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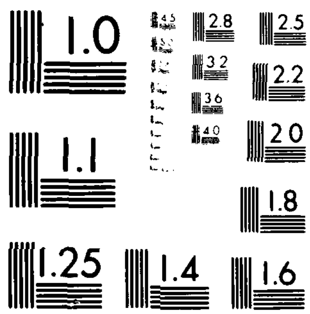

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THESIS

OPTIMIZATION OF A LOW ΔT RANKINE POWER SYSTEM

by

Raymond C. Schaubel

December 1980

Thesis Advisor: R. H. Nunn

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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

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

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
$\dot{E}$	power
f	friction factor
F	correction to LMTD
G	mass velocity
g	acceleration of gravity
g_c	conversion factor ($32.2\ lb_m \cdot ft / lb_f \cdot sec^2$)
h	specific state point enthalpy
$\bar{h}$	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

U	overall heat transfer coefficient
v	specific volume
V	velocity
X	quality of working fluid
Z	elevation
ϵ	heat exchange effectiveness
η	efficiency
ρ	density
μ	absolute or dynamic viscosity

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.

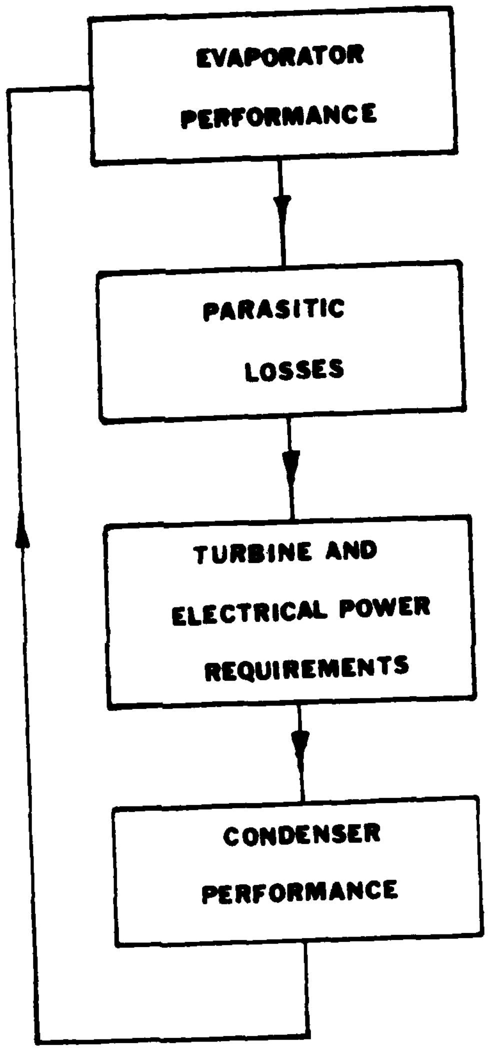


Figure 1. Power System Sequential Analysis

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

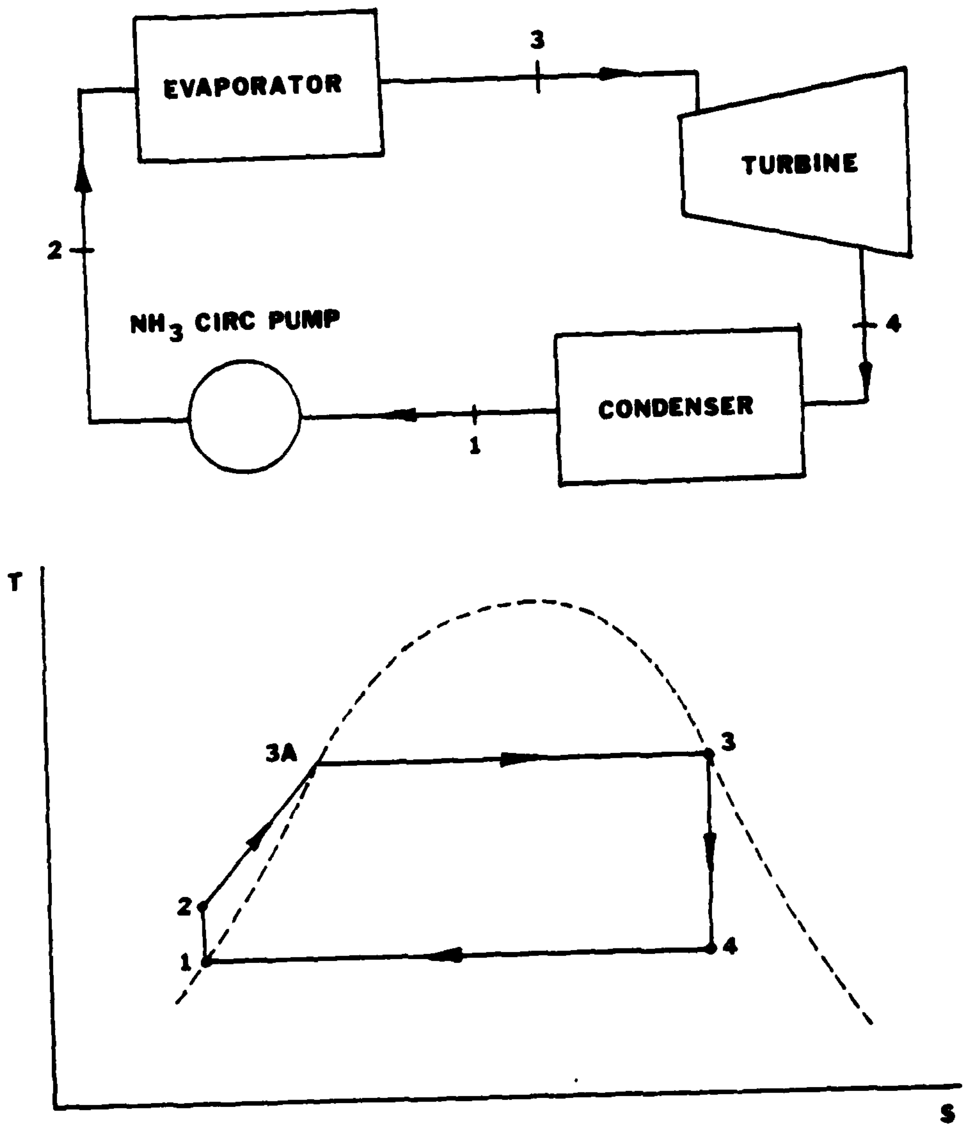


Figure 2. Idealized OTEC Rankine Cycle

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

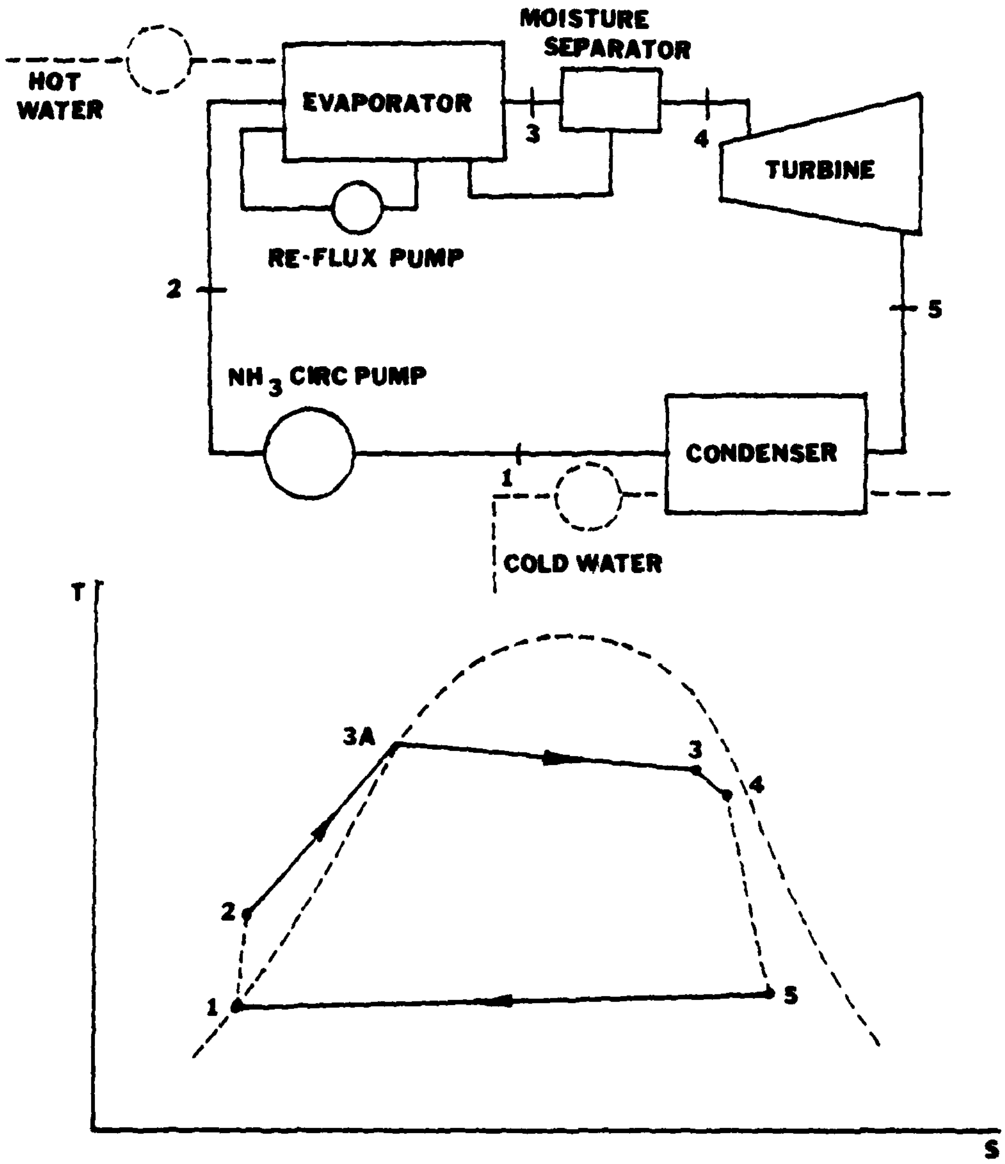


Figure 3. Actual OTEC Rankine Cycle

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 NH_3 piping.
  - Inner diameter of NH_3 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).

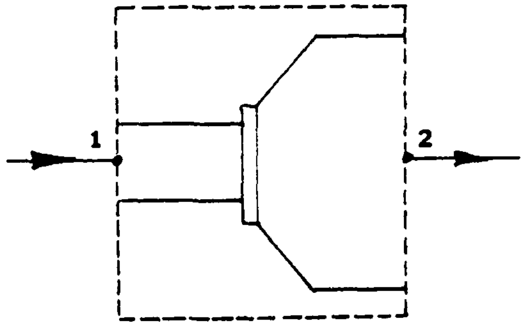
. Heat exchanger cost analysis.

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)



$$\dot{m}_1 \text{ (HOT PIPE)} = \dot{m}_2 \text{ (EVAPORATOR)}$$

and $$\dot{m} = \rho_{sw} \, A \, V \tag{1}$$

where A = cross-sectional area of the hot pipe.

V = salt water velocity through hot pipe.

ρ_{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. <u>Total Number of Evaporator Tubes, N_t</u>

Using equation (1), it follows that

$$\dot{m}_1 = \rho_{sw} \frac{\pi d_i^2}{4} V_t \, N_t \tag{2}$$

where ρ_{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 otpimization 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

$$A_t = \pi d_o L_t N_t \qquad (3)$$

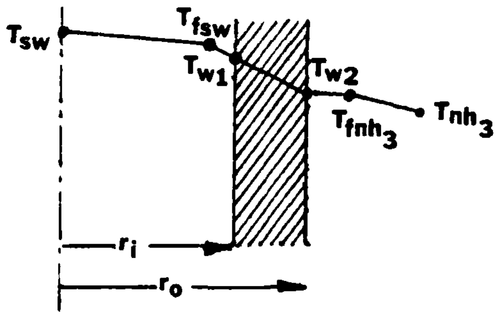
As previously, the outer tube diameter and tube length are initializing conditions and will be treated as design variables.

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

$$U_o = \frac{1}{\frac{A_o}{\eta_i h_{sw} A_i} + \frac{A_o}{A_i} R_{fsw} + \frac{d_o \ln d_o/d_i}{2K} + R_{fNH_3} + \frac{1}{\eta_o h_{NH_3}}} \tag{4}$$

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).

η_o, η_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 = total inner surface area (including fin and bare tube).[1]

$$\eta_i = 1 - \frac{A_{fni}}{A_i}\left(1 - \eta_{fi}\right)$$

$$\eta_o = 1 - \frac{A_{fno}}{A_o}\left(1 - \eta_{fo}\right)$$

where A_{fni} = total inner fin surface area.

A_i = total inner surface area (including fin and bare tube).

A_{fno} = total outer fin surface area.

A_o = total outer surface area (including fin and bare tube).

η_{fi} = fin efficiency of single interanl fin.

η_{fo} = fin efficiency of single external fin.

a. Tubeside Reynolds Number, Re_d

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

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}} \tag{5}$$

---

[1] Note that this analysis will hereafter consider smooth plain tube configurations only.

where μ_{sw} = dynamic viscosity of salt water.

ρ_{sw} = density of salt water.

