	.03177													9.69	.556
				EVAPORATOR	70.8	4.2	86984.	.57-03	3.67	319.6	177.6	.061	.0459	24.2	3.6		
				CONDENSER	53.2	3.5	44616.	.45-03	2.28	233.	148.9	.190	.1434	75.8	15.8		

Table 3. Output of Computer Program for Propane (Scheme 2/46F)

MAXIMIZATION SCHEME 2 -- TEMPERATURE OF SEAWATER 78°F AND 46°F

************************************ OCEAN THERMAL ENERGY CONVERSION SYSTEM SIMULATOR *********************************

TH=78.0DEG TL=46.0DEG KTW=11.56BTU/HR/FT/DEG DH=1200.FT POUT=9.7BTU/# OPTMAX=53.3PSI COST=5.05/# DP=25.5/TUBE

EFFTUR=.85 EFFGEN=.95 EFFPMPP=.90 EFFPMPSW=.76 WORKING FLUID (PROPANE) NPASS= 3

DIA (IN)	LENGTH (FT)	L/D	TWALL (IN)	TWF (F)	VSW (FPS)	RE*NUM	WALL SH (PSI)	DTSW (F)	HI*AV (BTU)	HTOT (BTU)	A*REQ (SQFT)	WTI (#/#)	COST (%)	PUMP (%)	RPOUT (#)	WPD BTU/hr$
1.00	14.4	173.	.01271												10.40	.519
		EVAPORATOR		75.9	4.1	33915.	.66-03	.70	284.	176.9	.103	.0310	48.4	7.4		
		CONDENSER		56.7	2.7	14313.	.36-03	1.99	250.	156.4	.110	.0339	51.6	11.5		
1.00	25.3	303.	.01271												10.40	.770
		EVAPORATOR		75.9	4.1	33915.	.66-03	1.16	284.	176.9	.103	.0310	47.8	6.5		
		CONDENSER		56.7	3.1	16358.	.45-03	2.96	252.	162.2	.113	.0339	52.2	9.5		
1.00	36.1	433.	.01271												9.67	.883
		EVAPORATOR		75.8	4.1	33915.	.66-03	1.57	284.	176.9	.103	.0310	50.1	6.2		
		CONDENSER		56.6	5.0	26582.	.11-02	2.87	246.	175.6	.102	.0309	49.9	18.2		
1.25	16.1	155.	.01589												10.56	.572
		EVAPORATOR		75.9	3.8	39391.	.55-03	.64	271.	166.2	.109	.0412	48.3	6.4		
		CONDENSER		56.7	2.4	14933.	.27-03	1.89	242.	146.3	.117	.0442	51.7	11.4		
1.25	28.2	271.	.01589												10.54	.799
		EVAPORATOR		75.9	4.2	43767.	.67-03	1.01	281.	174.5	.104	.0393	47.8	7.2		
		CONDENSER		56.7	3.2	21110.	.45-03	2.56	240.	156.2	.114	.0429	52.2	10.8		
1.25	40.3	387.	.01589												10.84	.926
		EVAPORATOR		75.9	4.2	43767.	.67-03	1.37	281.	174.5	.104	.0393	46.4	6.7		
		CONDENSER		56.7	3.2	21110.	.45-03	3.46	244.	157.7	.120	.0454	53.6	9.0		
1.25	52.4	503.	.01589												6.24	.664
		EVAPORATOR		75.8	4.6	48144.	.79-03	1.64	290.	182.1	.099	.0375	50.3	8.1		
		CONDENSER		56.6	7.5	60116.	.21-02	2.33	231.	176.0	.098	.0370	49.7	41.4		
1.50	17.7	141.	.01906												10.27	.585
		EVAPORATOR		75.9	3.9	48516.	.56-03	.57	268.	164.2	.111	.0500	48.1	7.0		
		CONDENSER		56.7	2.4	19501.	.27-03	1.67	233.	142.1	.119	.0539	51.9	12.9		
1.50	30.0	241.	.01906												10.55	.775
		EVAPORATOR		75.9	4.3	53907.	.67-03	.90	278.	172.3	.106	.0477	46.8	7.7		
		CONDENSER		56.7	2.8	22751.	.36-03	2.48	234.	147.9	.120	.0544	53.2	10.2		
1.50	44.2	354.	.01906												10.66	.872
		EVAPORATOR		75.9	4.3	53907.	.67-03	1.23	278.	172.3	.106	.0477	46.6	7.2		
		CONDENSER		56.7	3.2	26001.	.45-03	3.08	234.	152.7	.121	.0548	53.4	9.9		
1.50	57.5	460.	.01906												9.31	.860
		EVAPORATOR		75.9	4.3	53907.	.67-03	1.63	278.	172.3	.106	.0477	49.2	6.9		
		CONDENSER		56.7	5.3	42251.	.11-02	2.75	228.	164.3	.109	.0492	50.8	20.1		
1.50	70.7	566.	.01906												.73	.080
		EVAPORATOR		75.9	6.1	75470.	.12-02	1.48	311.	199.2	.093	.0421	49.2	15.4		
		CONDENSER		56.7	9.7	78002.	.32-02	2.08	221.	173.4	.096	.0435	50.8	34.5		
1.75	19.1	131.	.02224												9.99	.570
		EVAPORATOR		75.9	4.0	57863.	.56-03	.51	266.	162.3	.112	.0590	48.0	7.6		
		CONDENSER		56.7	2.5	23257.	.27-03	1.50	226.	138.8	.121	.0640	52.0	14.4		
1.75	33.4	229.	.02224												10.58	.726
		EVAPORATOR		75.9	4.0	57863.	.66-03	.86	266.	162.3	.112	.0590	47.9	6.4		
		CONDENSER		56.7	2.9	27134.	.36-03	2.24	227.	144.2	.122	.0642	52.1	11.2		
1.75	47.7	327.	.02224												10.49	.799
		EVAPORATOR		75.9	4.4	64292.	.68-03	1.12	276.	170.3	.107	.0563	46.7	7.7		
		CONDENSER		56.7	3.3	31010.	.44-03	2.79	227.	148.7	.122	.0644	53.3	10.7		

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Table 3. Continued

1.75	62.1	426.	.02224												10.45	.835
			EVAPORATOR	75.9	4.4	64292.	.68-03	1.40	276.	170.3	.107	.0563	46.6	7.3		
			CONDENSER	56.7	1.7	34886.	.55-03	3.21	227.	152.5	.123	.0646	53.4	11.3		
1.75	76.4	524.	.02224												7.10	.632
			EVAPORATOR	75.8	4.4	64292.	.68-03	1.67	276.	170.3	.107	.0562	50.1	7.1		
			CONDENSER	56.6	7.0	65096.	.17-02	2.43	219.	164.7	.106	.0559	49.9	36.0		
2.00	20.4	122.	.02542												10.43	.543
			EVAPORATOR	75.9	3.6	59914.	.46-03	.50	253.	152.1	.119	.0720	47.8	6.1		
			CONDENSER	56.7	2.1	22577.	.20-03	1.54	224.	130.5	.130	.0786	52.2	13.4		
2.00	35.7	214.	.02542												10.40	.669
			EVAPORATOR	75.9	4.1	67403.	.57-03	.79	264.	160.6	.113	.0682	47.9	6.8		
			CONDENSER	56.7	3.0	31607.	.36-03	2.04	221.	141.0	.123	.0742	52.1	12.1		
2.00	51.0	306.	.02542												10.41	.725
			EVAPORATOR	75.9	4.1	67403.	.57-03	1.09	264.	160.6	.113	.0682	46.8	6.3		
			CONDENSER	56.7	3.9	31607.	.35-03	2.79	223.	142.0	.129	.0775	53.2	9.7		
2.00	66.3	398.	.02542												10.19	.753
			EVAPORATOR	75.9	4.1	67403.	.57-03	1.37	264.	160.6	.113	.0682	46.9	6.0		
			CONDENSER	56.7	3.4	36122.	.45-03	3.19	223.	146.3	.128	.0772	53.1	10.1		
2.00	91.6	490.	.02542												9.23	.699
			EVAPORATOR	75.9	4.1	67403.	.57-03	1.62	264.	160.6	.113	.0682	49.7	5.8		
			CONDENSER	56.7	5.3	58609.	.11-02	2.71	217.	156.7	.114	.0684	50.3	21.7		
2.25	21.7	115.	.02859												10.11	.508
			EVAPORATOR	75.9	3.7	68547.	.46-03	.46	251.	150.6	.121	.0817	47.7	6.6		
			CONDENSER	56.7	2.1	25030.	.20-03	1.42	219.	128.2	.132	.0896	52.3	14.6		
2.25	37.9	202.	.02859												10.50	.612
			EVAPORATOR	75.9	4.1	77116.	.57-03	.73	262.	159.0	.114	.0775	46.7	7.2		
			CONDENSER	56.7	2.8	30946.	.27-03	2.09	218.	134.3	.131	.0885	53.3	11.0		
2.25	54.1	289.	.02859												10.67	.657
			EVAPORATOR	75.9	4.1	77116.	.57-03	1.01	262.	159.0	.114	.0775	46.9	6.6		
			CONDENSER	56.7	3.0	36162.	.35-03	2.58	218.	139.2	.130	.0878	53.1	10.4		
2.25	70.4	375.	.02859												10.66	.677
			EVAPORATOR	75.9	4.1	77116.	.57-03	1.27	262.	159.0	.114	.0775	47.0	6.3		
			CONDENSER	56.7	3.4	41328.	.45-03	2.95	218.	143.2	.129	.0874	53.0	10.8		
2.25	86.6	462.	.02859												10.23	.676
			EVAPORATOR	75.9	4.1	77116.	.57-03	1.52	262.	159.0	.114	.0775	47.9	6.1		
			CONDENSER	56.7	4.3	51659.	.67-03	3.02	216.	148.7	.124	.0841	52.1	14.1		
2.50	22.8	110.	.03177												9.89	.471
			EVAPORATOR	75.9	3.7	77319.	.46-03	.43	249.	149.2	.122	.0916	47.6	7.0		
			CONDENSER	56.7	2.2	29135.	.20-03	1.31	215.	126.1	.134	.1006	52.4	15.7		
2.50	39.9	192.	.03177												10.43	.558
			EVAPORATOR	75.9	4.2	86984.	.57-03	.68	260.	157.5	.115	.0869	46.7	7.6		
			CONDENSER	56.7	2.6	34962.	.27-03	1.95	214.	132.0	.132	.0993	53.3	11.9		
2.50	57.1	274.	.03177												10.53	.596
			EVAPORATOR	75.9	4.2	86984.	.57-03	.95	260.	157.5	.115	.0869	46.9	7.0		
			CONDENSER	56.7	3.1	40789.	.36-03	2.41	213.	136.7	.131	.0983	53.1	11.1		
2.50	74.2	356.	.03177												10.19	.612
			EVAPORATOR	75.9	4.2	86984.	.57-03	1.19	260.	157.5	.115	.0869	46.0	6.6		
			CONDENSER	56.7	3.1	40789.	.36-03	3.01	215.	137.5	.135	.1020	54.0	9.5		
2.50	91.3	438.	.03177												10.11	.620
			EVAPORATOR	75.9	4.2	86984.	.57-03	1.42	260.	157.5	.115	.0869	46.3	6.4		
			CONDENSER	56.7	3.5	46616.	.45-03	3.28	215.	141.4	.134	.1009	53.7	10.4		

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Table 4. Output of Computer Program for Propane (Scheme 1/38F)

