OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM. (U)

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THESIS

OPTIMIZATION OF A LOW ΔT RANKINE
POWER SYSTEM

by

Raymond C. Schaubel
December 1980

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18. KEY WORDS (Continue on reverse side if necessary and identify by block number)

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The Ocean Thermal Energy Conversion (OTEC) uses the low thermal energy potential available from ocean temperature gradients. A method is presented to analyze such systems and, for this purpose, a comprehensive simulation is developed. The simulation includes parasitic power requirements, losses due to interconnecting lines, and heat exchanger pressure drops. Cost functions are included and numerical optimization is employed to obtain optimal designs based upon minimum cost.
The analysis is converted to a computer code and coupled to the COPES/CONMIN optimization code to facilitate a fully-automated design where the computer makes the design decisions and performance trade-off studies. The final product is an optimum power system module design for the designated net electrical output required and the specified system and design constraints.

Preliminary results are presented for a range of system power levels. Optimum designs are obtained and compared for systems in which either titanium or aluminum tubes are used in the heat exchangers.
Optimization of a Low ΔT Rankine Power System

by

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PARTIAL LIST OF SYMBOLS

A  heat transfer surface area
A_f  tube bundle frontal area
A_{ff}  free-flow area
C_p  constant pressure specific heat
d  diameter
E  power
f  friction factor
F  correction to LMTD
G  mass velocity
g  acceleration of gravity
g_e  conversion factor (3.22 lb_m*ft/lb_f*sec^2)
h  specific state point enthalpy
\dot{h}  average heat transfer coefficient
k  thermal conductivity
K_m  mean salt water compressibility
L  tube or pipe length
\dot{m}  mass flow rate number
N_t  number of heat exchange tubes
Re  Reynolds number
P  static pressure
\dot{Q}  heat transfer rate
S  specific state point entropy
T  temperature
LMTD  log mean temperature difference
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<tr>
<td>$U$</td>
<td>overall heat transfer coefficient</td>
</tr>
<tr>
<td>$v$</td>
<td>specific volume</td>
</tr>
<tr>
<td>$V$</td>
<td>velocity</td>
</tr>
<tr>
<td>$X$</td>
<td>quality of working fluid</td>
</tr>
<tr>
<td>$Z$</td>
<td>elevation</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>heat exchange effectiveness</td>
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<tr>
<td>$\eta$</td>
<td>efficiency</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
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<td>$\mu$</td>
<td>absolute or dynamic viscosity</td>
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I. INTRODUCTION

A. BACKGROUND

Ocean Thermal Energy Conversion (OTEC) is a concept using the low thermal energy potential available from the ocean temperature gradient that exists between warm surface ocean water and cold water in deep ocean regions.

The idea of converting the stored ocean energy to useful power originated with French physicist Jacques d'Arsonval in 1881 [Ref. 1]. It was nearly a half-century later that the technical feasibility of ocean thermal energy conversion could be demonstrated. In 1926, George Claude used an open cycle power system to extract heat from surface water for indirect conversion of the thermal energy of a working fluid. Operating at a low pressure the working fluid was used to drive a turbine providing electrical power generation.

Though Claude's limited power system produced only 22 kilowatts of electricity while requiring approximately 80 kilowatts of power to drive its equipment, it stirred the scientific and research community to consider the attractiveness of ocean thermal energy conversion [Ref. 2].

Claude called for immediate action on his ocean thermal power system, because of the Federal Oil Conservation Board's dire predictions that the United States had only six years of oil production remaining. Obviously the dire predictions ascribed to by the Federal Oil Conservation Board did not
come true, but the oil crisis of that period heightened scientific interest in extracting energy from the ocean.

Now, 55 years later, the United States is faced with an energy crisis because of increasing industrial and social dependence on foreign petroleum. Dwindling supplies and erratic price hikes have rekindled interest in ocean thermal energy conversion, since it utilizes an inexhaustible supply of fuel.

Currently, the United States Department of Energy is attempting to develop the necessary technology and demonstrate the feasibility of large-scale OTEC power systems. However, there are major engineering development problems which must be solved before OTEC can be standardized and become a viable source of electrical power generation.

The single controlling factor which creates troublesome technical encounters is low thermal power system efficiency (one to four percent depending upon parasitic power requirements). Because the heat energy used by OTEC must be extracted from a small ocean temperature difference, extremely large volumes of surface water must pass through a proportionately sized evaporator to provide sufficient indirect heat energy to convert the working fluid into vapor to drive a turbine-generator for electrical power generation. Concurrently, to convert the turbine exhaust to a saturated liquid, completing the closed cycle, a condenser having compatible heat absorption capacity must be employed.
Economic handling of the volume of fluids required for the heat absorption, expansion, and heat rejection phases of the cycle requires close scrutiny of evaporator, turbine, condenser, and pump design to minimize the parasitic losses with respect to the generated electrical output. Because of the low thermal efficiency, relative to nuclear or fossil fuel-fired power plants, the margins for design and operating error in OTEC plants will be narrow.

With the advent of high-speed computers, numerical methods for solving these complex engineering problems with multiple design variables and constraints are now possible. The case for utilizing an optimizing scheme for not just one system component, but rather the complete power generation cycle, can easily be made. In effect, it would serve as a systems analysis tool, to optimize component design and cost, relative to a specific electrical output or to enable comparison and evaluation of competing OTEC designs.

B. OBJECTIVES

The objectives of this work are to develop a computer code for the Ocean Thermal Energy Conversion (OTEC) power system and to couple the analysis to a numerical optimization code to provide an optimum system design capability, considering both performance and economics.

This would create an optimum modular design relative to a specified objective function for a desired net electrical output, such as a 25 MW (net) power system. Such a design
would permit construction of higher capacity power systems using the optimized modules as substations of the total power plant. Cost savings, improved plant performance, redundancy, and reliability could be the immediate beneficiaries of such a venture.

C. OVERVIEW OF THE OTEC POWER SYSTEM ANALYSIS

To analyze the closed-cycle OTEC power system, the fundamental relationships of heat transfer, fluid mechanics and thermodynamics are used to simulate a variety of system component designs, which form the basis of the power system algorithm. The scope of this analysis will be limited to the OTEC power system and sea water systems only. Mooring systems, power delivery, hull, and cold pipe design will not be addressed.

The performance analysis will be divided into four sequential sections as shown in Figure 1, and discussed in detail in subsequent chapters of this thesis.

Input parameters (design constants) for the power cycle analysis will include:

- Required net electrical output.
- Salt water inlet temperature to the evaporator and condenser.
- Length of hot and cold salt water pipes.
- Heat exchanger tubing material (aluminum or titanium).
- Heat exchanger tube orientation and profile.
- Pump mechanical and motor efficiencies.
. Turbine mechanical efficiency.
. Generator mechanical and electrical efficiency.
. Biofouling control factor.
. Piping absolute roughness.
. Projected annual inflation rate for aluminum heat exchanger retubing.
Figure 1. Power System Sequential Analysis
II. POWER CYCLE DESCRIPTIONS

A. INTRODUCTION

This chapter will provide a brief description of the OTEC power system. First, looking at the ideal Rankine cycle, the fundamental thermodynamic concepts will be enumerated. Then the deviations from the ideal cycle will be presented, creating the configuration assumed for the present cycle analysis which will be amplified in detail by follow-on chapters.

B. IDEAL OTEC RANKINE CYCLE

The closed-cycle OTEC concept is based upon a Rankine power cycle that is driven by the low thermal energy potential available from the ocean temperature gradient that exists between warm surface water and cold deep water in ocean regions. The power cycle consists of a working fluid circulation pump, evaporator (heat absorption), turbine (expansion), and condenser (heat rejection), as shown in Figure 2. The majority of current OTEC designs are based upon ammonia as the working fluid -- a design decision that is adopted for this analysis.

Figure 2 also illustrates an ideal OTEC Rankine cycle, plotted on temperature-entropy coordinates. In the ideal cycle, the low pressure working fluid (state point 1) is isentropically pumped to the evaporator operating pressure (state point 2). The working fluid (ammonia) is then
Figure 2. Idealized OTEC Rankine Cycle
converted to a saturated vapor in the evaporator by indirect heat energy exchange from warm surface ocean water (state point 3). Mechanical power is generated by isentropic expansion of the saturated ammonia vapor through the turbine (state point 4).

After exiting the turbine, the wet, low-pressure vapor is converted to a saturated liquid in the condenser by indirect heat absorption from cold ocean water (state point 1), returning the cycle back to the working fluid circulation pump.

C. ACTUAL OTEC RANKINE CYCLE

In actuality there are numerous deviations from the ideal cycle which must be considered in this analysis. These are:

1. Turbine, generator and pump efficiencies.
2. Pressure drops in evaporator and condenser (tubeside and shellside).
3. Pressure drop across moisture separator.
4. Elevation change and frictional losses in piping: (a) re-flux pump piping, (b) piping from circulation pump to evaporator.
5. Evaporator outlet quality (85 to 95%).
6. Moisture separator outlet quality (99 to 99.5%).

The deviations from the ideal Rankine cycle described above are depicted in the flow diagram and temperature-entropy plot of Figure 3. In the actual OTEC Rankine cycle, the low pressure working fluid (state point 1) is pumped up to the evaporator operating pressure by the ammonia circulation pump with an adiabatic efficiency (state point 2). The working
Figure 3. Actual OTEC Rankine Cycle
fluid (ammonia) is then converted to a wet vapor with an evaporator outlet quality (85-95%) acting under a shellside pressure drop (state point 3). Evaporator outlet vapor then passes through a moisture separator to improve vapor quality (99-99.5%) creating a pressure drop (state point 4). Mechanical power is generated by the expansion of the moisture separator outlet vapor through the turbine with an adiabatic efficiency (state point 5). After exiting the turbine, the wet, low pressure vapor is converted to a saturated liquid in the condenser acting under a shellside pressure drop (state point 1), returning the cycle to the working fluid circulation pump.

This figure forms the thermodynamic basis for the OTEC power system analysis which follows.
III. EVAPORATOR AND MOISTURE SEPARATOR

A. INTRODUCTION

Several heat exchanger concepts have been proposed for closed-cycle OTEC systems. Among these designs are:

- Conventional shell and tube heat exchanger.
- Plate type heat exchanger.

Within these basic concepts, variations in design have been proposed, including:

- Orientation of tubes (horizontal or vertical).
- Heat exchanger tube material (i.e., titanium, aluminum).
- Method of tube enhancement (i.e., fluted, porous coatings).
- Location of tube enhancement (i.e., internal and/or external).
- Location of the vapor separator (i.e., internal or external).
- Location of the heat exchangers relative to the sea surface.
- Method of biofouling control.

The analysis to be presented for the evaporative heat exchanger will be based on the following design characteristics:

- Single-pass shell and tube heat exchanger.
- Internal vapor separator with a gravity drain to evaporator inlet.
- Horizontal orientation of tubes with an equilateral triangle or square tube profile.
- Smooth plain-tube configuration (no enhancements).
. Tube material (titanium or aluminum based on a 30-year life-cycle criterion).

. Biofouling control based upon an achievable fouling factor.

. Heat exchanger centerline located on sea surface.

As an overview of the evaporator-moisture separator analysis, the following major steps in the algorithm are listed in order of their execution (numbers in parentheses refer to equations developed in the subsequent analysis):

. Specification of system constants (see I.C.).

. Initialization of design variables (D.V.).

   .. Tube length.

   .. SW velocity through hot pipe.

   .. Inner diameter of hot pipe.

   .. Tube outer diameter.

   .. SW velocity through evaporator tubes.

   .. Inner diameter of NH3 piping.

   .. Inner diameter of NH3 re-flux piping.

   .. Tube profile pitch ratio.

. Salt water mass flow rate (1).

. Total number of tubes (2).

. Total heat transfer surface area (3).

. Assume an initial salt water bulk temperature (6), and ammonia heat transfer coefficient (9).

. Overall heat transfer coefficient (4).

. Number of transfer units (11).

. Heat exchanger effectiveness (13).

. Salt water outlet temperature (15).
Revised bulk temperature (16); iterate with (6).
Amount of heat absorption (17).
Log mean temperature difference (18).
Film temperature (19).
Initial ammonia mass flow rate (21) without the effects of moisture separator.
Initially assume state point 1 thermodynamic properties are ideal (21).
Thermodynamic pump work (23).
Tube profile, flow parameters across the tube bank (24, etc.).
Tube sheet diameter (30).
Evaporator shellside pressure drop for two phase flow (33).
Moisture separator pressure drop (38).
Properties at state points 3 and 4 (39-41).
Revised ammonia mass flow rate and velocity (50) includes the effects of the moisture separator; iterate with (31).
Revised ammonia heat transfer coefficient (51, etc.); iterate with (9).
Heat exchanger cost analysis.

In the following section, the basic steps summarized above will be described in detail.

B. ANALYSIS OF THE EVAPORATOR AND MOISTURE SEPARATOR

1. Salt Water Mass Flow rate, $\dot{m}_{sw}$

   The salt water mass flow rate through the hot pipe must be equivalent to the flow rate through the evaporator (assuming no leakage)
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where \( A \) = cross-sectional area of the hot pipe.

\( V \) = salt water velocity through hot pipe.

\( \rho_{sw} \) = density of salt water evaluated for an average hot pipe salt water temperature.

As previously stated, the diameter of the hot pipe and salt water velocity are among the initializing conditions of the optimization and will be treated as design variables.

2. Total Number of Evaporator Tubes, \( N_{e} \)

Using equation (1), it follows that

\[
\dot{M}_{1} = \rho_{sw} \int_{1}^{2} \frac{dV}{dV_{e}} \dot{V}_{e} N_{e}
\]

(2)

where \( \rho_{sw} \) = salt water density evaluated at the average bulk temperature initially assumed as the hot pipe salt water temperature.
\( d_i \) = tube inner diameter.
\( N_t \) = the number of tubes required to maintain the mass flow rate for an average salt water velocity per tube.

The total number of tubes can be determined by solving Eq. (2) for \( N_t \).

The diameter of the tube and average salt water velocity per tube are initialized for the analysis and will be treated as design variables by the optimization code.

3. Total Evaporator Heat Transfer Surface Area (Outer), \( A_t \)

Having determined the number of evaporator tubes, the total heat transfer surface area can be determined using initializing values of outer tube diameter and tube length.

For tubes without extended surfaces

\[
A_t = \pi d_o L_t N_t \tag{3}
\]

As previously, the outer tube diameter and tube length are initializing conditions and will be treated as design variables.

4. Overall Heat Transfer Coefficient, \( U \)

The quantity "U" provides a measure of the total thermal resistance in the flow path, based on either inside or outside surface area.

This analysis will be based on the value of \( U \) for the outside surface area derived from Eq. (3).
Using a resistance analysis, assuming one dimensional (radial) heat flow,

the overall heat transfer coefficient may be expressed as

\[
U_o = \frac{1}{\frac{A_o}{\eta_i h_{sw} A_i} + \frac{A_o}{A_i} R_{fsw} + \frac{d_o \ln d_o/d_i}{2K} + R_{fNH_3} + \frac{1}{\eta_o h_{NH_3}}}
\]

where
- \(h_{sw}\) = tubeside heat transfer coefficient.
- \(R_{fsw}\) = salt water fouling heat transfer resistance.
- \(K\) = thermal conductivity of the tube material.
- \(d_o, d_i\) = outer and inner tube diameter.
- \(R_{fNH_3}\) = ammonia fouling heat transfer resistance (assumed to be negligible).
- \(\eta_o, \eta_i\) = outer and inner total fin efficiency (for plain tube analysis, total fin efficiency equals 1).
- \(A_o\) = total outer surface area (including fin and bare tube).
\( A_i = \text{total inner surface area (including fin and bare tube)} \).\(^1\)

\[
\gamma_i = 1 - \frac{A_{fn_i}}{A_i} (1 - \gamma_{fi_i})
\]

\[
\gamma_o = 1 - \frac{A_{fn_o}}{A_o} (1 - \gamma_{fo_o})
\]

where

\( A_{fn_i} = \text{total inner fin surface area.} \)

\( A_i = \text{total inner surface area (including fin and bare tube).} \)

\( A_{fn_o} = \text{total outer fin surface area.} \)

\( A_o = \text{total outer surface area (including fin and bare tube).} \)

\( \gamma_{fi_i} = \text{fin efficiency of single internal fin.} \)

\( \gamma_{fo_o} = \text{fin efficiency of single external fin.} \)

a. Tubeside Reynolds Number, \( R_{es_i} \)

Since the heat transfer coefficient correlations for the evaporator and condenser are dependent on tubeside flow, Reynolds number must be calculated.

The tube Reynolds number is defined as

\[
R_{es_i} = \frac{\rho_i w_i \nu_i w_i c_i}{\mu_i w_i}
\]  \( (5) \)

\(^1\)Note that this analysis will hereafter consider smooth plain tube configurations only.
where \( \mu_{sw} \) = dynamic viscosity of salt water.
\( \rho_{sw} \) = density of salt water.

Initially, properties are evaluated for

\[ T_{bulk} = T_{sw} (INLET) \]  \hspace{1cm} (6)

Reynolds numbers greater than 2300 will be indicative of turbulent flow [Ref. 3]. Transition flow was considered laminar for numerical evaluation.

b. Salt Water Heat Transfer Coefficient, \( h_{sw} \)

The simple empirical relation proposed by Sieder and Tate [Ref. 3], expressed as

\[ Nu_d = 1.8 \left( \frac{Re_d Pr}{10} \right)^{0.8} \mu^{0.8} \left( \frac{c_p}{c_p^m} \right)^{0.14} \]  \hspace{1cm} (7)

was used for laminar heat transfer in tubes as defined by Eq. (5).

Nusselt and Prandtl numbers, \( Nu_d \) and \( Pr \), are defined as

\[ Nu_d = \frac{h_{sw} \mu}{\rho_{sw}} \]
\[ Pr = \frac{C_{p_{sw}} \mu_{sw}}{k_{sw}} \]

where \( \mu_{sw} \), \( C_{p_{sw}} \), and \( k_{sw} \) (dynamic viscosity, specific heat, and thermal conductivity) of salt water are evaluated at salt water bulk temperature.
The effect of the viscosity ratio term in Eq. (7)

\[ \left( \frac{\mu}{\mu_{\infty}} \right)^{0.14} \]

where \( \mu_{\infty} \) is salt water viscosity evaluated at tube wall temperature, is considered negligible and will hereafter be dropped from the expression of Eq. (7).

Relation (7) is based upon the following assumptions:

- fully developed flow in smooth tubes.
- fluid properties are evaluated at the bulk fluid temperature.

and is valid for the following condition

\[ \frac{Re_d Pr d}{L} > 10 \]

For fully developed turbulent flow in a tube as defined by Eq. (5), the Dittus-Boelter correlation [Ref. 3] expressed as

\[ Nu_d = 0.023 Re_d^{0.8 Pr} \]

was used. Nusselt and Prandtl numbers, \( Nu_d \) and \( Pr \), are previously defined by Eq. (9).

Relation (8) is based upon the following assumptions:

- fully developed flow in smooth tubes.
- fluid properties are evaluated at the bulk fluid temperature

and is valid for the following conditions:
Prandtl numbers ranging from 0.6 to 100.

moderate temperature differences between
the wall and fluid conditions.

c. Salt Water Fouling Heat Transfer Resistance

In this document, it will be assumed that the fouling resistance coefficient for tubeside salt water can be maintained at \(0.00025 \text{(hr}\cdot\text{ft}^2\cdot\text{F}/\text{BTU})\) using one of the following techniques:

- Chlorination.
- MAN Brush System.
- Amertap.
- Chemical cleaning

Pressure drops associated with cleaning techniques will not be considered in this analysis. Piping losses will be a function of tube length, inner diameter, salt water velocity and the absolute roughness of the tubing design material only.

d. Ammonia Shellside Heat Transfer Coefficient, \(h_{\text{NH}_3}\)

Initially, \(h_{\text{NH}_3}\) will be assumed

\[
\dot{h}_{\text{NH}_3} = 1000 \left( Btu/\text{hr}\cdot\text{ft}^2\cdot\text{F}^\circ \right)
\]  

(9)

since its value cannot be directly calculated during this phase of the analysis.

Using the thermal resistance expressed as

\[
R_I = \frac{d_o}{\dot{h}_i h_{SW} \cdot d_i}
\]
an initial value for the overall heat transfer coefficient may be calculated.

\[ U_o = \frac{1}{\frac{1}{R_1 + R_3} + \frac{1}{R_3 + R_5}} \] (10)

5. NTU-effectiveness Relations

The NTU-effectiveness relationships will be used to determine the evaporator outlet salt water temperature. Currently, all salt water properties have been based upon the initial assumption that

\[ T_{\text{bulk}} = T_{H_i} \text{ (SW INLET TO EVAP)} \]

The expression for the number of transfer units (NTU) which is a measure of the size of the heat exchanger is given by

\[ NTU = \frac{U_o A_t}{C_{m,\text{in}}} \]

where \( C_{\text{in}} \) is defined as capacity rate of the single phase flow in an evaporative or condensing two phase flow regime.
\[
C_{\text{min}} = m_{\text{sw}} C_{p, \text{sw}} \tag{11}
\]

Evaporator effectiveness can then be expressed as

\[
\varepsilon = 1 - e^{(-NTU)} \tag{12}
\]

for two phase flow regardless of the flow geometry.

Using the definition of effectiveness

\[
\text{Effectiveness} = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \tag{13}
\]

\[
\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{\Delta T_{\text{min}}}{\Delta T_{\text{max}}} = \frac{T_{\text{in}} - T_{\text{out}}}{T_{\text{in}} - T_{c_i}} \tag{14}
\]

The expression for \(\Delta T_{\text{min}}\) represents the single phase (salt water) flow and \(T_{c_i}\) represents ammonia inlet temperature to evaporator taken at state point 3A.

6. **Evaporator Salt Water Outlet Temperature and Bulk Temperature**

Using the relationships of Eqs. (12) and (14), the following expression may be formulated for salt water outlet temperature

\[
T_{H_o} = T_{H_i} - (T_{H_i} - T_{c_i})(1 - e^{(-NTU)})(1 - \varepsilon^{(-NTU)}) \tag{15}
\]
Concurrently, a revised evaporator average salt water temperature can be expressed as

\[ T_{\text{Bulk}} = \frac{T_{H_e} + T_{H_w}}{2} \]  

Using the revised value for average salt water temperature, iterate with equation (1) until the revised and current values of bulk temperature satisfy a specified convergence criterion.

7. Amount of Heat Absorption, \( \dot{Q} \)

Using the results of Eq. (16) and (12), the amount of heat absorption by the evaporator may be expressed as

\[ \dot{Q} = C_{m,n} (T_{H_e} - T_{H_w}) \]  

8. Log Mean Temperature Difference, \( \text{LMTD} \)

The NTU-effectiveness method can be used to determine the mean effective temperature difference \( \text{LMTD} \) across the evaporator (heat exchanger).

Using Eq. (17) and the definition of

\[ \dot{Q} = U_{e} A_{e} F_{\text{LMTD}} \]

with \( \dot{Q}_{\text{m,ix}} = C_{m,n} \Delta T_{\text{max}} \)

the log mean temperature difference across the evaporator may be expressed as
\[ LMTD = \frac{C_{m,n}(1-e^{(-NTU)})}{U_o A_T F} \times (T_{hi} - T_{ci}) \] (18)

where \( T_{ci} = T_{NH_3} \) evaluated at state point 3.

\( F \) = correction factor on \( LMTD \), equal to 1 for two phase flow.

9. Film Temperature for Property Evaluation, \( T_f \)

In order to evaluate the shellside ammonia heat transfer coefficient, working fluid properties (i.e., viscosity, specific heat, etc.) must be evaluated at the film temperature to validate critical heat transfer expressions.

By modifying the expression in Eq. (10) multiplying by a single tube outer area, a value for single tube conductance can be expressed as

\[ U_o A = \frac{A}{R_1 + R_2 + R_3 + R_5} \]

Subsequently, the average amount of heat transferred per tube would equate to

\[ \dot{Q} = U_o A (T_{bulk} - T_3) \]

where \( T_3 = T_{NH_3} \) evaluated at state point 3.

Again using the resistance analysis in Section 3, shellside wall temperature may be derived from

\[ T_{w2} = T_{bulk} - \dot{Q} \left( \frac{R_1 + R_2 + R_3}{A} \right) \]

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Knowing shellside wall temperature and the free-stream temperature, film temperature can be derived from their arithmetic mean.

\[ T_f = \frac{T_{w_2} + T_3}{2} \]  

(19)

10. Ammonia Mass Flow Rate, \( \dot{m}_{NH_3} \)

According to first law of thermodynamics for steady state, steady-flow conditions in the evaporator:

\[ \dot{m}_{NH_3} h_2 + Q = \dot{m}_{NH_3} h_3 \]  

(20)

from which the ammonia mass flow rate, \( \dot{m}_{NH_3} \), may be determined if the enthalpies at state points 2 and 3 are known.

If we initialize the lower and upper bounds of the analysis in terms of pressure \( P_1 \) and \( P_3 \), respectively, and initially assume that a saturated vapor leaves the evaporator, the following relations may be expressed...
where $h_1 = h_{r1} \rho_1$ represents enthalpy at state point 1 at the suction inlet to the working fluid circulation pump.

$h_3 = h_{g3} \rho_3$ represents enthalpy at (ideal)/state point 3 as a saturated vapor.

$T_1, T_3$ represent the respective saturation temperatures.

$\nu_1$ represents the specific volume at state point 1.

To summarize, the upper and lower pressure bounds of the system ($P_1$ and $P_3$) will be initialized in the analysis and treated as design variables by the optimization code. Temperature at state point 3 is initially assumed to be a saturated vapor (ideal $T_3$); however, the working fluid is subject to a shellside pressure drop as it passes across the evaporator with an outlet quality of 90-95%. Properties at state point 3 (actual) will be assessed in follow-on sections.
Assuming steady state, steady-incompressible flow, the change in kinetic and potential energies, and heat losses are negligible for isentropic conditions, and the isentropic pump work can be expressed as

\[ \dot{m}_{NH_3} h_1 + \dot{W}_{CP} = \dot{m}_{NH_3} h_2 \]  

(22)

After the isentropic pump work is calculated, the actual (adiabatic) pump work may be determined using pump efficiency, \( \eta_p \).

\[ \dot{W}_{CP} = \frac{\dot{W}_{CP_s}}{\eta_p} \]  

(23)

Actual outlet enthalpy at state point 2 may be determined using the results of Eq. (23) with Eq. (22) knowing the enthalpy at state point 1 from Eq. (21).

Using the results of Eqs. (21) and (22), the mass flow rate in Eq. (20) may be calculated as the average shell-side mass flow rate for the working fluid (ammonia).
11. Tube Profile, Flow across Tube Bank, and Tube Sheet Diameter

Since the heat-exchanger arrangements (evaporator and condenser) involve multiple rows of tubes, the geometric arrangement of the tube profiles is important in the determination of the heat transfer coefficient, the tube sheet diameter and the shell side pressure drop associated with two-phase flow (homogeneous model) [Ref. 4].

The following geometric arrangements are used:

\[ S_n = \text{pitch ratio} \times \text{outer tube diameter, equal to } S_p \]

\[ \rho_n = \text{pitch ratio; the distance between tube centers with respect to outer tube diameter.} \]

\[ A_p = \text{tube profile area (centerline to centerline) per tube.} \]

\[ S_n = \rho_n d_o \]  \hspace{1cm} (24)

\[ A_p = S_n^{\frac{1}{2}} \]  \hspace{1cm} (25)
Therefore, the tube profile area (centerline to centerline) per tube is equal to

\[ A_p = S_n S_p \quad (28) \]

The ratio of minimum flow area to the frontal area can be expressed as

\[ \frac{A_{mf}}{A_f} = \frac{S_n - d_o}{S_n} \quad (29) \]

Using the selected tube profile geometry, either in-line or staggered, and knowing the required number of tubes by equation (2), the tube sheet diameter for heat exchanger design can be assessed as follows:
\[ N_f A_p = \frac{\bar{V}_f T_{sd}}{4} \]

where \( T_{sd} \) = tube sheet diameter.

To estimate the shellside ammonia flow velocity the following control volume is introduced (ammonia circulation piping and the top portion of the evaporator).

If the mass flow rate remains unchanged across any boundary (continuity),

\[ \dot{m}_2 = \dot{m}_f \]

Furthermore, if we assume the evaporator has the means to evenly distribute liquid droplets across the top of the tube bundle (spray nozzles and baffling), the following expressions can be applied to estimate the mean droplet velocity approaching the bundle:

Let

\( (A_f)_{w, c} = A_f \gamma_f \)

where \( \gamma_f \) = percent of tube frontal area which is occupied by droplets.

The mass flow rates are

\[ \dot{m}_2 = \rho A_p V_p \]
\[ \dot{m}_t = \rho (A_f) \omega f V_f \]

where \( A_P \) = ammonia pipe cross-sectional area.
\( V_P \) = average ammonia velocity in the pipe.