Initially, properties are evaluated for

$$T_{BULK} = T_{sw}\ (INLET) \tag{6}$$

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

$$Nu_d = 1.86\left(Re_d\, Pr\right)^{1/3}\left(\frac{d_i}{L_t}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{7}$$

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

Nusselt and Prandtl numbers, Nu_d and Pr, are defined as

$$Nu_d = \frac{h_{sw}\, d_i}{k_{sw}}$$

$$Pr = \frac{C_{P_{sw}}\, \mu_{sw}}{k_{sw}}$$

where μ_{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)

$$\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where μ_w 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

$$Re_d \, Pr \, \frac{d}{L} > 10$$

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

$$Nu_d = 0.023 \, Re_d^{0.8} \, Pr^{0.4} \tag{8}$$

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 .00025 ($hr \cdot ft^2 F^\circ / BTU$) 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_{NH_3}

Initially, h_{NH_3} will be assumed

$$h_{NH_3} = 1000 \ (BTU/hr \cdot ft^2 \cdot F^\circ) \tag{9}$$

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

Using the thermal resistance expressed as

$$R_1 = \frac{d_o}{\eta_i \, h_{sw} \, d_i}$$

$$R_2 = \frac{d_o}{\eta_i h_{tsw} d_i}$$

$$R_3 = \frac{d_o \ln d_o / d_i}{2k}$$

$$R_5 = \frac{1}{\eta_o h_{NH_3}}$$

an initial value for the overall heat transfer coefficient may be calculated.

$$U_o = \frac{1}{R_1 + R_2 + R_3 + R_5} \tag{10}$$

5. <u>NTU-effectiveness Relations</u>

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

$$T_{BULK} = T_{Hi} \; (\text{SW INLET TO EVAP})$$

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

$$NTU = U_o A_t / C_{min}$$

where C_{min} is defined as capacity rate of the single phase flow in an evaporative or condensing two phase flow regime.

$$C_{min} = \dot{m}_{sw} C_{p_{sw}} \qquad (11)$$

Evaporator effectiveness can then be expressed as

$$\epsilon = 1 - e^{(-NTU)} \qquad (12)$$

for two phase flow regardless of the flow geometry.

Using the definition of effectiveness

$$\text{Effectiveness} = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \qquad (13)$$

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\Delta T_{min}}{\Delta T_{max}} = \frac{T_{H_i} - T_{H_o}}{T_{H_i} - T_{C_i}} \qquad (14)$$

The expression for ΔT_{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

$$T_{H_o} = T_{H_i} - (T_{H_i} - T_{C_i})\left(1 - e^{(-NTU)}\right) \qquad (15)$$

Concurrently, a revised evaporator average salt water temperature can be expressed as

$$T_{BULK} = (T_{H_i} + T_{H_o})/2 \tag{16}$$

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. <u>Amount of Heat Absorption, $\dot{Q}$</u>

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

$$\dot{Q} = C_{min}\left(T_{H_i} - T_{H_o}\right) \tag{17}$$

8. <u>Log Mean Temperature Difference, LMTD</u>

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

Using Eq. (17) and the definition of

$$\dot{Q} = U_o A_t F \, LMTD$$

with $\dot{Q}_{max} = C_{min} \, \Delta T_{max}$

the log mean temperature difference across the evaporator may be expressed as

$$LMTD = \frac{C_{min}(1-e^{(-NTU)})(T_{Hi}-T_{Ci})}{U_o A_t F} \tag{18}$$

where T_{Ci} = T_{NH3} evaluated at state point 3.

F = correction factor on $LMTD$, equal to 1 for two phase flow.

9. <u>Film Temperature for Property Evaluation, T_f</u>

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

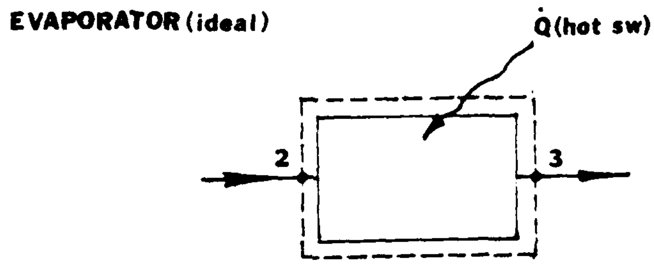
$$T_{W2} = T_{BULK} - \dot{Q}\left(\frac{R_1 + R_2 + R_3}{A}\right)$$

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

$$T_f = \frac{T_{w_2} + T_3}{2} \tag{19}$$

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

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



$$\dot{m}_{NH_3} h_2 + \dot{Q} = \dot{m}_{NH_3} h_3 \tag{20}$$

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

$$h_1 = h_f)_{P1} \qquad T_1 = T_{SAT})_{P1}$$
$$h_3 = h_g)_{P3} \qquad T_3 = T_{SAT})_{P3} \tag{21}$$

where h_1 = represents enthalpy at state point 1 at the suction inlet to the working fluid circulation pump.

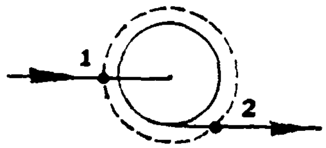
h_3 = represents enthalpy at (ideal)/state point 3 as a saturated vapor.

T_1, T_3 = represent the respective saturation temperatures.

v_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 otpimization 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.

AMMONIA CIRC PUMP



$$\dot{m}_{NH_3} h_1 + \dot{W}_{CP} = \dot{m}_{NH_3} h_2 \tag{22}$$

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

$$\dot{W}_{CP_S} = v_1 (P_2 - P_1) \quad \text{where } P_2 = P_{2S}$$

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

$$\dot{W}_{CP} = \frac{\dot{W}_{CP_S}}{\eta_P} \tag{23}$$

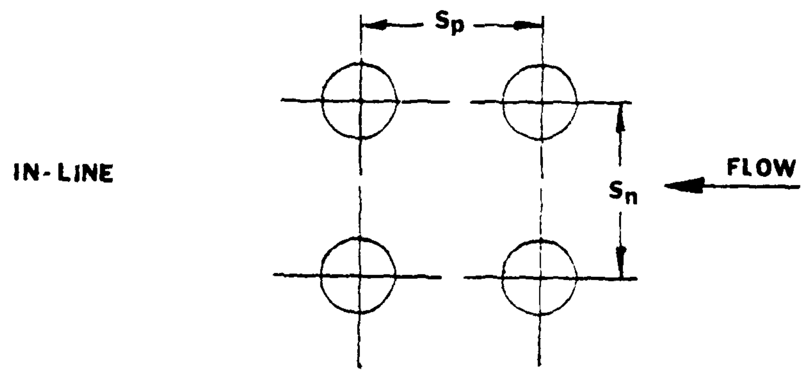
Actual outlet enthalpy at state point 2 may be determined using the results of Eq. (23) with Eq. (22) knowing the enthalpy at state point 1 from Eq. (21).

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:



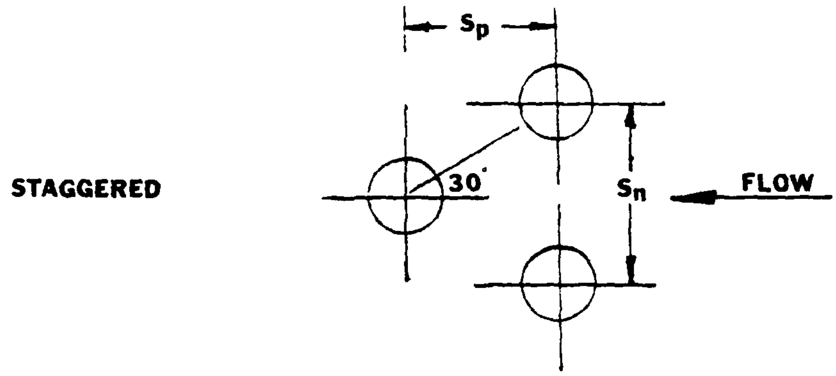
where S_n = pitch ratio x outer tube diameter, equal to Sp.

P_R = pitch ratio; the distance between tube centers with respect to outer tube diameter.

A_p = tube profile area (centerline to centerline) per tube.

$$S_n = P_R\, d_o \tag{24}$$

$$A_p = S_n^2 \tag{25}$$



$$S_n = 2 P_R d_o \sin 30^\circ \tag{26}$$

$$S_p = P_R d_o \cos 30^\circ \tag{27}$$

Therefore, the tube profile area (centerline to centerline) per tube is equal to

$$A_p = S_n S_p \tag{28}$$

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

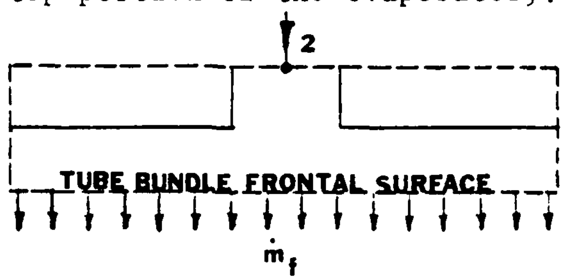
$$\frac{A_{ff}}{A_f} = \frac{S_n - d_o}{S_n} \tag{29}$$

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:

$$N_t A_p = \frac{\pi T_{SD}^{\,2}}{4} \qquad (30)$$

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 $$(A_f)_{LiQ} = A_f \eta$$

where η = percent of tube frontal area which is occupied by droplets.

The mass flow rates are

$$\dot{m}_2 = \rho A_p V_p$$

$$\dot{m}_f = \rho (A_f)_{Liq} V_f$$

where A_p = ammonia pipe cross-sectional area.

V_p = average ammonia velocity in the pipe.

Therefore

$$V_f = \frac{A_p}{(A_f)\eta_l} V_p$$

and since

$$\eta_l \simeq \frac{A_p}{A_f}$$

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.

$$V_p = V_f \tag{31}$$

Thus the assumption that $\eta_l = A_p/A_f$ 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):

$$A_f = T_{sD} L_t$$

$$A_{ff} = A_f \left(\frac{S_n - d_o}{S_n} \right) \tag{32}$$

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

$$G = \dot{m}_{NH_3} / A_{ff}$$

where $\dot{m}_{NH_3}$ represents the average ammonia mass flow rate.

12. <u>Pressure Drop of Two-Phase Flow across a Bank of Tubes, ΔP</u>

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

$$\Delta P_{TOT} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \tag{33}$$

For a given channel length, L_c , the pressure drop components can be represented by

$$\Delta P_{FRICTION} = \frac{f G^2 \bar{v}}{D_e \, 2 g_c} L_c \tag{34}$$

$$\Delta P_{MOMENTUM} = \frac{G^2 \bar{v}}{g_c}$$

$$\Delta P_{ELEVATION} = \frac{g}{\bar{v} \, g_c} L_c$$

and the total pressure drop, ΔP_{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 $L_c = T_{SD}$ (tube sheet diameter).

D_e = equivalent diameter of flow channel, defined by $D_e = P_R d_o - d_o$.

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

$$\bar{v} = v_f \left[1 + \frac{X}{v_f} \left(v_g - v_f \right) \right]$$

where X = quality of mixture (state point 3).

v_f = specific volume of liquid (state point 1).

v_g = 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:

$$f = \left\{ 0.25 + \frac{0.118}{\left[(S_n - d_o)/d_o \right]^{1.08}} \right\} Re_{max}^{-0.16} \tag{35}$$

and for in-line tube arrangements:

$$f = \left\{ 0.044 + \frac{0.08\, S_n/d_o}{\left[(S_n - d_o)/d_o \right]^{0.43 + 1.13\, d_o/S_n}} \right\} Re_{max}^{-0.15} \tag{36}$$

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

$$Re_{max} = \frac{\rho_f V_{max} d_o}{\mu_f} \tag{37}$$

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. <u>Pressure Drop Across the Moisture Separator. $\Delta P_{m.sep}$</u>

This portion of the analysis will simulate the use of a cyclone separator to improve te 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

$$A_{INLET} = 0.1\ T_{SD}\ L_t$$

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

$$\dot{m}_{NH_3} = \rho A_{INLET} V$$

where ρ = 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,

$$\Delta P_{m.SEP} = 20 \rho \frac{V^2}{2 g_c} \tag{38}$$

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

$$P_{3(NEW)} = P_3 - \Delta P_{EVAP} \tag{39}$$

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

$$P_4 = P_{3(NEW)} - \Delta P_{m.SEP} \tag{40}$$

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

$$\begin{aligned} h_{3f(NEW)} &= h_f \big]_{P_{3(NEW)}} & \qquad h_{4f} &= h_f \big]_{P_4} \\ h_{3g(NEW)} &= h_g \big]_{P_{3(NEW)}} & \qquad h_{4g} &= h_g \big]_{P_4} \end{aligned} \tag{41}$$

The subscript (NEW) 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)

$$h_3 = h_{3f} + X_3 (h_{3g} - h_{3f})$$
$$h_4 = h_{4f} + X_4 (h_{4g} - h_{4f}) \qquad (42)$$

15. <u>Revised Ammonia Mass Flow Rate and Velocity</u>

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.

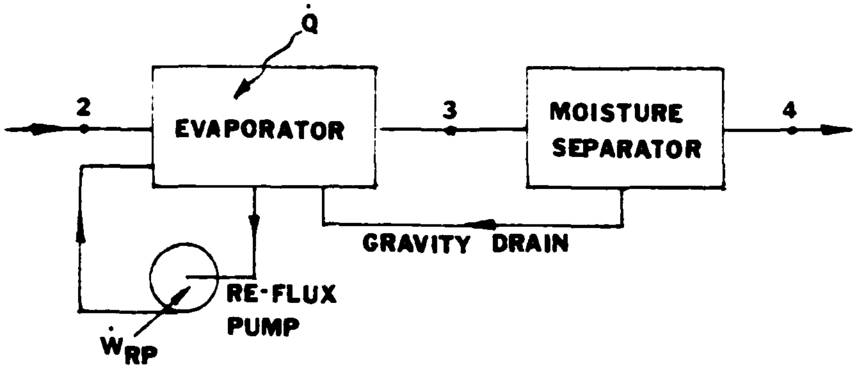
The diagramatic representation that follows better illustrates the heat absorption phase of the OTEC power system and will provide the basis for the analysis and optimization.

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

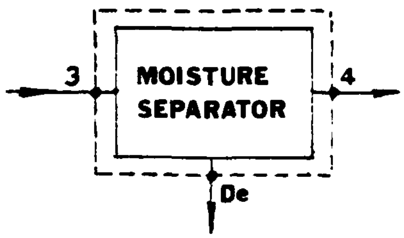
. Steady state.