MAXIMIZATION SCHEME 1 -- TEMPERATURE OF SEAWATER 78°F AND 38°F

************************************** OCEAN THERMAL ENERGY CONVERSION SYSTEM SIMULATOR ********************************

TH=78.0DEG TL=38.0DEG KTW=11.56BTU/HR/FT/DEG DH=1200.FT PQUI=12.4BTU/# OPTMAX=63.8PSI COST=5.0$/# DP=25.$/TUBE

EFFTUR=.85 EFFGEN=.95 EFFRPMP=.90 EFFRMPSW=.76 WORKING FLUID (PROPANE) NPASS=1

DIA (IN)	LENGTH (FT)	L/D (IN)	TWALL (IN)	TWF (F)	VSW (FPS)	RE*NUM	WALL SH (PSI)	DTSW (F)	HT*AV (BTU)	HTOT (BTU)	A*REQ (SQFT)	WTT (#/#)	COST ($)	PUMP ($)	HPOUT ($)	WFRD BTU/hr$
1.00	14.4	173.	.01271												9.72	.824
			EVAPORATOR	69.0	4.9	40698.	.91-03	2.36	456.	244.4	.036	.0109	22.4	3.7		
			CONDENSER	47.0	3.0	14198.	.45-03	1.57	261.	164.1	.125	.0376	77.6	14.6		
1.00	25.3	303.	.01271												9.87	1.217
			EVAPORATOR	69.0	4.9	40698.	.91-03	3.88	456.	244.4	.036	.0109	22.5	3.2		
			CONDENSER	47.0	3.8	17747.	.67-03	2.21	261.	172.4	.124	.0374	77.5	13.9		
1.00	36.1	433.	.01271												9.30	1.428
			EVAPORATOR	69.0	4.9	40698.	.91-03	5.22	456.	244.3	.036	.0109	22.9	3.1		
			CONDENSER	47.0	4.9	23072.	.11-02	2.50	259.	180.6	.121	.0365	77.1	18.6		
1.25	16.1	155.	.01589												9.47	.903
			EVAPORATOR	69.0	4.6	48144.	.79-03	2.14	438.	231.2	.038	.0143	23.0	3.3		
			CONDENSER	47.0	3.1	18322.	.45-03	1.35	249.	158.3	.128	.0481	77.0	16.9		
1.25	28.2	271.	.01589												9.87	1.260
			EVAPORATOR	69.0	5.1	52521.	.92-03	3.39	451.	240.7	.037	.0138	21.9	3.6		
			CONDENSER	47.0	3.5	20613.	.56-03	2.05	251.	163.1	.130	.0491	78.1	13.5		
1.25	40.3	387.	.01589												9.94	1.452
			EVAPORATOR	69.0	5.1	52521.	.92-03	4.59	451.	240.7	.037	.0138	21.6	3.3		
			CONDENSER	47.0	3.9	22903.	.67-03	2.59	252.	167.4	.133	.0500	78.4	13.2		
1.25	52.4	503.	.01589												7.11	1.243
			EVAPORATOR	69.0	4.6	48144.	.79-03	5.91	438.	231.2	.038	.0143	24.6	2.6		
			CONDENSER	47.0	7.0	41225.	.19-02	2.13	244.	180.7	.117	.0440	75.4	36.5		
1.50	17.7	141.	.01906												9.48	.920
			EVAPORATOR	69.0	4.8	59298.	.80-03	1.89	434.	228.1	.039	.0174	22.2	3.7		
			CONDENSER	47.0	2.8	19746.	.36-03	1.30	243.	149.7	.135	.0612	77.8	16.6		
1.50	30.9	247.	.01906												9.73	1.218
			EVAPORATOR	69.0	4.8	59298.	.80-03	3.15	434.	228.1	.039	.0174	22.6	3.1		
			CONDENSER	47.0	3.6	25388.	.56-03	1.81	241.	158.2	.132	.0597	77.4	15.1		
1.50	44.2	354.	.01906												10.01	1.362
			EVAPORATOR	69.0	5.2	64689.	.93-03	4.12	446.	237.4	.037	.0168	21.1	3.6		
			CONDENSER	47.0	3.6	25388.	.56-03	2.47	244.	159.4	.138	.0626	78.9	12.4		
1.50	57.5	460.	.01906												9.37	1.416
			EVAPORATOR	69.0	5.2	64689.	.93-03	5.11	446.	237.3	.037	.0168	22.0	3.4		
			CONDENSER	47.0	4.8	33851.	.90-03	2.53	241.	167.4	.132	.0596	78.0	17.6		
1.50	70.7	566.	.01906												3.04	.541
			EVAPORATOR	69.0	6.1	75470.	.12-02	5.63	468.	254.1	.035	.0157	23.1	4.8		
			CONDENSER	47.0	9.2	64491.	.29-02	1.87	243.	179.0	.115	.0521	76.9	66.7		
1.75	19.1	131.	.02224												9.30	.893
			EVAPORATOR	69.0	4.4	64292.	.68-03	1.79	411.	215.5	.041	.0215	22.9	3.2		
			CONDENSER	47.0	2.8	23551.	.36-03	1.16	236.	146.1	.138	.0726	77.1	18.5		
1.75	33.4	229.	.02224												9.81	1.139
			EVAPORATOR	69.0	4.9	70722.	.80-03	2.86	430.	225.2	.039	.0206	22.0	3.4		
			CONDENSER	47.0	3.3	26915.	.45-03	1.77	236.	151.0	.139	.0731	78.0	14.2		
1.75	47.7	327.	.02224												9.95	1.250
			EVAPORATOR	69.0	4.9	70722.	.80-03	3.91	430.	225.2	.039	.0206	21.9	3.1		
			CONDENSER	47.0	3.7	30279.	.54-03	2.23	236.	155.2	.140	.0737	78.1	13.4		

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Table 4. Continued

1.75	62.1	426.	.02224													9.90	1.299
				EVAPORATOR	69.0	4.9	70722.	.80-03	4.87	430.	225.2	.039	.0206	21.8	3.0		
				CONDENSER	47.0	4.1	33644.	.66-03	2.59	237.	158.7	.141	.0741	78.2	13.9		
1.75	76.4	524.	.02224													7.69	1.129
				EVAPORATOR	69.0	4.9	70722.	.80-03	5.76	430.	225.2	.039	.0206	23.6	2.9		
				CONDENSER	47.0	6.5	53830.	.15-02	2.22	731.	168.9	.127	.0668	76.4	31.6		
2.00	20.4	122.	.02542													9.40	.846
				EVAPORATOR	69.0	4.5	74893.	.68-03	1.63	414.	213.0	.041	.0249	22.0	3.4		
				CONDENSER	47.0	2.5	23514.	.27-03	1.18	232.	138.4	.146	.0883	78.0	17.5		
2.00	35.7	214.	.02542													9.63	1.047
				EVAPORATOR	69.0	5.0	82382.	.81-03	2.92	427.	222.6	.040	.0238	22.0	3.6		
				CONDENSER	47.0	3.3	31352.	.44-03	1.61	230.	147.6	.140	.0846	78.0	15.4		
2.00	51.0	306.	.02542													9.79	1.132
				EVAPORATOR	69.0	5.0	82382.	.81-03	3.59	427.	222.6	.040	.0238	21.9	3.3		
				CONDENSER	47.0	3.7	35272.	.55-03	2.04	230.	151.6	.141	.0850	78.1	14.5		
2.00	66.3	398.	.02542													10.03	1.169
				EVAPORATOR	69.0	5.0	82382.	.81-03	4.49	427.	222.6	.040	.0238	21.3	3.1		
				CONDENSER	47.0	3.7	35272.	.55-03	2.55	232.	152.5	.146	.0862	78.1	12.1		
2.00	81.6	490.	.02542													9.25	1.154
				EVAPORATOR	69.0	5.0	82382.	.81-03	5.32	427.	222.6	.040	.0238	22.2	3.0		
				CONDENSER	47.0	5.0	47029.	.92-03	2.50	229.	159.7	.138	.0834	77.8	19.0		
2.25	21.7	115.	.02859													9.27	.788
				EVAPORATOR	69.0	4.1	77116.	.57-03	1.59	397.	200.4	.044	.0297	22.8	2.8		
				CONDENSER	47.0	2.5	26903.	.27-03	1.08	227.	135.9	.149	.1007	77.2	19.0		
2.25	37.9	202.	.02859													9.85	.955
				EVAPORATOR	69.0	4.6	85684.	.69-03	2.54	411.	210.6	.042	.0283	22.0	3.0		
				CONDENSER	47.0	3.0	31386.	.35-03	1.63	221.	141.2	.148	.1002	78.0	14.2		
2.25	54.1	289.	.02859													9.92	1.023
				EVAPORATOR	69.0	5.0	94253.	.82-03	3.33	423.	220.1	.040	.0271	21.3	3.5		
				CONDENSER	47.0	3.4	35870.	.45-03	2.04	227.	145.6	.148	.1000	78.7	13.2		
2.25	70.4	375.	.02859													9.91	1.052
				EVAPORATOR	69.0	5.0	94253.	.82-03	4.18	423.	220.1	.040	.0271	21.3	3.3		
				CONDENSER	47.0	3.8	40354.	.55-03	2.36	227.	149.3	.147	.0999	78.7	13.5		
2.25	86.6	462.	.02859													10.07	1.061
				EVAPORATOR	69.0	5.0	94253.	.82-03	4.96	423.	220.1	.040	.0271	20.8	3.2		
				CONDENSER	47.0	3.8	40354.	.55-03	2.81	229.	150.2	.152	.1033	79.2	12.3		
2.50	22.8	110.	.03177													9.06	.728
				EVAPORATOR	69.0	4.2	86984.	.57-03	1.48	394.	198.4	.044	.0334	22.7	3.0		
				CONDENSER	47.0	2.6	30345.	.27-03	1.00	223.	133.6	.151	.1134	77.3	20.5		
2.50	39.9	192.	.03177													9.70	.870
				EVAPORATOR	69.0	4.7	96649.	.69-03	2.36	408.	208.4	.042	.0318	22.0	3.2		
				CONDENSER	47.0	3.0	35403.	.35-03	1.52	222.	138.7	.149	.1126	78.0	15.3		
2.50	57.1	274.	.03177													9.88	.927
				EVAPORATOR	69.0	4.7	96649.	.69-03	3.26	408.	208.4	.042	.0318	22.1	2.9		
				CONDENSER	47.0	3.4	40460.	.45-03	1.90	222.	143.0	.149	.1121	77.9	14.1		
2.50	74.2	356.	.03177													9.78	.949
				EVAPORATOR	69.0	5.1	******	.82-03	3.91	420.	217.7	.040	.0304	21.4	3.4		
				CONDENSER	47.0	3.9	45518.	.55-03	2.21	222.	146.5	.148	.1118	78.6	14.4		
2.50	91.3	438.	.03177													9.96	.956
				EVAPORATOR	69.0	5.1	******	.82-03	4.65	420.	217.7	.040	.0304	20.9	3.3		
				CONDENSER	47.0	3.9	45518.	.55-03	2.63	224.	147.3	.153	.1153	79.1	13.0		

Table 5. Output of Computer Program for Propane (Scheme 2/38F)

MAXIMIZATION SCHEME 2 -- TEMPERATURE OF SEAWATER 78°F AND 38°F

*************************** OCEAN THERMAL ENERGY CONVERSION SYSTEM SIMULATOR ***************************

TR=78.0DEG   TL=38.0DEG   KTW=11.5BTU/HR/FT/DEG   DH=1200.FT   POUT= 12.4BTU/W   DPTMAX= 63.8PSI   COST=5.0$/W   DP=25.3/TONE

EFFTUR=.85   EFFGEN=.95   EFFPHP=.90   EFFPMPSW=.76   WORKING FLUID (PROPANE)   NPASS= 3

DIA (IN)	LENGTH (FT)	L/D	TWALL (IN)	TWF (F)	VSW (FPS)	RE*NUM	WALL SH (PSI)	DTSW (F)	HI*AV (BTU)	HTOT (BTU)	A*REQ (SQFT)	WTI (#/#)	COST (%)	PUMP (%)	RPOUT (%)	WPD BTU/hrs
1.00	14.4	173.	.01271												9.81	.923
		EVAPORATOR		72.2	4.9	40698.	.91-03	1.58	388.	223.5	.050	.0149	34.2	5.1		
		CONDENSER		50.2	3.0	14198.	.45-03	2.08	244.	157.3	.095	.0287	65.8	11.3		
1.00	25.3	303.	.01271												9.95	1.362
		EVAPORATOR		72.2	4.9	40698.	.91-03	2.62	388.	223.4	.050	.0149	34.3	4.5		
		CONDENSER		50.2	3.8	17747.	.67-03	2.92	244.	165.0	.095	.0286	65.7	10.7		
1.00	36.1	433.	.01271												9.41	1.620
		EVAPORATOR		72.9	5.3	44090.	.11-02	3.03	384.	226.6	.052	.0157	37.2	5.6		
		CONDENSER		50.9	4.9	23072.	.11-02	3.47	240.	170.9	.088	.0265	62.8	13.7		
1.25	16.1	155.	.01589												9.64	1.019
		EVAPORATOR		72.9	4.6	48144.	.79-03	1.29	361.	207.6	.057	.0213	37.8	5.0		
		CONDENSER		50.9	3.1	18322.	.45-03	1.87	231.	150.7	.093	.0350	62.2	12.5		
1.25	28.2	271.	.01589												9.90	1.413
		EVAPORATOR		72.9	5.1	52521.	.92-03	2.03	370.	215.5	.055	.0206	36.7	5.4		
		CONDENSER		50.9	3.5	20613.	.56-03	2.86	232.	155.0	.094	.0355	63.3	9.9		
1.25	40.3	387.	.01589												9.97	1.634
		EVAPORATOR		73.0	5.1	52521.	.92-03	2.75	369.	215.3	.055	.0207	36.4	5.1		
		CONDENSER		51.0	3.9	22903.	.67-03	3.61	233.	158.8	.096	.0361	63.6	9.7		
1.25	52.4	503.	.01589												7.90	1.524
		EVAPORATOR		73.3	5.1	52521.	.92-03	3.22	363.	213.2	.057	.0216	40.9	5.1		
		CONDENSER		51.3	7.0	41225.	.19-02	3.03	224.	169.9	.083	.0312	59.1	26.3		
1.50	17.7	141.	.01906												9.61	1.042
		EVAPORATOR		72.9	4.8	59298.	.80-03	1.13	356.	204.7	.054	.0260	37.0	5.6		
		CONDENSER		50.9	2.8	19746.	.36-03	1.81	225.	142.7	.098	.0443	63.0	12.2		
1.50	30.9	247.	.01906												9.71	1.370
		EVAPORATOR		72.9	5.2	64689.	.93-03	1.81	366.	212.6	.056	.0251	36.7	5.4		
		CONDENSER		50.9	3.6	25388.	.56-03	2.52	224.	150.3	.096	.0433	63.3	11.1		
1.50	44.2	354.	.01906												9.81	1.536
		EVAPORATOR		73.0	5.2	64689.	.93-03	2.47	366.	212.5	.056	.0251	36.5	5.5		
		CONDENSER		51.0	4.0	28209.	.67-03	3.21	224.	153.8	.097	.0438	63.5	10.7		
1.50	57.5	460.	.01906												9.55	1.613
		EVAPORATOR		72.9	5.2	64689.	.93-03	3.09	366.	212.6	.056	.0251	36.8	5.2		
		CONDENSER		50.9	4.8	33851.	.90-03	3.53	224.	158.6	.096	.0432	63.2	13.0		
1.50	70.7	566.	.01906												4.90	.957
		EVAPORATOR		71.4	6.1	75470.	.12-02	3.13	376.	224.2	.055	.0249	40.4	7.8		
		CONDENSER		51.4	9.2	64681.	.29-02	2.67	215.	168.0	.081	.0360	59.6	47.4		
1.75	19.1	131.	.02224												9.53	1.018
		EVAPORATOR		72.9	4.4	64292.	.68-03	1.09	343.	193.9	.060	.0319	37.7	4.8		
		CONDENSER		50.9	2.8	23551.	.36-03	1.62	219.	139.4	.100	.0528	62.3	13.6		
1.75	33.4	229.	.02224												9.48	1.284
		EVAPORATOR		72.9	4.9	70722.	.80-03	1.72	353.	202.1	.058	.0308	36.8	5.1		
		CONDENSER		50.9	3.3	26915.	.45-03	2.47	219.	143.7	.100	.0529	63.2	10.5		
1.75	47.7	327.	.02224												10.00	1.409
		EVAPORATOR		73.0	4.9	70722.	.80-03	2.34	352.	201.8	.059	.0310	36.8	4.7		
		CONDENSER		51.0	3.7	30279.	.54-03	3.12	219.	147.4	.101	.0531	63.2	9.8		