Therefore

\[ V_f = \frac{A_P}{(A_f) \gamma_1} V_P \]

and since

\[ \gamma_1 \approx \frac{A_P}{A_f} \]

it follows that the average velocity of ammonia through the circulation pipe is equivalent to the average velocity of ammonia at the tube frontal area boundary.

\[ V_P = V_f \quad (31) \]

Thus the assumption that \( \gamma_1 = A_P / A_f \) is equivalent to the assumption of constant liquid kinetic energy in the transition from the pipe exit to the bundle entrance. Considering the minimum free-flow area for shellside flow passage, \( A_{ff} \) can be derived from Eqs. (29) and (30):

\[ A_f = T_{so} L_t \]

\[ A_{ff} = A_f \left( \frac{S_{nf} - d_i}{S_{nf}} \right) \quad (32) \]

where \( A_f \) = represents the flow frontal area.
\( L_t \) = tube length.
Using the calculated values of Eqs. (32) and (20), the mass velocity for the minimum free-flow area can be expressed

\[ G = \frac{n_{NH_3}}{A_{ff}} \]

where \( n_{NH_3} \) represents the average ammonia mass flow rate.

12. Pressure Drop of Two-Phase Flow across a Bank of Tubes, \( \Delta P \)

This portion of the analysis will use an analytical model for two-phase pressure drops applicable for a fog or spray flow pattern occurring at high void fractions -- the homogeneous model [Ref. 4].

The model asserts that if the pressure drop in the two-phase flow for a liquid-vapor mixture is relatively small compared to the absolute pressure, the flow is considered incompressible. Subsequently, the density of each phase is practically constant. During the process of phase change, the phase and velocity distributions are changed, and so is the momentum of the flow. Therefore, the pressure drop of a vertical two-phase flow consists of three components: friction loss, momentum change, and elevation pressure drop arising from the effects of the gravitational force field.

The local pressure gradient for a two-phase flow may be expressed as

\[ \Delta P_{TOT} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \]  (33)
For a given channel length, $L_c$, the pressure drop components can be represented by

\[
\begin{align*}
\Delta P_{\text{friction}} &= \frac{f G^2 \bar{v}}{D_e 2 y} L_c \\
\Delta P_{\text{momentum}} &= \frac{G^2 \bar{v}}{y} \\
\Delta P_{\text{elevation}} &= \frac{g}{\bar{v}} L_c
\end{align*}
\]  

(34)

and the total pressure drop, $\Delta P_{\text{EVAP}}$, is given by the sum of these expressions.

where $f$ = single-phase friction factor by Jakob expressed in Eqs. (35) and (36).

$L_c$ = channel flow length, defined for horizontal tubed evaporators as $L_c = \frac{T_S D}{D_e}$ (tube sheet diameter).

$D_e$ = equivalent diameter of flow channel, defined by $D_e = \frac{P_R D_e - c_{e}}{c_{e}}$.

$\bar{v}$ = mean specific volume defined by

\[
\bar{v} = \nu_{f} \left[ 1 + \frac{x}{\nu_{f}} \right] \left( \nu_{f} - \nu_{v} \right)
\]

where $x$ = quality of mixture (state point 3).

$\nu_{f}$ = specific volume of liquid (state point 1).

$\nu_{v}$ = specific volume of vapor (state point 3).
The basic assumptions of the homogeneous model (fog flow model) [Ref. 4] are:

1. equal linear velocities of vapor and liquid,
2. thermodynamic equilibrium between the two phases, and
3. a suitably defined single-phase friction factor is applicable to the two-phase flow.

Using assumption (3) and the correlations by Jakob [Ref. 3], a suitable single-phase friction factor can be calculated from previously defined tube profile relationships:

for staggered tube arrangements:

\[ f = \left\{ C_{1.25} + \frac{C_{1.18}}{\left(\frac{S_n - d_o}{d_o}\right)^{1.28}} \right\} \frac{1}{Re_{max}} ^{-0.16} \]  (35)

and for in-line tube arrangements:

\[ f = \left\{ C_{0.444} + \frac{C_{1.03} S_n/d_o}{\left[\left(\frac{S_n - d_o}{d_o}\right)^{0.43} + 1.13 d_o / S_n \right]^{0.43}} \right\} \frac{1}{Re_{max}} ^{-0.15} \]  (36)

where Reynolds number (max) is determined from the shellside ammonia flow and the nozzling effect of the tube geometry as expressed by

\[ V_{max} = V_f \left( \frac{S_n}{S_n - d_o} \right) \]

where \( V_f \) = the ammonia velocity at the tube frontal area boundary determined by equation (31).
Reynolds number for maximum shellside flow can be calculated using the following expression

\[ \text{Re}_{\text{max}} = \frac{\rho_i V_{\text{max}} c_i}{\mu_i} \]  

Eq. (37) and tube profile data can then be used to evaluate the single-phase friction factor, required for Eq. (34). All other components of the total pressure drop Eq. (33) can be determined from previously calculated data.

13. Pressure Drop Across the Moisture Separator, \( \Delta P_{m, \text{sep}} \)

This portion of the analysis will simulate the use of a cyclone separator to improve the evaporator outlet vapor quality. The flow pattern in a cyclone separator is complex and simplifying assumptions are inadequate to allow the calculation of the corresponding pressure drop, which can vary from 1 to 20 inlet velocity heads [Ref. 5]. Therefore, the worst case condition will be applied with an approximation for the fluid flow inlet area to the separator banks.

By approximating the inlet area as a fraction of the evaporator frontal area

\[ A_{\text{INLET}} = C_{\text{INLET}} T_{\text{SP}} L_t \]

the inlet fluid velocity can then be determined using the working fluid mass flow rate, Eq. (20).

\[ \dot{m}_{\text{NH}_3} = \rho A_{\text{INLET}} V \]

where \( \rho \) = density of ammonia at state point 3.
Therefore, if the pressure drop across the moisture separator is equal to 20 times the inlet velocity head,

$$\Delta P_{m,sep} = 20 \rho \frac{V^2}{2 g}$$  \hspace{1cm} (38)

14. Enthalpy at State Points 3 and 4

Since Eq. (33) represents the pressure drop across the evaporator shellside, the actual pressure at state point 3 or evaporator outlet may be determined from

$$P_{3(NEW)} = P_3 - \Delta P_{EVA}$$  \hspace{1cm} (39)

where $P_3$ was previously described as the pressure for a saturated vapor.

Similarly the actual pressure at state point 4, the moisture separator outlet, may be expressed as

$$P_4 = P_{3(NEW)} - \Delta P_{m,sep}$$  \hspace{1cm} (40)

Operating under the dome of the Temperature-Entropy diagram, the following properties are defined

$$h_{3f(NEW)} = h_{fP_{3(NEW)}} \hspace{1cm} h_{4f} = h_{fP_4}$$

$$h_{3g(NEW)} = h_{gP_{3(NEW)}} \hspace{1cm} h_{4g} = h_{gP_4}$$  \hspace{1cm} (41)
The subscript \( s_E \) representing a revised property will hereafter be dropped from the expressions in Eq. (41).

Assuming an evaporator outlet quality of 90-95\%, and a moisture separator outlet quality of 99-99.5\%, enthalpies at state points 3 and 4 may be determined using the relationships of Eqs. (41)

\[
\begin{align*}
\hat{h}_3 &= \hat{h}_{3r} + x_3 (\hat{h}_{3y} - \hat{h}_{3r}) \\
\hat{h}_4 &= \hat{h}_{4r} + x_4 (\hat{h}_{4y} - \hat{h}_{4r})
\end{align*}
\]

15. **Revised Ammonia Mass Flow Rate and Velocity**

Till now, we assumed that the shellside mass flow rate was given in accordance with the ideal system defined by Eq. (20); however, in actuality this is not the case.

The diagramatic representation that follows better illustrates the heat absorption phase of the OTEC power system and will provide the basis for the analysis and optimization.

Note, as in the previous control volume analysis, the following conditions are assumed.

- Steady state.
- Steady-incompressible flow.
- Change in potential and kinetic energies is negligible.
Analyzing the moisture separator as a separate control volume,

If we assume that there is no carry-over of vapor in the separator drain, then

\[ X_3 \dot{m}_3 = X_4 \dot{m}_4 \]

and

\[ \dot{m}_3 = \frac{X_4}{X_3} \dot{m}_4 \quad (43) \]

However, for reasons of flow continuity, the mass flow rate through the separator drain must be included in the control volume analysis; therefore

\[ \dot{m}_3 = \dot{m}_4 + \dot{m}_{de} \quad (44) \]
Substituting Eq. (43) into Eq. 41 and solving for \( \dot{m}_{de} \)
the following expression can be derived

\[
\dot{m}_{de} = \left( \frac{\frac{\dot{Q}^*}{\dot{W}_{RP}}}{\frac{T_2}{T_1}} - 1 \right) \dot{m}_4
\]

Looking at the evaporator as a separate control volume,

the energy balance is

\[
\dot{Q} + \dot{m}_2 h_2 + \dot{m}_2 \dot{h}_{12} + \dot{m}_0 \dot{h}_0 = \dot{m}_3 \dot{h}_3 + \dot{m}_{ref} \dot{h}_{ref}
\]

Assuming the change in enthalpy across the re-flux pump and the difference between the separator drain outlet and evaporator inlet are negligible, the energy balance becomes

\[
\dot{Q} + \dot{m}_2 h_2 + \dot{m}_{de} h_{de} = \dot{m}_3 h_3
\]
where $\dot{m}_D = \dot{m}_1$, fluid drained from the separator is

assumed to be a saturated liquid.

Furthermore, a mass balance of the evaporator control volume can be expressed as

$$\dot{m}_2 + \dot{m}_C + \dot{m}_D = \dot{m}_3 + \dot{m}_E$$

(47)

where $\dot{m}_2 = \dot{m}_C$.

Solving Eq. (47) for the mass flow rate at state point 3 and substituting into Eq. (46), with Eq. (45) yields the following expression

$$\dot{m}_3 \left[ 1 + \frac{\rho_3}{\rho_2} \left( \frac{T_3}{T_2} - 1 \right) \right] = \dot{m}_2 \frac{V}{A} \frac{L}{\varepsilon}$$

(48)

In addition, a mass balance for steady-state, steady-flow indicates that the mass flow rates at state points 3 and 1 are equal and therefore

$$\dot{m}_1 = \dot{m}_2$$

(49)

Using Eqs. 48 and 49, the revised mass flow rate at state point 2 may be determined. Concurrently, the revised average ammonia velocity acting on the tube profile geometry may be determined from this revised mass flow rate. Using the revised ammonia velocity acting on the tube profile geometry and iterating from Eq. 31 until
an acceptable convergence criterion is achieved provides the pressure drops across the evaporator and moisture separator, and the properties at state points 3 and 4 for a given film temperature. The result is more representative of the heat absorption phase in the OTEC power cycle than is the commonly used ideal analysis.

In addition, solving for the revised temperature at state point 3,
\[
T_3 = T_{\text{sat}} \bigg|_{P_3}
\]
and iterating through Eq. (18) revises the film temperature and subsequent working fluid properties.

16. Revised Shellside Ammonia Heat Transfer Coefficient

In the search for acceptable correlations to predict the average evaporative heat transfer coefficient, two analytical treatments were found that lent themselves to OTEC power system conditions.

The first of these correlations seeks to predict thin film evaporation heat transfer coefficient for horizontal tubes [Ref. 6]. Owens [Ref. 6] uses (1) the similarity between evaporation and condensation, (2) the correlation forms of local evaporation heat transfer coefficients for water on a vertical tube developed by Chun and Seban, and (3) the dependence of heat transfer on the vertical spacing of the tubes as was experimentally demonstrated by Liu, to arrive at the following correlations for non-boiling thin film evaporation:
for laminar flow

$$\frac{H}{d_o} = 2.2 \left( \frac{H}{d_o} \right) \left( \frac{U_f}{\sqrt{g^2 K_f}} \right)^{-1/3} \left( \frac{4 \Gamma}{U_f} \right)^{-1/3} \tag{51}$$

for turbulent flow

$$\frac{H}{d_o} = 0.185 \left( \frac{H}{d_o} \right) \left( \frac{U_f}{\sqrt{g^2 K_f}} \right)^{-1/3} \left( \frac{C_p U_f}{K_f} \right)^{0.5} \tag{52}$$

where \( \frac{H}{d_o} \) = vertical spacing with respect to tube outer diameter.

\( \Gamma \) = tube flow rate per unit length.

The laminar-turbulent transition point is defined by the intersection of Eqs. (51) and (52)

$$Re_{TR} = \frac{C_p U_f}{K_f} \left( \frac{4 \Gamma}{U_f} \right)^{-1.5}$$

The pseudo-Reynolds number for horizontal vertical falling film evaporation is defined by Ref. 7.

$$Re = \frac{4 \Gamma}{U_f}$$

The second correlation combines boiling and evaporation of liquid films on horizontal tubes, applicable for vertical banks of plain and enhanced tubes [Ref. 8].

53
The overall model for a single tube is expressed as

\[ \bar{h} = h_b + h_d \frac{L_d}{L} + h_c \left( 1 - \frac{L_d}{L} \right) \]  

(53)

where \( h_b \) = Rohsenow pool boiling correlation over the entire tube length given by

\[ h_b = \frac{C s_f}{\gamma_f^{2/3}} \left( \frac{\rho_\infty}{\rho_f} \right)^{0.22} \]  

(54)

with \( C s_f \) = function of the fluid-surface combination.

\( \Delta T \) = wall temperature minus free stream saturation temperature.

\( \gamma_f \) = surface tension

\( h_d \) = heat transfer coefficient in the developing region.

\[ h_d = \frac{3 \gamma_f f}{2 \rho \alpha} \]  

\[ L_d = \gamma_f^{4/3} \left( \frac{3 \gamma_f}{\rho \alpha} \right)^{1/3} \]  

and \( h_c \) = fully developed heat transfer coefficient given for laminar flow by

\[ h_c = 0.321 \left( \frac{2 - C_s^3}{k^3 \gamma_f^{2/3} \rho_f^{1/3}} \right)^{1/3} \left( \frac{4 \gamma_f}{\rho \alpha} \right)^{-0.22} \]  

(55)
and, for turbulent flow,

\[ h_c = 3.8 \times 10^3 \left( \frac{2 - \frac{\rho}{\mu^2}}{\kappa} \right)^{-1/3} \left( \frac{4\Gamma}{\kappa L_d} \right) \frac{C_2}{C_{1,5}} \]  \hspace{1cm} (56)

where \( L \) = circumferential length of heated surface.
\( \kappa \) = thermal diffusivity.
\( L_d \) = developing length around tube circumference.
\( \Gamma \) = flow rate per unit axial length of tube.

To apply Eq. (51) for a vertical bank of tubes, \( L \) is expressed as

\[ L = N_t \frac{\Gamma d_o}{2} \]

The laminar-turbulent transition point is defined by the intersection of Eqs. (55) and (56)

\[ Re_{\tau, K} = 5 \varepsilon \frac{\rho \Gamma}{\kappa} \left( \frac{\rho}{\kappa} \right)^{-1.06} \]

As before, the pseudo-Reynolds number is defined by Ref. 7

\[ Re = \frac{4\Gamma}{\kappa L_d} \]  \hspace{1cm} (57)

After using Eq. (57) to establish which flow regime the system is operating in, the revised heat transfer coefficient for non-boiling thin film evaporation or nucleate
boiling may be calculated and then iterated with the initial assumption for the shellside heat transfer coefficient, Eq. (9). This will have a convergence effect on variables which are a function of the shellside heat transfer coefficient, moving them closer to actual OTEC system performance characteristics.

The user should be aware that the predictions for the OTEC power system using ammonia have been for the case where no boiling occurs in the film. This condition is dictated by industrial preference for plain tube heat exchangers to minimize fouling and the characteristic of ammonia to wet surfaces well, flooding out nucleation sites. A number of enhancement techniques have been developed to create nucleate boiling, including a variety of tube configurations and surface preparations; however, a preference for them has not materialized. The nucleate boiling development in Eq. (51) which would be indicative of tube enhancement is provided for information only and will not be included in the optimization or summary of conclusions.

Having described the methods used to predict the shellside heat transfer coefficient, we can complete this chapter of the OTEC power system analysis by constructing the heat exchanger cost analysis.

17. Evaporator Cost Analysis

At the request of TRW, Wyatt Industries, a large exchanger fabricator, prepared cost estimates for three different sizes of vertically configured evaporators and condensers, based upon initial design specifications prepared
by TRW. Based upon these estimates, TRW developed sets of equations that represent the costs of various heat exchanger component parts for shell diameters ranging from 10-35 ft and 35-50 ft [Ref. 9].

The following are the TRW evaporator cost ($) equations as a function of outer tube diameter (inch), total number of tubes and tube-sheet diameter (ft) for tube-sheet diameters of 10-35 ft.

. Drilling time/tube sheet thickness

\[ t_d = 0.66 (d_o - c.5) \]

(58)

. Thickness of the tube sheet

\[ t_{TS} = c.56 \frac{C.TS}{T_{SD}} \]

(59)

. Tube sheet labor cost

\[ C_{TSL} = 156695 \left( \frac{N_t}{9630} \right) (t_d / c.66) (t_{TS} / 4) \]

(60)

. Tube sheet material cost

\[ C_{TSM} = 189436 \left( \frac{C.TS}{T_{SD}} \right)^{2.3} \]

(61)

. Tube installation cost

\[ C_{TI} = 34 N_t d_o \]

(62)

. Heat exchanger shell cost

\[ C_{HXS} = 177265 \left( \frac{L_s + 6}{3} \right) \left( \frac{T_{SD}}{18} \right)^2 \]

(63)
Ammonia distribution plate and battles cost

\[ C_{DPB} = 93 \times 65.75 \left( \frac{N_t}{9630} \right) \left( \frac{t_d/0.6 \theta}{T_{SP}/18} \right)^2 \]  

(64)

Bustle, flanges channels and flow plates cost

\[ C_{BFCF} = 308550 \left( \frac{T_{SP}/18}{18} \right)^2 \]  

(65)

Tube material cost

\[ C_{TH} = (E_1 t + E_2) N_t \frac{d_o}{1.5} \]  

(66)

where \( E_1 \) = curve fit of tube cost per foot.

\( E_2 \) = tube machining cost if required

Heat exchanger head costs

\[ C_{HXX} = 53240 \left( \frac{T_{SP}/18}{18} \right)^3 \]  

(67)

Water inlet, nozzles and supports cost

\[ C_{WNS} = 220310.75 \left( \frac{T_{SP}/18}{18} \right)^2 \]  

(68)

Tube welding costs (Titanium tubes)

for \( N_t \leq 36000 \)

\[ C_{TW} = 14.73 N_t \left( \frac{d_o/1.5}{1.5} \right)^{1.7} \]  

(69)

for \( N_t > 36000 \)

\[ C_{TW} = 0.8797 N_t \left( \frac{d_o/1.5}{1.5} \right)^{1.7} \]

The sum of cost Eqs. (60) through (69) would equal the cost to fabricate one OTEC evaporator with a tube sheet diameter of 10-35 feet (all the preceding component costs have been adjusted for current pricing at a 10% annual rate of inflation).
If our analysis is based on a 30-year life-cycle criterion, no adjustments are necessary to any component cost equation if titanium tubing is used due to its anti-corrosive qualities; however, using aluminum tubing (i.e., Al-5052), the expense of retubing must be considered to meet the criterion of a 10-year life cycle for aluminum tubing. This implies Eq. (61) and (66) must be modified to reflect the costs of retubing at the 10 and 20-year point in the cycle.

Aluminum tube installation cost

\[ C_{ATI} = C_{TI} \left[ t + (1 + i)^{10} + (1 + i)^{20} \right] \quad (70) \]

where \( i \) = projected inflationary rate (input by customer)

Aluminum tube material cost

\[ C_{ATM} = C_{TM} \left[ t + (1 + i)^{10} + (1 + i)^{20} \right] \quad (71) \]

For tube sheet diameters of 35-50 ft the following cost relationships apply [Ref. 9]:

. Equations for drilling time/tube sheet thickness (58), thickness of tube sheet (59), and tube material costs remain unchanged.

. Tube sheet labor and material cost (titanium)

\[ C_{TSL} = 55.189 N_{t}^{0.741} T_{SD}^{0.58} t_{d} \quad (72) \]

\[ C_{TSM} = 20.566 T_{SD}^{2.014} t_{d} \quad (73) \]
. Tube sheet labor and material cost (aluminum)

\[ C_{TSL} = 73.181 N_t^{0.741} T_{SO}^{0.68} t_d \tag{74} \]

\[ C_{TSM} = 354.3 T_{SO}^{1.61} t_{T3} \tag{75} \]

. Tube installation costs

\[ C_{TI} = 36.542 N_t^{0.7} d_t \tag{76} \]

. Heat exchanger shell cost

\[ C_{HXS} = 12.544 \left( L_t + 6 \right) T_{SO}^{2.06} \tag{77} \]

. Ammonia distribution plate and baffle costs

\[ C_{DPB} = 158.099 T_{SO}^{1.82} + 72.419 N_t^{0.873} t_d \tag{78} \]

. Bustle, flanges, channels, and flow plate costs

\[ C_{BFCF} = 472.977 T_{SO}^{2.12} \tag{79} \]

. Heat exchanger head cost

\[ C_{HXH} = 1725.31 T_{SO}^{1.45} \tag{80} \]

. Water inlet, nozzles and support cost

\[ C_{WINS} = 7445.297 T_{SO}^{6.1} \tag{81} \]
Tube welding costs (titanium tubes)
for \( N_t \leq 36 \text{ CCE} \):
\[
C_{TW} = 44.73 \, N_t^{1.53} \left( \frac{d_s}{1.5} \right)^{0.7}
\] (82)

for \( N_t > 36 \text{ CCE} \):
\[
C_{TW} = 0.8797 \, N_t^{1.03} \left( \frac{d_s}{1.5} \right)^{0.7}
\]

As indicated previously, the cost to fabricate one OTEC evaporator with a tube sheet diameter 35 to 50 ft is equal to the sum of component costs Eqs. (72) through (83) (all the preceding component costs have been adjusted for current pricing at a 10\% annual rate of inflation).

For an analysis based on a 30-year system life-cycle criterion, the additional costs for aluminum retubing must be considered and Eqs. (70) and (71) apply.
A. INTRODUCTION

This chapter describes in detail the programming analysis for parasitic losses which include: (1) pumping and pipe requirements for both cold and hot salt water systems, (2) pumping and pipe requirements for the working fluid (ammonia) circulation and re-flux systems, and (3) turbine generator losses due to inefficiencies. Hotel requirements have not been incorporated into the analysis, but could be included for the final design analysis.

Pumping power requirements will be determined through the use of the general energy equation between the inlet and outlet of the system control volume [Ref. 3].

\[
\int_{i}^{o} \frac{dP}{\nu} + \frac{V_i^2}{2} + g Z_i = \frac{V_o^2}{2} + g Z_o + \dot{W}_S + (Losses)_{i\to o}
\]

To determine the pumping power \( \dot{W}_S \) the following effects will be evaluated:

1. Density head.
2. Friction losses.
   . Intake piping.
   . Heat exchanger tubing.
   . Exit piping (if employed).
3. Thermodynamic pressure head.
4. Elevation head.


5. Minor losses.
   . Intake piping inlet configuration (contraction).
   . Intake piping screen (obstruction).
   . Flow through valves, elbows, etc.
   . Outlet piping (expansion).
   . Inlet to heat exchanger tubing (contraction).
   . Outlet from heat exchanger tubing (expansion).
   . Outlet of exit piping (if employed).

In the above pump head evaluations, the following inputs are specified:

   . Pipe lengths (hot, cold, ammonia circulation and re-flux piping).
   . Inner pipe diameters (initialized and treated as a design variable by the optimization code).
   . Absolute roughness corresponding to piping/tubing material (designer specified).
   . Fluid velocity (initialized and treated as a design variable by the optimization code).
   . Pump mechanical and electrical efficiencies.

As an overview of the parasitic pump loss analysis, the following major steps in the algorithm are listed in order of their execution:

   . Hot pipe salt water pump.
     .. Inlet piping friction losses (86).
     .. Minor piping losses due to inlet screen (87) and plenum design to evaporator core (88).
     .. Evaporator core minor losses (89, 90) and tubeside friction losses.
     .. Total pressure losses (92) and pumping head (93).
     .. Pumping power requirements (95).
.. Pump cost analysis (96).

. Cold pipe salt water pump.
  .. Initialize cold pipe inner diameter and SW velocity (design variables).
  .. Minor losses due to inlet ducting (97) and plenum design to condenser core (98).
  .. Inlet piping friction losses (99).
  .. Condenser core minor losses (100, 101) and tubeside friction losses (103).
  .. Density head (104).
  .. Total pressure losses (105).
  .. Pumping power requirement (107).
  .. Pump cost analysis (108).

. Ammonia circulation pump.
  .. Piping friction (109) and minor losses due to valving/elbows (110).
  .. Pressure drop across evaporator shellside (112).
  .. Thermodynamic head (113).
  .. Elevation head (114).
  .. Total pressure losses (115).
  .. Pumping power requirement (116).
  .. Pump cost analysis (118).

. Ammonia re-flux pump.
  .. Piping friction (119) and minor losses due to valving/elbows (120).
  .. Thermodynamic head due to pressure drop of saturated liquid ammonia across evaporator shellside (122).
  .. Elevation head (123).
  .. Total pressure losses (124).
In the following section, the basic steps summarized above will be described in detail.

B. ANALYSIS OF PARASITIC LOSSES

1. Hot Pipe Salt Water Pump, $\Sigma_{pl}$

The pressure losses due to piping friction and associated minor losses will be determined using the Darcy-Weisbach correlation [Ref. 10].

$$\Delta P = \sum_{i=1}^{n} \frac{\rho K_i V_i^2}{2 g_c}$$

where $K_i$ describes the resistance coefficient.

$V = \text{fluid velocity.}$

$$K_i = f \frac{L}{D}$$

where $f = \text{friction factor.}$

$\frac{L}{D} = \text{equivalent length in pipe diameters.}$

In order to determine the friction factor, the pipe flow Reynolds number must be calculated.

$$Re_d = \frac{\rho_{sw} V_{sw} d}{\mu_{sw}}$$
where \( \nu_{sw}, \rho_{sw} \) = properties of salt water at the hot pipe inlet temperature (assumed constant throughout the pipe).

\( \nu_{sw}, d_i \) = salt water velocity and inner pipe diameter (initialized and treated as design variables by the optimization code); velocity assumed constant over pipe length.

Pipe flow Reynolds number greater than 2300 will be considered turbulent.

For laminar flow

\[
f = \frac{64}{Re_d} \tag{85}
\]

For turbulent flow

\[
f = \frac{1.325}{\ln \left( \frac{\varepsilon}{3.74 d_i + 5.74/Re_d^{0.14}} \right)^2} \tag{86}
\]

where \( \varepsilon \) = absolute roughness corresponding to piping material selected.

Eq. (86) yields a friction factor within one percent of the Colebrook equation and is valid for the following conditions [Ref. 9].

\[
\begin{align*}
10^{-6} & \leq \frac{\varepsilon}{D} \leq 10^{-2} \\
5000 & \leq Re_d \leq 10^5
\end{align*}
\]
Considering the resistance coefficient for pipe minor losses:

Assume the inlet duct is the same size as the pipe inner diameter, but it is screened

\[ k' = 1.5 \]  \( \text{Eq. (87)} \)

Assume piping enters evaporator through an area which is abruptly changed [Ref. 11]

\[ k = \left[ 1 - \left( \frac{d_i}{T_{3D}} \right)^2 \right] \]  \( \text{Eq. (88)} \)

where \( T_{3D} = \) evaporator tube sheet diameter (assume tube sheet diameter is twice as large as the inner pipe diameter).

Summing the results of Eqs. (84), (87), and (88) to determine the total resistance coefficient, the pressure losses due to piping can then be determined using Eq. (85).

If a variety of valves or fittings are to be included with Eq. (84), Ref. 11 provides a representative listing of equivalent length-to-pipe-diameter values.

To analyze the pressure drop across the evaporator tubeside, we again use the Darcy-Weisbach correlation, but for different design assumptions.

Assume inlets to evaporator tubing are well rounded [Ref. 11]

\[ k = 0.5 \]  \( \text{Eq. (89)} \)
Assume outlets of evaporator tubing expand to an infinite reservoir [Ref. 10]

\[ K = 1.5 \]  

(90)

Using the Reynolds number in the previous chapter, Eq. (5), the corresponding friction factor Eq. (85) or (86), and resistance coefficient can be determined

\[ K_{CORE} = f \frac{L_e}{d_i} \]  

(91)

where \( L_e, d_i \) = evaporator tube length and inner tube diameter and are initialized and treated as design variables by the optimization code.

Summing the results of the resistance coefficient in Eqs. (89), (90) and (91), the pressure losses due to the evaporator design may be determined using the Darcy-Weisbach correlation Eq. (83).