. 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

$$X3\,\dot{m}_3 = X4\,\dot{m}_4$$

and

$$\dot{m}_3 = \frac{X4}{X3}\,\dot{m}_4 \qquad (43)$$

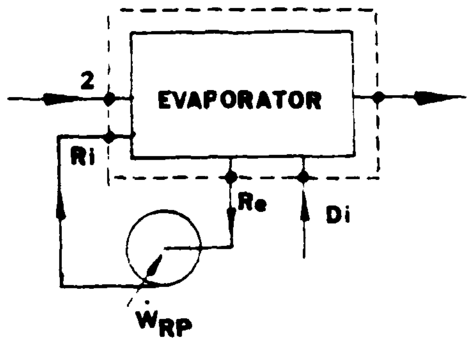
However, for reasons of flow continuity, the mass flow rate through the separator drain must be included in the control volume analysis; therefore

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_{DE} \qquad (44)$$

Substituting Eq. (43) into (44) and solving for $\dot{m}_{De}$, the following expression can be derived

$$\dot{m}_{De} = \left(\frac{x_4}{x_5} - 1 \right) \dot{m}_4 \quad (45)$$

Looking at the evaporator as a separate control volume,



the energy balance is

$$\dot{Q} + \dot{m}_2 h_2 + \dot{m}_{Ri} h_{Ri} + \dot{m}_{Di} h_{Di} = \dot{m}_3 h_3 + \dot{m}_{Re} h_{Re}$$

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

$$\dot{Q} + \dot{m}_2 h_2 + \dot{m}_{De} h_{De} = \dot{m}_3 h_3 \quad (46)$$

where $\dot{m}_{Dl} = \dot{m}_4 \left(\frac{x_3 - x_4}{x_3}\right)$ 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

$$\dot{m}_2 + \dot{m}_{K_i} + \dot{m}_{D_l} = \dot{m}_3 + \dot{m}_{K_e} \tag{47}$$

where $\dot{m}_{K_e} = \dot{m}_{K_i}$.

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

$$[illegible]\,\dot{m}_2 + \left(\frac{x_4}{x_3} - 1\right)\dot{m}_4 + \dot{m}_{Dl} = \dot{m}_2 \frac{x_4}{x_3} h_3 \tag{48}$$

In addition, a mass balance for steady-state, steady-flow indicates that the mass flow rates at state points 2 and 4 are equal and therefore

$$\dot{m}_4 = \dot{m}_2 \tag{49}$$

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,

$$T_3 = T_{SAT}\Big)_{P_3} \tag{50}$$

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

$$\bar{h} = 2.2\left(\frac{H}{d_o}\right)^{0.1}\left(\frac{\mu_f}{g\rho_f^{2}k_f^{3}}\right)^{-1/3}\left(\frac{4\Gamma}{\mu_f}\right)^{-1/3} \tag{51}$$

for turbulent flow

$$\bar{h} = 0.185\left(\frac{H}{d_o}\right)^{0.1}\left(\frac{\mu_f}{g\rho_f^{2}k_f^{3}}\right)^{-1/3}\left(\frac{C_p\mu_f}{k_f}\right)^{0.5} \tag{52}$$

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

Γ = tube flow rate per unit length.

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

$$Re_{TR} = 1680\left(\frac{C_p\mu_f}{k_f}\right)^{-1.5}$$

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

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

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

The overall model for a single tube is expressed as

$$\bar{h} = h_b + h_d \frac{L_d}{L} + h_c \left(1 - \frac{L_d}{L} \right) \tag{53}$$

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

$$h_b = \frac{\mu_f h_{fg}}{C_{sf} \sqrt[3]{\frac{g_c \sigma}{g \rho_f}}} \left(\frac{C_{pf}}{h_{fg} P_r} \right) \Delta T^2 \tag{54}$$

with C_{sf} = function of the fluid-surface combination.

ΔT = wall temperature minus free stream saturation temperature.

σ_f = surface tension

h_d = heat transfer coefficient in the developing region.

$$h_d = \frac{3}{8} c_p \frac{\Gamma}{L_d}$$

$$L_d = \frac{\Gamma^{4/3}}{4 \pi \rho \alpha} \left(\frac{3 \mu_f}{g \rho_f^2} \right)^{1/3}$$

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

$$h_c = 0.821 \left(\frac{\nu^2}{k^3 g} \right)^{-1/3} \left(\frac{4 \Gamma}{\mu_f} \right)^{-0.22} \tag{55}$$

and, for turbulent flow,

$$h_c = 3.8 \times 10^{-3} \left(\frac{\nu^2}{k^3 g}\right)^{-1/3} \left(\frac{4\Gamma}{\mu_f}\right)^{0.4} \left(\frac{\nu}{\alpha}\right)^{0.65} \tag{56}$$

where L = circumferential length of heated surface.

α = thermal diffusivity.

L_d = developing length around tube circumference.

Γ = flow rate per unit axial length of tube.

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

$$L = N_t \pi d_o / 2$$

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

$$Re_{TR} = 5800 \left(\frac{\nu}{\alpha}\right)^{-1.06}$$

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

$$Re = \frac{4\Gamma}{\mu_f} \tag{57}$$

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

$$t_d = 0.66\,(d_o - 0.5) \tag{58}$$

. Thickness of the tube sheet

$$t_{TS} = 0.56\,T_{SD}^{0.68} \tag{59}$$

. Tube sheet labor cost

$$C_{TSL} = 156695\,(N_t/9630)(t_d/0.66)(t_{TS}/4) \tag{60}$$

. Tube sheet material cost

$$C_{TSM} = 189.486\,T_{SD}^{2.3} \tag{61}$$

. Tube installation cost

$$C_{TI} = 34\,N_t\,d_o^{0.7} \tag{62}$$

. Heat exchanger shell cost

$$C_{HXS} = 177265\left(\frac{L_t + 6}{31}\right)\left(T_{SD}/18\right)^2 \tag{63}$$

. Ammonia distribution plate and battles cost

$$C_{DPB} = 93865.75\left(N_t/9630\right)\left(t_d/0.66\right)\left(T_{SD}/18\right)^2 \tag{64}$$

. Bustle, flanges channels and flow plates cost

$$C_{BFCF} = 308550\left(T_{SD}/18\right)^2 \tag{65}$$

. Tube material cost

$$C_{TM} = \left(E1\ L_t + E2\right) N_t \frac{d_o}{1.5} \tag{66}$$

where $E1$ = curve fit of tube cost per foot.

$E2$ = tube machining cost if required

. Heat exchanger head costs

$$C_{HXH} = 53240\left(T_{SD}/18\right)^3 \tag{67}$$

. Water inlet, nozzles and supports cost

$$C_{WNS} = 220310.75\left(T_{SD}/18\right)^2 \tag{68}$$

. Tube welding costs (Titanium tubes)

$$\begin{aligned} &\text{for } N_t \leq 36000 \\ &\quad C_{TW} = 14.73\, N_t^{1.03}\left(d_o/1.5\right)^{0.7} \\ &\text{for } N_t > 36000 \\ &\quad C_{TW} = 0.8797\, N_t^{1.3}\left(d_o/1.5\right)^{0.7} \end{aligned} \tag{69}$$

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

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

where i = projected inflationary rate (input by customer)

. Aluminum tube material cost

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

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)

$$C_{TSL} = 55.189\, N_t^{0.791}\, T_{SD}^{0.68}\, t_d \tag{72}$$

$$C_{TSM} = 29.566\, T_{SD}^{2.014}\, t_d \tag{73}$$

. Tube sheet labor and material cost (aluminum)

$$C_{TSL} = 73.181\, N_t^{0.791}\, T_{SD}^{0.68}\, t_d \tag{74}$$

$$C_{TSM} = 354.3\, T_{SD}^{1.61}\, t_{TS} \tag{75}$$

. Tube installation costs

$$C_{TI} = 36.542\, N_t\, d_o^{0.7} \tag{76}$$

. Heat exchanger shell cost

$$C_{HXS} = 12.544\, (L_t + 6)\, T_{SD}^{2.06} \tag{77}$$

. Ammonia distribution plate and baffle costs

$$C_{DPB} = 158.099\, T_{SD}^{1.82} + 72.419\, N_t^{0.873}\, t_d \tag{78}$$

. Bustle, flanges, channels, and flow plate costs

$$C_{BFCF} = 472.977\, T_{SD}^{2.12} \tag{79}$$

. Heat exchanger head cost

$$C_{HXH} = 1725.31\, T_{SD}^{1.45} \tag{80}$$

. Water inlet, nozzles and support cost

$$C_{WINS} = 7445.297\, T_{SD}^{1.1} \tag{81}$$

. Tube welding costs (titanium tubes)

for $N_t \leq 36000$

$$C_{TW} = 14.73\, N_t^{1.03} (d_o/1.5)^{0.7} \tag{82}$$

for $N_t > 36000$

$$C_{TW} = 0.8797\, N_t^{1.03} (d_o/1.5)^{0.7}$$

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.

IV. PARASITIC LOSSES

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].

$$\int_{o}^{i} \frac{dP}{\rho} + \frac{V_i^2}{2} + g Z_i = \frac{V_o^2}{2} + g Z_o + \dot{W}_s + (\text{Losses})_{i \to o}$$

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).

.. Pumping power requirements (126).

.. Pump cost analysis (127).

. Parasitic pump losses.

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, P_{HP}

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

$$\Delta P = \sum_{i=1}^{n} \rho \frac{K_i V^2}{2 g_c} \tag{83}$$

where K_i describes the resistance coefficient.

V = fluid velocity.

$$K_i = f \frac{L}{D} \tag{84}$$

where f = friction factor.

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

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

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

where ρ_{sw}, μ_{sw} = properties of walt water at the hot pipe inlet temperature (assumed constant throughout the pipe).

V_{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

$$f = \frac{1.325}{\left[\ln(\epsilon/3.7d_i + 5.74/Re_d{}^{0.9}\right]^2} \tag{86}$$

where ϵ = 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].

- $10^{-6} \leq \frac{\epsilon}{D} \leq 10^{-2}$
- $5000 \leq Re_d \leq 10^{8}$

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

$$K = 1.5 \tag{87}$$

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

$$K = \left[1 - \left(d_i / \overline{T_{SD}}\right)^2\right]^2 \tag{88}$$

where $\overline{T_{SD}}$ = 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. (83).

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]

$$K = 0.5 \tag{89}$$

. Assume outlets of evaporator tubing expand to an infinite reservoir [Ref. 10]

$$K = 1.0 \tag{90}$$

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

$$K_{CORE} = f \frac{L_t}{d_i} \tag{91}$$

where L_t, 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

$$\Delta P_{pump} = \Delta P_{PIPE\ SYSTEM} + \Delta P_{EVAP\ DESIGN} \tag{92}$$

converting to pumping head

$$H = \frac{g_c}{\rho_{sw}\, g} \Delta P_{pump} \tag{93}$$

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

$$P_{HP} = \frac{\dot{m}_{sw}}{\eta_P}\left(\frac{gH}{g_c}\right) \tag{94}$$

where η_P = pump mechanical efficiency (designer input).

$\dot{m}_{SW}$ = 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.

$$P_{HP(MW)} = \frac{P_{HP}}{\eta_M} \times \text{CONVERSION FACTOR} \tag{95}$$

where η_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

$$C_{pump} = \left[(D/1000)0.75 + 50\right] 1.21 \times 10^3 \tag{96}$$

where

$$D = \frac{\pi d_i^2}{4} V_{sw}$$

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, P_{cp}

Using Reynolds number

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

where ρ_{sw}, μ_{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].

$$K_{INLET} = 0.5 \tag{97}$$

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

$$K_{PLENUM} = \left[1 + (d_i / T_{SD})^2 \right]^2 \tag{98}$$

where T_{SD} = 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

$$K = f\left(\frac{L_p}{d_i} + \frac{L}{D}\right) + K_{INLET} + K_{PLENUM} \tag{99}$$

where L_p = length of cold pipe.

d_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.

. Assume inlets to evaporator tubing are well rounded.

$$K = 0.5 \tag{100}$$

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

$$K = 1.0 \tag{101}$$

Defining Reynolds number for condenser tubeside flow, while assuming

$$T_{BULK} = T_{COLD\,(INLET)} \tag{102}$$

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

where ρ_{SW}, μ_{SW} = properties evaluated at condenser tubeside bulk temperature (initially assumed equal to cold pipe inlet temperature).

V_{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

$$K_{CORE} = f \frac{L_t}{d_i} \tag{103}$$

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

$$\int_i^e \frac{dP}{\rho g}$$

with a density head defined as[2]

$$H_\rho = Z_e - Z_i + \int_i^e \frac{dP}{\rho g}$$

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

$$H_\rho = Z_e - Z_i + \frac{1}{\rho_0 g}(P_e - P_i)\left[1 - \frac{K_m}{2}(P_e + P_i)\right]$$

where K_m = mean compressibility of walt water, f(salinity, temperature and pressure).