Table 5. Continued

```
1.75   62.1   426.   .02224                                                              9.45    1.471
              EVAPORATOR   73.0   5.3   77151.   .94-03  2.79   362.   209.6   .057   .0298   35.8    5.6
              CONDENSER    51.0   4.1   33644.   .66-03  3.63   219.   150.7   .101   .0534   64.2   10.2

1.75   76.4   524.   .02224                                                              8.41    1.356
              EVAPORATOR   73.3   4.9   70722.   .80-03  3.26   346.   199.8   .061   .0329   40.7    4.6
              CONDENSER    51.3   6.5   53830.   .15-02  3.17   213.   159.0   .090   .0472   59.3   22.7

2.00   20.4   122.   .02542                                                              9.58     .967
              EVAPORATOR   72.9   4.5   74893.   .68-03   .98   340.   191.4   .062   .0371   36.8    5.2
              CONDENSER    50.9   2.5   23514.   .27-03  1.64   215.   132.3   .106   .0638   63.2   12.8

2.00   35.7   214.   .02542                                                              9.73    1.185
              EVAPORATOR   72.9   5.0   82382.   .81-03  1.57   350.   199.8   .059   .0356   36.7    5.4
              CONDENSER    50.9   3.3   31352.   .44-03  2.25   213.   140.6   .102   .0613   63.3   11.3

2.00   51.0   306.   .02542                                                              9.87    1.279
              EVAPORATOR   72.9   5.0   82382.   .81-03  2.18   350.   199.7   .059   .0356   36.7    5.0
              CONDENSER    50.9   3.7   35272.   .55-03  2.85   213.   144.2   .102   .0615   63.3   10.6

2.00   66.3   398.   .02542                                                             10.06    1.322
              EVAPORATOR   73.0   5.0   82382.   .81-03  2.70   349.   199.5   .059   .0358   36.0    4.8
              CONDENSER    51.0   3.7   35272.   .55-03  3.58   215.   145.0   .106   .0636   64.0    9.3

2.00   81.6   490.   .02542                                                              9.50    1.324
              EVAPORATOR   72.9   5.0   82382.   .81-03  3.24   350.   199.8   .059   .0355   37.0    4.6
              CONDENSER    50.9   5.0   47029.   .92-03  3.48   213.   151.5   .100   .0604   63.0   14.0

2.25   21.7   115.   .02859                                                              9.49     .905
              EVAPORATOR   72.9   4.6   85684.   .69-03   .90   337.   189.4   .062   .0422   36.7    5.5
              CONDENSER    50.9   2.5   26903.   .27-03  1.51   211.   129.8   .107   .0728   63.3   14.0

2.25   37.9   202.   .02859                                                              9.58    1.082
              EVAPORATOR   72.9   5.0   94253.   .82-03  1.45   347.   197.6   .060   .0405   36.7    5.7
              CONDENSER    50.9   3.4   35870.   .45-03  2.07   209.   137.9   .103   .0697   63.3   12.3

2.25   54.1   289.   .02859                                                              9.74    1.157
              EVAPORATOR   72.9   5.0   94253.   .82-03  2.01   347.   197.6   .060   .0405   36.7    5.3
              CONDENSER    50.9   3.8   40354.   .55-03  2.63   209.   141.3   .103   .0699   63.3   11.4

2.25   70.4   375.   .02859                                                              9.45    1.191
              EVAPORATOR   73.0   5.0   94253.   .82-03  2.51   347.   197.4   .060   .0406   36.1    5.0
              CONDENSER    51.0   3.8   40354.   .55-03  3.31   210.   142.0   .106   .0720   63.9    9.4

2.25   86.6   462.   .02859                                                              9.88    1.205
              EVAPORATOR   73.0   5.0   94253.   .82-03  2.99   347.   197.4   .060   .0406   36.1    4.8
              CONDENSER    51.0   4.2   44838.   .67-03  3.67   210.   144.9   .106   .0720   63.9   10.7

2.50   22.8   110.   .03177                                                              9.17     .840
              EVAPORATOR   72.9   4.2   86984.   .57-03   .90   324.   178.9   .065   .0493   37.5    4.5
              CONDENSER    50.9   2.6   30345.   .27-03  1.40   207.   127.8   .109   .0822   62.5   15.1

2.50   39.9   192.   .03177                                                              9.42     .988
              EVAPORATOR   72.9   4.7   96649.   .69-03  1.43   334.   187.5   .063   .0473   36.8    4.8
              CONDENSER    50.9   3.0   35403.   .35-03  2.12   206.   132.4   .108   .0814   63.2   11.2

2.50   57.1   274.   .03177                                                              9.84    1.049
              EVAPORATOR   73.0   5.1   ******   .82-03  1.86   344.   195.3   .061   .0456   36.1    5.5
              CONDENSER    51.0   3.4   40460.   .45-03  2.66   206.   136.2   .107   .0808   63.9   10.3

2.50   74.2   356.   .03177                                                              9.85    1.076
              EVAPORATOR   73.0   5.1   ******   .82-03  2.35   344.   195.3   .061   .0456   36.1    5.2
              CONDENSER    51.0   3.9   45518.   .55-03  3.09   206.   139.4   .107   .0806   63.9   10.5

2.50   91.3   438.   .03177                                                              9.99    1.086
              EVAPORATOR   73.0   5.1   ******   .82-03  2.79   344.   195.1   .061   .0458   35.6    5.1
              CONDENSER    51.0   3.9   45518.   .55-03  3.69   207.   140.1   .110   .0829   64.4    9.5
```

Appendix III. Rankine Cycle

The Rankine cycle is the standard format of all heat engines in which the working fluid in a closed system undergoes a cyclical change of phase.

Method of Description

The variation of the Rankine cycle used in this analysis excludes any superheat or reheating processes. The working fluid is assumed to be a simple compressible, pure substance devoid of magnetic or electrostatic properties. It is also assumed that there is no significant work done against surface tension. In this case the Rankine cycle consists of five distinct processes and may be defined by two temperatures. The cycle is shown graphically in Figure 6.
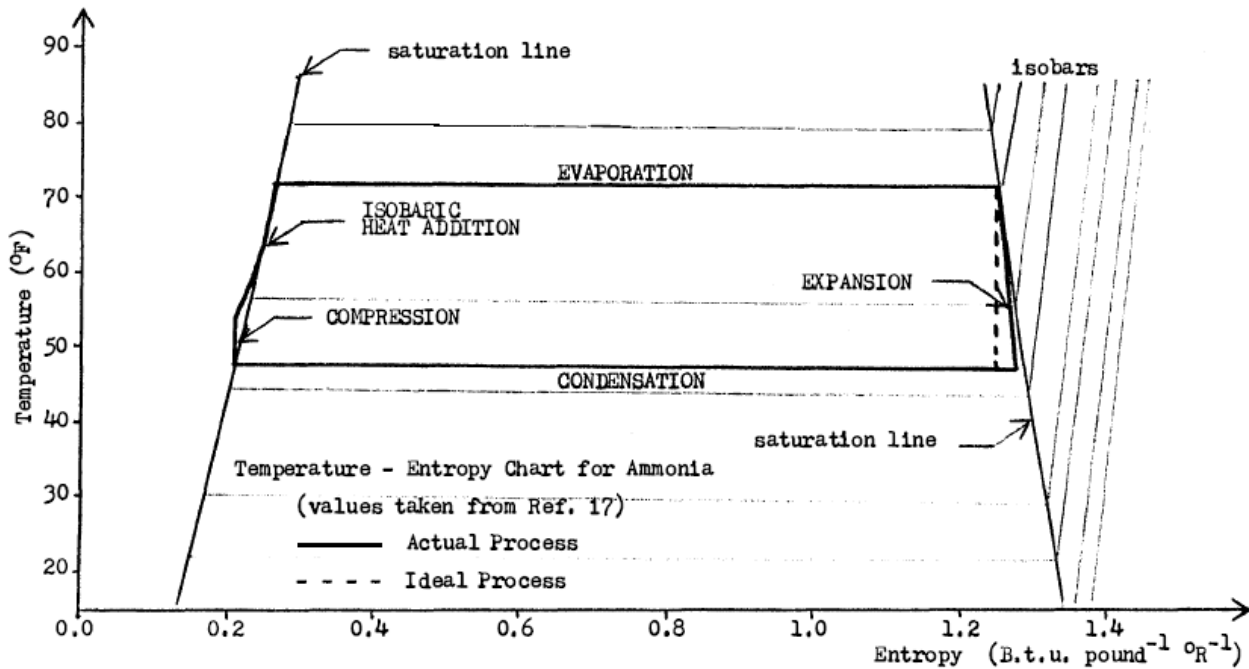
Figure 6. Temperature- Entropy Chart for Ammonia

The five processes in the Rankine cycle area evaporation (isothermal), expansion (polytropic), condensation (isothermal), compression (polytropic), and heat addition (isobaric). The corresponding five states of the fluid area saturated liquid, saturated vapor, vapor/liquid mixture, saturated liquid, and compressed liquid. During the evaporation process, the working fluid begins as a saturated liquid to which heat is added isothermally until it has become saturated vapor. The process of expansion involves a change of state and change of entropy. This change of entropy is due to the inefficiency of the turbine. Figure 7 shows this process graphically in detail, Equation 1 describes the process mathematically. The vapor/liquid mixture is then condensed isothermally to the point of saturated liquid. The saturated liquid is subsequently compressed to the pressure of the evaporator. During compression, there is also a change in entropy. This change in entropy is due to the inefficiency of the recirculating pump. The efficiency of the pump is assumed to be 90% Reference 10.

$$dh = \eta_{turbine} dh_s \qquad (1)$$

In defining the efficiency of a turbine , as the ratio of the actual power output to the power output were the process of the turbine to be isentropic , the following relationship can be derived from the graph of Figure 7. The efficiency of the turbine is assumed to be 85% while the generators are assumed to be 95% efficient.
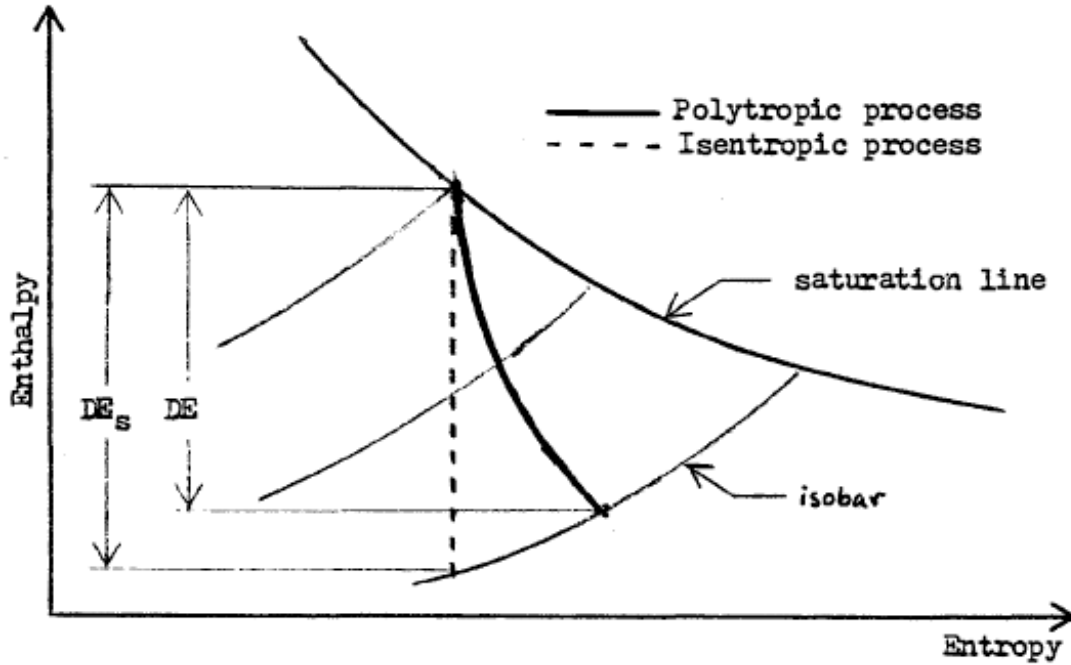
Figure 7. Polytropic Process

To justify the approximation of the work done by the recirculating pump by Equation 2, it is necessary to examine the differential equation:

$$dh = du + pdv + vdp$$

It can be noted that the relationship between the change in internal energy, work done, and the change in temperature is expressed by:

$$du = C\,dT = dh\left(1 - \eta_{pump}\right)$$

Combination of these two equations together with the fact that the change in specific volume of a liquid when compressed (under the moderate conditions experienced in this case) is negligible, the following equation results:

$$dh = vdp / \eta_{pump} \qquad (2)$$

The error in the fore mentioned assumption was calculated for the case of 10 times the variation of even the most extreme conditions to be expected in this system; and that error amounted to 0.04%. It may then be taken as a valid approximation of the actual process.

Determination of Definition Points

The complete Rankine cycle can be defined by two temperatures in this case. That being the gross temperature of the working fluid at the exit of the evaporator 'T1' and the gross temperature of the working fluid at the exit of the turbine 'T2'. The method used in the computer program of determining these two temperatures was broken down into two segments of operation. This was necessary due to excessive run time and the requirement of a human decision to initialize variables at the beginning of the re-entrant convergence routine.

The first segment of the determination process (termed maximization scheme #1) was based on the assumption that there was some sort of symmetry in the temperature differences made available to the various parts of the thermal circuit. Temperatures were selected by stepping in 0.5 degree intervals throughout the whole of possibilities, these results were then used to narrow the band of temperatures investigated. When the temperatures of the definition points of the Rankine cycle that most nearly corresponded to the maximum economic return factor under this symmetric assumption were found; they were used to initialize the second segment of the determination. The second segment (termed maximization scheme #2) operated under the assumption that the temperature differences were not symmetric. Improvement in the economic performance of the system was then accomplished by linear regression on the definition temperatures with the end result being the realization of another maximum above the first or the convergence of the cost of the condenser and evaporator.