The results of the piping losses and core design losses are equivalent to the hot pipe salt water pumping system requirements

\[ \Delta P_{pump} = \Delta P_{PIPE \_SYSTEM} + \Delta P_{EVAP \_DESIGN} \]  

(92)

converting to pumping head

\[ H = \frac{g_e}{\rho_{SW} g} \Delta P_{pump} \]  

(93)
Pumping power in terms of horsepower can be determined using the following expression

\[ P_{\text{HP}} = \frac{m_{\text{sw}}}{\eta_{\text{p}}} \left( \frac{gH}{g_c} \right) \]  \hspace{1cm} (94)

where \( \eta_{\text{p}} \) = pump mechanical efficiency (designer input).

\( m_{\text{sw}} \) = salt water mass flow rate determined in previous chapter, Eq. (2).

To equate parasitic pump losses to power input, Eq. (94) is converted to the motor load requirement in terms of megawatts electrical.

\[ P_{\text{HP(\text{MW})}} = \frac{P_{\text{HP}} \times \text{CONVERSION FACTOR}}{\eta_{\text{M}}} \]  \hspace{1cm} (95)

where \( \eta_{\text{M}} \) = pump motor efficiency (designer input).

Because of the high salt water flow rates and relatively low pumping heads, good engineering design would dictate the use of axial flow (propeller) type pumps.

Using the algorithm developed by TRW [Ref. 9] from data provided by Johnston Pump Co., and Process Equipment Co. (distributors of Ingersoll Rank and Johnston Pumps), the cost of salt water pumps can be expressed as

\[ C_{\text{pump}} = \left[ \left( \frac{D}{1000} \right) 0.75 + 5C \right] 1.21 \times 10^3 \]  \hspace{1cm} (96)
where

\[ D = \frac{n d_i^2}{4} v_{sw} \]

where \( d_i, v_{sw} \) = inner hot pipe diameter, salt water velocity
(initialized for analysis and treated as
design variables by the optimization code).

The above algorithm is valid for the following conditions

- vertical, wet pit, propeller type pumps with
  cast iron steel columns with protective epoxy coating, stainless steel shaft and bronze impeller.
- pump size from 155,000 through 750,000 GPM with total dynamic heads of 8 through 12 feet.

Eq. (96) has been adjusted for current pricing at a 10% annual rate of inflation.

2. Cold Pipe Salt Water Pump, \( \mathcal{C}_{cl} \)

Using Reynolds number

\[ \text{Re}_d = \frac{\rho_{sw} v_{sw} d_i}{\mu_{sw}} \]

where \( \rho_{sw}, \mu_{sw} \) = properties of salt water at the cold pipe inlet temperature (assumed constant throughout the pipe).

\( v_{sw}, d_i \) = salt water velocity and inner pipe diameter
(initialized and treated as design variable by the optimization code), velocity assumed constant over pipe length.
Pipe flow characteristics and friction factor can be identified. A pumping analysis will be developed for the cold pipe pump using the Darcy-Weisbach correlation, similar to the development in the preceding section.

Considering the resistance coefficient for minor pipe losses

Assume the inlet duct is well rounded [Ref. 11].

$$K_{\text{INLET}} = 0.5$$

(97)

Assume piping enters condenser through an area which is abruptly changed [Ref. 10].

$$K_{\text{PLENUM}} = \left[ 1 + \left( \frac{d_i}{T_{SD}} \right)^2 \right]^2$$

(98)

where $T_{SD} = \text{condenser tube sheet diameter}$ (assume tube sheet diameter is twice as large as the inner pipe diameter).

Assume one ninety-degree elbow is required in system [Ref. 11].

$$\frac{L}{D} = 30$$

Summing the results of Eqs. (84), (97), and (98), the total resistance coefficient can be expressed as

$$K = f \left( \frac{L}{d_i} + \frac{L}{D} \right) + K_{\text{INLET}} + K_{\text{PLENUM}}$$

(99)
where $L_p$ = length of cold pipe.

$\delta_i$ = inner diameter of cold pipe.

Pressure losses due to piping can then be determined using the Darcy-Weisbach, Eq. (83).

In analyzing the pressure drop across the condenser tubeside, the Darcy-Weisbach correlation is used again, but for different design assumptions.

1. Assume inlets to evaporator tubing are well rounded.

$$K = 0.5$$  \hspace{1cm} (100)

2. Assume outlet of condenser tubing expands to an infinite reservoir.

$$K = 1.0$$  \hspace{1cm} (101)

Defining Reynolds number for condenser tubeside flow, while assuming

$$T_{bulk} = T_{cold\ (inlet)}$$  \hspace{1cm} (102)

$$Re_d = \frac{\rho w \cdot V sw \cdot \delta_i}{\mu sw}$$

where $\rho w / \mu sw = $ properties evaluated at condenser tubeside bulk temperature (initially assumed equal to cold pipe inlet temperature).
\( \nu_{\text{sw}, d_i} \): average salt water velocity through tubing, inner condenser tube diameter (both are initialized and treated as design variables by the optimization code).

The corresponding friction factor, Eq. (85) or (86), and resistance coefficient can be determined

\[
K_{\text{core}} = f \frac{L_t}{d_i},
\]

(103)

where \( L_t, d_i \), the condenser tube length and inner tube diameter are initialized and treated as design variables by the optimization code.

Summing the results of the resistance coefficient in Eqs. (100), (101), and (103), the pressure losses due to the condenser design may be determined using the Darcy-Weisbach correlation, Eq. (83).

A complete analysis of cold pipe losses must also include the effect of density head and a corresponding increase in pumping power requirements.

For most engineering problems involving the flow of liquids through a pipe, where the temperature change in the pipe is small, the density of the fluid is considered to be a constant and the fluid is termed "incompressible." However, the flow problem in OTEC cold pipe systems is unique. We can continue to assume that there is negligible change in the fluid temperature, virtually unaffected by the ocean thermal gradients, because of the system's characteristic high mass
flow rates. However, the height of the water column (1500 to 3000 feet) inside the pipe requires the effect of fluid compressibility to be taken into consideration.

The effect of an increase in density with depth can be expressed by the integral

\[ \int_{i}^{e} \frac{dt}{\rho g} \]

with a density head defined as\(^2\)

\[ H_{\rho} = \frac{Z_{e} - Z_{i}}{\rho_{o} g} + \int_{i}^{e} \frac{dt}{\rho g} \]

Integrating the pressure-density variation, the density head reduces to [Ref. 12]

\[ H_{\rho} = \frac{Z_{e} - Z_{i}}{\rho_{o} g} + \frac{1}{\rho_{o} g} (P_{e} - P_{i}) \left[ 1 - \frac{K_{m}}{2} \left( \frac{P_{e}}{P_{i}} + 1 \right) \right] \]

where \( K_{m} \) = mean compressibility of water, \( f(\text{salinity, temperature and pressure}) \).

\( \rho_{o} \) = reference density at which \( K_{m} \) is evaluated.

Considering pressure at any depth obtained from the integral,

\[ P = -g \int_{i}^{e} \rho(z) dz \]

the density head can be rewritten as follows

\[ H_{\rho} = (Z_{e} - Z_{i}) - \frac{1}{\rho_{o} g} \int_{Z_{i}}^{Z_{e}} \rho(z) dz \left( 1 - \frac{K_{m}}{2} \int_{Z_{i}}^{Z_{e}} \frac{z_{i} + Z_{e}}{Z_{i} + Z_{e}} \right) \]

\(^2\)Note that \( Z \) is measured as positive upward so that ocean depth values \((Z_{e}, Z_{i})\) are negative and \((Z_{e} - Z_{i})\) is a positive quantity.
Rigorous procedures for calculating the density profile which is a function of temperature, salinity and pressure may be found in Ref. 13; however, they will not be discussed in this document.

For the purposes of simplification, the following solution technique was developed:

(1) If the liquid in the pipe is taken to have a constant density with respect to pressure, the compressibility approaches zero; the density head can then be expressed as

\[ H_\rho = (Z_e - Z_i) - \frac{1}{\rho_i} \int_{Z_i}^{Z_e} \rho(z) \, dz \]

(2) Converting the geometric term for elevation to an equivalent integral expression

\[ Z_e - Z_i = \frac{1}{\rho_i} \int_{Z_i}^{Z_e} \rho \, dz \]

The reference density is taken to be the inlet value so that

\[ \rho_i = \rho_i \]

and the density head can be rewritten as follows

\[ H_\rho = \frac{1}{\rho_i} \int_{Z_i}^{Z_e} (\rho_i - \rho(z)) \, dz \]

(3) Assuming a linear distribution of density with depth, due to temperature variations, as illustrated below
the following linear expression for density with respect to depth may be formulated, where $Z_e = 0$ for convenience.

$$\rho_i - \rho = (\rho_i - \rho_e)(1 - Z/Z_i)$$

(4) Applying the equation developed in section 3 to the density head integral above and integrating over the range of values for sea water depth ($z$), the following equation is derived as a linear approximation to the density variation of sea water with respect to depth

$$H_p = (\rho_i - \rho_e)\left(-\frac{Z_i}{2}\right)$$

where $\rho_i, \rho_e$ = curve fit evaluations of density for specified depths of sea water. Data extracted from Ref. 14.

The results of the piping losses, core design losses, and density head are equivalent to the cold pipe salt water pumping system requirements

$$\Delta P_{pump} = \Delta P_{pipe~system} + \Delta P_{cond~design} + \Delta P_{density} \quad (105)$$
Using Eq. (93), Eq. (105) can be converted to a pumping head. Similarly, pumping power in terms of horsepower can be determined using Eq. (44).

\[
P_{CP} = \frac{\dot{m}_{sw} (gH)}{\rho_w g_c}
\]

where

\[
\dot{m}_{sw} = \rho_{sw} \left( \frac{\pi d_i^2}{4} \right) V_{sw}
\]

and \( \rho_{sw} \) = density of salt water evaluated for a constant inlet temperature.

\( V_{sw}, d_i \) = cold pipe salt water velocity, and inner diameter (initialized and treated as design variables by the optimization code). Note salt water velocity through cold pipe is considered to be constant.

Pumping power can then be expressed in terms of megawatts electrical

\[
P_{CP(\text{MW})} = \frac{P_{CP} \times \text{CONVERSION FACTOR}}{\eta_M}
\]

where \( \eta_M \) = pump motor efficiency (designer input).

Using the same arguments for the selection of an axial flow (impeller type) pump, as used for the hot pipe salt water pump, the pump cost algorithm developed by TRW can be applied to the cold pipe salt water pump assuming...
the required conditions are validated.

\[ C_{\text{pump}} = \left[ \left( \frac{D}{1000} \right) 0.15 + 50 \right] 1.21 \times 10^{-3} \] (108)

Equation (108) has been adjusted for current pricing at a 10% annual rate of inflation.

3. **Ammonia Circulation Pump, \( \dot{P}_{\text{IRC}} \)**

The function of the ammonia circulation pump is to circulate and lift saturated liquid ammonia from the condenser hot well at state point 1 and increase its pressure to exceed the operating conditions in the evaporator at state point 2.

In order to evaluate these characteristics, the following pumping elements will be included in the analysis:

- Piping losses (friction and minor).
- Heat exchanger shellside pressure drop.
- Thermodynamic pressure head.
- Elevation head.

As in the preceding analysis, Reynolds number is used to determine pipe flow characteristics

\[ \text{Re}_d = \frac{\rho \, V \, d_i}{\mu} \]

where \( \rho, \mu \) = saturated liquid properties of ammonia for the temperature at state point 1 (assume any temperature increase from pump work is negligible).

\( d_i \) = inner pipe diameter (initialized and treated as a design variable by the optimization code).
\( V = \) ammonia flow velocity determined from the preceding chapter, Eq. (50).

The ammonia pipe friction factor can then be determined from Eqs. (85) or (86), and the piping friction resistance coefficient can be expressed as

\[
K = f \frac{L}{d},
\]

(109)

where \( L = \) ammonia circulation pipe length (designer input).

Considering the resistance coefficient for minor pipe losses, assume there are four ninety-degree elbows in the system

\[
K' = 4 \frac{L}{D}
\]

(110)

where \( \frac{L}{D} = \) equivalent length in pipe diameters for a standard elbow [Ref. 11].

Summing the results of Eqs. (109) and (110), piping losses (friction and minor) can be determined using the Darcy-Weisbach equation (93).

\[
\Delta P_{pipe} = \rho \left[ f \frac{L}{d} + 4 \frac{L}{D} \right] \frac{V^2}{2g_c}
\]

(111)

The heat exchanger shellside pressure drop is also included in the pumping head requirement because it serves as a resistance to flow.

79
Pressure drop across the evaporator shellside was determined using the two-phase flow model (homogeneous) expressed by Eq. (33)

\[ \Delta P_{\text{EVAP}} = \Delta P_{\text{TRICTION}} + \Delta P_{\text{MOMENTUM}} + \Delta P_{\text{ELEVATION}} \]  

Since the pump is required to lift the working fluid to a higher elevation and increase its operating pressure, the following elements must be included in the analysis:

. Thermodynamic head

\[ \Delta P_{\text{THermo}} = P_Z - P_1 \]  

represents the difference in thermodynamic operating pressure between state point 2 and state point 1.

. Elevation head

\[ \Delta P_{\text{ELEVATION}} = Z_Z - Z_1 \]  

\[ Z_1 = \text{datum.} \]
\[ Z_Z = \text{elevation of the evaporator inlet above datum (taken to be equal to evaporator tube sheet diameter plus 25).} \]

represents the lift head required to move the working fluid to a higher elevation.
The results of piping losses (111), evaporator pressure drop (112), the thermodynamic head (113) and elevation head (114) are equivalent to the ammonia circulation pump system requirements.

\[
\Delta P_{pump} = \Delta P_{pipe} + \Delta P_{evhp} + \Delta P_{thermic} + \Delta P_{elevation}
\]

(115)

Using Eq. (93) with ammonia properties, Eq. (113) can be converted to pumping head and finally expressed as pumping power (horsepower).

\[
P_{circ} = \frac{m_{NH_3} \text{gh} H}{\eta_p}
\]

(116)

where \(m_{NH_3}\) = mass flow rate of ammonia determined by Eq. (20) of the previous chapter.

\(\eta_p\) = pump mechanical efficiency (designer input).

Pumping power can then be expressed in terms of mega-watts electrical

\[
P_{circ(MW)} = \frac{P_{circ} \text{ conversion factor}}{\eta M}
\]

(117)

where \(\eta M\) = pump motor efficiency (designer input).

Because of high pumping head and moderate flow rates, good engineering design would dictate the use of a single suction centrifugal flow type pump.
Using the algorithm developed by Westinghouse Electric Co. [Ref. 15] from data provided by Bingham Pump Division, Portland, Oregon, the cost of the ammonia circulation pump can be expressed as

\[
C_{pump} = \left( \frac{\dot{m}_{NH_3} \nu_1}{20100} \right)^{0.64} \times 1.21 \times t_o^5
\]

(118)

where \( \dot{m}_{NH_3} \) = mass flow rate of ammonia \( (\text{lb}_m/\text{hr}) \)

\( \nu_1 \) = specific volume of saturated liquid ammonia at state point 1 \( (\text{ft}^3/\text{lb}_m) \)

Eq. (118) has been adjusted for current pricing at a 10% annual rate of inflation.

4. Ammonia Re-flux Pump, \( P_{\text{re-flux}} \)

The function of the re-flux pump is to recycle ammonia droplets which are not evaporated in the heat absorption process. Saturated liquid at approximately the heat exchanger's operating pressure is lifted from the evaporator drain to the ammonia feed inlet, for redistribution as droplets across the evaporator tube bundle. (Drainage mass flow rate is assumed to be equal to 30% of the evaporator inlet feed mass flow rate.)

In order to evaluate these characteristics, the following pump elements will be analyzed:

- Piping losses (friction and minor).
Thermodynamic pressure head.

Elevation head.

As in the preceding analysis, Reynolds number is used to determine pipe flow characteristics

\[ Re_d = \frac{\rho V d_i}{\mu} \]

where \( \rho, \mu \) = saturated liquid properties of ammonia for the average pressure across the evaporator.

\( d_i \) = inner pipe diameter (initialized and treated as a design variable by the optimization code).

\( V \) = ammonia flow velocity determined from the evaporator drainage mass flow rate assumed equal to 30\% of the evaporator inlet feed mass flow rate (assume velocity constant throughout the pipe).

The reflux pipe friction factor can be determined from Eqs. (85) or (86), and the piping resistance coefficient can be expressed as

\[ K = f \frac{L}{d_i} \quad (119) \]

where \( L \) = ammonia reflux pipe length (designer input).

Once again, considering the resistance coefficient for minor pipe losses assume there are four ninety-degree elbows in the system
where $\frac{L}{D}$ = equivalent length in pipe diameters from a standard elbow.

Summing the results of Eqs. (119) and (120), piping losses (friction and minor) can be determined using the Darcy-Weisbach, equation (83)

$$\Delta P_{p,irr} = \rho \left[ \frac{f L}{d_i} + 4 \frac{L}{D} \right] \frac{V^2}{2g_c} \quad (121)$$

In order to determine the thermodynamic pressure head, the pressure drop across the evaporator for the saturated ammonia liquid must be analyzed. Since the saturated vapor and liquid are in thermodynamic equilibrium, the results of Eq. (112) apply. Therefore

$$\Delta P_{liq} = P_3 - P_2$$

Therefore, the thermodynamic pressure head is equal to the pressure drop across the evaporator for the saturated ammonia liquid.

$$\Delta P_{thermo} = \Delta P_{liq} \quad (122)$$

Finally, the elevation head is equal to the elevation of the evaporator feed inlet with respect to datum, the drain outlet.
Therefore,

\[ \Delta P_{\text{ELEV}} = Z_2 - Z_1 \]  \hspace{1cm} (123)

where \( Z_1 \) = datum, drain outlet.
\( Z_2 \) = elevation of the evaporator inlet above datum
(taken to be equal to the evaporator tube sheet diameter plus 10).

The results of piping losses (121), the thermodynamic pressure head (122), and elevation head (123) are equivalent to the ammonia re-flux pump system requirements.

\[ \Delta P_{\text{pump}} = \Delta P_{\text{pipe}} + \Delta P_{\text{elevation}} + \Delta P_{\text{thermo}} \]  \hspace{1cm} (124)

As before, using Eq. (93), Eq. (124) can be converted to a pump head and finally expressed in terms of pumping power (horsepower).

\[ P_{\text{re-flux}} = \frac{\dot{m}_A}{\eta_p} \left( \frac{g H}{g_c} \right) \]  \hspace{1cm} (125)

where \( \dot{m}_A \) = drainage mass flow rate.
\( \eta_p \) = pump mechanical efficiency (designer input).

Pumping power can be expressed in terms of megawatts electrical

\[ P_{\text{re-flux (MW)}} = \frac{P_{\text{re-flux}}}{\eta_{\text{el}}} \text{ CONVERSION FACTOR} \]  \hspace{1cm} (126)
where $\eta'_M$ = pump motor efficiency (designer input).

Using the same arguments for the selection of a centrifugal pump, the pump cost algorithm developed by Westinghouse can also be applied to the ammonia re-flux pump.

\[
C_{\text{pump}} = \left( \frac{m'_{tr} v'_r}{80100} \right)^{0.64} 1.21 \times 10^5
\]  

(127)

where $m'_{tr}$ = mass flow rate of evaporator drainage ammonia (lbm/hr)

$v'_r$ = specific volume evaluated at the average evaporator pressure (ft$^3$/lbm)

Eq. (127) has been adjusted for current pricing at a 10% annual rate of inflation.

5. Parasitic Pump Losses

Parasitic pump losses is the summation of electrical auxiliary pumping requirements (hotel and maintenance loads not included) determined by Eqs. (95), (107), and (126).

\[
P_{\text{loss}} = P_{HP} + P_{CP} + P_{CIRC} + P_{RE-FLUX}
\]  

(128)
V. TURBINE AND ELECTRICAL POWER

A. INTRODUCTION

The turbine generator is one of the critical elements of the OTEC power system. Its energy conversion efficiency and efficiency of design have a major effect on the overall system performance. To illustrate this point, Ref. 16 reported that a three-point change in turbine efficiency from 85 to 88% results in a 3.6% increase in gross power, and a 5% increase in net power developed.

This chapter will describe the analysis to evaluate the expansion turbine thermodynamic properties and generator output. The use of these properties will determine the internal turbine efficiency and outlet quality subject to design and thermodynamic constraints. The relationship between the condenser operating pressure (design variable) and the turbine outlet quality will be used to initialize the heat rejection characteristics of the condenser.

General literature on turbomachinery designed for OTEC closed cycle systems indicates that a turbine having the following characteristics

- Double flow, axial inflow,
- Four stages of expansion,
- Operating at 1800 RPM,

provides the optimum aerodynamic design [Ref. 16]. However, it is not the intent of this thesis to analyze the geometry
and performance parameters of the turbine. Turbine geometry such as

- Specific speed and specific diameter,
- Wheel diameter,
- Rotational speed,
- Blade height,
- Blade stresses,

should be treated as a separate systems problem using optimization to improve state-of-the-art design.

Parasitic losses due to the following generator turbine inefficiencies will be evaluated in this section.

- Generator mechanical and electrical.
- Turbine mechanical.

As an overview of the turbine-generator analysis, the following major steps of the algorithm are listed in order of their execution:

- Gross electrical output with no parasitic losses (129).
- Enthalpy at state point 5 (130).
- Turbine outlet quality (131).
- Entropy at state point from a specified outlet quality (132).
- Quality and enthalpy at state point 5s (133, 134).
- Internal (adiabatic) turbine efficiency (135).
- Turbine cost analysis (137).
- Generator cost analysis (138).

In the following section, the basic steps summarized above will be described in detail.
B. ANALYSIS OF THE TURBINE AND ELECTRICAL POWER REQUIREMENTS

1. Gross Electrical Output and Inefficiency Losses

If the net electrical output required is indicated by (in terms of megawatts), the gross electrical load at the turbine shaft can be expressed as

\[ \dot{E}_g = \frac{\dot{E}}{\eta_{TM} \eta_{GEN}} + P_{Loss} \]  

(129)

where

\( P_{Loss} \) = parasitic pump losses determined by Eq. (128).
\( \eta_{TM} \) = turbine mechanical efficiency (designer input).
\( \eta_{GEN} \) = generator mechanical and electrical efficiency (designer input).

The loss of electrical output due to generator-turbine inefficiencies is equal to

\[ \dot{E}_{Loss} = \dot{E} \left( \frac{1}{\eta_{TM} \eta_{GEN}} \right) \]

2. Turbine Efficiency

The power developed across the turbine is

\[ \dot{E}_g = \dot{m} (h_5 - h_4) \]

where

\( \dot{m} \) = mass flow rate of ammonia given by Eq. (48).
\( h_4 \) = enthalpy at state point 4, Eq. (42).

From this, the enthalpy at state point 5 can be calculated.

If we initialize the operating pressure of the condenser in terms of \( P_S \), the following relations may be expressed
\[ h_{5g} = \left( h_g \right)_p \tag{130} \]

Therefore, it follows that the turbine outlet quality, \( x_{5s} \), can be determined from

\[ h_{5} = h_{5f} + x_{5s} ( h_{5g} - h_{5f} ) \tag{131} \]

Having established the moisture separator outlet pressure and temperature, Eqs. (40) and (41), the entropy at state point 4 can be determined for a known separator outlet quality (designer input) using the following relations

\[ S_{4f} = S_f \left( T_4 \right) \quad S_{4g} = S_g \left( T_4 \right) \]

\[ S_4 = S_{4f} + x_4 ( S_{4g} - S_{4f} ) \tag{132} \]

For isentropic turbine work,

\[ S_4 = S_{5s} \tag{133} \]

the quality at state point 5s may be determined using the following relations

\[ S_{5g} = S_g \left( T_5 \right) \quad S_{5f} = S_f \left( T_5 \right) \]

\[ S_{5s} = S_{5f} + x_{5s} ( S_{5g} - S_{5f} ) \tag{134} \]

Having determined the quality at state point 5s, the enthalpy can now be determined.
Using the results of Eqs. (41), (130), and (132), the internal turbine efficiency (adiabatic) can be determined, expressed by

\[
\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}}
\]  

(136)

To ensure a realistic selection of internal efficiency, the following constraints are attached to the optimization code:

- \( h_5 < h_{5q} \)
- \( x_{5s} < x_5 \)
- \( \eta_t \leq 90\% \)

3. **Turbine Cost Analysis**

The ammonia turbine cost is based on an algorithm developed by Westinghouse to estimate manufacturing costs [Ref. 15].

\[
C_{turb} = 2.42 \times 10^6 \left( 0.375 + \frac{\dot{E}_e}{136000 N_f} \right) F_f
\]

(137)

where

- \( \dot{E}_e \) = gross electrical output in KW.
- \( N_f = 2 \) (for a double flow turbine).
- \( F_f \) = flow price factor (1.0 for single-flow, 1.447 for double-flow).
The above algorithm is valid for the following conditions:

. Double flow, axial inflow.
. Multi-stage.
. Operating at 1800 RPM.

The generator cost will be based on an algorithm developed by TRW from data provided by selected manufacturers,

\[ C_{GEN} = \left( 0.23 \hat{E}_G + 0.3 \right) \times 1.21 \times 10^6 \]  

(138)

and is valid for the following conditions

. 1800 RPM rotor speed.
. power factor 0.8.

Eqs. (137) and (138) have been adjusted for current pricing at a 10\% annual rate of inflation.
VI. CONDENSER

A. INTRODUCTION

As indicated in the introduction to Chapter III, several heat exchanger concepts have been proposed for the closed-cycle OTEC system, with variations in their design.

The analysis to be presented for the condensing heat exchanger will be based upon the following design characteristics:

2. Horizontal/vertical orientation of tubes with an equilateral triangle or square tube profile.
4. Tube material (titanium or aluminum based on a 30-year life-cycle criterion).
5. Biofouling control based upon an achievable fouling factor.
6. Heat exchanger centerline located on sea surface.

As an overview of the condenser analysis, the following major steps in the algorithm are listed in order of their execution:

1. Initialization of design variables (DV).
   1. Tube length.
   2. SW velocity through condenser tubes.
   3. Outer tube diameter.
   4. Tube profile pitch ratio.
2. Amount of heat rejection (139).
3. Tubeside bulk temperature (142).
In the following section, the basic steps summarized above will be described in detail.

B. ANALYSIS OF THE CONDENSER

1. Amount of Heat Rejection, \( Q \)

Using the calculated value for enthalpy at state point 5, equation (131) from the previous chapter, the ideal
values at state point 1, Eq. (21), and the steady-state mass flow rate of ammonia, Eq. (48), the amount of heat rejected by the condenser can be expressed as

$$\dot{Q} = \dot{m}_{\text{NH}_3} \left( h_5 - h_1 \right)$$

(139)

2. **Tubeside Bulk Temperature**

As in condenser tubeside Reynolds number, salt water properties will be evaluated at bulk temperature, initially assumed equal to the cold pipe inlet temperature.

Using this premise, the condenser salt water capacity rate can be evaluated

$$\dot{C}_{\text{w}} = \dot{m}_{\text{w}} \cdot C_{\text{P},\text{w}}$$

(140)

where $C_{\text{P},\text{w}} = $ specific heat of salt water initially evaluated at the cold pipe inlet temperature.

$\dot{m}_{\text{w}} =$ mass flow rate of salt water through the cold pipe previously evaluated by Eq. (10c).

Using the results of Eqs. (139) and (140), and the known cold pipe inlet temperature, the condenser salt water outlet temperature may be evaluated from the basic expression

$$\dot{Q} = \dot{C}_{\text{w}} \left( T_{C_o} - T_{C_i} \right)$$

(141)

where $T_{C_o}, T_{C_i} =$ condenser salt water outlet and inlet temperatures, respectively.

Having determined the condenser salt water outlet temperature, the revised bulk temperature can be expressed as
OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM (U)
Using the revised condenser bulk temperature and iterating with Eq. (102) corrects the operating temperature for salt water properties which are essential to the analysis.

3. **Total Number of Condenser Tubes, \( N_t \)**

Since the mass flow rate of salt water through the cold pipe is equivalent to the mass flow rate through the condenser, according to the law of continuity,

\[
\dot{m}_{cp} = \dot{m}_{cond}
\]

it follows that the number of condenser tubes for a specified tube diameter, can be evaluated using the following expression:

\[
N = \rho_{sw} \frac{\pi d_i^2 \nu_t}{4} N_t
\]

where \( \rho_{sw} \) = average salt water density evaluated at bulk temperature.

\( d_i \) = inner tube diameter (initialized and treated as a design variable by the optimization code).

\( \nu_t \) = average salt water velocity through the condenser (initialized and treated as a design variable by the optimization code).
4. Log Mean Temperature Difference, \( \text{LMTD} \)

Using the result of Eq. (141), the known pipe salt water inlet temperature, and the inlet temperature of ammonia evaluated at state point 5, the \( \text{LMTD} \) of the condenser may be expressed as

\[
\text{LMTD} = \frac{T_c - T_i}{\ln \left( \frac{T_s - T_c}{T_s - T_i} \right)}
\]  

(144)

5. NTU-Effectiveness Relations

The number of transfer units which is a measure of the condenser size can be determined from the basic expression

\[
\text{NTU} = \frac{U_o A_o}{C_{\text{min}}}
\]  

(145)

where the conductance \( (U_oA_o) \) of the heat exchanger is a function of the heat absorbed and the \( \text{LMTD} \).

\[
\dot{Q} = (U_oA_o) \text{LMTD}
\]  

(146)

The condenser effectiveness can then be expressed as

\[
\zeta = 1 - \zeta_n \text{NTU}
\]  

(147)

for a two-phase flow, regardless of the flow geometry.