ρ_0 = 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

$$H_\rho = (Z_e - Z_i) - \frac{1}{\rho_0}\int_{Z_i}^{Z_e} \rho(z)\,dz \left\{1 - \frac{K_m}{2}\int_0^{Z_i + Z_e} \rho g\, dz\right\}$$

[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

$$z_e - z_i = \frac{1}{\rho_i}\int_{z_i}^{z_e} \rho_i \, dz$$

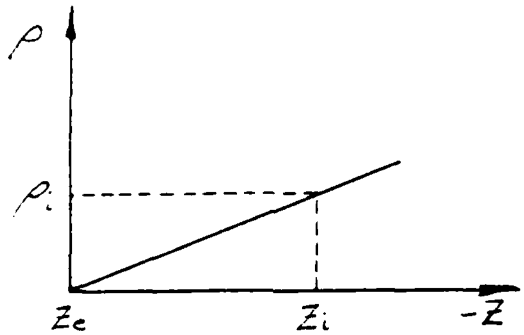
The reference density is taken to be the inlet value so that

$$\rho_i = \rho_o$$

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

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

where ρ_i, ρ_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

$$\Delta P_{pump} = \Delta P_{PIPE\ SYSTEM} + \Delta P_{COND\ DESIGN} + \Delta P_{DENSITY} \qquad (105)$$

Using Eq. (93), Eq. (105) can be converted to a pumping head. Similarly, pumping power in terms of horse-power can be determined using Eq. (44).

$$P_{CP} = \frac{\dot{m}_{SW}}{\eta_P}\left(\frac{gH}{g_c}\right)$$

where

$$\dot{m}_{SW} = \rho_{SW}\left(\frac{\pi d_i}{4}\right)^2 V_{SW} \qquad (106)$$

and ρ_{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

$$P_{CP(MW)} = \frac{P_{CP}}{\eta_M} \times \text{CONVERSION FACTOR} \qquad (107)$$

where η_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.

$$C_{pump} = \left[\left(P/1000 \right) 0.75 + 50 \right] 1.21 \times 10^{3} \qquad (108)$$

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

3. Ammonia Circulation Pump, P_{CIRC}

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

$$Re_d = \frac{\rho V d_i}{\mu}$$

where ρ, μ = 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

$$K = f \frac{L}{d_i} \tag{109}$$

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

$$K = 4 \frac{L}{D} \tag{110}$$

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 (83).

$$\Delta P_{pipe} = \rho \left[f \frac{L}{d_i} + 4 \frac{L}{D} \right] \frac{V^2}{2 g_c} \tag{111}$$

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

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

$$\Delta P_{EVAP} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \tag{112}$$

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

where $$\Delta P_{THERMO} = P_2 - P_1 \tag{113}$$

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

. Elevation head

where $$\Delta P_{ELEVATION} = Z_2 - Z_1 \tag{114}$$

Z_1 = datum.

Z_2 = 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 requireements.

$$\Delta P_{pump} = \Delta P_{PIPE} + \Delta P_{EVAP} + \Delta P_{THERMIC} + \Delta P_{ELEVATION} \tag{115}$$

Using Eq. (93) with ammonia properties, Eq. (115) can be converted to pumping head and finally expressed as pumping power (horsepower).

$$P_{CIRC} = \frac{\dot{m}_{NH_3}}{\eta_P} \left(\frac{gH}{g_c} \right) \tag{116}$$

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

η_P = pump mechanical efficiency (designer input).

Pumping power can then be expressed in terms of megawatts electrical

$$P_{CIRC(MW)} = \frac{P_{CIRC}}{\eta_M} \times \text{CONVERSION FACTOR} \tag{117}$$

where η_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

$$C_{pump} = \left(\frac{\dot{m}_{NH3}\ v_f}{80100}\right)^{0.64} 1.21 \times 10^{5} \qquad (118)$$

where $\dot{m}_{NH3}$ = mass flow rate of ammonia (lb_m/hr)

v_f = specific volume of saturated liquid ammonia at state point 1 (ft^3/lb_m)

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

4. Ammonia Re-flux Pump, $P_{RE-FLUX}$

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 ρ, μ = 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 re-flux pipe friction factor can be determined from Eqs. (85) or (86), and the piping resistance coefficient can be expressed as

$$K = f \frac{L}{d_i} \tag{119}$$

where L = ammonia re-flux pipe length (designer input).

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

$$K = 4 \frac{L}{D} \tag{120}$$

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)

$$\Delta P_{PIPE} = \rho \left[f \frac{L}{d_i} + 4 \frac{L}{D} \right] \frac{V^2}{2 g_c} \tag{121}$$

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.

$$\Delta P_{THERMO} = \Delta P_{LIQ} \tag{122}$$

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

Therefore,

$$\Delta P_{ELEV} = Z_2 - Z_1 \qquad (123)$$

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.

$$\Delta P_{pump} = \Delta P_{PIPE} + \Delta P_{ELEVATION} + \Delta P_{THERMO} \qquad (124)$$

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

$$P_{RE\text{-}FLUX} = \frac{\dot{m}_R}{\eta_P}\left(\frac{gH}{g_c}\right) \qquad (125)$$

where $\dot{m}_R$ = drainage mass flow rate.

η_p = pump mechanical efficiency (designer input).

Pumping power can be expressed in terms of megawatts electrical

$$P_{RE\text{-}FLUX\,(MW)} = \frac{P_{RE\text{-}FLUX}}{\eta_M} \times \text{CONVERSION FACTOR} \qquad (126)$$

where η_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.

$$C_{pump} = \left(\frac{\dot{m}_R \, v_f}{80100}\right)^{0.64} 1.21 \times 10^5 \tag{127}$$

where $\dot{m}_R$ = mass flow rate of evaporator drainage ammonia (lb_m/hr)

v_f = specific volume evaluated at the average evaporator pressure (ft^3/lb_m)

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).

$$P_{LOSS} = P_{HP} + P_{CP} + P_{CIRC} + P_{RE\text{-}FLUX} \tag{128}$$

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

$$\dot{E}_G = \frac{\dot{E}}{\eta_{TM}\,\eta_{GEN}} + P_{LOSS} \tag{129}$$

where P_{LOSS} = parasitic pump losses determined by Eq. (128).

η_{TM} = turbine mechanical efficiency (designer input).

η_{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

$$\dot{E}_G = \dot{m}\left(h_5 - h_4\right)$$

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_5, the following relations may be expressed

$$h_{5g} = h_g \Big)_{P5} \qquad h_{5f} = h_f \Big)_{P5} \tag{130}$$

Therefore, it follows that the turbine outlet quality, $X5$, can be determined from

$$h_5 = h_{5f} + X5\left(h_{5g} - h_{5f}\right) \tag{131}$$

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

$$S_{4f} = S_f \Big)_{T4} \qquad S_{4g} = S_g \Big)_{T4}$$

$$S_4 = S_{4f} + X4\left(S_{4g} - S_{4f}\right) \tag{132}$$

For isentropic turbine work,

$$S_4 = S_{5S} \tag{133}$$

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

$$S_{5g} = S_g \Big)_{T5} \qquad S_{5f} = S_f \Big)_{T5}$$

$$S_{5s} = S_{5f} + X5s\left(S_{5g} - S_{5f}\right) \tag{134}$$

Having determined the quality at state point 5s, the enthalpy can now be determined.

$$h_{5s} = h_{5f} + X_{5s}\left(h_{5g} - h_{5f}\right) \tag{135}$$

Using the results of Eqs. (41), (130), and (132), the internal turbine efficiency (adiabatic) can be determined, expressed by

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \tag{136}$$

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

- $h_5 < h_{5g}$
- $X_{5s} < X_5$
- $\eta_T \leq 90\%$

3. Turbine Cost Analysis

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

$$C_{TURB} = 2.42 \times 10^6 \left(0.375 + \dot{E}_G / 136000\, N_f\right) F_f \tag{137}$$

where $\dot{E}_G$ = 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,

$$C_{GEN} = \left(0.023\,\dot{E}_G + 0.3\right)1.21 \times 10^6 \qquad (138)$$

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:

. Single-pass shell and tube heat exchanger.

. Horizontal/vertical 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 condenser analysis, the following major steps in the algorithm are listed in order of their execution:

. Initialization of design variables (DV).

.. Tube length.

.. SW velocity through condenser tubes.

.. Outer tube diameter.

.. Tube profile pitch ratio.

. Amount of heat rejection (139).

. Tubeside bulk temperature (142).

- Total number of tubes (143).
- Log mean temperature difference (144).
- Conductance (146).
- Number of transfer units (145).
- Heat exchanger effectiveness (147).
- Initially assume a value for ammonia heat transfer coefficient (151).
- Single tube conductance (148).
- Average heat rejection per tube (152).
- Film temperature (153).
- Revised ammonia heat transfer coefficient (154, etc.); iterate with (151).
- Tube profile, flow parameters across the tube bank (158, etc.).
- Tube sheet diameter (163).
- Condenser shellside pressure drop for two-phase flow (166).
- Revised properties at state point 1 (171, 172); iterate with (21).
- Overall heat transfer coefficient (173).
- Total heat transfer surface area (174).
- Revised condenser tube length (175).
- Heat exchanger cost analysis.

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

B. ANALYSIS OF THE CONDENSER

1. Amount of Heat Rejection, $\dot{Q}$

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

$$\dot{Q} = \dot{m}_{NH_3} (h_5 - h_1) \tag{139}$$

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

$$C_{min} = \dot{m}_{cp} \, c_{p_{sw}} \tag{140}$$

where $c_{p_{sw}}$ = specific heat of salt water initially evaluated at the cold pipe inlet temperature.

$\dot{m}_{cp}$ = mass flow rate of salt water through the cold pipe previously evaluated by Eq. (107).

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

$$\dot{Q} = C_{min} (T_{c_o} - T_{c_i}) \tag{141}$$

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

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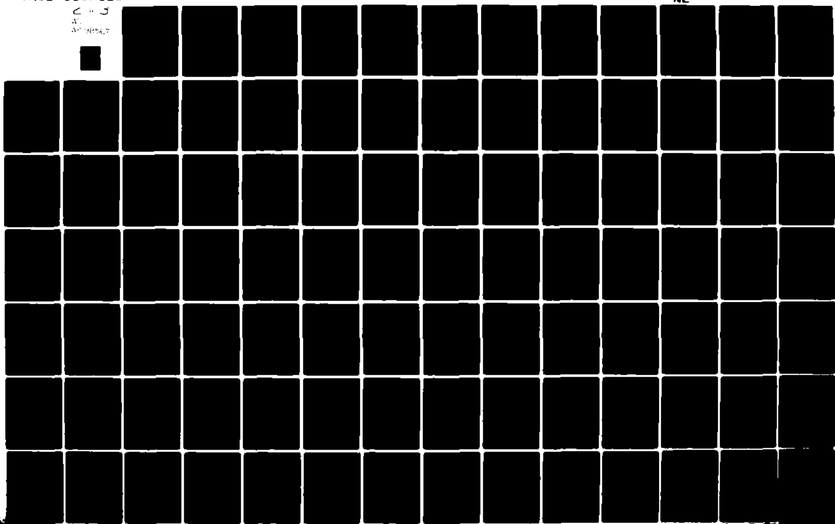

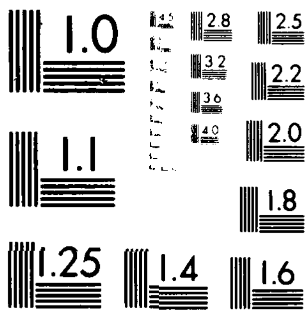

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$$T_B = \frac{T_{c_o} + T_{c_i}}{2} \tag{142}$$

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:

$$\dot{m} = \rho_{sw} \frac{\pi d_i^2}{4} V_t N_t \tag{143}$$

where ρ_{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).

V_t = average salt water velocity through the condenser (initialized and treated as a design variable by the optimization code).

4. Log Mean Temperature Difference, 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 LMTD of the condenser may be expressed as

$$LMTD = \frac{T_{c_o} - T_{c_i}}{\ln\left(\frac{T_5 - T_{c_i}}{T_5 - T_{c_o}}\right)} \tag{144}$$

5. NTU-Effectiveness Relations

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

$$NTU = \frac{U_o A_o}{C_{min}} \tag{145}$$

where the conductance $(U_o A_o)$ of the heat exchanger is a function of the heat absorbed and the LMTD.

$$\dot{Q} = (U_o A_o)\, LMTD \tag{146}$$

The condenser effectiveness can then be expressed as

$$\epsilon = 1 - e^{(-NTU)} \tag{147}$$

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

6. Single-Tube Conductance, U_oA_o

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

$$U_o A_o = \frac{1}{\frac{1}{\eta_i h_{sw} A_i} + \frac{1}{A_i} R_{fsw} + \frac{\ln d_o/d_i}{2\pi K L} + \frac{1}{A_o} R_{fNH_3} + \frac{1}{\eta_o h_{NH_3} A_o}} \qquad (148)$$

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

η_o, η_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 ρ_{sw}, μ_{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

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

Nusselt and Prandtl numbers are defined as

$$Nu_d = \frac{h_{sw} \, d_i}{k_{sw}} \qquad (149)$$

$$Pr = \frac{c_{p_{sw}} \, \mu_{sw}}{k_{sw}} \qquad (150)$$

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.

$$Nu_d = 0.023\, Re_d^{0.8} Pr^{0.3}$$

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 foulding resistance for tubeside salt water can be maintained at .00025 $(hr \cdot ft^2 \cdot F/BTU)$.

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

Initially, h_{NH_3} will be assumed

$$h_{NH_3} = 1000\ (BTU/hr \cdot ft^2 \cdot F) \tag{151}$$

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. <u>Film Temperature for Property Evaluation, T_f</u>

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).

$$\dot{Q} = U_o A_o \left(T_5 - T_{BULK}\right) \tag{152}$$

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

$$T_{W_2} = T_{BULK} + \dot{Q}\left(R_1 + R_2 + R_3\right)$$

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

$$T_f = \frac{T_{W_2} + T_5}{2} \tag{153}$$

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_{NH_3}

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

$$\bar{h} = 0.95 \left(\frac{K_f^3 \rho_f^2 g L}{\mu_f w} \right)^{1/3} \tag{154}$$

where w = estimate of ammonia mass flow rate across each tube.

K_f, ρ_f, μ_f = properties evaluated at film temperature.

L = 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]

$$\bar{h} = 1.045 \left(\frac{K_f^3 \rho_f g L}{\mu_f w} \right)^{1/3} \tag{155}$$

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]

$$Re = \frac{2\Gamma}{\mu_f}$$

where Γ = 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]:

$$\bar{h} = 1.28\left[1.47\left(\frac{\mu_f}{K_f^3 \rho_f^2 g}\right)^{-1/3}\left(\frac{4\Gamma}{\mu_f}\right)^{-1/3}\right] \tag{156}$$

where Γ = mass flow rate of condensate per tube over its diameter.