Appendix IV. Phase Change Heat Transfer

Heat transfer involving a change of phase is dependent, not only on the thermophysical properties of the fluid, but also on the excess temperature differential (the difference between the gross fluid temperature and the temperature of the surface). The determination of this excess temperature differential is crucial, since it absorbs some of the total potential available to the system.

Evaporation

There are three basic modes of evaporative heat transfer• natural convection, nucleate boiling, and film boiling. If heat flux is plotted vs. excess temperature (on log-log scales) for most fluids, the resulting graph would resemble Figure 8.
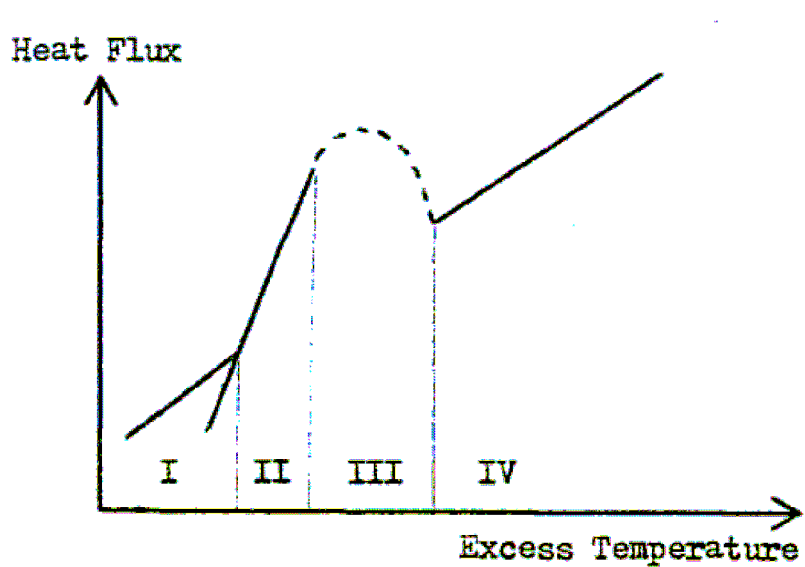
Figure 8. Heat Flux vs. Excess Temperature

In region I, the mode of heat transfer is natural convection, II is nucleate boiling, III is a transition zone, and IV is stable film boiling. Natural convection is characterized by vapor appearing at the surface of the liquid accompanied by an occasional bubble. Nucleate boiling, however, is evidenced by the formation of bubbles from ' nucleation sites' that rise in the fluid by virtue of a density instability. The formation rate of these bubbles increases until a film of vapor begins to cover the surface. Since the thermal conductivity of the vapor is less than that of the liquid, the ability of the fluid to 'carry away' heat from the surface is reduced. When the entire surface is insulated in this manner by the vapor, the transition from nucleate to stable film boiling is complete.

It can be noticed from Figure 8, that the region of maximum slope (thus maximum film coefficient) is that of nucleate boiling. This is a phenomenon that is taken advantage of in the modeling of the system: nucleate boiling is characterized by a very high heat flux for only a small temperature difference. The lower heat flux experienced during natural convection requires too much surface area in the evaporator to be financially feasible, thus the computer program was designed to reject those situations. While in the region near the transition zone from natural convection to nucleate boiling, both mechanisms may be present. It is assumed that the heat flux due to natural convection is far less significant than the heat flux due to nucleate boiling. The

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mode of stable film boiling is usually only experienced in cases of excess temperatures of orders of magnitude above even the maximum temperature difference available to the system. Film boiling is therefore, not a feasible consideration in this investigation.

Nucleate boiling was considered as the significant mode of evaporative heat transfer in the case of a ' pool ' type heat exchanger configuration (as described in detail in Appendix V). However, another type of beat exchanger configuration investigated is that of a falling film evaporator. Whether the subcooled liquid is sprayed over a bank of tubes or injected into a column, the correlations must be developed to express each individual case. The process then of selecting the correlations which would best express the heat transfer mechanisms in the system, began with a survey of the available literature and sorting out of the possible methods. A comparison and evaluation was made of each. Special attention was given to those correlations that have been experimentally verified with refrigerants.

There are many correlations that have been developed in an effort to describe and predict the phenomenon of nucleate boiling. These may be classified into three basic groups; the first two groups attempt to mathematically describe a particular model of the mechanism taking place. These two groups may be distinguished by the inclusion or exclusion of an arbitrary constant relating the effects of the interaction of the fluid and the surface. The third group is composed of those who make no attempt to model the mechanism, instead taking a purely statistical approach.

The first group of correlations, those that include an arbitrary surface parameter, can best be exemplified by the Rohsenow correlation (18). The second group , those which exclude a surface parameter, might best be exemplified by the correlation of Sterman (11) , and the third by Hughmark (11). While the first group might be very widely accepted, there enters the problem that this surface parameter must be determined experimentally. Very little information is available for the material/ fluid combinations of the rather exotics being considered for use in OTEC. There is little doubt that the third group may be the most reliable in some cases, but from a philosophical point of view it is the least desirable. Hughmark for instance makes no attempt to employ nondimensional quantities, nor is any consideration made for dimensionality. From this standpoint, there is left only the correlations of the second group.

However, there is another consideration that must be given, that of accuracy and reliability. A survey of some thirty correlations was made by Westwater in Reference 11. In this survey Westwater makes an evaluation of each correlation and gives bounds for expected errors. In his evaluation he states that errors of up to 100% result in some cases using the correlation of Rohsenow. He also states that errors of 100% are not uncommon to the correlation of Hughmark. In fact Westwater terms the present situation as an "unsatisfactory state of affairs". A selection was then made on the basis of which correlation seemed to have the least deviations and drawbacks. From this unfortunately rather negative outlook, a correlation for nucleate boiling was selected. that of McNelly (11).

There are three attractive aspects of this correlation. The first is the exclusion of a surface parameter. The justification of this exclusion is given by recognizing the limiting or controlling factors in the nucleation process. It is recognized that for commercial surfaces (that is not polished) there are sufficient nucleation sites if moderate heat flux is maintained. This is to say that, the nucleation sites in this situation are not a limiting factor, and actually exist in excess of that required to maintain stable nucleation. The limiting factor then is the excess temperature (or the available heat flux) and the ability of the fluid to transport the heat away from the surface.

The second attractive aspect of the correlation of McNelly is that it is given in complete nondimensional form (a Nusselt number related to various commonly accepted nondimensional quantities by a constant and exponents). The exponents and the constant were selected to give good fit with data for several liquids, those mentioned being mostly organic (of the type expect ed for use in OTEC). Thirdly, the characteristic length and the dynamic viscosity of the fluid, although they appear in the equation, cancel; thus, as suspected, do not affect the heat flux. The correlation in nondimensional form appears as shown:

$$Nu = 0.225\left(\frac{Dq}{Lu}\right)^{0.69}\left(\frac{pD}{s}\right)^{0.31}\left(\frac{R_L}{R_V}-1\right)^{0.33}\left(\frac{C\mu}{k}\right)^{0.69}$$

where: D = characteristic length
q = heat flux
L = latent heat
μ = dynamic viscosity of the liquid
p = pressure
s = surface tension
R_L = density of liquid
R_V = density of vapor
c = specific heat of liquid
k = thermal conductivity of liquid

This correlation can easily be rearranged to yield a conductance:

$$h = 0.00813\left(\frac{dTc}{L}\right)^{2.23}\left(\frac{R_L}{R_V}-1\right)^{1.065}\left(\frac{kp}{s}\right)$$

where: dT = excess temperature

It was this correlation that was used throughout the analysis of the system in which a 'pool' type of evaporator was assumed.

Location of a correlation for the heat flux to be expected for the case of a falling film over a bank of tubes was not quite so involved, due to the scarcity of information on the subject. No correlation was found that was specifically applicable, however, in examining the correlation of Nusselt for condensation of a film on a bank of tubes and Bromley (l2) (stable film boiling on horizontal tubes) the following can be noticed: The dimensional grouping of the thermophysical properties is the same, the exponent of these properties, and more importantly the exponent of the excess temperature is the same. The only differences being the constant and the phase evaluation of the density, dynamic viscosity, and thermal conductivity. It is only necessary to review the derivation of the equation by Nusselt to determine the applicability of the correlation in this case. In the case of stable film boiling, the vapor is assumed to be in contact with the surface at all times. Whereas in the case of condensation and falling film evaporation, the liquid is assumed to be in contact with the surface at all times. It can be seen that in the case of condensation and falling film evaporation (provided the surface is always wetted by the liquid) the fluid model from which the correlation is derived is precisely the same. Therefore it is

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assumed that the correlation of Nusselt is applicable in this case, an assumption that is reinforced by the correlation of Bromley. The only other possible difference being the constant that only differs by 14% (in the derivation of Nusselt this constant is a direct consequence of the solution). The values predicted by this expression are from 15 to 75% higher than that for nucleate boiling over the range of excess temperatures experienced in the system. The correlation of Nusselt (12):

$$h = 0.725 \left[\frac{\rho_L (\rho_L - \rho_V) gLk^3}{\mu D dt} \right]^{0.25}$$

where: ρ_L = density of liquid
ρ_V = density of vapor
g = gravitational constant
L = latent heat
k = thermal conductivity
μ = dynamic viscosity
D = diameter of the tubes
dt = excess temperature

Perhaps the most pertinent and recent work done in an attempt to correlate heat transfer to boiling refrigerants in vertical tubes has been done in the U. S. S.R. Kleis (8) investigated the boiling of oil free ammonia in a vertical tube. He was able to demonstrate a relationship to the well-known correlation of Kruzhilin (8) for nucleate pool boiling through a slight alteration. Kleis' correlation for ammonia may be reduced to:

$$h = 70 \, dt^{0.33} \left[\frac{BTU}{hr \; ft^2 \; °F} \right]$$

<u>Condensation</u>

The correlations used to predict the heat transfer coefficients in condensation are more widely accepted and do not require extensive justification for their selection. The correlation of Chen (12):

$$h = 0.728 \left[1 + 0.2 dTc \left(\frac{N-1}{L} \right) \right] \left[\frac{g\rho_L (\rho_L - \rho_V) Lk^3}{ND\mu dT} \right]^{0.25}$$

where: ρ_L = density of liquid
ρ_V = density of vapor
g = gravitational constant
L = latent heat
k = thermal conductivity
μ = dynamic viscosity
D = diameter of the tubes

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dt = excess temperature
N = number of horizontal tubes
c = specific heat of liquid

This correlation is reported to be in good agreement with experimental data provided the following condition is satisfied:

$$(N-1)\left(\frac{dT}{L}\right) < 2$$

In the event that this condition is not met, the computer model selects the correlation of Nusselt.

Appendix V. Modeling of Evaporator

In this analysis of the OTEC system, it is the evaporator that is considered to be the most complex subsystem. The difficulty in describing the processes taking place in the evaporator results from the number of degrees of variations expected within this single subsystem.

Description of Variations Expected

Although the standard four degrees of freedom (three space + time) determine the possible planes of variation, time and width are immediately rejected. Variations with width are not expected to be significantly present in the system; therefore, this assumption is not a critical or limiting one. However, variations with time are expected, mainly as a result of adjusting the operating point of the system to take maximum advantage of the external conditions. A transient analysis is always secondary to a steady state analysis. Since a steady state analysis of this type has not been sufficiently investigated and experimentally verified, any transient conditions resulting from unsteady state operation are left to future investigators (as inclusion of any such effects would only obscure the main points of this investigation). The two degrees of freedom remaining are those of variations throughout the length and depth of the evaporator. Variations are expected in fluid properties, surface temperature, seawater temperature, velocity and temperature profile of the seawater, and biogrowth.

Governing Relationships

The governing relationships become the boundary conditions of the mathematical solution of the problem. These relationships are summarized here:

1. The integration of the heat flux (or the total heat transferred) to the working fluid along any one tube or surface member1 must be equal and opposite to the change in enthalpy of the seawater flowing in that member.

2. The total heat added to the working fluid must be equal to the change in enthalpy of the vapor exiting the evaporator.

3. The thermal circuit must be continuous along any arbitrary path (restricted to two dimensions).

Assumptions and Limitations

The following assumptions were made to facilitate solution of the problem:

1. The inlet temperature of the seawater is assumed constant (with depth), although variation is allowed in the outlet temperature (from top to bottom of the heat exchanger).

2. The fluid properties of the seawater, after entering the heat exchanger, remain constant.

3. The temperature and velocity profile of the seawater is assumed to be fully developed and invariant with length (although variation is allowed with depth).

4. The biogrowth and corrosion product build-up are assumed to be constant throughout the heat exchanger.

5. Although a net migration of the working fluid along the tube in the direction of the seawater inlet will occur (due to increased excess temperature available and thus

greater evaporation), it is assumed that this effect will be minimized by partitions (which are necessary structurally).

6. Since only the endpoints of the solution of the two dimensional thermal circuit are required, it is assumed that as a direct result of governing relationship #3), that the two dimensional problem may be solved by determining the three endpoints of any two line segments in the thermal circuit.

Even with the great size of the heat exchangers to be used in OTEC, the temperature of the seawater is generally within 2 degrees over these ranges in depth (Reference 4). The exception might be the existence of a thermocline, a problem which has been solved in the design of some systems. The capability of variable buoyancy to allow the entire system to shift vertically as much as 50 feet in an attempt to locate the evaporators at the depth of maximum temperature has been proposed (Reference 1).

Results of this investigation show, that under the most favorable conditions, it is possible to effect a change in the temperature of the seawater of up to 5 degrees. Although some of the :fluid properties of the seawater (kinematic viscosity and Prandtl number in particular) change almost 7% over this range, the end effect on the thermal conductance is quite less (due to the fact that they have a mutually canceling affect in the correlation used).

Since the variations in the fluid properties of the seawater are small over the length of any single tube, as is the temperature, it is a fair assumption that the temperature and velocity profiles do not vary significantly. The range of Reynolds numbers encountered in this investigation would place the thermal and hydrodynamic entry lengths within the first foot of the tubes. Therefore, treatment of this region of the tubes in which the profiles are not fully developed would not result in significant changes in the solution.