Using the resistance analysis derived in Chapter III, Section 4 for an initialized tube length

$$L = L_i$$

the heat exchanger conductance for a single tube can be expressed as

$$U_o A_o = \frac{1}{\frac{1}{\nu_i h_{sw} A_i} + \frac{1}{A_i} R_{fsw} + \frac{1}{2 A_i} \ln \frac{d_o}{d_i} + \frac{1}{A_o} R_{fNH_3} + \frac{1}{\gamma_o h_{sw} A_o}} \quad (148)$$

where $h_{sw} = $ tubeside heat transfer coefficient.

$R_{fsw} = $ salt water fouling heat transfer resistance.

$K = $ thermal conductivity of the tube material.

$A_o, A_i = $ total outer and inner tube surface areas

(including fin and bare tube); tube length is initialized and treated as a design variable by the optimization code).

$R_{fNH_3} = $ ammonia fouling heat transfer resistance

$\gamma_o, \gamma_i = $ outer and inner total fin efficiency

a. Tubeside Reynolds Number

Since the salt water heat transfer correlation is dependent on tubeside flow, Reynolds number must be evaluated

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}}$$

where $\rho_{sw}, \mu_{sw} = $ salt water density and viscosity are evaluated for the fluid's bulk temperature.
\( d_i, v_{sw} \) = inner diameter and average salt water tube velocity.

Reynolds numbers greater than 2300 will be indicative of turbulent flow [Ref. 3].

b. Salt Water Heat Transfer Coefficient, \( h_{sw} \)

Once again the empirical relationship proposed by Sieder and Tate [Ref. 3] will be used for laminar heat transfer in tubes and as defined by

\[
Nu_d = 1.86 \left( \frac{Re_d \ Pr}{L} \right)^{1/3} \left( \frac{d_i}{L} \right)^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}
\]

Nusselt and Prandtl numbers are defined as

\[
Nu_d = \frac{h_{sw} d_i}{k_{sw}} \quad \text{(149)}
\]

\[
Pr = \frac{C_{p_{sw}} \mu_{sw}}{k_{sw}} \quad \text{(150)}
\]

where dynamic viscosity, specific heat, and thermal conductivity of salt water are evaluated at the salt water bulk temperature.

The effect of the viscosity ratio in the Sieder-Tate equation is considered negligible, and will hereafter be dropped from the expression. The assumptions and validity condition associated with the Sieder-Tate equation were stated in Chapter III, Section 4, and will not be repeated here.
For fully developed turbulent flow, again the Dittus-Boelter correlation [Ref. 3] was used.

\[ N_u = 0.023 \, R_e^{0.8} \, P_r^{0.3} \]

Nusselt and Prandtl numbers are previously defined by Eqs. (149) and (150). Assumptions and conditions for validity were stated in Chapter III, Section 4.

c. Salt Water Fouling Heat Transfer Resistance

As indicated previously, it will be assumed that the fouling resistance for tubeside salt water can be maintained at 0.00025 \((hr \cdot ft \cdot F/\text{BTU})\).

d. Ammonia Shellside Heat Transfer Coefficient, \(h_{NH_3}\)

Initially, \(h_{NH_3}\) will be assumed

\[ h_{NH_3} = 1000 \text{ (BTU/hr \cdot ft}^2\cdot F) \] (151)

since its value cannot be directly calculated during this phase of the analysis.

Using the following single-tube thermal resistance

\[ R_1 = \frac{1}{\eta_i \, h_{sw} \, \pi \, d_i \, L} \]

\[ R_2 = \frac{1}{\eta_i \, h_{fsw} \, \pi \, d_i \, L} \]

\[ R_3 = \frac{\ln d_o/d_i}{2 \, \pi \, K \, L} \]

\[ R_5 = \frac{1}{\eta_o \, h_{NH_3} \, \pi \, d_o \, L} \]
an initial value for single tube conductance (outer tube surface) may be calculated

\[ U_o A_o = \frac{1}{R_1 + R_2 + R_3 + R_5} \]

7. Film Temperature for Property Evaluation, \( T_f \)

In order to evaluate the shellside ammonia heat transfer coefficient, working fluid properties must be evaluated at the film temperature.

This can be accomplished by using the results of the single tube conductance, the tube side bulk temperature and the working fluid saturation temperature, expressed in the following equation for single tube heat transfer rate (average).

\[ \dot{Q} = U_o A_o (T_s - T_{bulk}) \] (152)

Again using the resistance analysis as in Chapter III, the shellside wall temperature may be expressed as

\[ T_{w_2} = T_{bulk} + \dot{Q} (R_1 + R_2 + R_3) \]

Knowing the shellside wall temperature and the free-stream temperature, the film temperature can be derived from their arithmetic mean

\[ T_f = \frac{T_{w_2} + T_s}{2} \] (153)

For purposes of this calculation, saturated temperature conditions at state point 5 are taken to represent
free-stream conditions, when in fact the two-phase process will experience a pressure drop and a corresponding drop in temperature.

8. Revised Shellside Ammonia Heat Transfer Coefficient, $h_{\text{NH3}}$

This analysis will include correlations for both horizontal and vertical heat exchangers.

In the horizontal-tubed condenser, Nusselt's correlation was used as a predictor [Refs. 7 and 17], for laminar flow

$$h = 0.95 \left( \frac{K_f g L}{\mu \nu} \right)^{1/3}$$

where $\nu = \text{estimate of ammonia mass flow rate across each tube.}$

$K_f/\rho_f \mu_f = \text{properties evaluated at film temperature.}$

$L = \text{tube length (initialized and treated as a design variable by the optimization code).}$

This correlation is probably conservative, since it does not consider turbulence due to high vapor velocity or splashing of condensate [Ref. 7].

For turbulent flow, Nusselt's correlation is increased by 10% as recommended by Jakob [Ref. 17]

$$h = 1.045 \left( \frac{K_f g L}{\mu \nu} \right)^{1/3}$$
The laminar-turbulent transition point is defined by a Reynolds number of 2100, where the pseudo-Reynolds number for film-type condensation on horizontal tubes is defined as [Ref. 7]

\[ Re = \frac{2 \Gamma}{\mu t} \]

where \( \Gamma \) = mass flow rate of condensate per tube over its length.

In the vertical tubed condenser, both Nusselt's and Kirkbride's correlations were used as predictors [Ref. 7].

For laminar flow, Nusselt's correlation is increased by a factor of 1.28 as recommended by McAdams [Ref. 7]:

\[ \Gamma_{\text{t}} = 1.28 \left[ 1.47 \left( \frac{\mu t}{K_f \rho_f^2 g} \right)^{-1/3} \left( \frac{4 \Gamma}{\mu t} \right)^{-1/3} \right] \] (156)

where \( \Gamma \) = mass flow rate of condensate per tube over its diameter.

For turbulent flow, Kirkbride's correlation is applied

\[ \Gamma_{\text{t}} = 0.0077 \left( \frac{\mu t^2}{K_f \rho_f^2 g} \right)^{-1/3} \left( \frac{4 \Gamma}{\mu t} \right)^{0.4} \] (157)

The laminar-turbulent transition point is defined by a Reynolds number of 1800, where the pseudo-Reynolds number for film-type condensation on vertical tubes is defined as [Ref. 7]

\[ Re = \frac{4 \Gamma}{\mu t} \]
After using the pseudo-Reynolds number to establish the flow in which regime the system is operating, the revised heat transfer coefficient for film-type condensation may be calculated and then iterated with the initial assumption for the shellside heat transfer coefficient, Eq. (151). Once again this will have a convergence effect on variables in which the shellside heat transfer coefficient is a function, moving closer to actual OTEC system operating point characteristics.

9. Tube Profile, Flow across Tube Bank, and Tube Sheet Diameter

Since the condenser tube bundle involves multiple rows of tubes, the geometry of the tube profile arrangement is important to determine the shellside heat transfer coefficient, the tube sheet diameter and the shellside pressure drop associated with the "homogenous" two-phase flow model [Ref. 4].

Using the same arrangements shown in Chapter III, Section 2,

\[ S_n = \frac{P_R}{d_o} \]  \hspace{1cm} (158)

\[ A_p = S_n^2 \]  \hspace{1cm} (159)
where $S_n = \text{pitch ratio} \times \text{outer tube diameter}$.  

$PR = \text{pitch ratio (initialized and treated as a design variable for the optimization code)}$.  

$A_p = \text{tube profile area per tube}$

\[ S_n = 2 \cdot PR \cdot d_0 \cdot \sin 30' \]  

\[ S_p = PR \cdot d_0 \cdot \cos 30' \]  

\[ A_p = S_n \cdot S_p \]  

the ratio of minimum flow area to the frontal area can be expressed as

\[ \frac{A_{ff}}{A_f} = \frac{S_n - d_0}{S_n} \]  

Using the selected tube profile geometry and knowing the number of condenser tubes by Eq. (143), the tube sheet diameter for the condenser design can be evaluated from the following expression.
where \( T_{SP} \) = Tube sheet diameter.

To analyze the shellside ammonia flow velocity, the following control volume is introduced (turbine generator discharge and top portion of the condenser).

Since the mass flow rate remains unchanged across any boundary

\[
\dot{m}_S = \dot{m}_f
\]

Furthermore, if we assume the condenser has the capability to evenly distribute vapor across the tube bundle (distribution baffles), the following development applies to the vapor coverage:

Let

\[
(A_f)_{VAP} = A_f \gamma
\]

where \( \gamma \) = percent of tube frontal area which is covered by vapor.

\[
\dot{m}_S = \rho_S A_S V_S
\]

\[
\dot{m}_f = \rho_f A_f V_f \gamma
\]
where \( A_5 \) = condenser inlet cross-sectional area.
\( V_5 \) = turbine discharge ammonia velocity.

Therefore

\[
V_f = \frac{\frac{C}{3} A_5}{\rho_f A_f \eta} V_5
\]

If \( \eta = \frac{C}{3} A_5 / A_f \rho_f \), it follows that the turbine discharge velocity is equal to the average velocity of ammonia at the tube frontal area boundary. A determination of the distribution fraction \( \eta \) requires a detailed knowledge of the design of the turbine/condenser interface. In the absence of this information it is assumed that

\[
V_f = V_5
\]

A similar argument could be presented for a vertical tubed condenser where turbine discharge is admitted to a distribution ring that bands the condenser tube bank. Exhaust vapor would travel radially through the tube bundle and then collect at the bottom after vertical film-condensation.

Again, in the absence of a detailed design, it is assumed that

\[
V_{side} = V_5
\]
Considering the minimum free-flow area for a horizontal tubed condenser, $A_{ff}$ can be derived using Eq. (163) and the projected frontal area.

$$A_f = T_{SD} L_t$$

$$A_{ff} = A_f \left( \frac{S_n - d_o}{S_n} \right)$$ (165)

where $A_f =$ the flow frontal area.

$L_t =$ tube length.

For vertical condensers

$$A_f = \pi T_{SD} \times \text{FRONTAL LENGTH OF VAPOR INLET FLOW}$$

Using the previously calculated value of the ammonia flow rate and Eq. (165), mass velocity for the minimum free flow area can be expressed as

$$G = \frac{m_4}{A_{ff}}$$ (166)

10. **Pressure Drop of Two-Phase Flow across a Bank of Tubes, $\Delta P$**

The pressure drop in the two-phase flow condensing heat exchanger will be determined using the homogeneous model introduced in Chapter III. The model will consist of three components -- friction loss, momentum change, and elevation pressure drop arising from the effects of gravity.

The local pressure drop for a two-phase flow may be expressed as
\[ \Delta P_{\text{cond}} = \Delta P_{\text{friction}} + \Delta P_{\text{momentum}} + \Delta P_{\text{elevation}} \]  

For a given channel length, \( L_c \), the pressure drop components can be expressed by

\[ \Delta P_{\text{friction}} = \frac{f G^2 \bar{v}}{D_e 2 g_c} L_c \]  

\[ \Delta P_{\text{momentum}} = \frac{G^2 \bar{v}}{g_c} \]  

\[ \Delta P_{\text{elevation}} = \frac{g}{\bar{v} g_c} L_c \]  

where

- \( f \) = single phase friction factor by Jakob expressed in Eq. (35) or (36).
- \( G \) = mass flow velocity determined from Eq. (166).
- \( L_c \) = channel flow length, defined for horizontal tubed condensers as \( L_c = T_{SD} \) (tube sheet diameter) and for vertical tubed condensers as \( L_c = L_t \) (tube length).
- \( D_e \) = equivalent diameter of flow channel, defined by \( D_e = Pr \, d_o - d_o \).
- \( \bar{v} \) = mean specific volume defined by

\[ \bar{v} = v_t \left[ 1 + \frac{x}{v_f} (v_g - v_t) \right] \]
where \( X \) = quality of mixture (state point 5).

\[ \mathcal{V}_f = \text{specific volume of liquid (state point 1)} \]

\[ \mathcal{V}_d = \text{specific volume of vapor (state point 5)}. \]

All components of the pressure drop model Eqs. (168, 169, and 170) can be determined using the preceding information.

**11. Revised Properties at State Point 1**

Since Eq. (167) represents the pressure drop across the condenser shellside, the actual pressure at state point 1 or condenser outlet may be determined from

\[ P_{1(NEW)} = P_1 - \Delta P_{COND} \]  \hspace{1cm} (171)

where \( P_1 \) is previously described as the condenser operating pressure for the ideal cycle.

Operating on the saturated liquid line on the Temperature-Entropy diagram, the following properties are defined:

\[ h_{1(NEW)} = h_f(P_{1(NEW)}), \quad T_{1(NEW)} = T_{SAT(P_{1(NEW)})} \]  \hspace{1cm} (172)

The subscript \( (NEW) \) representing a revised property will hereafter be dropped from the expression in Eq. (172).

Until now, we assumed the condenser outlet temperature and pressure were designed to operate as an ideal system, without a pressure drop. Therefore, using the revised temperature at state point 1 and iterating over the range from Eq. (21) until an acceptable convergence criterion is achieved, all the preceding variables as function of \( T_1 \) would be determined.
will be reevaluated to complete the closed-loop cycle of the simulated OTEC power system.

12. Overall Heat Transfer Coefficient, \( \dot{U}_o \)

The quantity "\( U \)" represents a measure of the total thermal resistances in the flow path. Therefore, using the tube conductance expressed in Eq. (148) which is divided by the outer heat transfer surface area of a single tube, the overall heat transfer coefficient for the condenser can be determined.

The thermal resistances are now expressed as

\[
R_1 = \frac{d_i}{\eta_i h_{sw} d_i}
\]
\[
R_2 = \frac{d_o}{\eta_i h_{fsr} d_i}
\]
\[
R_3 = \frac{d_o \ln d_o/d_i}{2k}
\]
\[
R_5 = \frac{1}{\eta_o h_{NH_3}}
\]

and the overall heat transfer coefficient for the condenser may be calculated using

\[
\dot{U}_o = \frac{1}{R_1 + R_2 + R_3 + R_5}
\]  

(173)
13. Total Condenser Heat Transfer Surface Area, $A_t$

Having determined the corrected number of condenser transfer units (145), salt water capacity rate (140) and overall heat transfer rate (173), the total condenser heat transfer area can be calculated from the $NTU$ expression

$$NTU = \frac{U_0 A_t}{C_{min}}$$  \hspace{1cm} (174)

14. Revised Condenser Tube Length

Using the total heat transfer surface area calculated from Eq. (174) and the total number of condenser tubes (143), the revised condenser tube length can be determined from the basic expression

$$A_t = N_t \pi d_o L_t(Revised)$$  \hspace{1cm} (175)

At this time, it is necessary to iterate the condenser design until the two values (initial and revised) of the tube length converge. This iteration may be accomplished by the COPES routine if the following constraint is defined

$$L_{Diff} = L_t(Revised) - L_t(Initial)$$

Minimization of this difference will cause continual adjustment of the required tube length, already treated as a design variable by the optimization code.
15. **Condenser Heat Exchanger Cost Analysis**

As indicated in Chapter III, TRW developed sets of equations to represent the costs of various heat exchanger component parts for shaft diameters ranging from 10-35 feet and 35-50 feet [Ref. 9].

The following are the TRW component cost equations for the condensing heat exchanger. Prior equation reference numbers will be substituted where equalities exist with the evaporative heat exchanger component cost expressions.

for tube sheet diameter 10-35 feet

- Drilling time/tube sheet thickness. \((58)\)
- Thickness of the tube sheet. \((59)\)
- Tube sheet labor cost. \((60)\)
- Tube sheet material cost. \((61)\)
- Tube installation cost. \((62)\)
- Heat exchanger drill cost. \((63)\)
- Ammonia distribution plate and baffles cost.
  \[
  C_{DPB} = 1534 x 10^{-5} t_{td} N_t T_{so}^{2.6} \tag{176}
  \]
- Bustle, flanges, channels and flow plate cost.
  \[
  C_{BFCF} = 1125 x 10^{-6} T_{so}^{2.6} \tag{177}
  \]
- Tube material cost.
  \[
  C_{TM} = (C1 L_e + C2) N_t d_e/15 \tag{178}
  \]
where $C_1$ = curve fit of tube material cost per foot.

$C_2$ = tube machining cost if required.

. Heat exchanger header cost. (67)

. Water inlet, nozzles and support cost.

\[
C_{wins} = 10106.475 T_{50}
\]  

(179)

. Tube welding costs (Titanium tubes). (69)

The sum of the preceding costs would equal the cost to fabricate one OTEC condenser with a tube sheet diameter of 10-35 feet (all the preceding component costs have been adjusted for current pricing at a 10% annual rate of inflation).

If our analysis is based on a 30-year life-cycle criterion, no adjustments are necessary to any component cost equation if titanium tubing is selected. However, using Al 5052-0, the expense of retubing must be considered to meet the 30-year life-cycle criterion, as in the case of the evaporation. For convenience, and possible subsequent modification, these considerations are repeated here.

Based upon the utility of Al 5052-0, two complete condenser retubings will be required to meet the basic 30-year criterion. This implies Eqs. (62) and (178) must be modified to reflect the costs of retubing at the 10 and 20 year point in the cycle.
Aluminum tube installation cost.

\[ C_{ATI} = C_{TE} \left[ 1 + (1+i)^{20} \right] \tag{180} \]

where \( i \) = projected annual inflationary rate (input by customer).

Aluminum tube material cost.

\[ C_{ATM} = C_{TM} \left[ 1 + (1+i)^{10} + (1+i)^{20} \right] \tag{181} \]

for tube sheet diameter 35-50 feet.

Drilling time/tube sheet thickness \( t_d \) \tag{58}

Thickness of the tube sheet. \( t \) \tag{59}

Tube sheet labor and material costs (titanium). \( (72, 73) \)

Tube sheet labor and material costs (aluminum). \( (74, 75) \)

Tube installation cost. \( (76) \)

Tube material cost. \( (178) \)

Heat exchanger shell cost. \( (77) \)

Ammonia distribution plate and baffles cost.

\[ C_{DPB} = 9.825 N_t^{0.475} t_d \tag{182} \]

Bustle, flanges, channels and flow plate.

\[ C_{BFCF} = 382.624 T_{SP}^{2.184} \tag{183} \]

Heat exchange head cost.

\[ C_{HXM} = 939.62 T_{SP}^{1.43} \tag{184} \]
Water inlet, nozzles, and supporters cost.

\[ C_{WINS} = 7453.6 T_{SD}^{1.056} \]  

(185)

Tube welding cost (titanium tubes).  

(82)

As indicated previously, the cost to fabricate one OTEC condenser with a tube sheet diameter of 35 to 50 feet is equal to the sum of component costs (note, all the preceding component costs have been adjusted for current pricing at a 10% annual inflation rate).

For an analysis based on a 30-year life-cycle criterion, the additional costs for replacing aluminum tubing must be considered and Eqs. (180) and (181) apply.
VII. NUMERICAL OPTIMIZATION

A. INTRODUCTION

Nearly all design processes attempt the minimization or maximization of some parameter or design objective. For the design to be acceptable, it must satisfy a set of constraints which impose limits or bounds on design parameters.

For the stated problem a computer program can be written to perform the basic analysis of the proposed design. If any parameters fall outside the prescribed bounds, the design engineer changes the parameters and re-runs the program. In effect, the computer code provides the analysis with the engineering making the actual design decisions.

A logical extension to the computer-aided approach is a fully automated design, where the computer also makes the actual design decisions and performs trade-off studies. The COPES program provides this automated design and trade-off capability by the use of the optimization program COPES/CONMIN [Ref. 18]. COPES is an acronym for Control Program for Engineering Synthesis, and CONMIN is an acronym for CONstrained function MINimization. Subsequently, a FORTRAN analysis program simulating a closed-cycle OTEC power system can be coupled to the COPES program for automated design, using some basic programming guidelines [Ref. 18].
B. COPES/CONMIN

There are many numerical optimization schemes available to the engineer. Methods employed by these schemes fall into four basic categories: random search, sequential unconstrained minimization, optimality criteria, and direct constrained optimization. The optimization program, selected for automated design analysis of the simulated OTEC power system, is based upon direct constrained optimization.

Before any discussion of the optimization technique, basic definitions are summarized for convenient reference [Ref. 19]:

- **Design variables** - those parameters which the optimization program is permitted to change in order to improve the program.
- **Objective function** - the parameter which is to be minimized or maximized during the optimization process.
- **Inequality constraint** - one-sided conditions which must be satisfied for an acceptable design.
- **Equality constraint** - condition which must be equaled for the design to be acceptable.
Side constraints - upper and lower bounds in a design variable.

Assuming that the FORTRAN analysis program has been developed and a particular objective function has been selected, the general optimization problem can be stated as [Ref. 20]:

Find the vector of design variables, \( \bar{X} \), to

Minimize \( F(\bar{X}) \) \hspace{1cm} (186)

Subject to the constraints:

\[
G_j(\bar{X}) \leq 0 \hspace{1cm} j = 1, N\text{CON} \hspace{1cm} (187)
\]

\[
H_j(\bar{X}) = 0 \hspace{1cm} j = 1, \text{NEQ} \hspace{1cm} (188)
\]

\[
\text{VL}_{B_i} \leq \bar{X}_i \leq \text{VU}_{B_i} \hspace{1cm} i = 1, \text{NDV} \hspace{1cm} (189)
\]

where \( \bar{X} \) = the vector containing the set of independent design variables.

\( F(\bar{X}) \) = the objective function to be minimized.

\( G_j(\bar{X}) \) = inequality constraint (\( N\text{CON} \) is the number of such constraints).

\( H_j(\bar{X}) \) = equality constraint (\( \text{NEQ} \) is the number of such constraints).

\( \text{VL}_{B_i}/\text{VU}_{B_i} \) = lower and upper bounds, respectively, on the design variables.

If all inequalities of Eqs. (187) and (189) are satisfied, the design is said to be feasible if any constraint is not satisfied, the design is infeasible. If the objective function
is a minimum and the design is feasible, it is said to be the optimal design.

In order to start the optimization algorithm, the initial set of design variables, $\mathbf{X}$, must be specified. It is desirable, but not essential, that the initial design variables provide a feasible solution. The optimization algorithm will then proceed in an iterative fashion using the following relationship

$$
\mathbf{X}^{q+1} = \mathbf{X}^q - \alpha \cdot \mathbf{S} \cdot q
$$

where $q$ = the iteration number.

$\alpha$ = scalar quantity which defines the move in the search direction.

$\mathbf{S}$ = vector search direction which will reduce the objective function (useable direction) without violating constraints (feasible direction).

To solve this problem, the optimization program COPES/CONMIN is used [Ref. 18]. CONMIN uses the Fletcher-Reeves algorithm for locally unconstrained problems [Ref. 20] and Zoutendijk's method of feasible directions (modified to improve efficiency and reliability and to deal with designs which do not initially satisfy all the constraints) for locally constrained problems [Ref. 21].

However, CONMIN does not handle equality constraints directly, but rather by means of penalty parameters. To achieve this, the objective function is augmented as follows
[Ref. 19]:

\[ F'(\bar{x}) = F(\bar{x}) - K \sum_{j=1}^{n=1} H_j \]  \hspace{1cm} (190)

and the equality condition of Eq. (188) is treated as an inequality constraint

\[ H_j(\bar{x}) \leq 0 \quad j = 1, N \in G \]

The penalty function approach effectively satisfies the equality constraint while maintaining the rapid convergence characteristics of the CONMIN program.

The numerical optimization problems of equations (186) through (190) are very general, allowing for any number of design variables and constraints. In assessing the value of optimization, the automated design provides a very attractive approach to numerical optimization; however, there are both advantages and limitations to these techniques [Ref. 20].

Advantages:

. Reduction in design time.
. Systematic design procedure.
. Applicable to a wide variety of design variables and constraints.
. Virtually always yields some design improvement.
. Not biased by engineering experience.
. Requires a minimal amount of man-machine interface.

Limitations:

. Computer times may increase dramatically as the number of design variables increases. A practical limit imposed by the current state of the art for most problems is 30 design variables.
Optimization techniques have no stored experience to draw upon; the validity of the result is limited to the validity of the analysis program.

The results of the optimization are as correct as the analysis program is theoretically precise.

Optimization algorithms used here cannot deal with discontinuous functions.

The optimization program will not always obtain a global design optimum and may require restarting from several different points to acquire reasonable assurance of obtaining the global optimum.

The analysis program must be properly structured to couple with the COPES/CONMIN optimization code.

C. DESIGNATED DESIGN VARIABLES, CONSTRAINTS AND OBJECTIVE FUNCTION

To assist in the interpretation of the enclosed OTEC power system FORTRAN analysis, the following summary identifies the design variables, constraint functions and objective function used in the analysis and subsequently operated upon by the COPES/CONMIN optimization code. These parameters are all contained in a labeled COMMON block in the computer code, referred to here as "GLOBAL COMMON." Specific GLOBAL COMMON location numbers and upper/lower bounds for operating parameters summarized below can be located in Appendix C.

Design Variables

- Inner cold pipe diameter
- Inner hot pipe diameter
- Inner ammonia circ pipe diameter
- Inner ammonia re-flux pipe diameter
Evaporator operating pressure
Condenser operating pressure
Outer condenser tube diameter
Outer evaporator tube diameter
Evaporator tube length
Condenser tube length
Condenser tube salt water velocity
Cold pipe salt water velocity
Evaporator tube salt water velocity
Hot pipe salt water velocity
Evaporator tube profile pitch ratio
Condenser tube profile pitch ratio

Constraint Functions
Operating system pressure ratio
Upper temperature bound of ammonia
Lower temperature bound of ammonia
Satisfactory enthalpy at state point 5
Satisfactory quality at state point 5
Satisfactory condenser tube length
Internal turbine efficiency
Evaporator tube sheet diameter
Condenser tube sheet diameter

Objective Function
Cost of major power system components
VIII. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

1. The use of an analysis code for OTEC power systems coupled to COPES/CONMIN optimization code provides a powerful tool to design an optimum power system for the desired net electrical output, measured against the objective function. Such a design could permit construction of higher capacity systems using the optimized modules as substations of the total power plant.

2. The analysis code coupled to COPES/CONMIN provides an excellent vehicle to evaluate proposed designs relative to a true optimum. Tables 1 through 4 illustrate the result of preliminary calculations using the analysis code with an objective function to minimize system cost. From these, the following conclusions can be drawn concerning horizontally oriented aluminum (Al-5052) and titanium-tubed heat exchanger power systems:

   a. The cost/KW output is nearly constant over the range of optimum designs for both titanium and aluminum tube heat exchangers.

   b. During testing for feasible plant designs in increments of 5 MW (net) electrical output, it was observed that a higher megawatt output plant could be achieved with titanium-tubed heat exchangers than for aluminum (Al-5052). For titanium-tubed heat exchangers, a 25 MW (net) power
system is a feasible design; however, aluminum-tubed systems could not provide a feasible design for the same output. Titanium tubed plants failed to produce a feasible design for a 30 MW (net) output power system. In both cases of infeasible design, the constraint which was consistently violated was turbine internal efficiency, set at 90% for current state-of-the-art design.

c. The energy conversion and efficiency of design of a turbine-generator has a major effect on the overall system performance as indicated in paragraph b above.

d. The cost/KW output for titanium-tubed heat exchangers is one third the cost/KW output for aluminum-tubed heat exchangers using a 30-year life-cycle criterion, with a 10% annual inflation rate and retubing at 10 and 20 year marks with AL-5052 tubing.

e. Aluminum-tubed heat exchangers have larger tube bundle volumes, with volumetric differences between aluminum and titanium varying from 26.1 to 11.8% for evaporators and 23.2 to 7.4% for condensers over the range of net power levels considered. In both cases volumetric differences diminish as the system's net electrical output increases to 20 megawatts.

f. COPES/CONMIN has provided optimum designs for each incremental output power level. By manipulating the specified design variables, subject to imposed constraints, COPES/CONMIN has created designs whose geometry and operating
parameters cannot be scaled on the basis of net power output (10 MW). Therefore, designs for component geometry at increasing power levels based upon such simplistic scaling criteria will not achieve an optimum design with respect to the cost objective function.