For turbulent flow, Kirkbride's correlation is applied

$$\bar{h} = 0.0077\left(\frac{\mu_f^2}{K_f^3 \rho_f^2 g}\right)^{-1/3}\left(\frac{4\Gamma}{\mu_f}\right)^{0.4} \tag{157}$$

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]

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

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,

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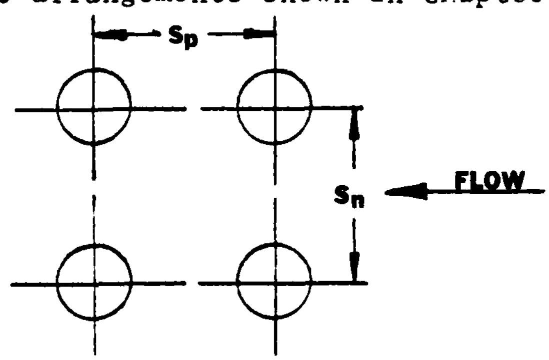


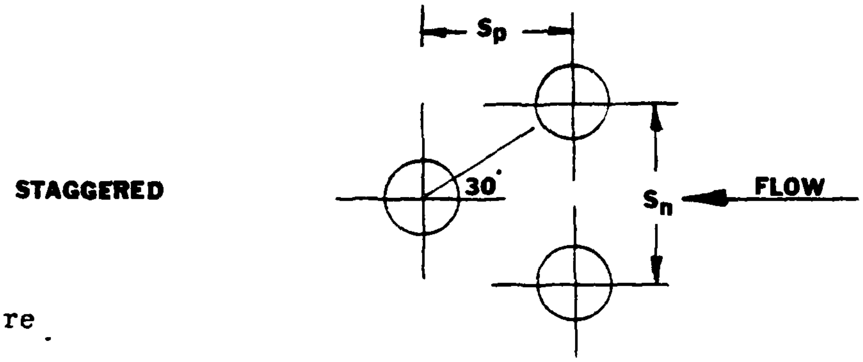
$$S_n = P_R \, d_o \tag{158}$$

$$A_p = S_n^2 \tag{159}$$

where S_n = pitch ratio x outer tube diameter.

P_R = pitch ratio (initialized and treated as a design variable for the optimization code).

A_p = tube profile area per tube



where

$$S_n = 2\, P_R\, d_o \sin 30^\circ \tag{160}$$

$$S_p = P_R\, d_o \cos 30^\circ \tag{161}$$

$$A_p = S_n\, S_p \tag{162}$$

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

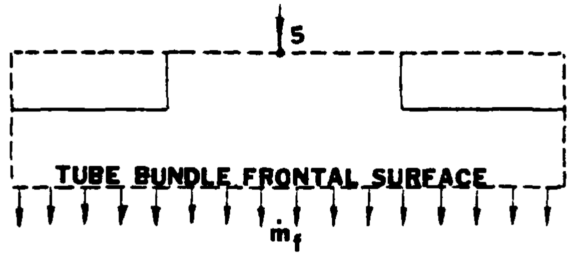
$$\frac{A_{ff}}{A_f} = \frac{S_n - d_o}{S_n} \tag{163}$$

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

$$N_t A_p = \frac{\pi T_{SD}^2}{4} \tag{164}$$

where T_{SD} = 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

$$\dot{m}_5 = \dot{m}_f$$

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 $$(A_f)_{VAP} = A_f \eta$$

where η = percent of tube frontal area which is covered by vapor.

$$\dot{m}_5 = \rho_5 A_5 V_5$$

$$\dot{m}_f = \rho_f A_f V_f \eta$$

where A_5 = condenser inlet cross-sectional area.

V_5 = turbine discharge ammonia velocity.

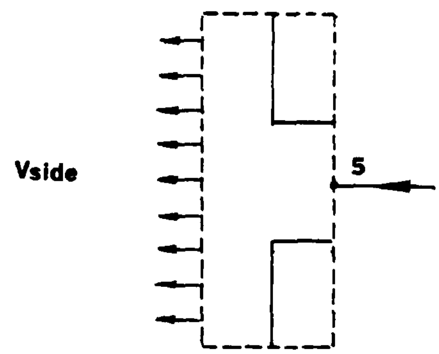
Therefore

$$V_f = \frac{\rho_5 A_5}{\rho_f A_f \eta} V_5$$

If $\eta = \rho_5 A_5 / A_f \rho_f$, 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 η 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.

$$A_f = T_{SD} \, L_t$$

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

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

$$G = \frac{\dot{m}_4}{A_{ff}} \qquad (166)$$

10. Pressure Drop of Two-Phase Flow across a Bank of Tubes, ΔP

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

$$\Delta P_{COND} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \tag{167}$$

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

$$\Delta P_{FRICTION} = \frac{f\, G^2 \bar{v}}{De\, 2\, g_c} L_c \tag{168}$$

$$\Delta P_{MOMENTUM} = \frac{G^2 \bar{v}}{g_c} \tag{169}$$

$$\Delta P_{ELEVATION} = \frac{g}{\bar{v}\, g_c} L_c \tag{170}$$

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

$$D_e = P_R\, d_o - d_o$$

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

$$\bar{v} = v_f \left[1 + \frac{x}{v_f} \left(v_g - v_f \right) \right]$$

where X = quality of mixture (state point 5).

v_f = specific volume of liquid (state point 1)

v_g = 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

$$P_{1(NEW)} = P_1 - \Delta P_{COND} \tag{171}$$

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:

$$h_{1(NEW)} = h_f \Big)_{P_{1(NEW)}} \quad T_{1(NEW)} = T_{SAT} \Big)_{P_{1(NEW)}} \tag{172}$$

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

will be reevaluated to complete the closed-loop cycle of the simulated OTEC power system.

12. Overall Heat Transfer Coefficient, 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

$$R_1 = \frac{d_o}{\eta_i h_{sw} d_i}$$

$$R_2 = \frac{d_o}{\eta_i h_{fsw} d_i}$$

$$R_3 = \frac{d_o \ln d_o/d_i}{2k}$$

$$R_5 = \frac{1}{\eta_o h_{NH_3}}$$

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

$$U_o = \frac{1}{R_1 + R_2 + R_3 + R_5} \tag{173}$$

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

$$NTU = \frac{U_o A_t}{C_{min}} \tag{174}$$

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

$$A_t = N_t \pi d_o L_{t\,(REVISED)} \tag{175}$$

At this time, it is necessary ot 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 shall 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.

$$C_{DPB} = 1.539 \times 10^{-2}\, t_d\, N_t\, T_{SD}^{\,2.0} \tag{176}$$

. Bustle, flanges, channels and flow plate cost.

$$C_{BFCF} = 1185.286\, T_{SD}^{\,2.0} \tag{177}$$

. Tube material cost.

$$C_{TM} = (C_1 L_t + C_2) N_t\, d_o/1.5 \tag{178}$$

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.

$$C_{WINS} = 10106.475\, T_{SD} \tag{179}$$

. 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 cast 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.

$$C_{ATI} = C_{TI}\left[1 + (1+i)^{10} + (1+i)^{20}\right] \quad (180)$$

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

. Aluminum tube material cost.

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

for tube sheet diameter 35-50 feet.

. Drilling time/tube sheet thickness (58)

. Thickness of the tube sheet. (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.

$$C_{DPB} = 9.825\, N_t^{0.479}\, t_d \quad (182)$$

. Bustle, flanges, channels and flow plate.

$$C_{BFCF} = 382.824\, T_{SD}^{2.184} \quad (183)$$

. Heat exchange head cost.

$$C_{HXH} = 939.62\, T_{SD}^{1.43} \quad (184)$$

. Water inlet, nozzles, and supporters cost.

$$C_{WINS} = 7453.6\, T_{SD}^{1.056} \tag{185}$$

. 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

$$\text{Minimize} \quad F(\bar{X}) \tag{186}$$

Subject to the constraints:

$$G_j(\bar{X}) \leq 0 \qquad j = 1, NCON \tag{187}$$

$$H_j(\bar{X}) = 0 \qquad j = 1, NEQ \tag{188}$$

$$VLB_i \leq \bar{X}_i \leq VUB_i \qquad i = 1, NDV \tag{189}$$

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 ($NCON$ is the number of such constraints).

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

VLB_i/VUB_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, $\bar{X}$, 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

$$\bar{X}^{q+1} = \bar{X}^{q} + \alpha * \bar{S}^{q}$$

where q = the iteration number.

α = scalar quantity which defines the move in the search direction.

$\bar{S}$ = 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]:

$$F'(\bar{X}) = F(\bar{X}) - K\sum_{j=1}^{NEQ} H_j \tag{190}$$

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

$$H_j(\bar{X}) \leq 0 \quad j = 1, NEQ$$

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 1: OTEC Power System Comparisons (Titanium Tubed Heat Exchangers)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.44 EØ9	2.01 EØ9	2.80 EØ9	3.46 EØ9
SW FLOW (LB_m/HR)	2.27 EØ8	3.34 EØ8	4.14 EØ8	4.86 EØ8
NH_3 FLOW (LB_m/HR)	2.73 EØ6	3.78 EØ6	5.29 EØ6	6.53 EØ6
OPER PRESS (LB_f/IN^2)	129.	130.0	127.7	127.1
OVL HT COEF (BTU/ HR·FT^2·F)	623.19	612.97	601.47	595.73
HT SURFACE (FT^2)	387,598	572,405	743,093	922,075
TUBE OUTER DIA (IN)	0.947	0.952	0.945	0.929
TUBE WALL THICK (IN)	0.025	0.025	0.025	0.025
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.47	1.52
TUBE LENGTH (FT)	43.66	42.18	42.27	42.56
TUBE SHEET DIA (FT)	21.96	27.21	32.45	36.76
TOT NR OF TUBES	35,806	54,449	71,034	89,105

TABLE 1. OTEC Power System Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
HT REJECT (BTU/HR)	1.39 EØ9	1.94 EØ9	2.71 EØ9	3.34 EØ9
SW FLOW (LB_m/HR)	2.23 EØ8	3.35 EØ8	4.72 EØ8	5.69 EØ8
NH_3 FLOW (LB_m/HR)	2.73 EØ6	3.79 EØ6	5.29 EØ6	6.53 EØ8
OPER PRESS (LB_f/IN^2)	88.16	88.15	87.46	87.49
OVL HT COEF (BTU/ HR·FT^2·F)	454.4	446.8	438.4	435.7
HT SURFACE (FT^2)	552,314	762,190	1,168,239	1,483,762
TUBE OUTER DIA (IN)	0.935	0.972	0.957	0.940
TUBE WALL THICK (IN)	0.025	0.025	0.025	0.025
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.48	1.51
TUBE LENGTH (FT)	58.57	57.42	58.32	59.09
TUBE SHEET DIA. (FT)	22.48	27.194	35.16	39.734
TOT NR OF TUBES	38,524	52,179	79,956	102,054

TABLE 1: OTEC Power System Comparisons (Continued)

PIPING SYSTEMS	10 MW	15 MW	20 MW	25 MW
SW HOT PIPE (300 FT LENGTH)				
INNER DIA (FT)	17.20	20.08	21.86	23.30
SW VEL (FT/SEC)	4.26	4.6	4.80	4.97
PRESS DROP (LB_f/IN^2)	0.280	0.322	0.348	0.371
SW COLD PIPE (3000 FT LENGTH)				
INNER DIA (FT)	16.1	18.62	21.35	22.97
SW VEL (FT/SEC)	4.94	5.33	5.72	5.95
PRESS DROP (LB_f/IN^2)	0.49	0.508	0.526	0.539
NH_3 CIRC PIPE (150 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	2.0
PRESS DROP (LF_f/IN^2)	8.46	9.87	11.33	12.53
NH_3 RE-FLUX PIPE (50 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	2.0
PRESS DROP (LB_f/IN^2)	8.46	9.87	11.33	12.53

TABLE 1: OTEC Power System Comparisons (Continued)

PUMP SYSTEMS	10 MW	15 MW	20 MW	25 MW
EVAP SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	11.0	9.89	9.42	9.26
CAPACITY (GAL/MIN)	444,338	653,260	808,625	930,352
COND SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	22.23	20.75	20.04	20.03
CAPACITY (GAL/MIN)	452,026	652,119	920,116	1,107,382
NH_3 CIRC PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	200.9	217.33	214.93	220.52
CAPACITY (GAL/MIN)	8709.3	12,101.7	16,880.2	20,586.8
NH_3 RE-FLUX PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	32.4	38.02	43.53	48.13
CAPACITY (GAL/MIN)	2684.3	3732.4	5201.4	6424.3

TABLE 1: OTEC Power System Comparisons (Continued)

EFFICIENCY OF OPERATION	10 MW	15 MW	20 MW	25 MW
TURBINE-GENERATOR (TURB MECH 99.8 PCT, GEN MECH AND ELECT 96.6 PCT)				
TURB INTERNAL (PCT)	87.16	89.88	88.22	89.80
OUTLET QUALITY (PCT)	96.6	96.77	96.9	96.9
POWER REQUIREMENTS (MEGAWATTS)				
TURB EFFIC LOSSES	0.373	0.559	0.745	0.932
EVAP SW PUMP	1.131	1.496	1.762	2.034
COND SW PUMP	2.334	3.143	4.283	5.151
NH_3 CIRC PUMP	0.281	0.412	0.583	0.738
NH_3 RE-FLUX PUMP	0.014	0.022	0.035	0.048
TURB-GEN GROSS	14.132	20.633	27.409	33.904
PCT PARASITIC POWER	26.6	24.6	24.31	23.51
THERMO CYCLE EFFIC (PCT)	2.45	2.65	2.53	2.56