Since very little of the inside of the tubes will be exposed to light, the biogrowth should be dependent only on the hydrodynamic and thermal conditions. The first investigation attempting to relate biogrowth to flow parameters in a true fluids engineering sense has been made by M. A. Wood (Ref. 7). In his investigation he points inconsistencies in the results of former works (mainly performed by biologists and not engineers); and leaves one with the conclusion that no real duplication of results, and certainly no predictions of value to the engineer, can be made using the work of these investigators. The result of Wood' s investigation is the demonstration of a distinct value of wall shear, above which biofoulers are not able to attach to the surface. It is this value (4×10^{-5} psi) that is used as a criterion for initial estimates of the velocities of the seawater in the heat exchanger. The results of Wood, while they show a sharp decline in the biogrowth rate and a leveling off to a rather constant value, do not permit the development of a correlation beyond this point. It is clear from his investigation that it is possible to discourage the formation of biofoulers in the heat exchangers. However, it is assumed that it will not be possible to totally eliminate their attachment only control it.

It is also obvious that a great deal of work is needed in this area; and until such work is done, it would be no less than speculation to attempt to formulate a complex correlation dealing with the biofouling problem from existing information. So that analysis could be made, it was necessary to determine a representative value for the thermal conductance of the biofouling layer. In doing so, all the value s reported by design proposals made at the Third "Workshop on Ocean Thermal Energy Conversion (Reference 1) were averaged and a value of 3333 (BTU/hr/ft/°F) was obtained. Since all of these proposals reported u sing some constant value

(usually arrived at by a subcontractor or independent investigator), it would seem evident that no further correlation can be justified at this time. The values used in this analysis of the thermal conductance of the corrosion product build-up were taken from Perry and Chilton, **Chemical Engineers' Handbook** (Reference 10) and are reportedly representative values under conditions most closely resembling those experienced in OTEC that were available.

Since the thermal circuit must be continuous along any arbitrary path (restricted by the two dimensional assumption), a path was selected which would not add any further assumptions as to the temperature distribution. This path is shown in Figure 9:
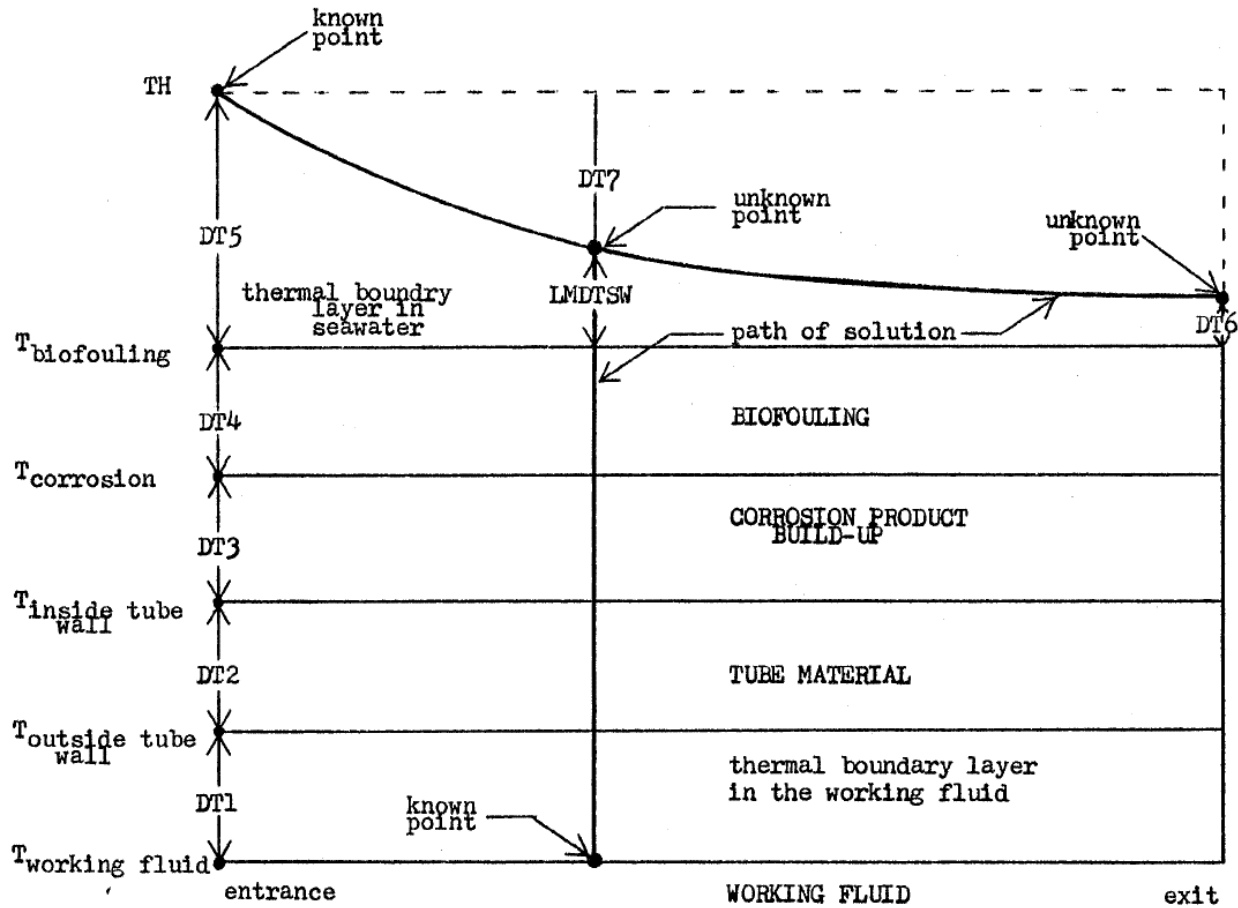
Figure 9. Temperature Differential Profile

of the Thermal Circuit in an Element of the Evaporator

The concept behind this path is to take advantage of the log mean temperature difference (IHTD) of the seawater in accounting for the lateral changes in the thermal circuit. The intersection of the two lines is not located in space; but is merely arrived at mathematically.

<u>Method of Solution</u>

The first two governing relationships are satisfied by the initial values assumed. These values are then used to arrive at an estimate of the intersection point. The discrepancy between the values found from the two initial estimates is reduced by numerical convergence techniques (which are described in Appendix VII). This operation can be used to determine the local

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average heat flux from any tube at any level of the evaporator (either starting at the specified temperature of the working fluid or working backwards from the temperature of the seawater).

The next step in the solution of the problem is to arrive at a figure for the total heat flux into the evaporator. Since the local values can be found (although time consuming), one might be tempted to use an average or even a log mean value. This is a common error made in the design of heat exchangers where evaporation takes place. Discussion of this problem, as well as an outline of the correct solution, is given by Afgan and Schlünder (Reference 5). Comparison of the actual solution to the differential equations (as reported by Schlünder), average, and log mean values, shows errors of up to +25% and -20% respectively.

The differential equations must be solved numerically; and therefore would require even longer computer run time. The technique used in this analysis is the fitting of an assumed solution form to known data points. A simple exponential form is assumed and fit to the endpoints of the known solution (that is the bottom and top of the evaporator). This expression can easily be integrated to produce an estimate of the total heat flux. A graphical comparison of the different possible solutions mentioned is shown in Fig. 10. Comparison of the integration of the numerical solution (using 20 point Gaussian quadrature) with the exponential assumption in most instances proved to be within 1% (and in all cases examined, proved to be more accurate than either the log mean or arithmetic mean).
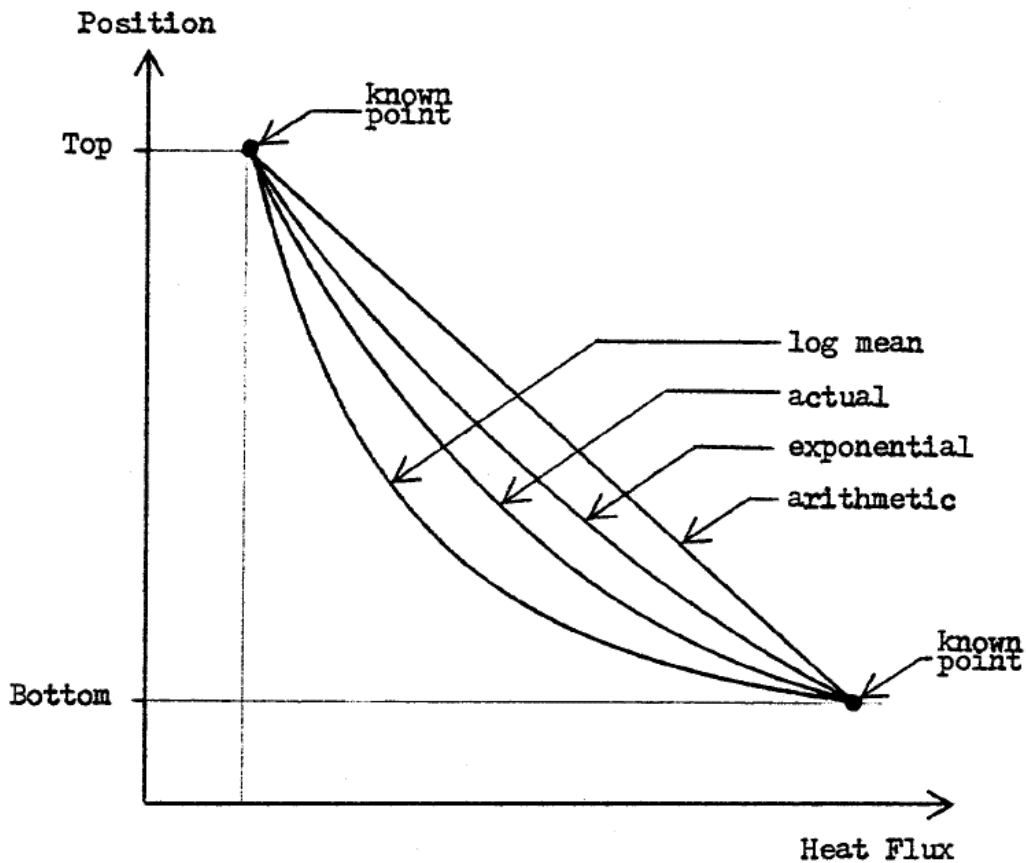
Figure 10. Heat Flux vs. Position in the Evaporator

Appendix VI. Fluid Properties

The properties of the following fluids are given at definition points equally spaced within the range of temperatures expected in the system (or where data is available). The temperatures are expected to range from 40°F to 80°F. All properties are given in English units, as they are prevalent in the current literature on OTEC.

Seawater

All properties of seawater are reported for 'normal' seawater which is accepted in this area as 35 parts per thousand (ppt).

DENSITY:

$$1.9940 \text{ (slugs ft.}^{-3}) @ 40F$$
$$1.9903 \text{ (slugs ft.}^{-3}) @ 60F$$
$$1.9844 \text{ (slugs ft.}^{-3}) @ 80F$$

  data from: Meyers, Holm, and McAllister, Handbook of Ocean and Underwater Engineering (New York, 1969)

KINEMATIC VISCOSITY:

$$1.6846 \times 10^{-5} \text{ (ft.}^2 \text{ sec.}^{-1}) @ 41F$$
$$1.2641 \times 10^{-5} \text{ (ft.}^2 \text{ sec.}^{-1}) @ 60F$$
$$9.8299 \times 10^{-5} \text{ (ft.}^2 \text{ sec.}^{-1}) @ 80F$$

  data from: Meyers, Holm, and McAllister, Handbook of Ocean and Underwater Engineering (New York, 1969)

THERMAL CONDUCTIVITY:

$$0.319 \text{ (B.t.u. hr.}^{-1} \text{ ft.}^{-1} \text{ }^{o}F^{-1}) @ 32F$$
$$0.346 \text{ (B.t.u. hr.}^{-1} \text{ ft.}^{-1} \text{ }^{o}F^{-1}) @ 68F$$
$$0.356 \text{ (B.t.u. hr.}^{-1} \text{ ft.}^{-1} \text{ }^{o}F^{-1}) @ 86F$$

  data from: Smith, Handbook of Marine Science, Vol. 1, (CRC Press, 1974)

SPECIFIC HEAT:

$$0.9524 \text{ (B.t.u. pound}^{-1} \text{ }^{o}F^{-1}) @ 32F$$
$$0.9536 \text{ (B.t.u. pound}^{-1} \text{ }^{o}F^{-1}) @ 59F$$
$$0.9548 \text{ (B.t.u. pound}^{-1} \text{ }^{o}F^{-1}) @ 77F$$

  data from: Smith, Handbook of Marine Science, Vol. 1, (CRC Press, 1974).

PRANDTL NUMBER:

13.35 @ 32F
7.209 @ 68F
5.788 @ 86F

  data from: Smith, Handbook of Marine Science, Vol. 1, (CRC Press, 1974).

DENSITY OF SATURATED LIQUID:

$\quad\quad$ 39.49 (pound ft.$^{-3}$) @ 40F

$\quad\quad$ 38.50 (pound ft.$^{-3}$) @ 60F

$\quad\quad$ 37.48 (pound ft.$^{-3}$) @ 80F

$\quad\quad$ data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973)

ENTHALPY OF SATURATED VAPOR:

$\quad\quad$ 623.0 (B.t.u. pound^{-1}) @ 40F

$\quad\quad$ 627.3 (B.t.u. pound^{-1}) @ 60F

$\quad\quad$ 630.7 (B.t.u. pound^{-1}) @ 80F

$\quad\quad$ data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973)

ENTHALPY OF SATURATED LIQUID:

$\quad\quad$ 86.8 (B.t.u. pound^{-1}) @ 40F

$\quad\quad$ 109.2 (B.t.u. pound^{-1}) @ 60F

$\quad\quad$ 132.0 (B.t.u. pound^{-1}) @ 80F

$\quad\quad$ date from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973)

ENTROPY OF SATURATE VAPOR:

$\quad\quad$ 1.2618 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 40F

$\quad\quad$ 1.2294 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 60F

$\quad\quad$ 1.1911 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 80F

$\quad\quad$ data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973)

ENTROPY OF SATURATED LIQUID:

$\quad\quad$ 0.1885 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 40F

$\quad\quad$ 0.2322 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 60F

$\quad\quad$ 0.2749 (B.t.u. pound^{-1} $^{\circ}$F$^{\text{ol}}$) @ 80F

$\quad\quad$ data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

SPECIFIC HEAT OF SATURATED LIQUID:

$\quad\quad$ 1.107 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 32F

$\quad\quad$ 1.126 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 50F

$\quad\quad$ 1.146 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 68F

$\quad\quad$ data from: Rosenow and Hartnett, Handbook of Heat Transfer, (New York, 1973).