B. RECOMMENDATIONS

1. Evaluate additional objective functions including:
   a. Minimize heat exchanger volumes.
   b. Minimize parasitic power losses.
   c. Maximize thermodynamic efficiency.
   d. Maximize net electrical output.

2. Perform a sensitivity analysis on power system design variables to evaluate their influence on component and system performance. This allows the designer to prioritize system components which can provide improvement in the objective function for a corresponding improvement in component design.

3. Considerable uncertainties are associated with the expressions used to estimate component performances (two-phase pressure drops, film coefficients, etc.). The code should be tested to determine the sensitivity of system design to these uncertainties.

4. Expand the code to include the use of enhanced heat transfer techniques and evaluate the influence of increased piping friction factors on pumping power requirements.
5. Evaluate proposed OTEC designs using proposed system parameter inputs, comparing both the basic analysis and the optimization output.

6. Select other analytical expressions for heat transfer coefficients to validate the performance and output of the existing code.

7. Evaluate the effect of a smaller thermal difference seen by the power system and its influence on a feasible design for a specific net electrical output.

8. Evaluate the cost aspects of using variable-pitch pumps versus fixed-blade for a variable thermal gradient environment.

9. Evaluate and verify the influence of incremental improvements (percent) in turbine internal/adiabatic efficiency with respect to gross and net electrical outputs and compare with the results reported in Ref. 16.
<table>
<thead>
<tr>
<th></th>
<th>10 MW</th>
<th>15 MW</th>
<th>20 MW</th>
<th>25 MW</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>HT ABSORB (BTU/HR)</strong></td>
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<td><strong>SW FLOW (LBₘ/HR)</strong></td>
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<td>3.34 E8</td>
<td>4.14 E8</td>
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<td><strong>NH₃ FLOW (LBₘ/HR)</strong></td>
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<td><strong>OPER PRESS (LBₜ/IN²)</strong></td>
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<td><strong>OVL HT COEF (BTU/HR•FT²•F)</strong></td>
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<td><strong>PITCH RATIO</strong></td>
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TABLE 1. OTEC Power System Comparisons (Continued)

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<tr>
<th></th>
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<th>15 MW</th>
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<td><strong>CONDENSER</strong></td>
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<td>HT REJECT (BTU/HR)</td>
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<td>OPER PRESS (LBf/IN²)</td>
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<td>PITCH RATIO</td>
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<td>TUBE LENGTH (FT)</td>
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<td>TOT NR OF TUBES</td>
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<td>52,179</td>
<td>79,956</td>
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### TABLE 1: OTEC Power System Comparisons (Continued)

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<th>PIPING SYSTEMS</th>
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<th>15 MW</th>
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<tr>
<td><strong>SW HOT PIPE (300 FT LENGTH)</strong></td>
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<td></td>
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<td>INNER DIA (FT)</td>
<td>17.20</td>
<td>20.08</td>
<td>21.86</td>
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<td>SW VEL (FT/SEC)</td>
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<td>PRESS DROP (LB/f/IN²)</td>
<td>0.280</td>
<td>0.322</td>
<td>0.348</td>
<td>0.371</td>
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<td><strong>SW COLD PIPE (3000 FT LENGTH)</strong></td>
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<td>INNER DIA (FT)</td>
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<td>18.62</td>
<td>21.35</td>
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<td>SW VEL (FT/SEC)</td>
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<td>5.33</td>
<td>5.72</td>
<td>5.95</td>
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<tr>
<td>PRESS DROP (LB/f/IN²)</td>
<td>0.49</td>
<td>0.508</td>
<td>0.526</td>
<td>0.539</td>
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<td><strong>NH₃ CIRC PIPE (150 FT LENGTH)</strong></td>
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<td>PRESS DROP (LB/f/IN²)</td>
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<tr>
<td>PRESS DROP (LB/f/IN²)</td>
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<td>9.87</td>
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<td>12.53</td>
</tr>
<tr>
<td>PUMP SYSTEMS</td>
<td>10 MW</td>
<td>15 MW</td>
<td>20 MW</td>
<td>25 MW</td>
</tr>
<tr>
<td>--------------</td>
<td>-------</td>
<td>-------</td>
<td>-------</td>
<td>-------</td>
</tr>
<tr>
<td><strong>EVAP SW PUMP (EFFICIENCY 85 PCT)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>HEAD (FT)</td>
<td>11.0</td>
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<td><strong>COND SW PUMP (EFFICIENCY 85 PCT)</strong></td>
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<td>HEAD (FT)</td>
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<td><strong>NH₃ CIRC PUMP (EFFICIENCY 75 PCT)</strong></td>
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### TABLE 1: OTEC Power System Comparisons (Continued)

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<td>Turbine-Generator (Turb Mech 99.8 Pct, Gen Mech and Elect 96.6 Pct)</td>
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<tr>
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<td>89.88</td>
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<td>89.80</td>
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<td>Outlet Quality (Pct)</td>
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<td>96.77</td>
<td>96.9</td>
<td>96.9</td>
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<td>Power Requirements (Megawatts)</td>
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<td>Turb Effic Losses</td>
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<td>Evap SW Pump</td>
<td>1.131</td>
<td>1.496</td>
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<td>Cond SW Pump</td>
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<td>Turb-Gen Gross</td>
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<td>Pct Parasitic Power</td>
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<td>COMPONENT Costs ($)</td>
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<td>15 MW</td>
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<td>25 MW</td>
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<tr>
<td>---------------------</td>
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<td>------------</td>
<td>------------</td>
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<tr>
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<td>NH3 FLOW (LBm/HR)</td>
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<td>OPER PRESS (LBf/IN²)</td>
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### TABLE 2: OTEC Power System Comparisons (Continued)

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**TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE**

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TABLE 2: OTEC Power System Comparisons (Continued)

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<td><strong>EVAP SW PUMP (EFFICIENCY 85 PCT)</strong></td>
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<td>CAPACITY (GAL/MIN)</td>
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TABLE 2: OTEC Power System Comparisons (Continued)

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<td>89.76</td>
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<td>96.93</td>
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<td>POWER REQUIREMENTS (MEGAWATTS)</td>
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<td>TURB EFFIC LOSSES</td>
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<td>2.70</td>
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### Table 2: OTEC Power System Comparisons (Continued)

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<th>COMPONENT COSTS ($)</th>
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<td>1,576,449</td>
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<td>GEN-TURBINE</td>
<td>758,998</td>
<td>932,173</td>
<td>1,122,921</td>
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<tr>
<td>EVAP SW PUMP</td>
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<td>667,809</td>
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<td>NI\textsubscript{3} CIRC PUMP</td>
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<td>OPTIMUM COST ($)</td>
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COST/ KW (NET) OUTPUT ($/KW)

|                  | 4885.25 | 4668.00 | 5207.47 |

139
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<th>Component</th>
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<th>25 MW</th>
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<td>HIT ABSORB (BTU/HR)</td>
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<td>SW FLOW (LB\textsubscript{m}/HR)</td>
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<td>OPER PRESS (LB\textsubscript{f}/IN\textsuperscript{2})</td>
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<td>NH\textsubscript{3} PRESS DROP (LB\textsubscript{f}/IN\textsuperscript{2})</td>
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**Tube Characteristics**

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<th>Value</th>
<th>Value</th>
<th>Value</th>
<th>Value</th>
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<td>TABLE 3: Heat Exchanger Comparisons (Continued)</td>
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<tr>
<td>EVAPORATOR</td>
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<td>10 MW</td>
<td>15 MW</td>
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<td>25 MW</td>
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<td>TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE</td>
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<td>PITCH RATIO</td>
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TABLE 3: Heat Exchanger Comparisons (Continued)

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### TABLE 4: Heat Exchanger Comparisons (Continued)

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<td>SW VEL (FT/SEC)</td>
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|           |       |       |       |       |
| **MOISTURE SEPARATOR** |       |       |       |       |
| OPER PRESS (LB/IN²) | 129.5 | 131.89 | 127.88 |       |
| OUTLET TEMP (DEG F) | 70.22 | 91.23 | 69.38 |       |
| OUTLET QUALITY (PCI) | 99.5 | 99.5 | 99.5 |       |
| NH₃ PRESS DROP (LB/IN²) | 0.24 | 0.307 | 0.522 |       |

|           |       |       |       |       |
| **CONDENSER** |       |       |       |       |
| HT REJECT (BTU/HR) | 1.47 E09 | 1.89 E09 | 2.71 E09 |       |
TABLE 4: Heat Exchanger Comparisons (Continued)

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TUBE CHARACTERISTICS

| OUTER DIA (IN) | 1.176 | 1.092 | 1.001 |
| WALL THICK (IN) | 0.065 | 0.065 | 0.065 |
| LENGTH (FT) | 64.04 | 57.46 | 57.24 |

TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE
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<td>EFFECTIVENESS</td>
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<td>h (FOULING)</td>
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<td>h (METAL)</td>
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<td>h (AMMONIA)</td>
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<td>SW PRESS DROP (LB_f/IN^2)</td>
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APPENDIX A

SAMPLE INPUT DATA FOR OTEC ANALYSIS

EVAPORATOR - HORIZONTAL

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<td>PITCH RATIO</td>
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CONDENSER - HORIZONTAL

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SALT WATER HOT PIPE

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SALT WATER COLD PIPE

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AMMONIA CIAC PIPE
PIPE I.D.  2.000(FT)  0.610(M)
PIPE LENGTH  15.000(FT)  4.570(M)
AMMONIA RE-FLUX PIPE
PIPE I.D.  2.000(FT)  0.610(M)
PIPE LENGTH  50.000(FT)  15.240(M)
PUMP AND GEN-TURB PERFORMANCE
EVAP SW PUMP
EFFICIENCY MECH  95.00(PCT)  MOTOR  95.00(PCT)
COND SW PUMP
EFFICIENCY MECH  95.00(PCT)  MOTOR  95.00(PCT)
AMMONIA CIAC PUMP
EFFICIENCY MECH  95.00(PCT)  MOTOR  95.00(PCT)
GEN-TURB EFFICIENCIES
GEN MECH & ELECT  95.00(PCT)
TURB MECH  99.80(PCT)
POWER REQUIREMENTS
NET POWER OUTPUT  15.000(MW)
### APPENDIX B

**SAMPLE OTEC ANALYSIS OPTIMIZATION OUTPUT DATA**

#### EVAPCRATCR - HORIZONTAL

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Heat Absorb</td>
<td>2205013700.0 (BTU/HR)</td>
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<tr>
<td>SW Flow</td>
<td>334081290.0 (LBM/HR)</td>
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<tr>
<td>SW Temp In</td>
<td>80.0000 (DEG F)</td>
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</tr>
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<td>SW Temp Out</td>
<td>73.724 (DEG F)</td>
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<tr>
<td>NH3 Flow</td>
<td>3789703.0 (LBM/HR)</td>
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<tr>
<td>OPER Pressure</td>
<td>130.037 (LBF/IN2)</td>
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<tr>
<td>EVAP SAT Temp</td>
<td>70.535 (DEG F)</td>
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<tr>
<td>OUTLET Temp</td>
<td>70.514 (DEG F)</td>
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<tr>
<td>OUTLET QUALITY</td>
<td>0.20 (PCT)</td>
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<tr>
<td>NH3 PRESS DROP</td>
<td>0.162 (LBF/IN2)</td>
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#### TUBE CHARACTERISTICS

<table>
<thead>
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<tr>
<td>Outer Dia</td>
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</tr>
<tr>
<td>WALL THICK</td>
<td>0.0251 (IN)</td>
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<tr>
<td>LENGTH</td>
<td>42.132 (FT)</td>
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#### MATERIAL - TITANIUM

#### TUBE PROFILE - STAGGERED EQUI-LATERAL

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<tr>
<td>SW VELOCITY</td>
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<tr>
<td>Film Temp</td>
<td>70.990 (DEG F)</td>
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<tr>
<td>Delta T Builing</td>
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<tr>
<td>LMT.D.</td>
<td>5.757 (DEG F)</td>
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#### EVAP EFFECTIVENESS

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<td>Evap effectiveness</td>
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#### NR OF TRANSFER UNITS

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<th>Unit</th>
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<tr>
<td>UVL HT CUEF</td>
<td>612.97 (BTU/HR, FT2,F)</td>
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<tr>
<td>H(WATER)</td>
<td>1115.97 (BTU/HR, FT2,F)</td>
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<tr>
<td>H(FOULING)</td>
<td>373.34 (BTU/HR, FT2,F)</td>
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<tr>
<td>H(METAL)</td>
<td>439.72 (BTU/HR, FT2,F)</td>
<td></td>
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<tr>
<td>H(AMMONIA)</td>
<td>4080.44 (BTU/HR, FT2,F)</td>
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<tr>
<td>Description</td>
<td>Value</td>
<td></td>
</tr>
<tr>
<td>---------------------------------</td>
<td>------------------------</td>
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<tr>
<td>HT SURFACE</td>
<td>572+0.35(J0(FT2)</td>
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<tr>
<td>TUBE SHEET JIA</td>
<td>27.213(FT)</td>
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<td>SW PRESS DROP</td>
<td>4.06L(LBF/IN2)</td>
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<td>MOISTURE SEPARATOR-INSIDE EVAP SHELL</td>
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<td>129.935(LBF/IN2)</td>
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<tr>
<td>OUTLET TEMP</td>
<td>70.333(DEC F)</td>
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<td>OUTLET QUALITY</td>
<td>39.50(PCT)</td>
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<tr>
<td>NH3 PRESS DROP</td>
<td>3.416(LBF/IN2)</td>
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**CONDENSER - NOILJNTAL**

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<tr>
<td>HT REJECT</td>
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<td>SW FLOW</td>
<td>33481552.0(JL/HR)</td>
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<td>SW TEMP IN</td>
<td>40.000(DEC F)</td>
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<td>SW TEMP OUT</td>
<td>46.055(DEC F)</td>
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<td>NH3 FLOW</td>
<td>3738708.0(LB/HR)</td>
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<tr>
<td>OPER PRESSURE</td>
<td>0.151(LBF/IN2)</td>
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<tr>
<td>COND SAT TEMP</td>
<td>49.351(DEC F)</td>
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<td>OUTLET TEMP</td>
<td>49.238(DEC F)</td>
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<td>NH3 PRESS DROP</td>
<td>0.206(LBF/IN2)</td>
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**TUBE CHARACTERISTICS**

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<tr>
<td>OUTER DIA</td>
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<td>WALL THICK</td>
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<td>MATERIAL - TITANIUM</td>
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<td>TUBE PROFILE - STAGGERED EQUILATERAL</td>
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<tr>
<td>PITCH RATIO</td>
<td>1.40</td>
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<td>ENHANCEMENT - PLAIN TUBE</td>
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<tr>
<td>SW VELOCITY</td>
<td>6.017(FT/S)</td>
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<td>T WALL(SHELLSIDE)</td>
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<td>FILM TEMP</td>
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<td>DELTA T COND</td>
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153
<table>
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<tr>
<td>CVL HT COEF</td>
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<tr>
<td>H(WATER)</td>
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<tr>
<td>H(FOULING)</td>
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<tr>
<td>H(METAL)</td>
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<td>H(AMMONIA)</td>
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<td>HT SURFACE</td>
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<td>TUBE SHEET DIA</td>
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<td>PIPE I.D.</td>
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<tr>
<td>SW PIPE VEL</td>
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<tr>
<td>SW FLOW</td>
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<tr>
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<td>SW SALINITY</td>
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<td>SW FLOW</td>
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<td>NH3 FLOW</td>
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<td>NH3 PRESS DROP</td>
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<td>NH3 PRESS DROP</td>
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### PUMP AND GEN-TURB PERFORMANCE

#### EVAP SW PUMP
- **Head Pressure**: 3.948 (ft) 3.017 (m)
- **Capacity**: 93259.3 (gallons/min) 247207.3 (liters/min)
- **Efficiency Mech**: 32.00 (PCT) 80.00 (PCT)

#### COND SW PUMP
- **Head Pressure**: 20.753 (ft) 7.227 (m)
- **Capacity**: 63219.2 (gallons/min) 240327.1 (liters/min)
- **Efficiency Mech**: 32.00 (PCT) 80.00 (PCT)

#### AMMONIA CIRC PUMP
- **Head Pressure**: 212.327 (ft) 64.717 (m)
- **Capacity**: 121.117 (gallons/min) 45305.3 (liters/min)
- **Efficiency Mech**: 75.00 (PCT) 98.00 (PCT)

#### AMMONIA RE-FLUX PUMP
- **Head Pressure**: 33.016 (ft) 11.397 (m)
- **Capacity**: 3732.4 (gallons/min) 1127.1 (liters/min)
- **Efficiency Mech**: 75.00 (PCT) 98.00 (PCT)

#### GEN-TURB EFFICIENCIES
- **Gen Mech Elect**: 36.6 (PCT)
- **Turb Mech**: 99.30 (PCT)
- **Turb Internal**: 39.83 (PCT)
- **Turb Outlet Quality**: 36.77 (PCT)

#### POWER REQUIREMENTS
- **Turb-Gens Gross**: 27603.313 (hp) 20.033 (mw)
- **Efficiency Losses**: 0.559 (mw)
- **Evap SW Pump**: 1964.831 (hp) 1.476 (mw)
- **Cond SW Pump**: 4129.313 (hp) 3.143 (mw)
- **NH3 Circ Pump**: 541.714 (hp) 0.412 (mw)
- **NH3 Re-Flux Pump**: 29.047 (hp) 0.022 (mw)

**Net Power Output**: 15.030 (mw)

#### PERCENT PARASITIC POWER
- **24.59 (PCT)**

#### THERMODYNAMIC CYCLE EFFICIENCY
- **2.65 (PCT)**

---

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COST OF COMPONENTS

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<th>Component</th>
<th>Cost (Dollars)</th>
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<td>Condenser</td>
<td>3367154.00</td>
</tr>
<tr>
<td>Gen-Turbine</td>
<td>1578776.00</td>
</tr>
<tr>
<td>Generator</td>
<td>737205.06</td>
</tr>
<tr>
<td>Evap SW Pump</td>
<td>653332.94</td>
</tr>
<tr>
<td>Cond SW Pump</td>
<td>552298.06</td>
</tr>
<tr>
<td>NH3 Circ Pump</td>
<td>136824.94</td>
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<tr>
<td>NH3 Re-Flux Pump</td>
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**Optimum Cost**

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<th>Cost Per Net KW Output</th>
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<td>2039219.00</td>
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<td>1389.95</td>
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### APPENDIX C