TABLE 1: OTEC Power System Comparisons (Continued)

COMPONENT COSTS ($)	10 MW	15 MW	20 MW	25 MW
EVAPORATOR	5,134,322.	8,223,672.	11,501,680.	13,026,794.
CONDENSER	5,952,658.	8,667,154.	13,011,939.	16,558,713.
GEN-TURBINE	1,495,085.	1,578,776.	1,666,011.	1,749,636.
GENERATOR	756,287.	937,205.	1,125,782.	1,306,554.
EVAP SW PUMP	463,736.	653,333.	794,355.	922,944.
COND SW PUMP	470,714.	652,298.	895,505.	1,065,449.
NH_3 CIRC PUMP	110,849.	136,825.	169,303.	193,848.
NH_3 RE-FLUX PUMP	52,190.	64,449.	79,701.	91,233.
OPTIMUM COST ($)	14,383,650.	20,849,216.	29,164,560.	34,823,904.
COST/KW (NET) OUTPUT ($/KW)	1438.36	1389.95	1458.23	1392.96

TABLE 2: OTEC Power System Comparisons (Aluminum Tubed Heat Exchanger)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.51 EØ9	1.97 EØ9	2.80 EØ9	INFEASIBLE DESIGN
SW FLOW (LB_m/HR)	2.53 EØ8	3.42 EØ8	3.89 EØ8	
NH_3 FLOW (LB_m/HR)	2.86 EØ6	3.72 EØ6	5.29 EØ6	
OPER PRESS (LB_f/IN^2)	129.57	131.98	128.01	
OVL HT COEF (BTU/ $HR \cdot FT^2 \cdot F$)	646.7	643.2	625.5	
HT SURFACE (FT^2)	391,370	610,261	777,553	
TUBE OUTER DIA (IN)	1.221	1.046	0.982	
TUBE WALL THICK (IN)	0.065	0.065	0.065	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.6	
TUBE LENGTH (FT)	47.68	42.42	41.63	
TUBE SHEET DIA (FT)	23.97	30.73	34.68	
TOT NR OF TUBES	25,669	52,512	72,662	

TABLE 2: *OTEC Power System Comparisons* (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
HT REJECT (BTU/HR)	1.47 EØ9	1.89 EØ9	2.71 EØ9	INFEASIBLE DESIGN
SW FLOW (LB_m/HR)	2.44 EØ8	3.04 EØ8	4.38 EØ8	
NH_3 FLOW (LB_m/HR)	2.86 EØ6	3.72 EØ6	5.30 EØ6	
OPER PRESS (LB_f/IN^2)	88.52	89.18	87.85	
OVL HT COEF (BTU/ $HR \cdot FT^2 \cdot F$)	454.57	453.54	446.48	
HT SURFACE (FT^2)	554,011	690,395	1,173,216	
TUBE OUTER DIA (IN)	1.176	1.092	1.001	
TUBE WALL THICK (IN)	0.065	0.065	0.065	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.5	
TUBE LENGTH (FT)	64.04	57.46	57.24	
TUBE SHEET DIA (FT)	24.15	28.61	36.82	
TOT NR OF TUBES	28,091	42,035	78,198	

TABLE 2: OTEC Power System Comparisons (Continued)

PIPING SYSTEMS	10 MW	15 MW	20 MW	25 MW
SW HOT PIPE (300 FT LENGTH)				INFEASIBLE DESIGN
INNER DIA (FT)	17.89	20.25	21.33	
SW VEL (FT/SEC)	4.38	4.63	4.74	
PRESS DROP (LB_f/IN^2)	0.295	0.326	0.341	
SW COLD PIPE (3000 FT LENGTH)				
INNER DIA (FT)	16.53	17.93	20.7	
SW VEL (FT/SEC)	4.93	5.23	5.64	
PRESS DROP (LB_f/IN^2)	0.478	0.504	0.523	
NH_3 CIRC PIPE (150 FT LENGTH)				
INNER DIA (FT)	1.99	2.0	2.0	
PRESS DROP (LB_f/IN^2)	13.78	15.94	17.89	
NH_3 RE-FLUX PIPE (50 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	
PRESS DROP (LB_f/IN^2)	8.996	10.788	11.916	

TABLE 2: OTEC Power System Comparisons (Continued)

PUMP SYSTEMS	10 MW	15 MW	20 MW	25 MW
EVAP SW PUMP (EFFICIENCY 85 PCT)				INFEASIBLE DESIGN
HEAD (FT)	10.42	10.27	10.06	
CAPACITY (GAL/MIN)	494,239	669,211	760,748	
COND SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	21.32	20.80	21.05	
CAPACITY (GAL/MIN)	474,918	592,656	851,978	
NH_3 CIRC PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	203.5	218.32	216.63	
CAPACITY (GAL/MIN)	9144.3	11,895.6	16,911.6	
NH_3 RE-FLUX PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	34.33	41.27	45.66	
CAPACITY (GAL/MIN)	2818.4	3669.9	5210.5	

TABLE 2: OTEC Power System Comparisons (Continued)

EFFICIENCY OF OPERATION	10 MW	15 MW	20 MW	25 MW
TURBINE-GENERATOR (TURB MECH 99.8 PCT, GEN MECH AND ELECT 96.6 PCT)				
TURB INTERNAL (PCT)	83.37	89.76	88.05	INFEASIBLE DESIGN
OUTLET QUALITY (PCT)	97.06	96.73	96.93	
POWER REQUIREMENTS (MEGAWATTS)				
TURB EFFIC LOSSES	0.373	0.559	0.745	
EVAP SW PUMP	1.191	1.590	1.770	
COND SW PUMP	2.351	2.863	4.166	
NH_3 CIRC PUMP	0.299	0.416	0.588	
NH_3 RE-FLUX PUMP	0.015	0.024	0.037	
TURB-GEN GROSS	14.229	20.452	27.306	
PCT PARASITIC POWER	27.10	23.92	24.03	
THERMO CYCLE EFFIC (PCT)	2.34	2.70	2.53	

TABLE 2: OTEC POWER SYSTEM COMPARISONS (CONTINUED)

COMPONENT COSTS ($)	10 MW	15 MW	20 MW	25 MW
EVAPORATOR	20,848,960.	34,297,296.	44,203,408.	INFEASIBLE DESIGN
CONDENSER	24,633,360.	31,812,640.	55,404,448.	
GEN-TURBINE	1,496,339.	1,576,449.	1,664,688.	
GENERATOR	758,998.	932,173.	1,122,921.	
EVAP SW PUMP	509,021.	667,809.	750,878.	
COND SW PUMP	491,487.	598,335.	833,670.	
NH_3 CIRC PUMP	114,361.	135,329.	169,505.	
NH_3 RE-FLUX PUMP	53,845.	63,756.	79,790.	
OPTIMUM COST ($)	48,852,480.	70,020,000.	104,149,488.	
COST/KW (NET) OUTPUT ($/KW)	4885.25	4668.00	5207.47	

TABLE 3: Heat Exchanger Comparisons (Titanium Tubed)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.44 EØ9	2.01 EØ9	2.80 EØ9	3.46 EØ9
SW FLOW (LB_m/HR)	2.27 EØ8	3.34 EØ8	4.14 EØ8	4.86 EØ8
SW TEMP IN (DEG F)	80.0	80.0	80.0	80.0
SW TEMP OUT (DEG F)	73.36	73.72	72.92	72.56
NH_3 FLOW (LB_m/HR)	2.73 EØ6	3.78 EØ6	5.29 EØ6	6.53 EØ6
OPER PRESS (LB_f/IN^2)	129.0	130.1	127.72	127.1
SAT TEMP (DEG F)	70.11	70.59	69.54	69.27
OUTLET TEMP (DEG F)	70.06	70.51	69.47	69.19
OUTLET QUALITY (PCT)	92	92	92	92
NH_3 PRESS DROP (LB_f/IN^2)	0.105	0.162	0.165	0.174
TUBE CHARACTERISTICS				
OUTER DIA (IN)	0.947	0.952	0.945	0.929
WALL THICK (IN)	0.025	0.025	0.025	0.025
LENGTH (FT)	43.66	42.18	42.27	42.56

TABLE 3: Heat Exchanger Comparisons (Continued)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.47	1.52
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.3	6.03	5.80	5.64
T WALL (DEG F)	71.09	71.47	70.51	70.22
FILM TEMP (DEG F)	70.57	70.99	69.99	69.71
DELTA T BOILING (DEG F)	1.027	0.952	1.041	1.032
LMTD	6.00	5.76	6.31	6.34
EFFECTIVENESS	0.671	0.667	0.677	0.693
NTU	1.112	1.098	1.13	1.182
OVL HT COEF (BTU/ $HR \cdot FT^2 \cdot F$)	623.19	612.97	601.47	595.73
h (WATER)	1156.01	1115.97	1081.6	1059.15
h (FOULING)	3787.7	3788.5	3787.35	3784.37

TABLE 3: Heat Exchanger Comparisons (Continued)

	10 MW	15 MW	20 MW	25 MW
EVAPORATOR				
h (METAL)	4413.77	4404.7	4415.43	4429.66
h (AMMONIA)	4015.73	4088.4	4040.26	4090.74
SW PRESS DROP (LB_f/IN^2)	4.593	4.061	3.824	3.727
MOISTURE SEPARATOR				
OPER PRESS (LB_f/IN^2)	128.9	129.94	127.55	126.92
OUTLET TEMP (DEG F)	69.92	70.33	69.22	68.89
OUTLET QUALITY (PCT)	99.5	99.5	99.5	99.5
NH_3 PRESS DROP (LB_f/IN^2)	0.311	0.416	0.578	0.681
CONDENSER				
HT REJECT (BTU/HR)	1.44 EØ9	2.01 EØ9	2.80 EØ9	3.46 EØ9
SW FLOW (LB_m/HR)	2.27 EØ8	3.34 EØ8	4.14 EØ8	4.86 EØ8

TABLE 3: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
SW TEMP IN (DEG F)	40.0	40.0	40.0	40.0
SW TEMP OUT (DEG F)	46.29	46.06	46.00	46.16
NH_3 FLOW (LB_m/HR)	2.73 EØ6	3.79 EØ6	5.29 EØ6	6.53 EØ6
OPER PRESS (LB_f/IN^2)	88.16	88.15	87.46	87.49
SAT TEMP (DEG F)	49.36	49.36	49.95	48.97
OUTLET TEMP (DEG F)	49.3	49.24	49.84	48.85
NH_3 PRESS DROP (LB_f/IN^2)	0.127	0.206	0.173	0.195
TUBE CHARACTERISTICS				
OUTER DIA (IN)	0.935	0.972	0.957	0.940
WALL THICK (IN)	0.025	0.025	0.025	0.025
LENGTH (FT)	58.57	57.42	58.32	59.09
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.48	1.51

TABLE 3: *Heat Exchanger Comparisons* (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.12	6.02	5.72	5.60
T WALL (DEG F)	48.39	48.33	48.04	48.07
FILM TEMP (DEG F)	48.84	48.79	48.44	48.46
DELTA T COND (DEG F)	0.895	0.901	0.808	0.781
LMTD	5.56	5.68	5.29	5.17
EFFECTIVENESS	0.678	0.655	0.679	0.696
NTU	1.133	1.065	1.136	1.191
OVL HT COEF (BTU/ $HR \cdot FT^2 \cdot F$)	454.4	446.83	438.4	435.73
h (WATER)	719.6	704.4	678.0	669.3
h (FOULING)	3785.5	3792.0	3789.4	3786.4
h (METAL)	4424.2	4393.6	4405.6	4420.1
h (AMMONIA)	3118.9	3058.6	3171.5	3219.8
SW PRESS DROP (LB_f/IN^2)	6.31	5.64	5.30	5.28

TABLE 4: Heat Exchanger Comparisons (Aluminum Tubed)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.51 EØ9	1.97 EØ9	2.80 EØ9	INFEASIBLE DESIGN
SW FLOW (LB_m/HR)	2.53 EØ8	3.42 EØ8	3.89 EØ8	
SW TEMP IN (DEG F)	80.0	80.0	80.0	
SW TEMP OUT (DEG F)	73.74	73.99	72.47	
NH_3 FLOW (LB_m/HR)	2.86 EØ6	3.72 EØ6	5.29 EØ6	
OPER PRESS (LB_f/IN^2)	129.57	131.98	128.01	
SAT TEMP (DEG F)	70.35	71.4	69.67	
OUTLET TEMP (DEG F)	70.32	71.4	69.61	
OUTLET QUALITY (PCT)	92.0	92.0	92.0	
NH_3 PRESS DROP (LB_f/IN^2)	0.070	0.087	0.136	
TUBE CHARACTERISTICS				
OUTER DIA (IN)	1.221	1.046	0.982	
WALL THICK (IN)	0.065	0.065	0.065	

TABLE 4: Heat Exchanger Comparisons (Continued)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
LENGTH (FT)	47.68	42.42	41.63	INFEASIBLE DESIGN
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.50	
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.6	6.2	5.89	
T WALL (DEG F)	71.49	72.25	70.63	
FILM TEMP (DEG F)	70.90	71.80	70.12	
DELTA T BOILING (DEG F)	1.163	0.885	1.02	
LMTD	6.0	5.03	5.8	
EFFECTIVENESS	0.649	0.699	0.729	
NTU	1.047	1.199	1.307	
OVL HT COEF (BTU/ HR·FT^2·F)	646.7	643.2	625.5	
h (WATER)	1090.0	1052.5	1011.5	

TABLE 4: Heat Exchanger Comparisons (Continued)