SPECIFIC HEAT OF SATURATED VAPOR:

$\quad\quad$ 0.395 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 40F

$\quad\quad$ 0.418 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 60F

$\quad\quad$ 0.491 (B.t.u. pound^{-1} $^{\circ}$F^{-1}) @ 80F

$\quad\quad$ data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

CONDUCTIVITY OF SATURATED LIQUID:
$$0.312 \ (B.t.u. \ hr.^{-1} \ ft.^{-1} \ {}^{\circ}F^{-1}) \ @ \ 32F$$
$$0.307 \ (B.t.u. \ hr.^{-1} \ ft.^{-1} \ {}^{\circ}F^{-1}) \ @ \ 50F$$
$$0.301 \ (B.t.u. \ hr.^{-1} \ ft.^{-1} \ {}^{\circ}F^{-1}) \ @ \ 68F$$

data from: Rosenow and Hartnett, Handbook of Heat Transfer, (New York, 1973).

PRANDTL NUMBER OF SATURATED LIQUID:
2.05 @ 32F
2.04 @ 50F
2.02 @ 68F

data from: Rosenow and Hartnett, Handbook of Heat Transfer, (New York, 1973).

SATURATION PRESSURE:
73.32 (psia) @ 40F
107.6 (psia) @ 60F
153.0 (psia) @ 80F

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

DYNAMIC VISCOSITY OF SATURATED LIQUID:
$$0.431 \ (pound \ hr.^{-1} \ ft.^{-1}) \ @ \ 40F$$
$$0.381 \ (pound \ hr.^{-1} \ ft.^{-1}) \ @ \ 60F$$
$$0.337 \ (pound \ hr.^{-1} \ ft.^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

SURFACE TENSION (IN SATURATED STATE):
$$1.603 \times 10^{-3} \ (pound \ ft.^{-1}) \ @ \ 53F$$
$$1.240 \times 10^{-3} \ (pound \ ft.^{-1}) \ @ \ 93.4F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

Propane

DENSITY OF SATURATED LIQUID:
$$32.7340 \ (pound \ ft.^{-3}) \ @ \ 40F$$
$$31.7500 \ (pound \ ft.^{-3}) \ @ \ 60F$$
$$30.7030 \ (pound \ ft.^{-3}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

DENSITY OF SATURATED VAPOR:
$$0.7416835 \ (pound \ ft.^{-3}) \ @ \ 40F$$
$$1.01037 \ (pound \ ft.^{-3}) \ @ \ 60F$$
$$1.35382 \ (pound \ ft.^{-3}) \ @ \ 80F$$
$$0.373214 \ (pound \ ft.^{-3}) \ @ \ 0F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

ENTHALPY OF SATURATED VAPOR:

$$202.516 \ (\text{B.t.u. pound}^{-1}) \ @ \ 40F$$
$$207.527 \ (\text{B.t.u. pound}^{-1}) \ @ \ 60F$$
$$202.284 \ (\text{B.t.u. pound}^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

ENTHALPY OF SATURATED LIQUID:

$$45.722 \ (\text{B.t.u. pound}^{-1}) \ @ \ 40F$$
$$57.976 \ (\text{B.t.u. pound}^{-1}) \ @ \ 60F$$
$$70.605 \ (\text{B.t.u. pound}^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

ENTROPY OF SATURATED VAPOR:

$$0.41265 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 40F$$
$$0.41035 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 60F$$
$$0.40854 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

ENTROPY OF SATURATED LIQUID:

$$0.09896 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 40F$$
$$0.12258 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 60F$$
$$0.14602 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

SPECIFIC HEAT OF SATURATED LIQUID:

$$0.5932 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 40F$$
$$0.5714 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 60F$$
$$0.4810 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

SPECIFIC HEAT OF SATURATED VAPOR:

$$0.4507 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 40F$$
$$0.4810 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 60F$$
$$0.5174 \ (\text{B.t.u. pound}^{-1} \ {}^{o}F^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

CONDUCTIVITY OF SATURATED LIQUID:

$$0.0618 \ (\text{B.t.u. hr.}^{-1} \ \text{ft.}^{-1} \ {}^{o}F^{-1}) \ @ \ 40F$$
$$0.0587 \ (\text{B.t.u. hr.}^{-1} \ \text{ft.}^{-1} \ {}^{o}F^{-1}) \ @ \ 60F$$
$$0.0567 \ (\text{B.t.u. hr.}^{-1} \ \text{ft.}^{-1} \ {}^{o}F^{-1}) \ @ \ 80F$$

data from: ASHRAE, Handbook of Fundamentals, (ASHRAE, 1973).

SATURATION PRESSURE:
        38.3443 (psia) @ 0F
        78.5210 (psia) @ 40F
        107.5323 (psia) @ 60F
        143.7665 (psia) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

DYNAMIC VISCOSITY OF SATURATED LIQUID:
        0.313 (pound hr.$^{-1}$ ft.$^{-1}$) @ 40F
        0.286 (pound hr.$^{-1}$ ft.$^{-1}$) @ 60F
        0.255 (pound hr.$^{-1}$ ft.$^{-1}$) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

SURFACE TENSION (IN SATURATED STATE):
        4.52×10^{-4} (pound ft.$^{-1}$) @ 77F

    datum from: Bolez and Tuve, _Handbook of Tables for Engineering Science_, (CRC Press, 1976).

Ethane

DENSITY OF SATURATED LIQUID:
        25.06 (pound ft.$^{-3}$) @ 40F
        22.95 (pound ft.$^{-3}$) @ 60F
        19.75 (pound ft.$^{-3}$) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

DENSITY OF SATURATED VAPOR:
        1.738 (pound ft.$^{-3}$) @ 0 F
        3.266 (pound ft.$^{-3}$) @ 40F
        4.621 (pound ft.$^{-3}$) @ 60F
        7.087 (pound ft.$^{-3}$) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

ENTHALPY OF SATURATED VAPOR:
        403.90 (B.t.u. pound^{-1}) @ 0 F
        404.50 (B.t.u. pound^{-1}) @ 40F
        401.30 (B.t.u. pound^{-1}) @ 60F
        391.40 (B.t.u. pound^{-1}) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

ENTHALPY OF SATURATED LIQUID:
        281.00 (B.t.u. pound^{-1}) @ 40F
        299.30 (B.t.u. pound^{-1}) @ 60F
        323.70 (B.t.u. pound^{-1}) @ 80F

    data from: ASHRAE, _Handbook of Fundamentals_, (ASHRAE, 1973).

ENTROPY OF SATURATED VAPOR:

$$1.5234 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 40F$$
$$1.5064 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 60F$$
$$1.4751 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 80F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

ENTROPY OF SATURATED LIQUID:

$$1.2762 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 40F$$
$$1.3100 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 60F$$
$$1.3505 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 80F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

SPECIFIC HEAT OF SATURATED LIQUID:

$$0.6931 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 0 \; F$$
$$0.7345 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 20F$$
$$0.8091 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 40F$$
$$0.9314 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 60F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

SPECIFIC HEAT OF SATURATED VAPOR:

$$0.6060 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 20F$$
$$0.7413 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 40F$$
$$0.9040 \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 60F$$
$$1.041 \; \; (B.t.u. \; pound^{-1} \; {}^{\circ}F^{-1}) \; @ \; 70F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

CONDUCTIVITY OF SATURATED LIQUID:

$$0.0501 \; (B.t.u. \; hr.^{-1} \; ft.^{-1} \; {}^{\circ}F^{-1}) \; @ \; 40F$$
$$0.0458 \; (B.t.u. \; hr.^{-1} \; ft.^{-1} \; {}^{\circ}F^{-1}) \; @ \; 60F$$
$$0.0391 \; (B.t.u. \; hr.^{-1} \; ft.^{-1} \; {}^{\circ}F^{-1}) \; @ \; 80F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

SATURATION PRESSURE:

$$385.00 \; (psia) \; @ \; 40F$$
$$494.20 \; (psia) \; @ \; 60F$$
$$630.70 \; (psia) \; @ \; 80F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

DYNAMIC VISCOSITY OF SATURATED LIQUID:

$$0.135 \; (pound \; hr.^{-1} \; ft.^{-1}) \; @ \; 40F$$
$$0.114 \; (pound \; hr.^{-1} \; ft.^{-1}) \; @ \; 60F$$
$$0.088 \; (pound \; hr.^{-1} \; ft.^{-1}) \; @ \; 80F$$

data from: ASHRAE, *Handbook of Fundamentals*, (ASHRAE, 1973).

SURFACE TENSION (IN SATURATED STATE):
see Appendix VII, part 2 for derived value.

Appendix VII. Numerical Approximation of Fluid Properties

The fluid properties have been approximated numerically to allow incorporation into the computer program of continuous expressions over the range of expected temperatures.

Selection of Methods and Error Bounds

The method of polynomial regression was chosen to express all fluid properties (except surface tension) for three basic reasons: First, and foremost, is the rapidity of evaluation by the computer. The execution times of various mathematical operations were compared. From this comparison it was determined that addition and multiplication (even if done repeatedly) were the least time consuming. Polynomials of up to degree five were examined (it was noticed that often polynomials of higher order resulted in less accuracy). The second reason being the ease of correlation and comparison. Finally, polynomial regression is strictly a statistical technique and does not involve extensive theory exterior to the main thrust of this investigation.

The criteria for the error bounds was seta to be as precise but simplistic as possible in the case of lim1ted data (if datum points were not available other than those used to determine the expression), or to be within one digit of the final significant figure given by the same reference at all points within the upper and lower bound. This last criterion was satisfied for applicable cases by a parabolic or cubic polynomial.

Correlation of Surface Tension

Due to a limited number of data points determining the surface tension of the working fluids in question; it was necessary to find a theory that would be reliable in extrapolating the known data beyond the range of temperatures specified by the reference. A survey was made of several techniques; and examples of calculations relating to known data were evaluated. From this evaluation it was determined that the most reliable correlation investigated was that of Brock and Bird (14). This method allows the approximation of the surface tension of any fluid, provided other critical data are known. The result of the approximation can be summed up by the calculation of a quantity that might be referred to as 'σ_0'. This quantity can be used to determine the surface tension at any temperature from the following expression:

$$\sigma = \sigma_0 \left(1 - \frac{T}{T_{CRIT}} \right)^{\frac{11}{9}}$$

The most significant correlation being drawn is the relationship of the surface tension at any reduced temperature (temperature/critical temperature) by an eleven-ninth's power law. This correlation using the eleven-ninth's power law was checked against the known values for ammonia and proved to be precise.

At the time of compilation, it was not possible to locate reliable information concerning the interfacial tension of ethane liquid and vapor in the saturated state. The method of Brock and Bird was therefore used to determine the value of ' SIGMA? ' • In justification of this, the same values corresponding to other saturated hydrocarbons of the paraffin series (namely hexane, octane, and propane) along with other fluids (toluene and Freon-12) were determined and compared to their known values. The errors calculated were t hexane 0. 22%, octane 1.1%, propane 4.16% (which was the largest error), toluene 1.8%, and Freon-12 2.3%. It was the claim

of Brock and Bird, that the method yields an average error less than 3%. By demonstration this would seem to be reasonable.

Seawater

DENSITY:
$$RHO = 1.9948 + 9.\times10^{-5} * T - 2.75\times10^{-6} * T^2$$

KINEMATIC VISCOSITY:
$$RNUE = 3.1014\times10^{-5} - 4.3407\times10^{-7} * T + 2.0708\times10^{-9} * T^2$$

THERMAL CONDUCTIVITY:
$$RK = 0.2876 + 1.113\times10^{-3} * T - 3.733\times10^{-6} * T^2$$

SPECIFIC HEAT:
$$C = 0.9519 + 4.938\times10^{-7} * (T^2 - T)$$

PRANDTL NUMBER:
$$PR = 22.503 - 0.3403 * T + 1.697\times10^{-3} * T^2$$

Ammonia

DENSITY OF SATURATED LIQUID:
$$RL = 41.38 - 0.04575 * T - 3.75\times10^{-5} * T^2$$

DENSITY OF SATURATED VAPOR:
$$RV = 0.1374 + 1.043\times10^{-3} * T + 4.541\times10^{-5} * T^2$$

ENTHALPY OF SATURATED VAPOR:
$$HV = 611.7 + 0.3275 * T - 1.125\times10^{-3} * T^3$$

ENTHALPY OF SATURATED LIQUID:
$$HL = 43.20 + 1.070 * T + 5.\times10^{-4} * T^2$$

ENTROPY OF SATURATED VAPOR:
$$SV = 1.3329 - 1.8825\times10^{-3} * T + 2.625\times10^{-6} * T^2$$

ENTROPY OF SATURATED LIQUID:
$$SL = 0.0981 + 2.31\times10^{-3} * T - 1.25\times10^{-6} * T^2$$

SPECIFIC HEAT OF SATURATED LIQUID:
$$CPL = 1.076 + 9.29\times10^{-4} * T + 1.543\times10^{-6} * T^2$$

SPECIFIC HEAT OF SATURATED VAPOR:
$$CPV = 0.3490 + 1.15\times10^{-3} * T$$

CONDUCTIVITY OF SATURATED LIQUID:
$$RKL = 0.318 - 1.512\times10^{-4} * T - 1.534\times10^{-6} * T^2$$