SAMPLE COPES OPTIMIZATION AND SENSITIVITY ANALYSIS DATA

```plaintext
$BLOCK A (TITLE CARD)
OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM

$BLOCK 3 (PROGRAM CONTROL PARAMETERS)
2,16,16

$BLOCK C (INTEGER OPT CONTROL PARAMETERS)
5,3,0,5

$BLOCK D (FLOATING PT OPT PROC PARAMETERS)
3.0

$BLOCK E (TOT NR DESIGN VAR, DESIGN OBJ IDENT AND SIGN)
16,27,-1.0

$BLOCK F (DESIGN VARIABLE BOUNDS, INIT VALUES & SCALE FACTOR)
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1.0,1.0+2.0
1.0,1.0+2.0
1.0,1.0+2.0
1.0,1.0+2.0
1.0,1.0+2.0
1.0,1.0+2.0
1.0,1.0+2.0
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157
$\text{BLOCK H (NR OF CONSTRAINED PARAMETERS)}$

$\text{BLOCK I (CONSTRAINT IDENT AND BOUNDS)}$

$\text{BLOCK P (SENSITIVITY OBJECTIVES)}$

$\text{BLOCK Q (SENSITIVITY VARIABLE BOUNDS)}$
THESIS PRESENTATION: MODEL ANALYSIS OF THE CLOSED-CYCLE OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM
BY LCDR RAYMOND C. Schaubel, USN

NOMENCLATURE

AC(TDOC) - SUBROUTINE TO COST COND AL TUBING/FT($/FT)
AC(TDOC) - SUBROUTINE TO COST EVAP AL TUBING/FT($/FT)
AC(TDOC) - SUBROUTINE TO COST ENHANCED COND AL TUBE($/FT)
AAE - SUBROUTINE TO SIZE ENHANCED EVAP AL TUBE THICK(IN)
AAE(TDOC) - SUBROUTINE TO SIZE ENHANCED COND TUBING(IN)
AEE(TDOC) - SUBROUTINE TO COST ENHANCED EVAP AL TUBE($/FT)
AF - EVAP TUBE BANK MINIMUM FREE-FLOW AREA(FT2)
AG - PROJECTED ANNUAL INFLATION RATE (DECIMAL)
AO - OUTER SURFACE AREA OF A SINGLE EVAP TUBE(FT2)
AP(TDOC) - SUBROUTINE TO SIZE COND AL TUBE WALL THICK(IN)
AR - VERT. TUBED COND INLET DISTRIBUTION BAND
BX - 778.2 FT.LBF/BTU
CE - CONDENSER TUBE MACHING COST($)
CF - CONDENSER TUBE FRONTAL AREA(FT2)
CAF - CONDENSER TUBE FREE-FLOW AREA(FT2)
CAG - OUTSIDE SURFACE AREA PER COND TUBE(FT2)
CAREA - TOTAL SURFACE AREA OF CONDENSER TUBE PROFILE(IN2)
CBASE - HOR. DIST BETWEEN TUBES-CENTER TO CENTER(IN)
CBPFE - BUSTLES, FLANGES AND CHANNELS COST($)
CDPEFE - EVAP FLANGES, CHANNELS AND FLOW PLATES COST($)
CCF - TOTAL CONDENSER FABRICATION COST($)
CCSWP - COST OF COND. OR SW COLD PIPE PUMP($)
CDPHER - CONDENSER 4TH DIST PLATE AND BAFFLES COST($)
CDPHE - EVAP NH3 DISTRIBUTION PLATE AND BAFFLES COST($)
CESELECT - COST OF ELECT POWER GENERATION($)
CELEH - TWO-PHASE EFFECT OF COND ELEVATION(LBF/IN2)
CESWP - COST OF EVAP SW PUMP($)
CF - CONDENSER TUBE SIDE FRICTION FACTOR
CFRIG - TWO-PHASE EFFECT OF COND PIPE FRICTION(LBF/IN2)
CFI - CONDENSER FILM TEMPERATURE(DEG F)
CGC - CONVERSION OF CFT TO S.I. UNITS(DEG C)
CGEN - COST OF AMMIPELA GENERATOR($)
CGF - COND MASS VEL FOR MIN FREE-FLOW AREA(LBM/FT2.SEC)
CHC - CONDENSER HEAT EXCH HEAD COST($)
CHE - EVAP HEAT EXCHER HEADS COST($)
CHSC - CONDENSER HEAT EXCH SHELL COST($)
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<td>CTTW2</td>
<td>CONDENSER SHELLSIDE WALL TEMPERATURE (DEG F)</td>
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<td>CTTW2C</td>
<td>CONVERSION OF CTTW TO S.I. UNITS (DEG C)</td>
</tr>
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<td>CTTW3</td>
<td>CONDENSER TUBE WELDING COST ($)</td>
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<td>CTTW4</td>
<td>EVAP TUBE WELDING COST ($)</td>
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<td>CCHWSE</td>
<td>EVAP WATER INLET NOZZLES AND SUPPORTS COST ($)</td>
</tr>
<tr>
<td>CCHWSC</td>
<td>COND WATER INLETS NOZZLES &amp; SUPPORTS COST ($)</td>
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<tr>
<td>CBLKTL</td>
<td>ABSOLUTE DIFFERENCE BETWEEN TBLK2 &amp; RTBLK2 (DEG F)</td>
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<td>CBLKTC</td>
<td>DIFFERENCE BETWEEN REVISED AND ASSUMED TBulk TEMP</td>
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<td>CCOND</td>
<td>TOTAL PRESSURE LOSSES ACROSS CONDENSER (LBF/IN2)</td>
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<td>CDCOND</td>
<td>CONVERSION OF CCOND TO S.I. UNITS (KPA)</td>
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<td>CDOREC</td>
<td>CONDENSER TUBE FRICTION LOSSES (LBF/IN2)</td>
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<td>EVAPORATOR TUBE FRICTION LOSSES (LBF/IN2)</td>
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C TYPEC CONDENSER TYPE SELECTION
C TYPEE - EVAPORATOR TYPE SELECTION
C UAC - CONDENSER TUBE CONDUCTANCE(BTU/LBM.F)
C UACC - OVERALL CONDENSER HT COEFFICIENT(NPU01200)
C UCC - CONVERSION OF UC TO S.I. UNITS(W/M2.C)
C UE - EVAP OVERALL HT COEFF(DIFFER)(BTU/HR.FT2.F)
C UEC - CONVERSION OF UE TO S.I. UNITS(W/M2.C)
C VAPC - SPECIFIC VOL COND SHELLSIDE SAT NH3 LIQ(FT3/LBM)
C VAPE - SPECIFIC VOL SAT NH3 VAPOR EVAP SHELLSIDE(FT3/LBM)
C VAVGC - AVG SPECIFIC VOL COND SHELLSIDE NH3 FLOW(FT3/LBM)
C VAVGE - AVERAGE NH3 SPECIFIC VOL EVAP SHELLSIDE(FT3/LBM)
C VEF - SPECIFIC VOLUME AT STATE PT I(FT3/LBM)
C VFA - SPECIFIC VOLUME OF PUMPED RE-FLUX AMMONIA(FT3/LBM)
C VHEAD - MOISTURE SEPARATOR INLET VEL HEAD(LBF/IN2)
C VISS - SUBROUTINE SW VISCOITY(LBM/FT.HR)
C VISSW - CONDENSER SW VISCOITY(LBM/HR.FT)
C VISSWE - EVAP SW VISCOITY(LBM/FT.HR)
C VISSW(TLKC) - SUBROUTINE SW VISCOITY(LBM/FT.HR)
C VISSW(TLKE) - SUBROUTINE SW VISCOITY(LBM/FT.HR)
C VISSW(THEL) - SUBROUTINE SW VISCOITY(LBM/FT.HR)
C VISSWHEL - SUBROUTINE SW VISCOITY(LBM/FT.HR)
C VISSHCP - VISCOITY OF COLD PIPE SW(LBM/HR.FT)
C VISSHPC - VISCOITY OF COLD PIPE SW(LBM/FT.HR)
C VISSHPH - VISCOITY OF COLD PIPE SW(LBM/FT.HR)
C VLIQC - SPECIFIC VOL COND SHELLSIDE SAT NH3 LIQ(FT3/LBM)
C VLIQE - SPECIFIC VOL SAT NH3 LIQ EVAP SHELLSIDE(FT3/LBM)
C VMAXC - MAXIMUM CONDENSER SHELLSIDE NH3 VELOCITY(FT/SEC)
C VMAXE - MAX VELOCITY OF EVAP SHELLSIDE AMMONIA(FT/SEC)
C VNHC - CONDENSER SHELLSIDE AMMONIA VELOCITY(FT/SEC)
C VHNHC - AVERAGE EVAP SHELLSIDE AMMONIA(FT/SEC)
C VHNH - REVERSE AMMONIA SHELLSIDE VELOCITY(FT/SEC)
C VHNHCP - VELOCITY OF PIPED AMMONIA(FT/SEC)
C VHNHP - VELOCITY OF PIPED AMMONIA(FT/SEC)
C VHNHRP - RE-FLUX AMMONIA VELOCITY(FT/SEC)
C VHNHS - NH3 VELOCITY AT MOISTURE SEPARATOR INLET(FT/SEC)
C VSNH3C - CONDENSER AMMONIA VISCOITY(LBM/HR.FT)
C VSNH3C(F) - CONDENSER FOR AMMONIA VISCOITY(LBM/HR.FT)
C VSNH3F - VISCOITY OF AMMONIA IN EVAP(LBM/HR.FT)
C VSNH3(EFT) - SUBROUTINE FOR VISCOITY AMMONIA(LBM/HR.FT)
C VSNHP - VISCOITY OF PIPED AMMONIA(LBM/HR.FT)
C VSNHRP - RE-FLUX AMMONIA VISCOITY(LBM/HR.FT)
C VSMC - CONDENSER SW FLOW VELOCITY(FT/SEC)
C VSNC - CONVERSION OF VSMC TO S.I. UNITS(M/SEC)
C VSMC - CONDENSER SW FLOW VELOCITY(FT/SEC)
C WSMPC - CONVERSION OF VSMC TO S.I. UNITS(M/SEC)
C VSMC - CONDENSER SW FLOW VELOCITY(FT/SEC)
C WSME - EVAPORATOR SW FLOW VELOCITY(FT/SEC)
C VSMSEC - CONVERSION OF VSMCP TO S.I. UNITS(M/SEC)
C VSMHC - CONDENSER SW FLOW VELOCITY(FT/SEC)
C WSMP - CONVERSION OF VSMC TO S.I. UNITS(M/SEC)
C WC - SHELLSIDE NH3 FLOW RATE PER TUBE(LBM/HR)
C WE - AVG MASS FLOW RATE PER EVAP TUBE(LBM/HR)
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<td>X5P</td>
<td>Conversion X5 (Pct)</td>
<td>NPU00460</td>
</tr>
<tr>
<td>X5SP</td>
<td>Conversion of X5SP (Pct)</td>
<td>NPU00570</td>
</tr>
<tr>
<td>X5SP</td>
<td>Conversion of X5SP (Pct)</td>
<td>NPU00580</td>
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C THESIS PRESENTATION: MODEL ANALYSIS OF THE CLOSED-CYCLE
C OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM
C BY LTOR RAYMOND C. SCHAUBEL, USN
C SUBROUTINE ANALIZ (ICALC)  COMMON /GLOBCN/ DICT, DIHP, DNH4, DNH3R, PEVAP, PCOND, TDOC, TODE, TLE, TIC, VSCW, VSCE, VSCP, VSHE, VSWP, EPR, CPR, PSYST, DTEMP1, DTEMP2, DTEMP3, DH5, D2X5, DILF, ETURBP, TSDE, TSDE, OBJ
C IF (ICALC.GT.1.) GO TO 170
C
C CONSTANTS
C
BUC=778.2
CONC=35
CONH=35
CHW=7.46.E-06
G=32.174
GC=32.174
GAL=7.481
HPC=550.
PTE=3.141592654
C
C SALT WATER TEMPERATURE INPUTS (DEG F)
C
TIC=40.
THIE=80.
C
C INITIALIZED DESIGN VARIABLES
C
DICT=19.6
DIHP=19.3
DNH4=2.0
DNH3R=2.0
PCOND=89.
PEVAP=130.
TDOC=1.
TODE=1.
TLE=40.
VSCW=6.
VSCP=5.5
VSHE=6.
VSWP=4.5
EPR=1.5
CPR=1.5
C
C EVAPORATOR/CONVENER TRIAL SELECTIONS
C
TYPE=1. HORIZONTAL EVAPORATOR OWENS CORRELATION
C
TYPE=2. HORIZONTAL EVAPORATOR NON-BOILING CORRELATION
C TYPEE=3. HORIZONTAL EVAPORATOR BOILING CORRELATION
C TYPEC=1. HORIZONTAL CONDENSER
C TYPEC=2. VERTICAL CONDENSER
C TYPEC=1.
C TYPEE=1.
C ENHANCEMENT SELECTION
C EHFE=1. PLAIN TUBE EVAPORATOR
C EHFC=1. PLAIN TUBE CONDENSER
C EHFE=2. LINDE-PROMOTER EVAPORATOR (NOT CODED)
C EHFC=2. LINDE-PROMOTER CONDENSER (NOT CODED)
C EHFC=1.
C EHFE=1.
C TUBE FIN EFFICIENCY
C EFFIE=1.0
C EFFOE=1.0
C EFFOC=1.0
C EFFIC=1.0
C TUBE PROFILE SELECTION
C PROF=1. STAGGERED EQUI-LATERAL
C PROF=2. IN-LINE EQUI-SIDED
C PROF=1.
C SYSTEM PIPE GEOMETRY AND MATERIAL
C ECC=150.E-06
C ECP=500.E-06
C ECE=150.E-06
C EHP=500.E-06
C ELBOW=30.
C ENH3P=500.E-06
C TLP=3000.
C TLHP=300.
C TNH3P=150.
C TNLHP=50.
C TUBING MATERIAL SELECTION
C TMATL=1. ALUMINUM
C TMATL=2. TITANIUM
C TMATL=1.
C TKW=77.
C PROJECTED INFLATION RATE FOR AL TUBE REINSTALLATION (PCT)
C     AINT = 10.
C EVAP/MOISTURE SEPARATOR QUALITY REQUIREMENTS (PCT)
C     X3P = 92.
X4P = 99.5
C SALT WATER FOULING COEFFICIENT (HR.FT.F/BTU)
C     SWFC = .00025
C PUMP AND GEN-TURB EFFICIENCIES
C     EEP = 96.6
EEP = 96.6
EMC = 98.
EME = 98.
EMNH3 = 98.
EMNH3 = 98.
EPC = 85.
EPC = 85.
EPNH3 = 75.
EPNH3 = 75.
ETRP = 99.8

C NET OUTPUT POWER REQUIRED (MW)
C     ELECT = 30.
C INPUT DATA SI UNIT CONVERSION
C     DICPC = .3048 * DICP
DICPC = .3048 * DICP
DINHC = .3048 * DINH3
DINHC = .3048 * DINH3
PCONDC = 6.89476E-03 * PCOND
PCONDC = 6.89476E-03 * PCOND
PEVAPC = 6.89476E-03 * PEVAP
PEVAPC = 6.89476E-03 * PEVAP
TDUC = 25.4 * TDUC
TDUC = 25.4 * TDUC
TDECC = 25.4 * TDEC
TDECC = 25.4 * TDEC
TKWC = 1.73707 * TKW
TKWC = 1.73707 * TKW
TLC = 3048 * TLC
TLC = 3048 * TLC
TLHC = 3048 * TLHC
TLHC = 3048 * TLHC
TLNH3 = .3048 * Tlnh3p
TLNPC = 3048 * TLNHRP
TLCPC = 3048 * TLCPP
VSWCP = 3048 * VSWCP
VSWEC = 3048 * VSWEC
VSMCP = 3048 * VSMCP
VSWHPC = 3048 * VSWHCP

C SUMMARY OF INPUT DATA

WRITE (6, 1150)
IF (TYPEE.EQ.2) GO TO 10
WRITE (6, 1160)
GO TO 20
C CONTINUE
WRITE (6, 1170)
CONTINUE
WRITE (6, 1180) TDQET, TDDECC
WRITE (6, 1190) TLE, TLEC
WRITE (6, 1200) VSE, VSMCC
WRITE (6, 1210) PEVAP, PEVAPC
IF (TMATL.EQ.2) GO TO 30
WRITE (6, 1220) TKW, TKWC
GO TO 40
C CONTINUE
WRITE (6, 1230) TKW, TKWC
CONTINUE
WRITE (6, 1240)
GO TO 60
C CONTINUE
WRITE (6, 1250)
CONTINUE
WRITE (6, 1260) EPR
IF (HEPE.EQ.2) GO TO 70
WRITE (6, 1270)
GO TO 80
C CONTINUE
WRITE (6, 1280)
CONTINUE
IF (TYPEE.EQ.2) GO TO 90
WRITE (6, 1290)
GO TO 100
C CONTINUE
WRITE (6, 1300)
CONTINUE
WRITE (6, 1310) TDQC, TDQCC
WRITE (6, 1320) TLE, TLEC
WRITE (6, 1330) VSWC, VSWCC
WRITE (6,1340) PCOND,PCONDC
IF (TMATL.EQ.2.) GO TO 110
WRITE (6,1350) TKW,TKWC
GO TO 120
110 CONTINUE
WRITE (6,1360) TKW,TKWC
120 CONTINUE
IF (PROF.EQ.2.) GO TO 130
WRITE (6,1370)
GO TO 140
130 CONTINUE
WRITE (6,1380)
140 CONTINUE
WRITE (6,1390) EPR
IF (EHFC.EQ.2.) GO TO 150
WRITE (6,1400)
GO TO 160
150 CONTINUE
WRITE (6,1410)
160 CONTINUE
WRITE (6,1420)
WRITE (6,1430) DIHP,DIHPC
WRITE (6,1440) TLP,H,TLHPC
WRITE (6,1450) VSWHP,VSHPC
WRITE (6,1460) THIE,THIEC
WRITE (6,1470) CONH
WRITE (6,1480) DICP,DICPC
WRITE (6,1490) TLCP,TLPC
WRITE (6,1500) VSWP,VSWPC
WRITE (6,1510) TCIC,TCICC
WRITE (6,1520) CONC
WRITE (6,1530)
WRITE (6,1540)
WRITE (6,1550) DINH3,DINH3C
WRITE (6,1560) TLNHP,TLNHPC
WRITE (6,1570)
WRITE (6,1580) DINHR,DINHRM
WRITE (6,1590) TLRHP,TLRTPC
WRITE (6,1600)
WRITE (6,1610)
WRITE (6,1620) EPE,EME
WRITE (6,1630)
WRITE (6,1640) EPC,EMC
WRITE (6,1650)
WRITE (6,1660) EPNH3,EMNH3
WRITE (6,1670)
WRITE (6,1680) EEP
WRITE (6,1690) ETRP
WRITE (6,1700) 
WRITE (6,1710) ELECT 
RETURN 

170 CONTINUE

**** EXECUTION OF INPUT DATA ****
TUBE WALL THICKNESS (INCH) AND TUBE COST PER FT ($/FT) 
ALUMINUM TUBING

IF (TMIL.GT.1.5) GO TO 210
IF (EHFC.GT.1.5) GO TO 180
TTE=AP(TDOE)
E1=AC(TDOE)
E2=0.0
GO TO 190

180 CONTINUE
TTE=AE(TDOE)
E1=AEC(TDOE)
E2=0.0
GO TO 190

190 CONTINUE
IF (EHFC.GT.1.5) GO TO 200
TTC=AP(TDOC)
C1=AC(TDOC)
C2=0.0
GO TO 250

200 CONTINUE
TTC=AE(TDOC)
C1=AEC(TDOC)
C2=0.0
GO TO 250

TITANIUM TUBING

210 CONTINUE
IF (EHFE.EQ.2.) GO TO 220
TTC=TP(TDOE)
E1=TC(TDOE)
E2=1.21
GO TO 230

220 CONTINUE
TTC=TE(TDOE)
E1=TEC(TDOE)
E2=1.21
GO TO 240

230 CONTINUE
IF (EHFE.EQ.2.) GO TO 240
TTC=TP(TDOC)
C1=TC(TDOC)

A 241
A 242
A 243
A 244
A 245
A 246
A 247
A 248
A 249
A 250
A 251
A 252
A 253
A 254
A 255
A 256
A 257
A 258
A 259
A 260
A 261
A 262
A 263
A 264
A 265
A 266
A 267
A 268
A 269
A 270
A 271
A 272
A 273
A 274
A 275
A 276
A 277
A 278
A 279
A 280
A 281
A 282
A 283
A 284
A 285
A 286
A 287
A 288
C 2=1.21
GO TO 250
CONTINUE
TTC=T(E(TDOC)
C1=T(C(TDOC)
C2=1.21
CONTINUE

C DESIGN VARIABLE DIMENSION CONVERSION
C
P1=PCOND
H1=HF(PCOND)
T1=TSAT(PCOND)
T3A=TSAT(PEVAP)
T5=TSAT(PCOND)
TDIC=TDCC-2.*TTC
TDICC=TDIC/12.
TDIE=TDCE-2.*TTE
TDIEC=TDIE/12.
TDCC=TDCC/12.
TDCE=TDCE/12.

C CONSTRAINT FOR SAT SYSTEM PRESSURE RATIO
C
PRSYST=PEVAP/PCOND
C CONSTRAINT FOR SAT UPPER TEMP BOUND
C
DTEMP1=THIE-T3A
C CONSTRAINT FOR SAT LOWER TEMP BOUND
C
DTEMP2=T1-TCIC
C CONSTRAINT FOR CONDENSER OUTLET TEMPERATURE BOUND
C
DTEMP5=T1-TCOC

******************************************************************************

* *
* EVaporator SEction *
* *
* *
******************************************************************************
INITIALLY ASSUME AN EVAP TUBE LENGTH(FT)

ASSUME A HT COEF FOR EVAP SHELLSIDE
HTNH3E=1000.

HOT PIPE SW MASS FLOW RATE(LBM/HR)
Roswh=RHOSW(THIE)
Flowh=3600.*Roswh*PIE*D1HP**2*VSWHP/4.

EVAP SW DENSITY(INITIALLY ASSUMEBULK=THIE)

TBLKE=THIE
CONTINUE
Rhosw=Rhosw(TBLKE)
TKSWE=TKS(TBLKE)

TOTAL NUMBER OF EVAP TUBES
TNET=4.*Flowh/(3600.*RHOSW*PIE*TD1EC**2*VSWE).

TUBE SHEET DIAMETER(FT)

TUBE PROFILE - STAGGERED

IF PROF.EQ.2.1 GO TO 270
EHT=EPr*TD1E*0.5
EBASE=EPr*TD1E*0.866
ETAREA=EHT*EBASE*2.
SN=2.*EHT
SPE=2.*EBASE-TOOE
SPEC=SPE/12.
GO TO 280

CONTINUE

TUBE PROFILE - IN-LINE

EPLONG=EPr*TD1E
SN=EPLONG
EPLAT=EPr*TD1E
ETAREA=EPLONG*EPLAT
SPE=EPr*TD1E-TOOE
SPEC=SPE/12.

CONTINUE

ETAREA=ETAREA*TNET
TSDE=(4.*ETAREA/PIE)**0.51/12.

C TOTAL HT AREA OF EVAP(FT2 OR M2)
C THTAE=TNET*PIE*TDOEC*TLE
C CMIN FOR EVAP(BTU/HR.*F OR W/C)
C CPSWE=CPSW(TBLKE)
C CMIN=FLOHP*CPSWE
C C OVERALL HT COEF(BTU/HR.*FT2.*F OR W/M2.*C)
C TBLKE=TBLKE*459.69
C C REYNOLDS NUMBER FOR EVAP TUBESIDE
C VISSWE=VISSW(TBLKE)
C RNWE=3600.*RNWE*VSWE*TDIEC/VISSWE
C C PRANDTL NUMBER FOR EVAP TUBESIDE
C PNSWE=CPSWE*VISSWE/TKSWE
C C HT COEF FOR EVAP TUBESIDE
C IF (RNWE.GT.2300.) GO TO 290
C C LAMINAR FLOW USING SIEDER-TATE CORRELATION
C HTSWE=1.86*TKSWE*(RNWE*PNSWE)**.3333*(TDIEC/TLE)**.3333/TDIEC
C GO TO 300
C 290 CONTINUE
C C TURBULENT FLOW USING DITUS-BOELTER CORRELATION
C (INITIALLY ASSUME TBLKE=THIE)
C HTSWE=.023*TKSWE*RNWE*.8*PNSWE*.4/TDIEC
C 300 CONTINUE
C C THERMAL RESISTANCE FOR SW(HR.*FT2.*F/BTU)
C TR1E=TDOEC/(EFFIE-HTSWE*TDIEC)
C C THERMAL RESISTANCE FOR SW FOULING(HR.*FT2.*F/BTU)
C HTFSWE=L./SHFC
C TR2E=TDOEC/(EFFIE-HTFSWE*TDIEC)
C C THERMAL RESISTANCE FOR WALL THICKNESS(HR.*FT2.*F/BTU)
TR3E=TDQEC*A(LOG(TDOE/TDIE)+2.*TKW)

THermal resistance for NH3 fouling(HR.FT2,F/ BTU)
considered negligible

THermal resistance for NH3
EFFOE=1.
TR5E=1./(EFFOE*HTNH3E)

Pseudo HT COEF for SW(BTU/HR.FT2,F; OR W/M2.C)
HSWE=1./TR1E

Pseudo HT COEF for SW fouling(BTU/HR.FT2,F OR W/M2.C)
HFSWE=1./TR2E

Pseudo HT COEF for wall THICK(BTU/HR.FT2,F OR W/M2.C)
HWE=1./TR3E

Pseudo HT COEF for ammonia(BTU/HR.FT2,F OR W/M2.C)
HNH3E=1./TR5E

Overall HT COEF calculation-OUTTER surface(BTU/HR.FT2,F OR W/M2.C)
UE=1./(TR1E+TR2E+TR3E+TR5E)

Number of transfer units for evap(NTU)
ENTU=UE*HTAE/CMINE

Evap effectiveness(Epsilon)
EPSE=1.-EXP(-ENTU)

Evap sw outlet Temp(F OR C)
THOE=THIE-(THIE-T3A)*(1.-EXP(-ENTU))

Revised sw avg BULK Temp(F)
RTBLKE=(THOE+THIE)/2.
C TEST FOR A SAT TBULK TEMPERATURE
C
DBLKT=ABS(TBLKE-RTBLKE)
SCALB=ABS(TBLKE)
IF (SCALB-LT.0.01) SCALB=0.1
DELB=DBLKT/SCALB
IF (DELB.LT.0.001) GO TO 310
TBLKE=RTBLKE
GO TO 260
310 CONTINUE
TBLKE=RTBLKE
C FILM TEMP FOR PROPERTY EVALUATION
C INITIALLY ASSUME T3(IDEAL)=T3(Actual)
C
THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE(UA)
(BTU/HR.F)
AO=PIE*TDHEC*TLE
THERMAL RESISTANCE SW(HR.F/BTU)
CTR1E=TR1E/AO
THERMAL RESISTANCE SW FOULING(HR.F/BTU)
CTR2E=TR2E/AO
THERMAL RESISTANCE WALL THICKNESS(HR.F/BTU)
CTR3E=TR3E/AO
THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)
NEGLIGIBLE
THERMAL RESISTANCE NH3(HR.F/BTU)
CTR5E=TR5E/AO
HEAT TRANSFERED PER TUBE(BTU/HR)
QET=(TBLKE-T3)/(CTR1E+CTR2E+CTR3E+CTR5E)
SHELLSIDE WALL TEMPERATURE(F)
ETW2=TBLKE-QET*(CTR1E+CTR2E+CTR3E)
C TUBESIDE WALL TEMPERATURE(F)
C ETW1=TBK-E-QET*(CTR1E+CTR2E)
C EVAP FILM TEMP CALCULATION(F)
C EFT=(ETW2+T3)/2.
C DELTA T TEMPERATURE(F)
C DELTAE=ETW2-T3
C AMOUNT OF HEAT ADDITION(BTU/HR OR W)
C QE=CMINE*(THIE-THOE)
C LOG MEAN TEMPERATURE DIFFERENCE OF EVAP(F OR C)
C INITIALLY ASSUME T3(Ideal)=T3(ACTUAL)
C ELMTD=(1.-EXP(-ENTU))*CMINE*(THIE-T3)/(UE*THAE)
C ISENTROPIC NH3 PUMP WORK(BTU/LBM)
C INITIALLY ASSUME P1(Ideal)=P1(ACTUAL)
C VF1=1./RFNH3(T1)
C WPSNH3=VF1*(PEVAP-P1)*144./BTUC
C THERMODYNAMIC NH3 PUMP WORK(BTU/LBM)
C EPNH3C=EPNH3/100.
C WPNH3=WPSNH3/EPNH3C
C WORKING FLUID PROPERTIES
C CPHNH3=CPNH3(EFT)
C VSNH3=VSNH3(EFT)
C TKNH3=TKNH3(EFT)
C RONH3=RFNH3(EFT)
C ENTHALPY AT STATE PT 2(BTU/LBM)
C H2=H1+WPNH3
C ENTHALPY AT STATE PT 3A(BTU/LBM)
C H3A=HF(PEVAP)
C INITIALIZE NH3 MASS FLOW RATE(LBM/HR)
C H3=HG(PEVAP)
C FLOH3=QE/(H3-H2)
323 CONTINUE
C PRESS DROP EVAP SHELLSIDE(LBF/IN2)
C ASSUME VISCOITY(TWALL)=VISCOITY(TBULK)
C MAX VELOCITY THRU MIN-FLOW AREA F(TUBE PROFILE)
C RNH3P=RFNH3IT1)
C VNH3E=4.*F(LNH3/(3600.*RNH3P+PIE*DIH3**2)
C VMAXE=VNH3E*(SN/(SN-TDOE))
C REYNOLDS NO. FOR MAX SHELLSIDE FLOW
C REMAX=3600. *RNH3P*VMAXE*TDOE/(12.*VSNH3E)
C EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
C IF (PROF.EQ.2.) GO TO 330
C EFF=10.25+0.18/(SN-TDOE/TDOE)**1.08*REMAX**1**.16
C GO TO 340
330 CONTINUE
C EFF=10.47/(0.168+SN/TDOE)/15(SN-TDOE/TDOE)**1.13+TDOE/SN)**RE
C MAX**1.16
C CONTINUE
C MASS VELOCITY FOR MIN FREE-FLOW AREA(LBF/FT2-SEC)
C EAF=TSDE**TLE
C EE=EFR*TSDE/(SN-TDOE/SN)
C EGF=FLOH3/3600.*EAF
C CALCULATION OF PRESS DROP EVAP SHELLSIDE(LBF/IN2)
C USİNG THE HOMOGENEOUS TWO-PHASE MODEL
C X3A=1.0
C EDE=EFR*TSDE-TDOE/12.
C VLIQE=1./RFNH3(T3A)
C VAPE=1./RGNH(PEVAP)
C VVAGE=VLIQE*(1.**X3A*(VAPE-VLIQE)/VLIQE)
C EFRICT=EFR**2*VAVE*TSDE/144.*EDE**2.*GC1
C EMOM=(EGF**2*VAVE)/(144.*GC)
C EEL=EGR**TSDE/144.*GC)
C DSEVAP=EFRICT+EMOM+EEL
C ENTHALPY AT STATE PT 3(BTU/LBM)
C ASSUME QUALITY EVAPORATOR OUTLET X3P
C
X3=X3P/100.
P3=PEVAP-DSEVAP
H3F=Hf(P3)
H3G=Hg(P3)
H3=H3F+X3*(H3G-H3F)
C
C PRESSURE DROP ACROSS THE MOISTURE SEPARATOR(LBF/IN2)
C
RONH3S=RGNH3(P3)
ESPACE=0.1*TSDE*TE
VNH3S=FLONH3/(3600.*RONH3S*ESPACE)
VHEAD=RONH3S*VNH3S**(2/12.*GC*144.)