	10 MW	15 MW	20 MW	25 MW
EVAPORATOR				INFEASIBLE DESIGN
h (FOULING)	3574.3	3503.1	3470.5	
h (METAL)	13,444.8	13,312.9	13,252.2	
h (AMMONIA)	3640.3	4097.4	4058.5	
SW PRESS DROP (LB_f/IN^2)	4.32	4.22	4.11	
MOISTURE SEPARATOR				
OPER PRESS (LB_f/IN^2)	129.5	131.89	127.88	
OUTLET TEMP (DEG F)	70.22	71.23	69.38	
OUTLET QUALITY (PCT)	99.5	99.5	99.5	
NH_3 PRESS DROP (LB_f/IN^2)	0.24	0.307	0.522	
CONDENSER				
HT REJECT (BTU/HR)	1.47 EØ9	1.89 EØ9	2.71 EØ9	

TABLE 4: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
SW FLOW (LB_m/HR)	2.44 EØ8	3.04 EØ8	4.38 EØ9	INFEASIBLE DESIGN
SW TEMP IN (DEG F)	40.0	40.0	40.0	
SW TEMP OUT (DEG F)	46.30	46.53	46.49	
NH_3 FLOW (LB_m/HR)	2.86 EØ6	3.72 EØ6	5.30 EØ6	
OPER PRESS (LB_f/IN^2)	88.52	89.18	87.83	
SAT TEMP (DEG F)	49.52	49.97	49.17	
OUTLET TEMP (DEG F)	49.52	49.90	49.08	
NH_3 PRESS DROP (LB_f/IN^2)	0.088	0.111	0.148	
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

TABLE 4: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
PITCH RATIO	1.4	1.46	1.5	INFEASIBLE DESIGN
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.31	6.23	5.87	
T WALL (DEG F)	48.51	48.85	48.25	
FILM TEMP (DEG F)	49.02	49.38	48.67	
DELTA T COND (DEG F)	1.020	1.055	0.832	
LMTD	5.820	6.062	5.175	
EFFECTIVENESS	0.661	0.660	0.715	
NTU	1.082	1.078	1.254	
OVL HT COEF (BTU/ $HR \cdot FT^2 \cdot F$)	454.57	453.56	446.48	
h (WATER)	670.0	668.56	641.97	
h (FOULING)	3557.0	3523.7	3480.59	
h (METAL)	13,414.54	13,351.18	13,271.11	
h (AMMONIA)	2842.96	2853.65	3131.0	
SW PRESS DROP (LB_f/IN^2)	5.92	5.66	5.74	

APPENDIX A

SAMPLE INPUT DATA FOR OTEC ANALYSIS

EVAPORATOR - HORIZONTAL		
TUBE O.D.	1.000(IN)	25.400(MM)
TUBE LENGTH	40.000(FT)	12.192(M)
SW TUBE VEL	6.000(FT/S)	1.829(M/S)
OPER PRESSURE	130.000(LBF/IN2)	0.896(MPA)
TUBE MATERIAL - TITANIUM		
THERMAL COND(K)	9.500(BTU/HR.FT.F)	16.502(W/M.C)
TUBE PROFILE - STAGGERED EQUI-LATERAL		
PITCH RATIO	1.50	
ENHANCEMENT - PLAIN TUBE		
CONDENSER - HORIZONTAL		
TUBE O.D.	1.000(IN)	25.400(MM)
TUBE LENGTH	56.500(FT)	17.221(M)
SW TUBE VEL	6.000(FT/S)	1.829(M/S)
OPER PRESSURE	89.000(LBF/IN2)	0.614(MPA)
TUBE MATERIAL - TITANIUM		
THERMAL COND(K)	9.500(BTU/HR.FT.F)	16.502(W/M.C)
TUBE PROFILE - STAGGERED EQUI-LATERAL		
PITCH RATIO	1.50	
ENHANCEMENT - PLAIN TUBE		
SALT WATER HOT PIPE		
PIPE I.D.	19.300(FT)	5.883(M)
PIPE LENGTH	300.000(FT)	91.440(M)
SW PIPE VEL	4.500(FT/S)	1.372(M/S)
SW INLET TEMP	80.000(DEG F)	26.667(DEG C)
SW SALINITY	35. 0/000	
SALT WATER COLD PIPE		
PIPE I.D.	19.600(FT)	5.974(M)
PIPE LENGTH	3000.000(FT)	914.400(M)
SW PIPE VEL	5.500(FT/S)	1.676(M/S)
SW INLET TEMP	40.000(DEG F)	4.444(DEG C)
SW SALINITY	35. 0/000	

```
  AMMONIA CIRC PIPE
    PIPE I.D.            2.000(FT)                 0.610(M)
    PIPE LENGTH        150.000(FT)                45.720(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  85.00(PCT)         MOTOR 98.00(PCT)
    COND SW PUMP
      EFFICIENCY MECH  85.00(PCT)         MOTOR 98.00(PCT)
    AMMONIA CIRC PUMP
      EFFICIENCY MECH  75.00(PCT)         MOTOR 98.00(PCT)
GEN-TURB EFFICIENCIES
      GEN MECH&ELECT   96.60(PCT)
      TURB MECH        99.80(PCT)
  POWER REQUIREMENTS
    NET POWER OUTPUT             15.000(MW)
```

APPENDIX B

SAMPLE OTEC ANALYSIS OPTIMIZATION OUTPUT DATA

```
EVAPORATOR - HORIZONTAL
  HT ABSORB      2005013760.0(BTU/HR)          587.601(MW)
  SW FLOW          334081280.0(LBM/HR)   151535904.0(KG/HR)
    SW TEMP IN         80.000(DEG F)            26.667(DEG C)
    SW TEMP OUT        73.724(DEG F)            23.180(DEG C)
  NH3 FLOW           3788703.0(LBM/HR)      1718519.0(KG/HR)
  OPER PRESSURE      130.097(LBF/IN2)          896.987(KPA)
    EVAP SAT TEMP      70.585(DEG F)            21.436(DEG C)
    OUTLET TEMP        70.514(DEG F)            21.397(DEG C)
    OUTLET QUALITY      92.00(PCT)
  NH3 PRESS DROP       0.162(LBF/IN2)            1.115(KPA)
  TUBE CHARACTERISTICS
    OUTTER DIA          0.952(IN)               24.180(MM)
    WALL THICK          0.025(IN)                0.639(MM)
    LENGTH             42.182(FT)               12.857(M)
    MATERIAL - TITANIUM
    TUBE PROFILE - STAGGERED EQUI-LATERAL
      PITCH RATIO      1.40
    ENHANCEMENT - PLAIN TUBE
    SW VELOCITY         6.026(FT/S)              1.837(M/S)
  T WALL(SHELLSIDE) 71.466(DEG F)               21.926(DEG C)
  FILM TEMP           70.990(DEG F)             21.661(DEG C)
  DELTA T BOILING      0.952(DEG F)              0.529(DEG C)
  L.M.T.D.             5.757(DEG F)              3.198(DEG C)
  EVAP EFFECTIVENESS              0.667
  NR OF TRANSFER UNITS            1.093
  OVL HT COEF         612.97(BTU/HR.FT2.F)   3480.58(W/M2.C)
    H(WATER)         1115.97(BTU/HR.FT2.F)   6336.71(W/M2.C)
    H(FOULING)       3788.54(BTU/HR.FT2.F)  21512.09(W/M2.C)
    H(METAL)         4409.72(BTU/HR.FT2.F)  25039.29(W/M2.C)
    H(AMMONIA)       4088.44(BTU/HR.FT2.F)  23215.00(W/M2.C)
```

HT SURFACE	572405.00(FT2)	53178.12(M2)
TUBE SHEET DIA	27.213(FT)	8.294(M)
TOT NR OF TUBES	54449.	
SW PRESS DROP	4.061(LBF/IN2)	28.003(KPA)

MOISTURE SEPARATOR-INSIDE EVAP SHELL

OPER PRESSURE	129.935(LBF/IN2)	895.872(KPA)
OUTLET TEMP	70.333(DEG F)	21.296(DEG C)
OUTLET QUALITY	99.50(PCT)	
NH3 PRESS DROP	0.416(LBF/IN2)	2.868(KPA)

CONDENSER - HORIZONTAL

HT REJECT	1935472896.0(BTU/HR)	567.221(MW)
SW FLOW	334871552.0(LBM/HR)	151894368.0(KG/HR)
SW TEMP IN	40.000(DEG F)	4.444(DEG C)
SW TEMP OUT	46.055(DEG F)	7.808(DEG C)
NH3 FLOW	3788708.0(LBM/HR)	1718519.0(KG/HR)
OPER PRESSURE	88.151(LBF/IN2)	607.781(KPA)
COND SAT TEMP	49.361(DEG F)	9.645(DEG C)
OUTLET TEMP	49.238(DEG F)	9.577(DEG C)
NH3 PRESS DROP	0.206(LBF/IN2)	1.423(KPA)

TUBE CHARACTERISTICS

OUTTER DIA	0.972(IN)	24.683(MM)
WALL THICK	0.025(IN)	0.642(MM)
LENGTH	57.416(FT)	17.500(M)

MATERIAL - TITANIUM

TUBE PROFILE - STAGGERED EQUI-LATERAL

PITCH RATIO 1.40

ENHANCEMENT - PLAIN TUBE

SW VELOCITY	6.017(FT/S)	1.834(M/S)
T WALL(SHELLSIDE)	48.331(DEG F)	9.073(DEG C)
FILM TEMP	48.785(DEG F)	9.325(DEG C)
DELTA T COND	0.907(DEG F)	0.504(DEG C)
L.M.T.D.	5.683(DEG F)	3.157(DEG C)
COND EFFECTIVENESS	0.655	
NR OF TRANSFER UNITS	1.065	

CVL HT COEF	446.83(BTU/HR.FT2.F)	2537.13(W/M2.C)
H(WATER)	704.35(BTU/HR.FT2.F)	3999.47(W/M2.C)
H(FOULING)	3792.00(BTU/HR.FT2.F)	21531.75(W/M2.C)
H(METAL)	4393.63(BTU/HR.FT2.F)	24947.90(W/M2.C)
H(AMMONIA)	3058.65(BTU/HR.FT2.F)	17367.63(W/M2.C)
HT SURFACE	762190.25(FT2)	70809.69(M2)
TUBE SHEET DIA	27.194(FT)	8.289(M)
TOT NR OF TUBES	52179.	
SW PRESS DROP	5.636(LBF/IN2)	38.861(KPA)
SALT WATER HOT PIPE		
PIPE I.D.	20.077(FT)	6.120(M)
PIPE LENGTH	300.000(FT)	91.440(M)
SW PIPE VEL	4.597(FT/S)	1.401(M/S)
SW FLOW	334081280.0(LBM/HR)	151535904.0(KG/HR)
SW INLET TEMP	80.000(DEG F)	26.667(DEG C)
SW SALINITY	35. 0/000	
SW PRESS DROP	0.322(LBF/IN2)	2.217(KPA)
SALT WATER COLD PIPE		
PIPE I.D.	18.622(FT)	5.676(M)
PIPE LENGTH	3000.000(FT)	914.400(M)
SW PIPE VEL	5.334(FT/S)	1.626(M/S)
SW FLOW	334871552.0(LBM/HR)	151894368.0(KG/HR)
SW INLET TEMP	40.000(DEG F)	4.444(DEG C)
SW SALINITY	35. 0/000	
SW PRESS DROP	0.508(LBF/IN2)	3.501(KPA)
AMMONIA CIRC PIPE		
PIPE I.D.	2.001(FT)	0.610(M)
PIPE LENGTH	150.000(FT)	45.720(M)
NH3 FLOW	3788708.0(LBM/HR)	1718519.0(KG/HR)
NH3 PRESS DROP	15.033(LBF/IN2)	103.650(KPA)
AMMONIA RE-FLUX CIRC PIPE		
PIPE I.D.	2.000(FT)	0.610(M)
PIPE LENGTH	50.000(FT)	15.240(M)
NH3 FLOW	1136612.0(LBM/HR)	515555.3(KG/HR)
NH3 PRESS DROP	9.874(LBF/IN2)	68.079(KPA)

PUMP AND GEN-TURB PERFORMANCE

EVAP SW PUMP

HEAD PRESS	9.898(FT)	3.017(M)
CAPACITY	653259.5(GAL/MIN)	2472587.0(LIT/MIN)
EFFICIENCY MECH	85.00(PCT)	MOTOR 98.00(PCT)

COND SW PUMP

HEAD PRESS	20.753(FT)	9.227(M)
CAPACITY	652119.2(GAL/MIN)	2468271.0(LIT/MIN)
EFFICIENCY MECH	85.00(PCT)	MOTOR 98.00(PCT)

AMMONIA CIRC PUMP

HEAD PRESS	212.327(FT)	64.717(M)
CAPACITY	12101.7(GAL/MIN)	45305.0(LIT/MIN)
EFFICIENCY MECH	75.00(PCT)	MOTOR 98.00(PCT)

AMMONIA RE-FLUX PUMP

HEAD PRESS	33.016(FT)	11.537(M)
CAPACITY	3732.4(GAL/MIN)	14127.1(LIT/MIN)
EFFICIENCY MECH	75.00(PCT)	MOTOR 98.00(PCT)

GEN-TURB EFFICIENCIES

GEN MECH&ELECT	96.60(PCT)
TURB MECH	99.80(PCT)
TURB INTERNAL	89.88(PCT)

TURB OUTLET QUALITY 96.77(PCT)

POWER REQUIREMENTS

TURB-GEN GROSS	27668.313(HP)	20.633(MW)
EFFICIENCY LOSSES		0.559(MW)
EVAP SW PUMP	1964.851(HP)	1.496(MW)
COND SW PUMP	4129.313(HP)	3.143(MW)
NH3 CIRC PUMP	541.714(HP)	0.412(MW)
NH3 RE-FLUX PUMP	29.097(HP)	0.022(MW)

NET POWER OUTPUT		15.000(MW)

PERCENT PARASITIC POWER 24.59(PCT)

THEMODYNAMIC CYCLE EFFICIENCY 2.65(PCT)