SATURATION PRESSURE:
$$PSAT = 38.12 + 0.324 * T + 0.0139 * T^2$$

DYNAMIC VISCOSITY OF SATURATED LIQUID:
$$DVL = 0.549 - 3.25X10^{-3} * T + 7.5X10^{-6} * T^2$$

SURFACE TENSION:
$$STEN = 0.006983 * ((1. - ((T + 460.)/731.))^{1.222})$$

Propane

DENSITY OF SATURATED LIQUID:
$$RL = 34.513 - 0.04133 * T - 7.875 * T^2$$

DENSITY OF SATURATED VAPOR:
$$RV = 0.4286 + 0.004089 * T + 9.3454X10^{-5} * T^2$$

ENTHALPY OF SATURATED VAPOR:
$$HV = 191.73 + 0.2823 * T - 3.175X10^{-4} * T^2$$

ENTHALPY OF SATURATED LIQUID:
$$HL = 22.639 + 0.55708 * T + 5.3125X10^{-4} * T^2$$

ENTROPY OF SATURATED VAPOR:
$$SV = 0.41872 - 1.762X10^{-4} * T + 6.125X10^{-7} * T^2$$

ENTROPY OF SATURATED LIQUID:
$$SL = 0.05118 + 1.2035X10^{-3} * T - 2.25X10^{-7} * T^2$$

SPECIFIC HEAT OF SATURATED LIQUID:
$$CPL = 0.5743 + 1.475X10^{-4} * T + 8.125X10^{-6} * T^2$$

SPECIFIC HEAT OF SATURATED VAPOR:
$$CPV = 0.4084 + 7.525X10^{-4} * T + 7.62X10^{-6} * T^2$$

CONDUCTIVITY OF SATURATED LIQUID:
$$RKL = 0.0713 - 2.925X10^{-4} * T + 1.37X10^{-6} * T^2$$

SATURATION PRESSURE:
$$PSAT = 42.167 + 0.5477 * T + 0.0090286 * T^2$$

DYNAMIC VISCOSITY OF SATURATED LIQUID:
$$DVL = 0.355 - 8X10^{-4} * T - 5.X10^{-6} * T^2$$

SURFACE TENSION:
$$STEN = 0.00336 * ((1. - ((1. - ((T + 460.)/666.))^{1.222})$$

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Ethane

DENSITY OF SATURATED LIQUID:
$$RL = 26.01 + 0.03075 * T - 0.001362 * T^2$$

DENSITY OF SATURATED VAPOR:
$$RV = 1.738 + 0.04539 * T - 6.278 \times 10^{-4} * T^2 + 1.12 \times 10^{-5} * T^3$$

ENTHALPY OF SATURATED VAPOR:
$$HV = 403.9 - 0.03208 * T + 0.003906 * T^2 - 6.823 \times 10^{-5} * T^3$$

ENTHALPY OF SAUTRATED LIQUID:
$$HL = 262.7 + 0.1525 * T + 7.625 \times 10^{-3} * T^2$$

ENTROPY OF SATURATED VAPOR:
$$SV = 1.5145 + 9.375 \times 10^{-4} * T - 1.7875 \times 10^{-5} * T^2$$

ENTROPY OF SATURATED LIQUID:
$$SL = 1.2287 + 8.525 \times 10^{-4} * T + 8.375 \times 10^{-6} * T^2$$

SPECIFIC HEAT OF SATURATED LIQUID:
$$CPL = 0.6931 + 1.482 \times 10^{-3} * T + 2.339 \times 10^{-5} * T^2 + 3.021 \times 10^{-7} * T^3$$

SPECIFIC HEAT OF SATURATED VAPOR:
$$CPV = 0.3529 + 0.01802 * T - 3.288 \times 10^{-4} * T^2 + 3.025 \times 10^{-6} * T^3$$

CONDUCTIVITY OF SATURATED LIQUID:
$$RKL = 0.0515 + 9.5 \times 10^{-5} * T - 3. \times 10^{-6} * T^2$$

SATURATION PRESSURE:
$$PSAT = 248.5 + 2.048 * T - 0.03413 * T^2$$

DYNAMIC VISCOSITY OF SATURATED LIQUID:
$$DVL = 0.162 - 4.25 \times 10^{-4} * T - 6.25 \times 10^{-6} * T^2$$

SURFACE TENSION:
$$STEN = 0.003382 * ((1. - ((T + 460.)/550.1))^{1.222})$$

Appendix VIII. Computer Program

A computer program was developed to effect rapid analysis of the various segments of the system over a range of selected conditions. The task of assemblage and interfacing of the segments was accomplished by successive iteration and numerical convergence techniques. Once the system is assembled, a survey of the operating points, internal and external parameters, and cost estimates can be made to determine the configuration of maximum economy.

Description of Technique

The computer program was originally coded in FORTRAN V and consists of a main program, seven primary subroutines, and seventeen secondary subroutines. The main program controls the input/output, defines the system segments, reassembles the segments, and effects maximization. The seven primary subroutines assist the main program in the analysis of the overall system and the analysis of the components. The seventeen secondary subroutines serve to supply the fluid properties (five pertaining to the seawater and twelve pertaining to the working fluid).

The computer program is d1v.ided into these twenty-five sections for three basic reasons: 1) versatility, 2) the necessity of using similar operations at several points throughout the program, and 3) to allow rapid change of working fluids. Although the program as it appears here is structured to analyze shell and tube heat exchangers, it can easily be altered to handle other configurations without disrupting the flow of logic.

The main program is composed of an input and initialization section, five nested loops, and an output section. The input and initialization section defines the external parameters, initializes variables, and determines internal parameters (such as the gross properties of the seawater, theoretical maximums, and normalization factors). The five nested loops can be broken down into two parts& the outermost three loops which are external to the Rankine Cycle Simulator and the innermost two which are completely within the Rankine Cycle Simulator and do not affect its definition points. The loops are defined by the change of the following variables (in order of increasing inwardness) a diameter of the tubes, length of the tubes, temperatures of the working fluid (definition points of the Rankine cycle), velocity of the seawater in the evaporator, and the velocity of the seawater in the condenser. Maximization occurs in all but the outermost two loops, thus the output section lists only the results of three levels of maximization.

Subroutine 'EVAP' which analyses the evaporator, consists of five distinct parts: initialization, simulation of the heat transfer process at the bottom of the evaporator, simulation of the heat transfer process at the top of the evaporator, re-entrant convergence routine, and computation of dependent variables. The initialization section defines estimates (from information gathered from previous runs) of the solution to the operating points, thermal conductances, and overall heat flux. The simulation sections compute estimates of the thermal circuits and approximate the excess temperatures available to the evaporation process (with this, subroutine 'EVAPK' returns values for the boiling heat conductances). The convergence routine consists of four nested loops that improve the accuracy of the thermal circuit. The computation section determines such dependent variables as overall heat flux, size requirements of the evaporator, changes in temperature of the seawater, and pumping work.

Subroutine 'COND' which analyzes the condenser, operates in very much the same way as 'EVAP'. The major difference between the two is simpler description of the process allowed by the essentially isothermal assumption as described in Appendix IV.

The flowchart of the computer program (immediately following) shows the flow of logic and the order of computation. The flowchart is broken down by section and shows only the main program and two major subroutines (as the logic of the other subroutines is too simplistic to warrant separate flowcharts).

```
┌─────────────────────────────────────────┐
│ DEFINE   OVERALL   SYSTEM   PARAMETERS   │
│                                          │
│     Thermal Gradient Available           │
│     Depth from Which Cold Water          │
│          Must Be Raised                  │
│     Properties of the Heat Exchanger     │
│          Material                        │
│     Assumed Efficiencies                 │
│     Cost Factors                         │
│     Minimum Wall Shear for Tolerable     │
│          Biogrowth Rate                  │
│     'NPASS'                              │
│     Determine Gross Properties of        │
│          Seawater                        │
│     Define the Theoretical Available     │
│          Power                           │
└─────────────────────────────────────────┘
                    │
                    ▼
┌─────────────────────────────────────────┐
│ PRINT   OUTPUT   HEADINGS   AND   INPUT  │
│              INFORMATION                 │
└─────────────────────────────────────────┘
                    │
                    ▼
┌─────────────────────────────────────────┐
│ SELECT   TUBE   DIAMETER                 │
│                                          │
│     Determine Corresponding Required     │
│          Wall Thickness                  │◄──────┐
│     Determine Minimum Velocities         │       │
└─────────────────────────────────────────┘       │
                    │                              │
                    ▼                              │
                  ╱   ╲                            │
               ╱         ╲                         │
            ╱  IF(D.GT.0.21) ╲ ─────────► STOP     │
               ╲         ╱                         │
                  ╲   ╱                            │
                    │                              │
                    ▼                              │
┌─────────────────────────────────────────┐       │
│ SELECT   TUBE   LENGTH                   │◄──────┤
└─────────────────────────────────────────┘       │
                    │                              │
                    ▼                              │
```

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Subroutine 'EVAP'

```
┌─────────────────────────────────────┐
│  INITIALIZE  VARIABLES              │
└─────────────────────────────────────┘
                  │
                  ▼
┌─────────────────────────────────────┐
│  CALCULATE  'HE1'                   │◄──────────┐
└─────────────────────────────────────┘           │
                  │                                │
                  ▼                                │
┌─────────────────────────────────────┐           │
│  APPROXIMATE  'Q1','DTSW1'          │           │
└─────────────────────────────────────┘           │
                  │                                │
                  ▼                                │
┌─────────────────────────────────────┐           │
│  IMPROVE  ESTIMATE  OF  'Q1'        │◄──────┐    │
│     CALCULATE  'DTSW1','LDTSW1','DT6'│       │    │
└─────────────────────────────────────┘       │    │
                  │                            │    │
                  ▼                            │    │
┌─────────────────────────────────────┐       │    │
│  IMPROVE  ESTIMATE  OF  'DT6'       │◄──┐    │    │
└─────────────────────────────────────┘   │    │    │
                  │                        │    │    │
                  ▼                        │    │    │
              ╱────────╲                   │    │    │
            ╱            ╲                  │    │    │
          ╱  IF(I.LE.NPASS) ╲──────────────┘────┘    │
          ╲                ╱                          │
            ╲            ╱                            │
              ╲────────╱                              │
                  │                                   │
                  ▼                                   │
┌─────────────────────────────────────┐              │
│  CALCULATE  'DT7'                   │              │
└─────────────────────────────────────┘              │
                  │                                   │
                  ▼                                   │
              ╱────────╲                              │
            ╱            ╲                             │
          ╱  IF(N.LE.NPASS) ╲──────────────────────────┘
          ╲                ╱
            ╲            ╱
              ╲────────╱
                  │
                  ▼
┌─────────────────────────────────────┐
│  IMPROVE  ESTIMATE  OF  'HB1'       │◄──────┐
└─────────────────────────────────────┘       │
                  │                            │
                  ▼                            │
```

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IF(.N.LE.NPASS)

CALCULATE 'HE2'

APPROXIMATE 'Q2','DTSW2'

IMPROVE ESTIMATE OF 'Q2'
    CALCULATE 'DTSW2','LDTSW2','DT6'

IMPROVE ESTIMATE OF 'DT6'

IF(I.LE.NPASS)

CALCULATE 'DT7'

IF(N.LE.NPASS)

IMPROVE ESTIMATE OF 'HB2'

IF(N.LE.NPASS)

IF(M.LE.NPASS)

DETERMINE OPERATING POINTS
CALCULATE DEPENDANT VARIABLES

RETURN

Subroutine 'COND'


```
┌─────────────────────────────────┐
│ INITIALIZE  VARIABLES           │
└─────────────────────────────────┘
                 │
                 ▼
┌─────────────────────────────────┐
│ CALCULATE  'HK'                 │◄──────┐
└─────────────────────────────────┘       │
                 │                         │
                 ▼                         │
┌─────────────────────────────────┐       │
│ APPROXIMATE  'QC'               │       │
└─────────────────────────────────┘       │
                 │                         │
                 ▼                         │
┌─────────────────────────────────┐       │
│ IMPROVE  ESTIMATE  OF  'QC'     │◄────┐ │
│   CALCULATE  'DTSWC','LMTDSW','DT6' │  │ │
└─────────────────────────────────┘     │ │
                 │                       │ │
                 ▼                       │ │
┌─────────────────────────────────┐     │ │
│ IMPROVE  ESTIMATE  OF  'DT6'    │◄──┐ │ │
└─────────────────────────────────┘   │ │ │
                 │                     │ │ │
                 ▼                     │ │ │
            ╱─────────╲               │ │ │
           ╱ IF(I.LE.  ╲──────────────┘ │ │
           ╲ NPASS)    ╱────────────────┘ │
            ╲─────────╱──────────────────┘
                 │
                 ▼
┌─────────────────────────────────┐
│ CALCULATE  'DT7'                │
└─────────────────────────────────┘
                 │
                 ▼
            ╱─────────╲
           ╱ IF(N.LE.  ╲───────────────────┐
           ╲ NPASS)    ╱                    │
            ╲─────────╱                     │
                 │                          │
                 ▼                          │
┌─────────────────────────────────┐        │
│ CALCULATE  'DTC','HC'           │        │
└─────────────────────────────────┘        │
                 │                          │
                 ▼                          │
┌─────────────────────────────────┐        │
│ IMPROVE  ESTIMATE  OF  'HC'     │◄───┐   │
└─────────────────────────────────┘    │   │
                 │
                 ▼
```