DSDM=20.*VHEAD
C
C MOISTURE SEPARATOR DISCHARGE DRAIN ENTHALPY (BTU/LBM)
C
P4=P3-DSDM
T4=TSAT(P4)
PAVGD=(P3+P4)/2.
HDE=Hf(PAVGD)
C
C ENTHALPY AT STATE PT 4 (BTU/LBM)
C
C ASSUME QUALITY OF MOISTURE SEPARATOR OUTLET X4P
X4=X4P/100.
H4F=Hf(P4)
H4G=Hg(P4)
H4=H4F+X4*(H4G-H4F)
C
C REVISED AMMONIA FLOW RATE (LBM/HR)
C
FLONH3=QE/((X4/X3)*H3-H2-(X4/X3-1.)*HUE)
C
C REVISED AMMONIA SHELLSIDE VELOCITY (FT/SEC)
C
VNH3ER=4.*FLONH3/(3600.*RONH3P*PIE*DINH3**2)
C
C TEST FOR SAT VNH3E (FT/SEC)
C
DVNH3E=ABS(VNH3ER-VNH3E)
SCALV=ABS(VNH3ER)
IF (SCALV.LT.0.1) SCALV=0.1
DELV=DVNHN3E/SCALV
IF (DELV.LT.0.001) GO TO 350
VNH3E=VNH3ER
GO TO 320
350 CONTINUE
VNH3E=VNH3ER
C
REVISED TEMP AT STATE PT 3(10G F)
C
T3R=TSAT(P3)
C
TEST FOR SAT T3(10G F)
C
DTEMP3=ABS(T3R-T3)
SCALT=ABS(T3R)
IF (SCALT.LT.0.1) SCALT=0.1
DELT3=DTEMP3/SCALT
IF (DELT3.LT.0.001) GO TO 360
T3=T3R
GO TO 260
360 CONTINUE
T3=T3R
C
PRANDTL NUMBER
C
PNH3E=CPNH3E*VSNH3E/TKNH3E
C
EVAPORATOR TYPE IDENTIFICATION
C
TYPEE=1:HORIZONTAL OWENS CORRELATION
C
TYPEE=2:HORIZONTAL NON-BOILING CORRELATION
C
TYPEE=3:HORIZONTAL BOILING CORRELATION
C
IF (TYPEE.GT.1) GO TO 380
C
HORIZONTAL NON-BOILING USING OWENS CORRELATION
C
TRANSITION REYNOLDS NUMBER
C
TRNE1=1680.*1CPNH3E*VSNH3E/TKNH3E**(-1.5)
C
REYNOLDS NUMBER(PSEUDO)
C
WE=FLONH3/NTET
RNH3EH=4.*WE/(TLE*VSNH3E)
IF (RNH3EH.GT.TRNE1) GO TO 370
C
LAMINAR FLOW USING OWENS CORRELATION
C
HNNH3ER=2.2*(SPEC/TDOEC)**.1*(VSNH3E**2/(3600.*2*G*RONH3E**2*TKNH3
1E**3))**(-.3333)*RNNH3EH**(-.3333)
GO TO 420
370 CONTINUE
TURBULENT FLOW USING OWENS CORRELATION

\[ \text{MN}3\text{E} = \text{.185 (SPEC/TDOEC)**1} \times (\text{VSNH3E}**2 / (3600.*2*GRONH3E*2*TKNH13E**3)) \times (-.3333) \times \text{PNH3E**2} \times 5 \]

GO TO 420

IDENTIFY PARAMETERS FOR TYPEE = 2 OR 3

FLOW RATE AVERAGED PER TUBE

\[ \text{WE} = \text{FLONH3/TNET} \]

FLOW RATE PER UNIT AXIAL LENGTH OF TUBE

\[ \text{WEN} = \text{WE} / (\text{PIE} \times \text{TDOEC} / 2) \]

TUBE LENGTH FOR DEVELOPING FLOW (FT)

\[ \text{TLD} = \text{WEA} \times (4.31 / (4.14 \times \text{IKNH3E/CPNH3E})) \times (3. \times \text{VSNH3E} / (3600.**2 \times \text{RONH3E} \times \text{TEN}**2 \times \text{G}))) \times .3333 \]

TUBE LENGTH FOR FULLY DEVELOPED FLOW (FT)

\[ \text{TLF0} = \text{TNET**2*5} \times \text{PIE} \times \text{TDOEC} / 2 \]

IF (TYPEE.GT.2.1) GO TO 400

HORIZONTAL NON-BOILING LORENZ AND YUNG CORRELATIONS

TRANSITION REYNOLDS NUMBER FOR CORRELATIONS

\[ \text{TRNE2} = 5800. \times (\text{CPNH3E*VSNH3E/TKNH3E})**(-1.06) \]

REYNOLDS NUMBER (PSEUDO-VERTICAL)

\[ \text{RNH3E} = 4. \times \text{WE} / (\text{PIE} \times \text{TDOEC} \times \text{VSNH3E}) \]

IF (RNH3E.GT.TRNE2) GO TO 390

LAMINAR FLOW USING LORENZ AND YUNG CORRELATIONS

CONVECTION IN DEVELOPING REGION

\[ \text{HTDR} = 3. \times \text{CPNH3E} \times \text{WE} / \text{TLF0} \]

CONVECTION IN FULLY DEVELOPED REGION

\[ \text{HTFD} = 0.821 \times (\text{VSNH3E}**2 / (3600.**2 \times \text{RONH3E}**2 \times \text{TKNH3E}**3 \times \text{G}))**(-.3333) \times A \]

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LAMINAR FLOW CALCULATIONS
HNN3E=HTDR+HTFDR
GO TO 420
CONTINUE
TURBULENT FLOW USING LORENZ AND YUNG CORRELATIONS
CONVECTION IN DEVELOPING REGION
HTDR=3.*CPNH3E*WEA/TLFD
CONVECTION IN FULLY DEVELOPED REGION
HTFDR=3.8E-03*(VSNH3E**2/(3600.*2*RONH3E**2*TKNH3E**3*G1)**(-.333)
131*RNH3E**.4*PNH3E**.65
TURBULENT FLOW CALCULATION
HNN3E=HTDR+HTFDR
GO TO 420
CONTINUE
HORIZONTAL BOILING USING LORENZ AND YUNG CORRELATIONS
TRANSITION REYNOLDS NUMBER FOR CORRELATIONS
TRNE=5400.*(CPNH3E*VSNH3E/TKNH3E)**(-1.061)
REYNOLDS NUMBER(PSEUDO-VERTICAL)
RNNH3E=4.*WE/(PIE*TDVCC*VSNH3E)
IF (RNNH3E**.51*TRNE) GO TO 410
LAMINAR FLOW USING LORENZ AND YUNG CORRELATIONS
CONVECTION IN DEVELOPING REGION
HTDR=3.*CPNH3E*WEA/TLFD
CONVECTION IN FULLY DEVELOPED REGION
HTFDR=.821*(VSNH3E**2/(3600.*2*RONH3E**2*TKNH3E**3*G1)**(-.333)*
IRNH3E**(-.22)*((1.-TLD/TLFD))
BOILING USING LORENZ AND YUNG CORRELATION
HFNH3E=HF(P3) A 817
HGNH3E=HG(P3) A 818
TENS=1.6038998E-03 A 819
CSF=.0154 A 820
HTB=VSNH3E*(HGNH3E-HFNH3E)/(CSF**3*(TENS/RDNH3E)**.5)*(CPNH3E/(1(HG
1NH3E-HFNH3E)*PNNH3E))**3*DELTAE**2 A 821

LAMINAR FLOW CORRELATION A 822

410 CONTINUE A 823

TURBULENT FLOW USING LORENZ AND YUNG CORRELATIONS A 824

CONVECTION IN DEVELOPING REGION A 825

HTDR=3.*CPNH3E*WEA/TLFD A 826

CONVECTION IN FULLY DEVELOPED REGION A 827

HTFDR=3.8E-03*(VSNH3E**2/(3600.***2*RDNH3E**2*TKNH3E**3*G1)**(-.33
13)*RNNH3EV**.4*PNH3E**.65 A 828

BOILING USING LORENZ AND YUNG CORRELATION A 829

420 CONTINUE A 830

TURBULENT FLOW CALCULATIONS A 831

430 CONTINUE A 832

TEST FOR SAT HTNH3E A 833

DHTNH3=ABS(HNH3ER-HTNH3E) A 834
SCALH=ABS(HNH3ER) A 835
IF (SCALH.LT.0.1) SCALH=0.1 A 836
DELH=DHTNH3/SCALH A 837
IF (DELH.LT.0.001) GO TO 430 A 838
HTNH3E=HNH3ER A 839
GO TO 260 A 840

CONTINUE A 841

CONTINUE A 842

CONTINUE A 843

CONTINUE A 844

CONTINUE A 845

CONTINUE A 846

CONTINUE A 847

CONTINUE A 848

CONTINUE A 849

CONTINUE A 850

CONTINUE A 851

CONTINUE A 852

CONTINUE A 853

CONTINUE A 854

CONTINUE A 855

CONTINUE A 856

CONTINUE A 857

CONTINUE A 858

CONTINUE A 859

CONTINUE A 860

CONTINUE A 861

CONTINUE A 862

CONTINUE A 863

CONTINUE A 864
HTNH3E=HNH3ER
C REVISED EVAP HEAT TRANSFER AREA(FT2)
C THTAER=ENTU*CMINE/UE
C REVISED EVAP TUBE LENGTH(FT)
C TLER=THTAER/(PIE*TDCE*TNET)
C TEST FOR SAT TUBE LENGTH
C COST CF EVAPORATOR UNIT($)
IF (TSDE.GT.35.) GO TO 480
C EVAPORATOR TUBE SHEET DIAMETER(10-35)FT
C DRILLING TIME/TUBE SHELL THICK(MIN/IN)
C DTE=0.66*(TDCE-.5)
C THICKNESS OF TUBE SHEET(IN)
C TTSE=0.56*TSDE**0.68
C TUBE SHEET LABOR COST($)
C CTSLE=156695.*(TNET/9630.)*(DTE/0.66)*(TTSE/4.)
C TUBE SHEET MATERIAL COST($)
C CTSME=189.486*TSDE**2.3
IF (TMTL.EQ.1.) GO TO 440
C TUBE MATERIAL COST($)
C CTME=(E1*TLE+E2)*TNET*(TDCE/1.5)
C TUBE INSTALLATION COST($)
C CTIE=34.*TNET*TDCE**0.7
GO TO 450
CONTINUE
440
C TUBF MATERIAL COST($)
C CTME=(E1*TLE+E2)*TNET*(TDCE/1.5)*(1.+(1.+AINT/100.)*10+(1.+AINT/100.)**20)

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OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM. (U)

DEC 80 R C SCHAUBEL
TUBE INSTALLATION COST($)
CTIE=34.*TNET*TDOE**0.7*(1.+(1.+AINT/100.)*1.5)*10+(1.+AINT/100.)*20
CONTINUE
HEAT EXCH SHELL COST($)
CHSE=177265.*(TLE+6.1/31.)*(TSDE/18.)**2
NH3 DIST PLATE AND Baffles($)
CDPBE=93865.75*(TNET/9630.1)*(DTE/0.66)*(TSDE/18.)**2
BUSTLE, FLANGES, CHANNELS, AND FLOW PLATES COST($)
CBFCFE=308550.*(TSDE/18.)**2
HEAT EXCH HEADS COST($)
CHE=53240.*(TSDE/18.)**3
WATER INLET, NOZZLES AND SUPPORTS COST($)
CWINSW=220310.75*(TSDE/18.)**2
TUBE WELDING COST($)
CTWE=0.0
IF (TMLT.EQ.1.) GO TO 470
IF (TNET.GT.760000) GO TO 460
CTWE=14.73*TNET**0.03*(TDOE/1.5)**0.7
GO TO 470
CONTINUE
CTWE=0.8797*TNET**1.3*(TDOE/1.5)**0.7
CONTINUE
CEVAP=CTSE+CTME+CTIE+CHSE+CBFCFE+CHE+CWINSW+CTWE+CDPBE
GO TO 550
CONTINUE
EVAPORATOR TUBE SHEET DIAMETER(35-50)FT
DRILLING TIME/TUBE SHELL THICK(MIN/IN)
DTE=0.66*(TDOE-.5)
THICKNESS OF TUBE SHEET(IN)
TTSE=0.56*TSDE**0.68

C TUBE SHEET MATERIAL AND LABOR COSTS($)
C IF (TML.T.EQ.1.) GO TO 490
C TSTLE=55.189*TSDE**0.791*TSDE**0.68*DTE
C TSTM=29.566*TSDE**2.014*TTSE
C GO TO 500
C CONTINUE
C TSTLE=73.81*TSDE**0.791*TSDE**0.68*DTE
C TSTM=354.3*TSDE**1.61*TTSE
C CONTINUE
C IF (TML.T.EQ.1.) GO TO 510
C TUBE MATERIAL COST($)
C TCM=(E1*TME+E2)*TNET*(TDDE/1.5)
C TUBE INSTALLATION COST($)
C CTIE=36.542*TNET*TDDE**0.7
C GO TO 520
C CONTINUE
C TUBE MATERIAL COST($)
C TCM=(E1*TME+C2)*TNET*(TDDE/1.5)*(1+(1.+AINT/100.1)**10+(1.+AINT/100.1)**20)
C TUBE INSTALLATION COST($)
C CTIE=36.542*TNET*TDDE**0.7*(1+(1.+AINT/100.1)**10+(1.+AINT/100.1)**20)
C CONTINUE
C HEAT EXCH SHELL COST($)
C CHSE=12.544*(TLE+6.)*TSDE**2.06
C NH3 DIST PLATE AND BAFLEES COST($)
C CDPBE=158.099*TSDE**1.82+72.419*TNET**0.873*DTE
C BUSTLE,FLANGE,CHANNELS AND FLOW PLATE COSTS($)
C CBFCFE=472.977*TSDE**2.12
C HEAT EXCH HEAD COST($)
CHE=1725.31*TSDE**1.45
WATER INLET, NOZZLES AND SUPPORTS COST($) 
CWINSE=7445.297*TSDE**1.1
TUBE WELDING COSTS($) 
C
CTWE=0.0
IF (TMATL.EQ.1.) GO TO 540
IF (INET.GT.36000) GO TO 530
CTWE=14.73*INET**1.03*(TDOE/1.5)**0.7
GO TO 540
530 CONTINUE
CTWE=0.8797*INET**1.3*(TDOE/1.5)**0.7
540 CONTINUE
CEVAP=(CTSL+CTSM+CTM+CTIE+CHSE+CDOB+CBFCE+CHE+CWINSE+CTWE)
550 CONTINUE

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* *
* PARASITIC LOSS SECTION *
* *
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EVAPORATOR SALT WATER PUMP OR HOT PIPE PUMP
DELTA P EVAP SW PUMP 
RSHWP=RHOSWHTIE
THIETHIE+459.69
VISHWP=VISSWHTIE
DELTA P SW HOT PIPE USING DARCEY-WEISBACH CORRELATION 
(REYNOLDS NUMBER FOR HOT PIPE FLOW 
RNSWHP=3600.*RSHWP*VSHWP*DIHP/VISHWP 
FRICITION FACTOR FOR LAMINAR FLOW 
IF (RNSWHP.GT.2300.) GO TO 560 
FFHP=64./RNSWHP 
GO TO 570

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CONTINUE

FRICITION FACTOR TURBULENT FLOW USING STREETER CORR

FFHP=1.325/(ALOGEHP/(3.7*DIHP)+5.74/RNSWHP**.91)**2

CONTINUE

INTAKE DUCT LOSSES(assume k=1.5 FOR INLET SCREEN AND
ENTERS PLENUM PRIOR TO EVAP WHERE AREA IS ABRUPTLY
CHANGED TO 2 TIMES THE PIPE DIAMETER(ILBF/IN2)

HPKI=1.5
HPKE=1.-(DIHP/(2.*DIHP))**2)**2
DDUCTH=(HPKI+HPKE)*RNSWHP*VSHWP**2/(2.*GC*144.)

PIPE FRICTION LOSSES(ILBF/IN2)

RKHP=FFHP*ILHP/DIHP
DPIPEH=RKHP*RNSWHP*VSHWP**2/(2.*GC*144.)

OUTLET DUCT LOSSES(assume NO OUTLET PIPING)

DELTA P PIPE LOSS CALCULATION(ILBF/IN2)

DPIPH=DDUCTH+DPIPEH

DELTA P EVAPORATOR(assume NO OUTLET PIPING) USING DARCY-
WEISBACH CORRELATION(ILBF/IN2)

MINOR ENTRY/EXIT LOSSES(assume k=0.05 WELL ROUNDED
TUBE ENTRANCE AND KE=1. EXPANSION TO AN INFINITE
RESERVOIR)(ILBF/IN2)

RKI=.05
RKE=1.

DPIHE=(RKI+RKE)*RHSWE*VSWE**2/(2.*GC*144.)

DELTA P EVAPORATOR CORE(ILBF/IN2)

REYNOLDS NUMBER FOR EVAP(DETERMINED IN EVAP SECTION)

FRICITION FACTOR LAMINAR FLOW

IF (RNSWEGT,2300.) Go to 580
FFE=64./RNSW
Go to 590

CONTINUE
FRICITION FACTOR TURBULENT FLOW USING STREEETER CORR

FFE=1.325/(ALOG(10C/(3.7*TDLIE)+5.74/RNSEE**.9))**2

CONTINUE

DELTA P EVAP CORE CALCULATED(LBF/IN2)

RKHE=FFE*TE/TLIE

DCOREE=RKHE*RNSEE*VSEE**2/(2.*GC**144.)

DELTA P EVAPORATOR CALCULATED(LBF/IN2)

DEVAP=DPMINE+DCOREE

DELTA P EVAP SW PUMP CALCULATION(LBF/IN2 OR FT)

DPMPE=DDUCAH+DPipeH+DPMINE+DCOREE

DPMPEC=144.*DPipe*G/(ROSWH*G)

POWER EVAP SW PUMP(HP)

EPEC=EPE/100.

PWREP=FLOMP*DPMPEC/G/(EPEC*HPC*3600.*GC)

POWER EVAP SW PUMP MOTOR(MW)

EMEC=EME/100.

PWREP=M/CMW/EMEC

DISCHARGE RATE EVAP SW PUMP(FT3/SEC OR GAL/MIN)

QEPMP=PIE*DIHP**2*VSWH/4.

QEPMP=QEPMP*60.*GAL

COST OF EVAP SW PUMP($) 

CESWP=(QEPMP/1000.)*.75+50.*1.21E+03

CONDENSER SALW WATER PUMP OR COLD PIPE PUMP

DELTA P CONDENSER SW PUMP(ASSUME TSW(IN)=TSW(OUT),V1=V2)

ROSWCP=Rhosw+TCIC

TCICR=TCIC+450.69

VSWCP=Vissw+TCIC

FLOC=3600.*ROSWCP*PIE*DIIP**2*VSWCP/4.

DELTA P SW COLD PIPE(ASSUME V1=V2, TSW(IN)=TSW(OUT) USING
C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C REYNOLDS NUMBER FOR COLD PIPE FLOW
C RNSWCP=3600.*ROSWCP*VSWCP*DICP/VISWCP
C
C FRICTION FACTOR LAMINAR FLOW
C IF (RNSWCP-.GT.2300.) GO TO 600
C FFCP=64./RNSWCP
C GO TO 610
C 600 CONTINUE
C C FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C FFCP=1.325/(ALOG(ECP/(13.7*DICP)+5.74/RNSWCP**.9))**2
C 610 CONTINUE
C
C MINOR ENTRY/EXIT LOSSES(ASSUME K=0.05 WELL ROUNDED
C PIPE ENTRANCE AND ENTERS PLENUM PRIOR TO CONDENSER
C WHERE AREA IS ABRUPTLY CHANGED TO 2 TIMES PIPE DIAMETER(LBF/IN2)
C CPK1=.05
C CPKE=1-(DICP/12.**DICP)***2
C DDUCT=(CPK1*CPKE)*ROSWCP*VSWCP**2/(2.*GC*144.)
C
C PIPE FRICTION LOSSES(LBF/IN2)
C RKCP=FFCP*(TLCP/DICP+1.80)
C DPIPEC=RKCP*ROSWCP*VSWCP**2/(2.*GC*144.)
C
C DELTA P PIPE LOSS CALCULATION(LBF/IN2)
C DPIPC=DDUCTC+DPIPEC
C
C DELTA P DUE TO SW DENSITY(LBF/IN2)
C RDEPTH=RHOSWD*(TLCP)
C RAVG=(RDEPTH+64.184)/2.
C DHEAD=((RDEPTH-64.184)/RDEPTH)*TLCP/2.
C DDENC=G*RAVG*DHEAD/(144.*GC)
C
C DELTA P CONDENSER(ASSUME NO OUTLET PIPING) USING
C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C INITIALLY ASSUME TBLK=TSW(IN)
C TBLKC=TCIC
CONTINUE
TBLKCR=TBLKC+59.69
RHO=SWC=RHO=SWC(TBLKC)
MINOR ENTRY/EXIT LOSSES (ASSUME KI=.05 WELL ROUNDED
TUBE ENTRANCE AND KE=1 EXPANSION TO AN INFINITE
RESEVOIR (LBF/IN2)
RKI=.05
RKE=1
GPHG=(RKI+RKE)*RHO=SWC*SWC*2/12.*GC*144.
DELTA P CONDENSER CORE (LBF/IN2)
REYNOLDS NUMBER FOR COND TUBESIDE
VISSWC=VISSW(TBLKC)
RNSWC=3600.*RHO=SWC*SWC*TDICC/VISSWC
FRICITION FACTOR LAMINAR FLOW
IF (RNSWC GT 2300.) GO TO 630
FFC=64./RNSWC
GO TO 640
CONTINUE
FFC=1.325/(ALOG(ECC/(3.7*TDICC)+5.74/RNSWC**.9))**2
CONTINUE
DELTA P COND CORE CALCULATION (LBF/IN2)
RKHC=FFC*TLC/TDICC
DCOREC=RKHC*RHO=SWC*SWC*2/12.*GC*144.
DELTA P CONDENSER DUE TO CORE ELEVATION (LBF/IN2)
IF (TYPEC GT 1.) GO TO 650
DPELEC=0.
GO TO 660
CONTINUE
VERTICAL TUBED CONDENSER SW INLET ENTERS AT THE TOP
OF THE HEAT EXCHANGER
DPELEC=0.
CONTINUE
DELTA P CONDENSER CALCULATION (LBF/IN2)
C DCOND=DPMINC+DCOREC+DPELEC
C
C DELTA P COND SW PUMP CALCULATION(LBF/IN2 OR FT)
C
C DPMPC=DDUTC+DDENC+DPMINC+DCOREC+DPIPEC
C DPMPC=144.*DP4PC*GC/IR05*CP*G
C
C POWER COND SW PUMP(HP)
C
C EPCC=EPCC/100.
C PWRCP=FLOCP*DPMPCC*G/(EPCC*HPC*3600.*GC)
C
C POWER COND SW PUMP MOTOR(MW)
C
C EMCC=EMC/100.
C PWRCPM=PWRCP*CMW/EMCC
C
C DISCHARGE RATE OF COND SW PUMP(FT3/SEC OR GAL/MIN)
C
C QCPMP=PIE*DP2P**2*VSNCP/4.
C QCPMP=QCPMP*60.*GAL
C
C COST OF COND SW PUMP($)
C
C CCSWP=QCPMPC/1000.*0.75+50.1*1.21E+03
C
C AMMONIA CIRCULATION PUMP
C
C DELTA P NH3 CIRC PUMP
C VSNH3P=VSNH3(T1)
C RONH3P=RFNH1(T1)
C
C DELTA P NH3 PUMP(ASSUME V1=V2)USING DARCEY-WEISBACH
C CORRELATION(LBF/IN2)
C
C NH3 PIPE FLOW VELOCITY(FT/SEC)
C
C VNH3P=4.*FLONH3/(3600.*RONH3P*PIE*DIMH3**2)
C
C REYNOLDS NUMBER FOR NH3 PIPE FLOW
C
C RNH3P=3600.*RONH3P*VNH3P*DIMH3/VSNH3P
C
C FRICTION FACTOR LAMINAR FLOW
C
C IF (RNH3P.GT.2300.1) GO TO 670
C FFNH3P=64./RNH3P
GO TO 680
CONTINUE

FRICITION FACTOR TURBULENT FLOW USING STREEFER CORR
FFNH3P=1.325/(ALOG(ENH3P/(3.7*DIH3)+5.74/RNH3P**.9))**2

CONTINUE

PIPE FRICTION LOSSES(LBF/IN2)
RKNH3P=FFNH3P*(TLNH3P/DIH3+4.*ELBOW)
DPNH3=RKNH3P*RDNH3P*VNH3P**2/(2.*GC*144.)

DELTA P CORE EVAPORATOR
DPCORE=DEVAP+DSDEN

DELTA P DUE TO PIPING ELEVATION(LBF/IN2)
EZ1=0.
EZ2=TSDE+25
DPELEV=RDNH3P*G*E22-EZ1)/G*144.)

DELTA P PIPE LOSS CALCULATION(LBF/IN2)
DPIN=DPNH3+DPELEV

DELTA P THERMODYNAMICALLY(LBF/IN2)
DPTHER=PEVAP-PCOND

DELTA P NH3 CIRC PUMP (LBF/IN2 OR FT)
DPMPN=DPNH3+DPCORE+DPELEV+DPTHER
DPMPNC=144.*DPMPN*GC/(RDNH3P*G)

POWER NH3 CIRC PUMP (HP)
EPNH3C=EPNH3/100.
PWRNP=FLON3*DPMPNC*G/(EPNH3C*HPC*3600.*GC)

POWER NH3 CIRC PUMP (WAT/HR)
EMNH3C=EMNH3/100.
PWRNP=EMNH3C/CFN/EMNH3C

DISCHARGE RATE NH3 CIRC PUMP (FT3/SEC OR GAL/MIN)
DELTA P PIPE LOSSES(LBF/IN2)
DPIPNR=DPNHPR*DPELER
DELTA P NH3 RE-FLUX THERMODYNAMICALLY(LBF/IN2)
USING A SINGLE PHASE PRESSURE MODEL
RONHEV=RNH3(PEVAP)
DPTHRF=2.*EFF*EGF**2*TNET**0.5/(144.*RONHEV*GC)
DELTA P NH3 RE-FLUX PUMP(LBF/IN OR FT)
DPMPNR=DPNHPR*DPELER+DPTHRF
DPPNRC=144.*DPMPNR*GC/(RNH3R*G)
POWER NH3 RE-FLUX PUMP(HP)
EPNHRC=EPNHR/100.
PWRP=FLONHR*DPPNRC*G/(EPNHRC*HPC*3600.*GC)
POWER NH3 RE-FLUX PUMP MOTOR(MW)
EMNHRC=EMNHR/100.
PWRM=PWRP*CMW/EMNHRC
DISCHARGE RATE NH3 RE-FLUX PUMP(FT3/SEC OR GAL/MIN)
QRPMP=PI*DNH3R**2*VNH3RP/4.
QRPMPC=QRPMP*60.*GAL
COST OF NH3 RE-FLUX PUMP($) 
VFR=1./RFNH3(TAVGE)
CNH3RP=(FLONHR*VFR/80100.)*0.64*1.21E+05
PARASITIC PUMP LOSSES(MW)
PARAL=PWRPM+PWRCPM+PWRNP+PWRM
************************************************************
*
* TURBINE AND ELECTRICAL POWER SECTION *
* 
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GROSS ELECTRICAL LOAD(MW)

NET ELECTRICAL OUTPUT DESIRED(ELEC-
PROVIDED IN INITIAL PARAMETERS)(MW)

ELECTRICAL LOADING AS EFFECTED BY EFFICIENCY(MW)

EEPC=EEP/100.
ETRPC=ETRP/100.
WELECT=ELECT/(EEPC*ETRPC)
WELOSS=WELECT-ELECT

GROSS ELECT LOADING INCL PARASITIC LOSSES(MW)

WELEG=WELECT+PARAL

POWER GENERATOR-TURBINE(HP)
PWRTR=1341.0*WELEG

TURBINE EFFICIENCY REQUIREMENT(PCT)

ENTHALPY AT STATE PT 5(BTU/LBM)

H5=H4-(3412.2E+03*WELEG/FLONH3)

CONSTRAINT FOR SAT STATE PT 5

H5G=H5G(RECOND)
DH5=H5G-H5

QUALITY OF NH3 AT STATE PT5(PCT)

H5F=H5F(RECOND)
X5=(H5-H5F)/(H5G-H5F)
X5P=X5*100.

QUALITY OF NH3 EXHAUST AT STATE PT 55(PCT)

$4F=S4F(T4)$
$4G=S4G(T4)$
$4S4F=X4*(S4G-S4F)$
$5F=S5F(T5)$
$5G=S5G(T5)$
$X5S=($4-S5F)/(S5G-S5F)$
$X5SP=X5S*100.$
CONSTRAINT FOR A SAT QUALITY AT STATE PT 5S
DX5=X5-X5S
ENTHALPY AT STATE PT 5S
H5S=H5F+X5S*(H5G-H5F)
TURBINE EFFICIENCY CALCULATION(PCT)
ETURB=(H4-H5)/(H4-H5S
ETURBP=ETURB*100.
COST OF NH3 TURBINE- GENERATOR($)

PPF1.447
DF=2.0
CTURB=(0.375*(WELECG*1000.)/(136000*DF))*PPF*2.42E+06
CGEN=(WELECG*.023+.31)*1.21E+06
CELECT=CGEN*CTURB

***************
* CONDENSER SECTION *
*
***************

AMOUNT OF HEAT REJECTION(BTU/HR OR MW)
QC=FLONH3*(H5-H1)
CMIN FOR CONDENSER(BTU/HR.F)
CPSWC=CPSWITBLKC)
CMINC=FLOCP*CPSWC
COND SW OUTLET TEMP(F OR C)
TCOC=TCIC+QC/CMINC
TOTAL NUMBER OF CONDENSER TUBES
RHOSWC=RHOSWITBLKC)
TNCT=4.*FLOCP/(3600.*RHOSWC*PIE*TDICC**2*VSWC)
C TUBE SHEET DIAMETER(FT)
C TUBE PROFILE - STAGGERED
C IF (PROF.EQ.2.) GO TO 710
C H=CTR*TDCC*0.5
C BASE=CTR*TDCC*0.866
C AREA=CTR*CBASE*2.
C NCC=2.*CTR
C GO TO 720
C CONTINUE
C TUBE PROFILE - IN-LINE
C CPLONG=CTR*TDCC
C PLAT=CPLONG*CPLAT
C AREA=CPLONG*CPLAT
C CONTINUE
C CAREA=CTAREA*TNTC
C TSO=(14.*CAREA/PIE)**0.5/12.
C LOG MEAN TEMP DIFFERENCE OF CONDENSER(F OR C)
C CLMTD=(T1-TCOC-(T1-TCIC))/ALOG((T1-TCOC)/(T1-TCIC))
C COND SW AVG BULK TEMP(F)
C RTBLKC=TCOC+TCIC/2.
C TEST FOR SAT TBLKC
C DBLKTC=ABS(TBLKC-RTBLKC)
C SCALBC=ABS(TBLKC)
C IF (SCALBC.LT.0.1) SCALBC=0.1
C DLBC=DBLKTC/SCALBC
C IF (DLBC.LT.0.001) GO TO 730
C TBLKC=RTBLKC
C GO TO 720
C CONTINUE
C TBLKC=RTBLKC
C TBLKCR=TBLKC+459.69
C CONDENSER CONDUCTANCE(BTU/HR.F)
C UAC=QC/CLMTD
C NUMBER OF TRANSFER UNITS FOR COND(NTU)
C CNTU=UAC/CMINC
C COND EFFECTIVENESS(EPSON)
C EPSC=1.-EXP(-CNTU)
C INITIALLY ASSUME A TLC
C OVERALL HEAT TRANSFER COEFFICIENT(BTU/HR.FT2.F OR W/M2)
C INITIALLY ASSUME HTNH3
C
740  HTNH3C=1000.
740  CONTINUE
C REYNOLDS NUMBER FOR COND TUBESIDE
C (PROPERTIES EVAL AT TBULK)
C VISSWC=VISSW(TBLKC)
C RHOSWR=RHOSW(TBLKC)
C RNSWC=3600.*(RHOSWR*VSWC*TDICC/VISSWC
C PRANDTL NUMBER FOR COND TUBESIDE
C CPSWC=CPSW(TBLKC)
C TKSWC=TKSW(TBLKC)
C PNSWC=CPSWC*VISSWC/TKSWC
C HEAT TRANSFER COEF FOR COND SW TUBESIDE
C (BTU/HR.FT2.F OF W/M2)
C IF (RNSWC.GT.2300.) GO TO 750
C LAMINAR FLOW USING SEIDEN-TATE CORRELATION(ASSUME
C VISCOSITY(TBULK)=VISCOSITY(TWALL)
C HTSWC=1.86*TKSWC*(RNSWC*PNSWC)**.3333*(TDICC/TLC)**.3333/TDICC
C GO TO 760
750  CONTINUE
C TURBULENT FLOW USING DITTUS-BOELTER CORRELATION
C HTSWC=.023*TKSWC*RNSWC**.8*PNSWC**.3/TDICC
C 760  CONTINUE
C FILM TEMP FOR PROPERTY EVALUATION
C THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE
C UA = OUTSIDE(BTU/HR.F)
THERMAL RESISTANCE SW(HR.F/BTU)
CTR1C=1/(EFFIC*HTSWC*PIE*TDICC*TLC)
THERMAL RESISTANCE FOR SW FOULING(HR.F/BTU)
HTSWC=1./SWF
CTR2C=1./(EFFIC*HTSWC*PIE*TDICC*TLC)
THERMAL RESISTANCE FOR WALL THICKNESS(HR.F/BTU)
CTR3C=ALOG(TDOCC/TDICC)/(2.*PIE*TKW*TLC)
THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)
(CONSIDERED NEGLIGIBLE)
THERMAL RESISTANCE FOR NH3(HR.F/BTU)
CTR5C=1./(EFFOC*HTNH3C*PIE*TDOCC*TLC)
HEAT TRANSFERED PER TUBE(BTU/HR)
QCT=(T1-TBLKC)/(CTR1C+CTR2C+CTR3C+CTR5C)
TUBE SIDE WALL TEMP(F)
CTWL=TBLKC+QCT*(CTR1C+CTR2C)
SHELLSIDE WALL TEMP(F)
CTW2=TBLKC+QCT*(CTR1C+CTR2C+CTR3C)
COND FILM TEMP CALCULATION(F)
CFT=(CTW2+T1)/2.
CONDENSER DELTA T TEMP(F)
DELTC=T1-CTW2
COND SHELLSIDE HEAT TRANSFER COEF(BTU/HR.FT2.F OR W/M2)
VSNH3C=VSNH3(CFT)
RONH3C=RFNH3(CFT)
TKNH3C=TKNH3(CFT)
REYNOLDS NUMBER (PSEUDO-HORIZ COND)
WC=FLONH3/TNCT
RNH3CH=2.*WC/(TLC*VSNH3C)
REYNOLDS NUMBER (PSEUDO-VERT COND) REF: MCADAMS
RNH3CV=4.*WC/(PIE*TDCC*VSNH3C)
CONDENSER TYPE IDENTIFICATION
TYPEC=1 HORIZONTAL CONDENSER
TYPEC=2 VERTICAL CONDENSER
IF (TYPEC.GT.1.) GO TO 780
HORIZONTAL TYPE CONDENSER
IF (RNH3CH.GT.2100.) GO TO 770
LAMINAR FLOW USING NUSSELT CORRELATION
MNH3CR=.95*(((3600.*2*TKNH3C**3*RONH3C**2*G*TLC)/(VSNH3C*WC))**.33
1333 GO TO 800 CONTINUE
770 TURBULENT FLOW USING NUSSELT CORR MOD BY MCADAMS
MNH3CR=1.045*(((3600.*2*TKNH3C**3*RONH3C**2*G*TLC)/(VSNH3C*WC))**.33
1333 GO TO 800 CONTINUE
780 VERTICAL TYPE CONDENSER
IF (RNH3CV.GT.1800.) GO TO 790
LAMINAR FLOW USING NUSSELT CORR MOD BY MCADAMS
CORRF=1.28
MNH3CR=CORRF*1.47*(VSNH3C**2/(3600.*2*TKNH3C**3*RONH3C**2*G))**(-1.3333)*RNH3CV**(-.3333)
1333 GO TO 800 CONTINUE
790 TURBULENT FLOW USING KIRKBRIDE CORRELATION
MNH3CR=.0077*(VSNH3C**2/(3600.*2*TKNH3C**3*RONH3C**2*G))**(-.3333)
1)WHN3CV**.4
800 CONTINUE
C TEST FOR SAT HTNH3C
C DHNH3C=ABS([HNN3CR-HTNH3C]
SCALC=ABS([HNN3CR])
IF (SCALC.LT.0.1) SCALC=0.1
DELC=DNH3C/SCALC
IF (DELC.LT.0.001) GO TO 810
HTNH3C=HNN3CR
GO TO 740
810 CONTINUE
HTNH3C=HNN3CR
C PRESSURE DROP ACROSS CONDENSER SHELLSIDE(LBF/IN2)USING
C TWO-PHASE MODEL(HOMOGENEOUS)
C MAX VELOCITY THRU MINIMUM FLOW AREA FTUBE PROFILE
RONH3T*RGNH3(PCOND)
VNH3C*4.*FLONH3/(3600.*RONH3T*PIE*TSOC**21
VMAXC=VNH3C*(SNC/(SNC-TDOC))
C REYNOLDS NUMBER FOR MAXIMUM SHELLSIDE FLOW
C REMAXC=3600.*RONH3C*VMAXC*TDQC/112.*VSNH3C)
C EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
C IF (PROF.EQ.2.) GO TO 820
CFF=10.25+0.118)/(1SNC-TDOC)/TDQC)**1.08*REMAXC**(-.16)
GO TO 830
820 CONTINUE
CFF=10.44+0.08*SNC/TDOC)**1.043+1.13*TDQC/SNC)
1*REMAXC**(-1.51)
830 CONTINUE
C MASS VELOCITY FOR MINIMUM FREE-FLOW AREA(LBM/FT2.SEC)
C IF (TYP0C.GT.1.) GO TO 840
CAF=TSOC*ILC
CL=TSDC
GO TO 850
840 CONTINUE
BAND=ILC
CAF=PIE*TSDC*BAND
CL=ILC
850 CONTINUE
CAFF = CAFF*(1/(SNC - TDCC)*SNC)
CGF = FLOHM3/(3600.*CAFF)

CALCULATION OF CONDENSER SHELLSIDE PRESSURE DROP(LBF/IN2)
USING THE HOMOGENEOUS TWO-PHASE MODEL

EDC = (CPRTODC - TDCC)/12.
VLIQC = 1./RFNM3[T5]
VAPC = 1./RGNM3(PCOND)
VAVGC = VLIQC*1.**5*(VAPC - VLIQC)/VLIQC
CFRICT = [CGF - CGF**2*VAVGC*CL]/(144.*EDC*2.*GC)
CHOM = [CGF**2*VAVGC]/(144.*GC)
CELEV = [CGF*CL]/(144.*GC)
SCOND = CFRICT + CHOM + CELEV

PROPERTIES AT STATE PT 1
P1 = PCOND - OSCOND
HI = HF(P1)
T1R = TSAT(P1)

TEST FOR SAT TL(DEG F)

OTEMP = ABS(T1R - T1)
SCALT1 = ABS(T1R)
IF (SCALT1.LT.0.1) SCALT1 = 0.1
DELT4 = OTEMP/SCALT1
IF (DELT4.LT.0.001) GO TO 860
T1 = T1R
GO TO 260

860 CONTINUE
T1 = T1R

THERMAL RESISTANCES FOR OVERALL HEAT TRANSFER COEF - U
OUTSIDE(HR.FT2.F/ BTU)

OUTSIDE TUBE SURFACE AREA(FT2)
CAO = PIE*TDCC*TLCC

THERMAL RESISTANCE FOR SW (HR.FT2.F/ BTU)
TR1C = CAO*CTR1C

THERMAL RESISTANCE FOR SW FOULING (HR.FT2.F/ BTU)
TR2C = CAO*CTR2C
C THERMAL RESISTANCE FOR WALL THICKNESS(HR.*FT2.*F/BTU)
C TR3C=CA0+CTR3C
C THERMAL RESISTANCE FOR NH3 FOULING(HR.*FT2.*F/BTU)
(CONSIDERED NEGLIGIBLE)
C THERMAL RESISTANCE FOR NH3(HR.*FT2.*F/BTU)
C TR5C=CA0+CTR5C
C OVERALL HEAT TRANSFER COEF U - OUTSIDE(BTU/HR.*FT2.*F
OR W/M2)
C UC=1./(TRIC+TR2C+TR3C+TR5C)
C PSEUDO HT COEF FOR SW(BTU/HR.*FT2.*F OR W/M2.*C)
C HSWC=1./TRIC
C PSEUDO HT COEF FOR SW FOULING(BTU/HR.*FT2.*F OR W/M2.*C)
C HFSWC=1./TR2C
C PSEUDO HT COEF FOR WALL THICK(BTU/HR.*FT2.*F OF W/M2.*C)
C HWC=1./TR3C
C PSEUDO HT COEF FOR NH3(BTU/HR.*FT2.*F OR W/M2.*C)
C HNH3C=1./TR5C
C TOTAL CONDENSER HEAT TRANSFER AREA(FT2 OR M2)
C THTAC=CNTU*CMINC/UC
C REVISITED CONDENSER TUBE LENGTH(FT)
C TLCR=THTAC/(PIE*TDOCC*TNCT)
C CONSTRAINT FOR A SAT TUBE LENGTH
C DTLC=TLT-TLCR
C COST OF CONDENSER UNIT($)
C IF (TSOC.GT.35.) GO TO 910
CONDENSER TUBE SHEET DIAMETER (10-35) FT

DRILLING TIME/TUBE SHELL THICK (MIN/IN)
DTC = 0.66*(TDCC - 0.5)

THICKNESS OF TUBE SHEET (IN)
TTSC = 0.56*TSDC**0.68

TUBE SHEET LABOR COSTS ($)

CTSCLC = 156.695*(TNCT/9630.)*(DTC/0.66)*(TTSC/4.0)

TUBE SHEET MATERIAL COST ($)

CITMC = 1.89486*TSDC**2.3
IF (TNATL.EQ.1) GO TO 870

TUBE MATERIAL COST ($)

CTMC = (C1*TLCC+C2)*TNCT*(TDCC/1.5)

TUBE INSTALLATION COST ($)

CTIC = 3.4*TNCT*TDCC**0.7
GO TO 880

CONTINUE

TUBE MATERIAL COST ($)

CTMC = (C1*TLCC+C2)*TNCT*(TDCC/1.5)*(1+(1.0*AIN1/100.)*10+(1.0*AIN1/100.)*20)

TUBE INSTALLATION COST ($)

CTIC = 3.4*TNCT*TDCC**0.7*(1.0+(1.0*AIN1/100.)*10+(1.0*AIN1/100.)*20)

CONTINUE

HEAT EXCH SHELL COST ($)

CHSC = 17726.5*(TLC+6.1)/31.1*(TSOC/18.1)**2

NH3 DIST PLATE AND BAFFLES COST ($)

CDPBC = 1.539E-02*DTC*TNCT*TSDC**2

BUSTLE, FLANGES, CHANNELS AND FLOW PLATES COST ($)
C CBFCFC=1185.286*TSDC**2
C HEAT EXCH HEAD COST($)
C CHC=53.240.**(TSDC/18.)**3
C WATER INLETS, NOZZLES AND SUPPORTS COST($)
C CWINSC=10106.475*TSDC
C TUBE WELDING COST($)
C
C CTWC=0.0
C IF (TMATL.EQ.1.0) GO TO 900
C IF (TNCT.GT.36000.) GO TO 890
C CTWC=14.73*TNCT**1.03*(TDOC/1.5)**0.7
C GO TO 900
C 890 CONTINUE
C
C CTWC=0.8797*TNCT**1.3*(TDOC/1.5)**0.7
C CONTINUE
C
C CCOND=(CTSLC+CTSMC+CTMC+CTIC+CHSC+CDPBC+CBFCFC+CHC+CWINSC+CTWC)
C GO TO 900
C CONTINUE
C 910 CONTINUE
C
C CONDENSER TUBE SHEET DIAMETER(35-50)FT
C DRILLING TIME/TUBE SHELL THICKNESS(MIN/IN)
C DTC=0.66*(TDOC-0.5)
C THICKNESS OF TUBE SHEET(IN)
C TTSC=0.56*TSDC**0.68
C TUBE SHEET MATERIAL AND LABOR COST($)
C IF (TMATL.EQ.1.0) GO TO 920
C CTSLC=55.189*TNCT**0.791*TSDC**0.68*DTC
C CTSMC=25.366*TSDC**2.014*TTSC
C GO TO 930
C 920 CONTINUE
C
C CTSLC=73.81*TNCT**0.791*TSDC**0.68*DTC
C CTSMC=354.3*TSDC**1.61*TTSC
C CONTINUE
C IF (TMATL.EQ.1.0) GO TO 940
C TUBE MATERIAL COST($)
C
CTMC=(C1*TC+C2)*TNCT*(TDCC/1.5)
CUBE INSTALLATION COST($)
CTIC=36.542*TNCT*TDCC**0.7
GO TO 950
CONTINUE
C
TUBE MATERIAL COST($)
CTMC=(C1*TC+C2)*TNCT*(TDCC/1.5)*((1.+(1.+AINT/100.))**10+(1.+AINT/100.))**20
CUBE INSTALLATION COST($)
CTIC=36.542*TNCT*TDCC**0.7*(1.+(1.+AINT/100.))**10+(1.+AINT/100.))**120
CONTINUE
HEAT EXCH SHELL COST($)
CHSC=12.544*(TC+C6.)*TSDC**2.06
NHX DIST PLATE AND BAFFLES COST($)
CDPC=9.8252*TNCT**0.978*DTCC
BUSTLE, FLANGES, CHANNELS AND FLOW PLATES COST($)
CBFCC=(383.824*TSDC**2.184
HEAT EXCH HEAD COST($)
CCH=938.62*TSDC**1.43
WATER INLET, NOZZLES AND SUPPORTERS COST($)
CWINCC=(7453.6*TSDC**1.056
TUBE WELDING COST($)
CTWC=0.0
IF (TMTL.EQ.1.0) GO TO 970
IF (TNCT.GT.360000.) GO TO 960
CTWC=14.73*TNCT**1.03*(TDCC/1.5)**0.7
GO TO 970
960 CONTINUE
CTWC=0.8797*TNCT**1.3*(TDGC/1.5)**0.7
CONTINUE
CCOND=(CSTLC+CSTMG+CTMC+CTIC+CHSC+GDPBC+CBFCFC+CHC+GWINSC+CTWC)
CONTINUE

******************************************************************************
**
**  SYSTEM EVALUATION SECTION
**
**
******************************************************************************

PERCENT OF PARASITIC POWER
PPP=(PARAL/MELEC)*100.

THERMODYNAMIC CYCLE EFFICIENCY (PCT)

WNET=MELEC-PARAL
QIN=QE/(3412.2E+03)
TCE=(WNET/QIN)*100.

******************************************************************************
**
**  OBJECTIVE FUNCTION SECTION
**
**
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OBJ=CEVAP+CCOND+CELECT+CESWP+CCSWP+CNH3P
CPKW=OBJ/CELECT*1000.
IF (ICALC.LT.5.) RETURN

INPUT/OUTPUT DATA SI UNIT CONVERSION
CFIC=5.0*(CFI-32.1)/9.
CLMTGC=5.0*CLMTD/9.
CTWGC=5.0*(CTW2-32.1)/9.
DCOND=6.89476*DCOND
DEVAP=6.89476*DEVAP
DELTEC=5.0*DELTAC/9.
DELTCC=5.0*DELTAC/9.
DICPC=.3048*DICP

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1010 GO TO 1020
1010 CONTINUE
1020 WRITE (6,1900)
1040 CONTINUE
1070 CONTINUE
1080 CONTINUE
1090 WRITE (6,1910)
1030 IF (PROF.EQ.2.) GO TO 1030
1040 WRITE (6,1920)
1050 GO TO 1050
1060 WRITE (6,1930)
1070 WRITE (6,1940)
1080 WRITE (6,1950)
1090 WRITE (6,1960)
1100 WRITE (6,1970)
1110 WRITE (6,1980)
1120 WRITE (6,1990)
1130 WRITE (6,2000)
1140 WRITE (6,2010)
1150 WRITE (6,2020)
1160 WRITE (6,2030)
1170 WRITE (6,2040)
1180 WRITE (6,2050)
1190 WRITE (6,2060)
1200 WRITE (6,2070)
1210 WRITE (6,2080)
1220 WRITE (6,2090)
1230 WRITE (6,2100)
1240 WRITE (6,2110)
1250 WRITE (6,2120)
1260 WRITE (6,2130)
1270 WRITE (6,2140)
1280 WRITE (6,2150)
1290 WRITE (6,2160)
1300 IF (TYPE.EQ.2.) GO TO 1070
1310 WRITE (6,2170)
1320 GO TO 1080
1330 WRITE (6,2180)
1340 WRITE (6,2190)
1350 WRITE (6,2200)
1360 WRITE (6,2210)
1370 WRITE (6,2220)
WRITE (6, 2230) FLON43, FLNH3C
WRITE (6, 2240) PCOND, FCND
WRITE (6, 2250) T5, T5C
WRITE (6, 2260) T1, T1C
WRITE (6, 2270) DSCOND, DSCNC
WRITE (6, 2280) TDOC, TDOC
WRITE (6, 2290) TDC, TDC
WRITE (6, 2310) TCR, TCR
IF (TMAT.EQ.2.) GO TO 1030
WRITE (6, 2320) GO TO 1100
1090 CONTINUE
WRITE (6, 2330)
1100 CONTINUE
IF (PROF.EQ.2.) GO TO 1110
WRITE (6, 2340) GO TO 1120
1110 CONTINUE
WRITE (6, 2350)
1120 CONTINUE
WRITE (6, 2360) CPR
IF (EMF.EQ.2.) GO TO 1130
WRITE (6, 2370) GO TO 1140
1130 CONTINUE
WRITE (6, 2380)
1140 CONTINUE
WRITE (6, 2390) VSWC, VSWCC
WRITE (6, 2400) CTWC, CTWCC
WRITE (6, 2410) CFT, CFTC
WRITE (6, 2420) DELTAC, DELTCC
WRITE (6, 2430) CLMTDC, CLMTDC
WRITE (6, 2440) EPC
WRITE (6, 2450) CNTU
WRITE (6, 2460) UCC
WRITE (6, 2470) HSWC, HSWCC
WRITE (6, 2480) HFSWC, HFSWCC
WRITE (6, 2490) HNC, HNC
WRITE (6, 2500) HN3C, HN3CC
WRITE (6, 2510) HTAC, HTACC
WRITE (6, 2520) TSDC, TSDCC
WRITE (6, 2530) TNCT
WRITE (6, 2540) DCON, DCOND
WRITE (6, 2550) DIHP, DIHP
WRITE (6, 2570) TLHP, TLHP
WRITE (6, 2580) VSHP, VSHPC
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A2248
A2249
A2250
A2251
A2252
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A2256
WRITE (6,3070) PWRCP, PWRCP
WRITE (6,3080) PWRNP, PWRNP
WRITE (6,3090) PWRNP, PWRNP
WRITE (6,3100) ELECT
WRITE (6,3110) PPE
WRITE (6,3120) TCE
WRITE (6,3130) CEVAP
WRITE (6,3140) CCONP
WRITE (6,3150) CTURB
WRITE (6,3160) CGEN
WRITE (6,3170) CESNP
WRITE (6,3180) CH3P
WRITE (6,3190) CCH3P
WRITE (6,3200) OBJ
WRITE (6,3210) CPKW
RETURN

1150 FORMAT (1HY,19X,21INITIAL DESIGN VALUES)
1160 FORMAT (1HY,2X,23HEVAPORATOR - HORIZONTAL)
1170 FORMAT (1HY,2X,24HEVAPORATOR - VERTICAL)
1180 FORMAT (1HY,4X,9HTUBE O.D., 7X, F8.3, 4H(IN), 12X, F8.3, 4H(MM))
1190 FORMAT (1HY,4X,11HTUBE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))
1200 FORMAT (1HY,4X,11HSW TUBE VEL., 5X, F8.3, 6H(FT/S), 10X, F8.3, 5H(M/S))
1210 FORMAT (1HY,4X,13HOPER PRESSURE, 3X, F8.3, 9H(LBF/IN2), 7X, F8.3, 5H(MPA))
1220 FORMAT (1HY,4X,24HTUBE MATERIAL - ALUMINUM, //, 6X, 15HTHERMAL COND(K
1230 FORMAT (1HY,4X,24HTUBE MATERIAL - TITANIUM, //, 6X, 15HTHERMAL COND(K
1240 FORMAT (1HY,4X,21HTUBE PROFILE - STAGGERED EQUI-LATERAL)
1250 FORMAT (1HY,4X,23HTUBE PROFILE - IN-LINE EQUI-SIDED)
1260 FORMAT (1HY,6X,11PITCH RATIO, 6X, F5.2)
1270 FORMAT (1HY,4X,24HENHANCEMENT - PLAIN TUBE)
1280 FORMAT (1HY,4X,25HENHANCEMENT - LINDE-PROMOTER)
1290 FORMAT (1HY,4X,22HCONDENSER - HORIZONTAL)
1300 FORMAT (1HY,2X,20HCONDENSER - VERTICAL)
1310 FORMAT (1HY,4X,9HTUBE O.D., 7X, F8.3, 4H(IN), 12X, F8.3, 4H(MM))
1320 FORMAT (1HY,4X,11HTUBE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))
1330 FORMAT (1HY,4X,11HSW TUBE VEL., 5X, F8.3, 6H(FT/S), 10X, F8.3, 5H(M/S))
1340 FORMAT (1HY,4X,13HOPER PRESSURE, 3X, F8.3, 9H(LBF/IN2), 7X, F8.3, 5H(MPA))
1350 FORMAT (1HY,4X,24HTUBE MATERIAL - ALUMINUM, //, 6X, 15HTHERMAL COND(K
1360 FORMAT (1HY,4X,24HTUBE MATERIAL - TITANIUM, //, 6X, 15HTHERMAL COND(K
1) \( F_8, 3, 13(BTU/HR.FT.F) 3X, F_8, 3, 7(H/M.C) \)
1792 FORMAT (1HO, 4X, 8HNM3, FLOW, 2X, F14.1, 8H(LBM/HR), 1X, F14.1, 7H(KG/HR))  A2401
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1820 FORMAT (1HO, 6X, 11HOUTLET TEMP., 3X, F8.3, 7H(DEC F), 8X, F8.3, 7H(DEC C))  A2404
1830 FORMAT (1HO, 6X, 11HOUTLET QUALITY, 4X, F5.2, 5H(PCT))  A2405
1840 FORMAT (1HO, 4X, 14HNM3 PRESS DROP, 2X, F8.3, 9H(LBF/IN2), 6X, F8.3, 5H(KPA)  A2406
1850 FORMAT (1HO, 4X, 20MTUBE CHARACTERISTICS)  A2407
1860 FORMAT (1HO, 6X, 10HOUTTER DIA., 4X, F8.3, 4H(IN), 11X, F8.3, 4H(MM))  A2408
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1890 FORMAT (1HO, 6X, 10HNM3 MATERIAL - ALUMINUM)  A2411
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2030 FORMAT (1HO, 4X, 11HEVAP HT COEF., 5X, F8.2, 14H(BTU/HRFT2.F), 1X, F8.2, 8H  A2425
12W/M2.C)  A2426
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12W/M2.C)  A2430
2070 FORMAT (1HO, 6X, 10HNM3 AMMONIA, 4X, F8.2, 14H(BTU/HRFT2.F), 1X, F8.2, 8H 12W/M2.C)  A2431
2080 FORMAT (1HO, 4X, 10HNM3 SURFACE, 2X, F12.2, 5H(FT2), 6X, F12.2, 4H(M2)  A2432
2090 FORMAT (1HO, 4X, 14TUBE SHEET DIA., 2X, F8.3, 4H(FT2), 11X, F8.3, 3H(M))  A2433
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2920 FORMAT ((H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN))) A2547
2930 FORMAT ((H0,6X,1OHEFFICIENCY,1X,4HEMCH,2X,F5.2,5H(PCT),7X,5HMOTOR,11X,F5.2,5H(PCT))) A2548
2940 FORMAT ((H0,6X,2OHAMMONIA RE-FLUX PUMP)) A2549
2950 FORMAT ((H0,6X,10HEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(4))) A2550
2960 FORMAT ((H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN))) A2551
2970 FORMAT ((H0,6X,1OHEFFICIENCY,1X,4HEMCH,2X,F5.2,5H(PCT),7X,5HMOTOR,11X,F5.2,5H(PCT))) A2552
2980 FORMAT ((H0,6X,12HGEN-TURB EFFICIENCIES)) A2553
2990 FORMAT ((H0,6X,14HGEN MECH/ELECT,5X,F5.2,5H(PCT))) A2554
3000 FORMAT ((H0,6X,9HTURB MECH,9X,F5.2,5H(PCT))) A2555
3010 FORMAT ((H0,6X,13HTURB INTERNAL,5X,F5.2,5H(PCT))) A2556
3020 FORMAT ((H0,6X,19HTURB OUTLET QUALITY,1X,F5.2,5H(PCT))) A2557
3030 FORMAT ((H0,6X,18POWER REQUIREMENTS)) A2558
3040 FORMAT ((H0,6X,14HTURB-GEN GROSS,10X,F8.3,4H(HP),12X,F8.3,3H(NW))) A2559
3050 FORMAT ((H0,6X,17HEFFICIENCY LOSSES,2IX,F8.3,4H(HP),12X,F8.3,3H(NW))) A2560
3060 FORMAT ((H0,6X,12HEVAP SW PUMP,4X,F8.3,4H(HP),12X,F8.3,3H(NW))) A2561
3070 FORMAT ((H0,6X,12COND SW PUMP,4X,F8.3,4H(HP),12X,F8.3,3H(NW))) A2562
3080 FORMAT ((H0,6X,13NH3 CIRC PUMP,3X,F8.3,4H(HP),12X,F8.3,3H(NW))) A2563
3090 FORMAT ((H0,6X,16NH3 RE-FLUX PUMP,8X,F8.3,4H(HP),12X,F8.3,3H(NW))) A2564
3100 FORMAT ((H0,6X,1H)) A2565
3110 FORMAT ((H0,6X,10X,16HRET POWER OUTPUT,18X,F8.3,3H(NW))) A2566
3120 FORMAT ((H0,6X,12PERCENT PARASITIC POWER,10X,F5.2,5H(PCT))) A2567
3130 FORMAT ((H0,6X,29HHEMODYNAMIC CYCLE EFFICIENCY,4X,F5.2,5H(PCT))) A2568
3140 FORMAT ((H0,6X,18HHEMODYNAMIC COMPONENTS)) A2569
3150 FORMAT ((H0,6X,12HEVAPORATOR,1X,F12.2,9H(DOLLARS))) A2570
3160 FORMAT ((H0,6X,19HCAPACITY,15X,F12.2,9H(DOLLARS))) A2571
3170 FORMAT ((H0,6X,11HTURB TURBINE,13X,F12.2,9H(DOLLARS))) A2572
3180 FORMAT ((H0,6X,9HCAPACITY,15X,F12.2,9H(DOLLARS))) A2573
3190 FORMAT ((H0,6X,12HEP SW PUMP,12X,F12.2,9H(DOLLARS))) A2574
3200 FORMAT ((H0,6X,12COND SW PUMP,12X,F12.2,9H(DOLLARS))) A2575
3210 FORMAT ((H0,6X,13NH3 CIRC PUMP,11X,F12.2,9H(DOLLARS))) A2576
3220 FORMAT ((H0,6X,16NH3 RE-FLUX PUMP,8X,F12.2,9H(DOLLARS))) A2577
3230 FORMAT ((H0,6X,20X)) A2578
3240 FORMAT ((H0,6X,12HHPYRAM COST,6X,F12.2,9H(DOLLARS))) A2579
3250 FORMAT ((H0,6X,22HHPYRAM COST PER NET KW OUTPUT,4X,F12.2,17H(DOLLARS))) A2580

END
RETURN
END

C SALT WATER DENSITY (DEPTH) (LBM/FT3)

FUNCTION RHOSWD (Z)
RHOSWD=6.18049+9.8878E-05*Z
RETURN
END

C SALT WATER SPECIFIC HEAT (BTU/LBM.F)

FUNCTION CPSW (T)
CP=0.947314+3.8183E-04*T-8.023E-06*T**2+28.468E-08*T**3-3.261E-10
1*T**4
RETURN
END

C SALT WATER VISCOSITY (LBM/HR.FT)

FUNCTION VISSW (T)
VISSW=8.806895-1.988746*T+2.567299E-03*T**2-1.804352E-05*T**3+5.26
1E-08*T**4
RETURN
END

C SALT WATER THERMAL CONDUCTIVITY (BTU/HR.FT.F)

FUNCTION TKSW (T)
TKSW=0.2787615+2.62894E-03*T-5.2014E-05*T**2+5.6873E-07*T**3-2.330
191E-09*T**4
RETURN
END

C AMMONIA SAT TEMP (F)

FUNCTION TSAT (P)
TSAT=-28.505313+1.2445557*P-.0049090885*P**2+.000009203093*P**3
RETURN
END

C ENTHALPHY OF SAT AMMONIA VAPOR (BTU/LBM)

FUNCTION HG (P)
HG=603.58813*+3893361*P-.002002459*P**2+4.01727E-06*P**3
RETURN
END

C ENTHALPHY OF SAT AMMONIA LIQUID (BTU/LBM)

FUNCTION HF (P)
HF=9.739724*+1.3956686*P-.005461178*P**2+1.0190106E-05*P**3
RETURN
END

C VISCOSITY OF SAT AMMONIA LIQUID (LBM/HR.FT)
FUNCTION VSNH3 (T)
VSNH3=.55203968261-.003405489426*T+1.01587304E-05*T**2-1.5046298E-
10*B*T**3
RETURN
END

FUNCTION CPNH3 (T)
CPNH3=1.0829682542+.0041058198*T+1.7658742E-06*T**2+1.6203699E-08
1*T**3
RETURN
END

FUNCTION TKNH3 (T)
TKNH3=.33917857143-.000720535715*T-2.2321429E-07*T**2
RETURN
END

FUNCTION SF (T)
SF=.09771914+.002334056*T-1.66882E-06*T**2+2.1649E-09*T**3
RETURN
END

FUNCTION SG (T)
SG=1.33473105-.001973293*T+.0808E-06*T**2-7.5887E-09*T**3
RETURN
END

FUNCTION RFNH3 (T)
RFNH3=41.415499+.0490092*T+.00039001*T**2-5.2019E-07*T**3
RETURN
END

FUNCTION RGNH3 (P)
RGNH3=.01242535+.0032785773*P-2.52513E-07*P**2+.9285E-10*P**3
RETURN
END

TITANIUM TUBE WALL THICKNESS - PLAIN(INCH)
FUNCTION TP (D)
TP=.007966159+.040237327*D-.035418004*D**2+.015039976*D**3-.002418
12547*D**4
RETURN
END

C

TITANIUM TUBE COST/FT - PLAIN($/FT)

FUNCTION TC (D)
TC=3.43889-8.9432814*D+10.9182422*D**2-4.68667*D**3+7.2236862*D**4
RETURN
END

C

TITANIUM TUBE WALL THICKNESS - LINDE-PROMOTER(INCH)

FUNCTION TE (D)
TE=.04163034-.03507642*D+.0239336*D**2-.00178528*D**3-.000241899*D
RETURN
END

C

EVAP TITANIUM TUBE COST/FT - LINDE-PROMOTER($/FT)

FUNCTION TEC (D)
TEC=1.51007568-1.001975385*D+1.723921769*D**2
RETURN
END

C

COND TITANIUM TUBE COST/FT - LINDE-PROMOTER($/FT)

FUNCTION TCC (D)
TCC=-1.836825813+3.7220155039*D
RETURN
END

C

ALUMINUM TUBE WALL THICKNESS - PLAIN(INCH)

FUNCTION AP (D)
AP=0.065
RETURN
END

C

ALUMINUM TUBE COST/FT - PLAIN($/FT)

FUNCTION AC (D)
AC=+.09288381+1.2279558*D+.9161318*D**2+.51160016*D**3-.09134991*D
RETURN
END

C

ALUMINUM TUBE WALL THICK - LINDE-PROMOTER(INCH)

FUNCTION AE (D)
AE=.15958172-.2879843*D+.2942726*D**2-.1182109*D**3+.01837821*D**4
RETURN
END

C

EVAP ALUMINUM TUBE COST/FT - LINDE-PROMOTER($/FT)
FUNCTION AEC (D)
AEC = 2.157473 - 5.206014*D + 7.336678*D**2 - 3.656495*D**3 + 7.204888*D**4
RETURN
END

COND ALUMINUM TUBE COST/FT - LINDE-PROMOTER($/FT)

FUNCTION ACC (D)
ACC = 1.799865 - 4.543195*D + 6.556578*D**2 - 3.305876*D**3 + 6.622916*D**4
RETURN
END
LIST OF REFERENCES


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