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I have completely rewritten the program in C and modernized the structure:

```c
#define _CRT_SECURE_NO_DEPRECATE
#include <stdio.h>
#include <conio.h>
#include <stdlib.h>
#define _USE_MATH_DEFINES
#include <math.h>

char*fluid="Propane";
double ktw=11.56;
double den=280.9;
double hbioe=1./0.005;
double hbioc=1./0.005;
double hcorre=1./0.002;
double hcorrc=1./0.002;
double wsmin=4E-5;

int iprt=0;
double cost=10.;
double cstfct=2.;
double depth=1200.;
double dpp=125.;
double effgen=0.95;
double effpmp=0.90;
double efftur=0.85;
double eswpmp=0.76;
double th=78.;
double tl=45.;

double
ac,ae,carnot,cl,costc,coste,cswc,cswe,d,de,dhpump,dptmax,drho,dtp;
double
eh,el,h1,h2,h3,h4,hl,hs2,hv,hw,kc,ke,l,od,p1,p2,p3,p4,pavabl,pc,pe;
double
ph,pl,pout,prc,pre,r3,rhohgh,rholow,rnuc,rnue,rpout,s1,sl2,ss2,sv2,t1;
double
t2,t3,t4,tw,uc,ue,v3,vc,vcmin,ve,vemin,wpd,wsc,wse,wtic,wtie,wtpera,xs
2;

double ff(double Re)
  {
  if(Re>1E5)
    return 1.02/pow(log(Re),2.5);
  if(Re>4000.)
    return 0.316/pow(Re,0.25);
  else if(Re>2000.)
    {
    double f1,f2;
    f1=64./Re;
    f2=0.316/pow(Re,0.25);
    return f1+(f2-f1)*(Re/2000.-1.);
```

```c
     }
   return 64./Re;
   }


double rhosw(double T)
   {
   return 32.2*((-2.75E-6*T+9E-5)*T+1.9948);
   }


double prnum(double T)
   {
   return (1.097E-3*T-0.3403)*T+22.503;
   }


double consw(double T)
   {
   return (-3.733E-6*T+1.113E-3)*T+0.2876;
   }


double visct(double T)
   {
   return (2.0708E-9*T-4.3047E-7)*T+3.1014E-5;
   }


double cpsw(double T)
   {
   return 0.9519+4.938E-7*T*(T-1.);
   }


double rhov(double T)
   {
   return ((2.88474E-7*T+4.15291E-5)*T+7.08902E-3)*T+0.373214;
   }


double rhol(double T)
   {
   return (-7.875E-5*T-0.04133)*T+34.513;
   }


double psat(double T)
   {
   return ((0.1991E-5*T+0.0054448)*T+0.75477)*T+38.3443;
   }


double enthv(double T)
   {
   return (-3.175E-4*T+0.2833)*T+191.73;
   }


double enthl(double T)
   {
   return (5.3125E-4*T+0.55708)*T+22.639;
```

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```c
    }

double entrv(double T)
    {
    return (6.125E-7*T-1.7625E-4)*T+0.41872;
    }

double entrl(double T)
    {
    return (-2.25E-7*T+1.2035E-3)*T+0.05118;
    }

double cpv(double T)
    {
    return (7.625E-6*T+7.525E-4)*T+0.4084;
    }

double cpl(double T)
    {
    return (8.125E-6*T+1.475E-4)*T+0.5743;
    }

double condl(double T)
    {
    return (1.375E-6*T-2.925E-4)*T+0.0713;
    }

double dvisl(double T)
    {
    return (-5E-6*T-8.5E-4)*T+0.355;
    }

double sten(double T)
    {
    return 0.00336*pow(1.-(T-460.)/666.,1.2222);
    }

double evapk(double tv,double tl,double dt)
    {/* local boiling heat transfer coefficient using McNelley's
correlation */
    double cl,hfg,hl,hv,ps,rkl,rl,rv,s;
    hv=enthv(tv);
    hl=enthl(tl);
    hfg=hv-hl;
    rv=rhov(tv);
    rl=rhol(tl);
    ps=psat(tv);
    cl=cpl(tl);
    s=sten(tl);
    rkl=condl(tl);
    return 0.00813*pow(dt*cl/hfg,2.23)*pow((rl/rv)-
1.,1.065)*ps*144.*rkl/s;
```

```c
  }

void evap(double th,double*t1,double*t4,double h1,double h4,double hw,
  double l,double d,double rhohgh,double rnue,double pre,double ke,
  double cswe,double*ReE,double*wse,double*dtswe,double*hb,double*ue,
  double ve,double*ae,double*tubes,double*ppe,double hbioe,double
hcorre)
  {
  int ibis;
  double
acs,dt1,dt2,dt3,dt4,dtperq,fe,hb1,hb2,hee,q1,q2,q3,q4,qe,qm,qx,tsw,ue1
,ue2;

/* model the heat transfer process in the evaporator */
/* Reynolds number, friction factor, convective heat transfer
coefficient,
   and wall shear stress */

  *ReE=ve*d/rnue/12.;
  fe=ff(*ReE);
  hee=12.*0.023*pow(*ReE,0.8)*pow(pre,0.3)*ke/d;
  *wse=fe*rhohgh*ve*ve/2./144./32.174/4.;

/* change in temperature of the seawater per unit heat transfered */

  dtperq=4.*12.*l/rhohgh/cswe/d/ve/3600.;

/* evaporative heat transfer coefficients */

  dt1=th-*t4;
  hb1=evapk(*t1,*t4,dt1/2.);
  dt2=th-*t1;
  hb2=evapk(*t1,*t1,dt2/2.);

/* overall heat transfer coefficients */

  ue1=1./(1./hb1+1./hee+1./hcorre+1./hbioe+1./hw);
  ue2=1./(1./hb2+1./hee+1./hcorre+1./hbioe+1./hw);

/* heat transfer process at the top of the evaporator using bisection
search */

  qm=0.;
  qx=ue1*dt1;
  for(ibis=1;ibis<=20;ibis++)
    {
    q1=(qm+qx)/2.;
    tsw=th-q1*dtperq;
    dt3=th-tsw;
    if(dt3>0.)
      {
      q3=ue1*(dt1-dt3)/log(dt1/dt3);
```

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```
      if(q1<=q3)
         {
         qm=q1;
         continue;
         }
      }
   qx=q1;
   }

/* heat transfer process at the bottom of the evaporator using
bisection search */

   qm=0.;
   qx=ue2*dt2;
   for(ibis=1;ibis<=20;ibis++)
      {
      q2=(qm+qx)/2.;
      tsw=th-q2*dtperq;
      dt4=th-tsw;
      if(dt4>0.)
         {
         q4=ue2*(dt2-dt4)/log(dt2/dt4);
         if(q2<=q4)
            {
            qm=q2;
            continue;
            }
         }
      qx=q2;
      }

/* mean heat transfer and seawater temperature */

   qe=(q1-q2)/log(q1/q2);
   *ue=(ue1-ue2)/log(ue1/ue2);
   *hb=(hb1-hb2)/log(hb1/hb2);
   *dtswe=dtperq*qe;

/* required specific area */

   *ae=(h1-h4)/qe;
   *tubes=12.*(*ae)/M_PI/d/l;
   acs=M_PI*d*d*(*tubes)/144./4.;

/* required pumping power */


*ppe=(rhohgh*ve*acs)*(ve*ve/2.)*((l/(d/12.))*fe+1.5)*3600./778.3/32.2;
   }

double condk(double t2,double dt,double od)
   {
```

```c
    double cp,dr,dv,g,hfg,hfgp,hl,hv,rct,rkl,rl,rows,rv;

/* local condensation heat transfer coefficient */

    rows=20.;
    g=4.1698e8;
    hv=enthv(t2);
    hl=enthl(t2);
    hfg=hv-hl;
    cp=cpv(t2);
    hfgp=hfg+0.375*cp*dt;
    rv=rhov(t2);
    rl=rhol(t2);
    dr=rl-rv;
    rkl=condl(t2);
    dv=dvisl(t2);
    rct=(rows-1.)*cp*dt/hfg;

/* Chen's correlation */

    if(rct<2.)
      return 0.728*(1.+0.2*cp*dt*(rows-
1.)/hfg)*pow(g*rl*dr*rkl*rkl*rkl*hfgp/rows/od/dv/dt,0.25);

/* Nusselt's correlation */

    return 0.725*pow(rl*dr*g*hfg*rkl*rkl*rkl/od/rows/dv/dt,0.25);
    }

void cond(double tl,double*t2,double h2,double h3,double hw,double
l,double d,
    double od,double rholow,double drho,double depth,double rnuc,double
prc,
    double kc,double cswc,double*hc,double*uc,double*ReC,double
vc,double*dtswc,
    double*wsc,double*ac,double*tubes,double*ppc,double hbioc,double
hcorrc)
    {
    int ibis;
    double acs,dt1,dt2,dtperq,fc,hec,qc,qm,qq,qx,tsw;

/* model the heat transfer process in the condenser */
/* Reynolds number, friction factor, convective heat transfer
coefficient,
    and wall shear stress */

    *ReC=vc*d/rnuc/12.;
    fc=ff(*ReC);
    hec=12.*0.023*pow(*ReC,0.8)*pow(prc,0.4)*kc/d;
    *wsc=rholow*vc*vc*fc/2./144./32.2/4.;

/* change in temperature of the seawater per unit heat transferred */
```

```
    dtperq=4.*12.*l/rholow/cswc/d/vc/3600.;

/* condensation heat transfer coefficients */

    dt1=*t2-tl;
    *hc=condk(*t2,dt1/2.,od/12.);

/* overall heat transfer coefficient */

    *uc=1./(1./(*hc)+1./hw+1./hcorrc+1./hbioc+1./hec);

/* heat transfer in the condenser using bisection search */

    qm=0.;
    qx=*uc*dt1;
    for(ibis=1;ibis<=20;ibis++)
      {
      qc=(qm+qx)/2.;
      tsw=tl+qc*dtperq;
      dt2=*t2-tsw;
      if(dt2>0.)
        {
        qq=*uc*(dt1-dt2)/log(dt1/dt2);
        if(qc<=qq)
          {
          qm=qc;
          continue;
          }
        }
      qx=qc;
      }

/* requied heat exchange area */

    *dtswc=dtperq*qc;
    *ac=(h2-h3)/qc;
    *tubes=12.*(*ac)/M_PI/d/l;
    acs=M_PI*d*d*(*tubes)/144./4.;

/* required pumping power */

    *ppc=(rholow*vc*acs)*(((vc*vc/2.)*((l/(d/12.))*fc+1.5))
        +(drho*32.174*depth/2./rholow))*3600./778.3/32.2;
    *uc=1./(1./(*hc)+1./hw+1./hec+1./hbioc+1./hcorrc);
    }

void otec1()
    {
    double
ctubes,dols,dtswc,dtswe,etubes,hb,hc,pmptot,pnet,ppc,ppe,ReC,ReE;
```

```
evap(th,&t1,&t4,h1,h4,hw,l,d,rhohgh,rnue,pre,ke,cswe,&ReE,&wse,&dtswe,
&hb,&ue,ve,&ae,&etubes,&ppe,hbioe,hcorre);

cond(tl,&t2,h2,h3,hw,l,d,od,rholow,drho,depth,rnuc,prc,kc,cswc,&hc,&uc
,&ReC,vc,&dtswc,&wsc,&ac,&ctubes,&ppc,hbioc,hcorrc);

  wtie=wtpera*ae;
  wtic=wtpera*ac;

/* pumping power and the net power per pound/hr */

  pmptot=(ppe+ppc)/eswpmp;
  pnet=(de*effgen-dhpump-pmptot)/3.412;

/* specific cost of the heat exchangers */

  dols=cost*cstfct*(wtie+wtic)+dpp*(etubes+ctubes);

/* watts/$ for this configuration and operation */

  wpd=pnet/dols;
  }

double otec2()
  {
  p1=psat(t1);
  p2=psat(t2);
  h1=enthv(t1);
  s1=entrv(t1);
  ss2=s1;
  sv2=entrv(t2);
  sl2=entrl(t2);
  xs2=(ss2-sl2)/(sv2-sl2);
  hv=enthv(t2);
  hl=enthl(t2);
  hs2=hl+xs2*(hv-hl);
  de=efftur*(h1-hs2);
  h2=h1-de;
  p3=p2;
  t3=t2;
  h3=hl;
  r3=rhol(t3);
  v3=1./r3;
  p4=p1;
  dhpump=v3*((p4-p3)/effpmp)*144./778.3;
  cl=cpl(t3);
  dtp=dhpump*(1.-effpmp)/cl;
  t4=t3+dtp;
  h4=h3+dhpump;
  otec1();
  return wpd;
```

```c
      }

void ListProperties()
   {
   double T;
   printf("Properties of Seawater\n");
   printf(" T   rho    Cp    k       nu       Pr\n");
   for(T=30.;T<81.;T+=5.)
      printf("%2.0lf %5.2lf %5.3lf %5.3lf %9.7lf
%5.2lf\n",T,rhosw(T),cpsw(T),consw(T),visct(T),prnum(T));
   printf("\nProperties of %s\n",fluid);
   printf(" T Psat  rhof  rhog  Hf   Hg     Sf    Sg    CpL   CpV
mu     k      sigma\n");
   for(T=30.;T<81.;T+=5.)
      printf("%2.0lf %5.1lf %5.2lf %5.3lf %4.1lf %5.1lf %5.4lf %5.4lf
%5.3lf %5.3lf %5.3lf %6.4lf %7.5lf\n",

T,psat(T),rhol(T),rhov(T),enthl(T),enthv(T),entrl(T),entrv(T),cpl(T),c
pv(T),dvisl(T),condl(T),sten(T));
   }

int main(int argc,char**argv,char**envp)
   {
   double ld;
   FILE*fo;

   if((fo=fopen("OTEC.OUT","wt"))==NULL)
      {
      fprintf(stderr,"can't create output file\n");
      return(1);
      }

/* properties of seawater */

   rhohgh=rhosw(th);
   rholow=rhosw(tl);
   drho=rholow-rhohgh;
   pre=prnum(th);
   prc=prnum(tl);
   ke=consw(th);
   kc=consw(tl);
   rnue=visct(th);
   rnuc=visct(tl);
   cswe=cpsw(th);
   cswc=cpsw(tl);

   for(ld=10.;ld<1001.;ld*=pow(10.,0.02))
      {
      fprintf(fo,"%lG",ld);
      for(d=1.;d<2.76;d+=0.25)
         {
         l=ld*d;
```

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```
/* tube wall thickness and the weight per square foot */

        tw=d/18.;
        od=d+2.*tw;
        wtpera=den*(od*od-d*d)/d/48.;
        hw=12.*ktw/tw;

/* required velocities to meet biofouling criteria */


vemin=6.338*(pow(wsmin*144.*32.2/rhohgh,0.5714))*pow(d/12./rnue,0.1429
);

vcmin=6.338*(pow(wsmin*144.*32.2/rholow,0.5714))*pow(d/12./rnuc,0.1429
);

/* these values roughly correspond to the optimum */

        t1=th-9.;
        t2=tl+4.;
        vc=vcmin;
        ve=vemin;
        fprintf(fo," %lG",otec2());
        }
     fprintf(fo,"\n");
     }

  fclose(fo);

  return(0);
  }
```

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