COST OF COMPONENTS

EVAPORATOR	8223672.00(DOLLARS)
CONDENSER	3567154.00(DOLLARS)
GEN-TURBINE	1578776.00(DOLLARS)
GENERATOR	937205.06(DOLLARS)
EVAP SW PUMP	653332.94(DOLLARS)
COND SW PUMP	652298.06(DOLLARS)
NH3 CIRC PUMP	136824.94(DOLLARS)
NH3 RE-FLUX PUMP	64448.34(DOLLARS)

OPTIMUM COST	20849216.00(DOLLARS)
COST PER NET KW OUTPUT	1389.95(DOLLARS)

APPENDIX C

SAMPLE COPES OPTIMIZATION AND SENSITIVITY ANALYSIS DATA

```
$BLOCK A (TITLE CARD)
OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM
$BLOCK B (PROGRAM CONTROL PARAMETERS)
2,16,16
                2                16               16
$BLOCK C (INTEGER OPT CONTROL PARAMETERS)
5,0,0,5
                5                0                0                5
$BLOCK D (FLOATING PT OPT PROG PARAMETERS)
0.0
          0.0
0.0
          0.0
$BLOCK E (TOT NR DESIGN VAR,DESIGN OBJ IDENT AND SIGN)
16,27,-1.0
          16               27             -1.0
$BLOCK F (DESIGN VARIABLE BOUNDS,INIT VALUES & SCALE FACTOR)
1.0,1.0+20
          1.0      1.0+20
1.0,1.0+20
          1.0      1.0+20
1.0,1.0+20
          1.0      1.0+20
1.0,1.0+20
          1.0      1.0+20
85.0,148.0
          85.0      148.0
85.0,148.0
          85.0      148.0
0.5,2.5
          0.5        2.5
0.5,2.5
          0.5        2.5
10.0,1.0+20
          10.0     1.0+20
10.0,1.0+20
          10.0     1.0+20
2.0,10.0
          2.0       10.0
2.0,10.0
          2.0       10.0
2.0,10.0
          2.0       10.0
2.0,10.0
          2.0       10.0
1.4,3.0
          1.4        3.0
1.4,3.0
          1.4        3.0
$BLOCK G (DESIGN VARIABLE IDENT)
1,1,1.0
            1            1            1.0
2,2,1.0
            2            2            1.0
3,3,1.0
            3            3            1.0
4,4,1.0
            4            4            1.0
5,5,1.0
            5            5            1.0
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```
6,6,1.0
          6           6           1.0
7,7,1.0
          7           7           1.0
8,8,1.0
          8           8           1.0
9,9,1.0
          9           9           1.0
10,10,1.0
         10          10           1.0
11,11,1.0
         11          11           1.0
12,12,1.0
         12          12           1.0
13,13,1.0
         13          13           1.0
14,14,1.0
         14          14           1.0
15,15,1.0
         15          15           1.0
16,16,1.0
         16          16           1.0
$BLOCK H (NR OF CONSTRAINED PARAMETERS)
5
          5
$BLOCK I (CONSTRAINT IDENT AND BOUNDS)
17
         17
1.,0.0,3.,0.0
         1.          0.0          3.          0.0
18,20
         18          20
0.1,0.0,1.0+20,0.0
        0.1          0.0      1.0+20          0.0
21,23
         21          23
0.,0.0,1.0+20,0.0
         0.          0.0      1.0+20          0.0
24
         24
30.,0.0,90.,0.0
        30.          0.0         90.          0.0
25,26
         25          26
10.,0.0,1.0+20,0.0
        10.          0.0      1.0+20          0.0
$BLOCK P (SENSITIVITY OBJECTIVES)
27,0
         27           0
1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18,19,20,
          1           2           3           4           5
          9          10          11          12          13
         17          18          19          20          21
         25          26          27
21,22,23,24,25,26,27
          6           7           8
         14          15          16
         22          23          24
$BLOCK Q (SENSITIVITY VARIABLE BOUNDS)
1,6
          1           6
18.,10.,15.,18.,20.,25.
        18.         10.         15.         18.         20.         25.
2,6
          2           6
18.,10.,15.,18.,20.,25.
        18.         10.         15.         18.         20.         25.
3,6
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           3             6
2.,0.5,1.,2.,3.,4.
           2.           0.5          1.          2.          3.          4.
4,6
           4             6
2.,0.5,1.,2.,3.,4.
           2.           0.5          1.          2.          3.          4.
5,6
           5             6
130.,128.,129.,130.,131.,132.
         130.          128.        129.        130.        131.        132.
6,6
           6             6
89.,87.,88.,89.,90.,91.
          89.           87.         88.         89.         90.         91.
7,6
           7             6
1.,.5,.75,1.,1.25,1.5
           1.            .5         .75          1.        1.25         1.5
8,6
           8             6
1.,.5,.75,1.,1.25,1.5
           1.            .5         .75          1.        1.25         1.5
9,6
           9             6
40.,30.,35.,40.,45.,50.
          40.           30.         35.         40.         45.         50.
10,6
          10             6
55.,45.,50.,55.,60.,65.
          55.           45.         50.         55.         60.         65.
11,6
          11             6
7.,5.,6.,7.,8.,9.
           7.            5.          6.          7.          8.          9.
12,6
          12             6
4.,2.,3.,4.,5.,6.
           4.            2.          3.          4.          5.          6.
13,6
          13             6
7.,5.,6.,7.,8.,9.
           7.            5.          6.          7.          8.          9.
14,6
          14             6
4.,2.,3.,4.,5.,6.
           4.            2.          3.          4.          5.          6.
15,6
          15             6
1.6,1.4,1.5,1.6,1.7,1.8
          1.6           1.4         1.5         1.6         1.7         1.8
16,6
          16             6
1.6,1.4,1.5,1.6,1.7,1.8
          1.6           1.4         1.5         1.6         1.7         1.8
END
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C
C
C THESIS PRESENTATION: MODEL ANALYSIS OF THE CLOSED-CYCLE
C OCEAN THERMAL ENERGY CONVERSION(OTEC) POWER SYSTEM
C BY LCDR RAYMOND C. SCHAUBEL,USN
C                                                                       NPU04410
C                                                                       NPU02500
C                          NOMENCLATURE                                 NPU00910
C                                                                       NPU04010
C                                                                       NPU03460
C AC(TDOC) - SUBROUTINE TO COST COND AL TUBING/FT($/FT)
C AC(TDOE) - SUBROUTINE TO COST EVAP AL TUBING/FT($/FT)                 NPU04030
C ACC(TDOC) - SUBROUTINE TO COST ENHANCED COND AL TUBE($/FT)            NPU04120
C AE - SUBROUTINE TO SIZE ENHANCED EVAP AL TUBE THICK(IN)               NPU04070
C AE(TDOC) - SUBROUTINE TO SIZE ENHANCED COND TUBING(IN)                NPU04110
C AEC(TDOC) - SUBROUTINE TO COST ENHANCED EVAP AL TUBE($/FT)            NPU04080
C AF - EVAP TUBE BANK MINIMUM FREE-FLOW AREA(FT2)                       NPU05660
C AINT - PROJECTED ANNUAL INFLATION RATE(DECIMAL)                       NPU01500
C AO - OUTER SURFACE AREA OF A SINGLE EVAP TUBE(FT2)                    NPU04930
C AP(TDOC) - SUBROUTINE TO SIZE COND AL TUBE WALL THICK(IN)             NPU04090
C AP(TDOE) - SUBROUTINE TO SIZE EVAP AL TUBE WALL THICK(IN)             NPU04020
C BAND - VERT. TUBED COND INLET DISTRIBUTION BAND                       NPU00850
C BTUC - 778.2 FT.LBF/BTU                                               NPU03370
C C1 - CONDENSER TUBING COST/FT($/FT)                                   NPU04130
C C2 - CONDENSER TUBE MACHING COST($)                                   NPU04140
C CAF - CONDENSER TUBE FRONTAL AREA(FT2)                                NPU01730
C CAFF - CONDENSER FREE-FLOW AREA(FT2)                                  NPU01740
C CAO - OUTSIDE SURFACE AREA PER COND TUBE(FT2)                         NPU02100
C CAREA - TOTAL SURFACE AREA OF CONDENSER TUBE PROFILE(IN2)             NPU01120
C CBASE - HORIZ. DIST BETWEEN TUBES-CENTER TO CENTER(IN)                NPU01080
C CBFCFC - BUSTLES,FLANGES AND CHANNELS COST($)                         NPU02330
C CBFCFE - EVAP FLANGES,CHANNELS AND FLOW PLATES COST($)                NPU05420
C CCOND - TOTAL CONDENSER FABRICATION COST($)                           NPU02380
C CCSWP - COST OF COND. OR SW COLD PIPE PUMP(DOLLARS)                   NPU06510
C CDPBC - CONDENSER NH3 DIST PLATE AND BAFFLES COST($)                  NPU02320
C CDPBE - EVAP NH3 DISTRIBUTION PLATE AND BAFFLES COST($)               NPU05410
C CELECT - COST OF ELECT POWER GENERATION($)                            NPU00630
C CELEV - TWO-PHASE EFFECT OF COND ELEVATION(LBF/IN2)                   NPU01830
C CESWP - COST OF EVAP SW PUMP(DOLLARS)                                 NPU06180
C CFF - CONDENSER SHELLSIDE FRICTION FACTOR                             NPU01720
C CFRICT - TWO PHASE EFFECT OF COND PIPE FRICTION(LBF/IN2)              NPU01810
C CFT - CONDENSER FILM TEMPERATURE(DEG F)                               NPU01470
C CFTC - CONVERSION OF CFT TO S.I. UNITS(DEG C)                         NPU02540
C CGEN - COST OF AMMONIA GENERATOR($)                                   NPU00620
C CGF - COND MASS VEL FOR MIN FREE-FLOW AREA(LBM/FT2.SEC)               NPU01750
C CHC - CONDENSER HEAT EXCH HEAD COST($)                                NPU02340
C CHE - EVAP HEAT EXCHANGER HEADS COST($)                               NPU05430
C CHSC - CONDENSER HEAT EXCH SHELL COST($)                              NPU02300
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C CHSE - EVAP HEAT EXCHANGER SHELL COST($)                          NPU05400
C CHT - VERT. DIST BETWEEN TUBES-CENTER TO CENTER(IN)               NPU01050
C CL - COND TUBE PROFILE CHANNEL LENGTH(FT)                         NPU00840
C CLMTD - CONDENSER LOG MEAN TEMPERATURE DIFFERENCE(DEG F)          NPU01150
C CLMTDC - CONVERSION OF CLMTD TO S.I.UNITS(DEG C)                  NPU02530
C CMINC - CAPACITY RATE OF CONDENSER SW(BTU/HR.F)                   NPU00700
C CMINE - EVAP SW CAPACITY RATE(BTU/HR.F)                           NPU04670
C CMOM - TWO-PHASE EFFECT OF COND DROP MOMENTUM(LBF/IN2)            NPU01820
C CMW - 746 MW/HP                                                   NPU03400
C CNH3P - COST OF AMMONIA CIRC PUMP(DOLLARS)                        NPU06720
C CNH3RP - COST OF AMMONIA RE-FLUX PUMP($)                          NPU00270
C CNTU - NUMBER OF CONDENSER TRANSFER UNITS                         NPU01210
C CONC - COLD PIPE SALINITY CONCENTRATION                           NPU03380
C CONH - HOT PIPE SALINITY CONCENTRATION                            NPU03390
C CORRF - CORRECTION FACTOR FOR REVISED NH3 HT COEF                 NPU02060
C CPKE - CONDENSER OUTLET RESISTANCE COEFFICIENT                    NPU06260
C CPKI - CONDENSER INLET RESISTANCE COEFFICIENT                     NPU06250
C CPLAT - VERT. DIST BETWEEN TUBES-CENTER TO CENTER(IN)             NPU01110
C CPLONG - HORIZ. DIST BETWEEN TUBES-CENTER TO CENTER(IN)           NPU01100
C CPNH3E - SPECIFIC HEAT OF AMMONIA IN EVAP(BTU/LBM.F)
C CPNH3(EFT) - SUBROUTINE SPECIFIC HEAT AMMONIA(BTU/LBM.F)          NPU05110
C CPR - CONDENSER TUBE PROFILE PITCH RATIO(IN)                      NPU03660
C CPSWC - SPECIFIC HEAT OF COND SW(BTU/LBM.F)                       NPU00680
C CPSWE - EVAP SW SPECIFIC HEAT(BTU/LBM.F)                          NPU04650
C CPSW(TBLKC) - SUBROUTINE SW SPECIFIC HEAT(BTU/LBM.F)              NPU00810
C CPSW(TBLKE) - SUBROUTINE SW SPECIFIC HEAT(BTU/LBM.F)              NPU04660
C CRKE - CONDENSER OUTLET RESISTANCE COEFFICIENT                    NPU06360
C CRKI - CONDENSER INLET RESISTANCE COEFFICIENT                     NPU06350
C CSF - FLUID-SURFACE COMBINATION VALUE                             NPU05280
C CTAREA - CONDENSER TUBE PROFILE AREA-CENTER TO CENTER(IN2)        NPU01090
C CTIC - CONDENSER TUBE INSTALLATION COST($)                        NPU02290
C CTIE - EVAP TUBE INSTALLATION COST($)                             NPU05390
C CTMC - CONDENSER TUBE MATERIAL COST($)                            NPU02280
C CTME - EVAP TUBE MATERIAL COST($)                                 NPU05380
C CTR1C - THERMAL RESISTANCE FOR COND.TUBE SW(HR.F/BTU)             NPU01390
C CTR2C - THERMAL RESISTANCE FOR COND SW FOULING(HR.F/BTU)          NPU01410
C CTR3C - THERMAL RESISTANCE FOR WALL THICK(HR.F/BTU)               NPU01960
C CTR5C - THERMAL RESISTANCE FOR AMMONIA(HR.F/BTU)                  NPU01430
C CTR1E - THERMAL RESIST EVAP(TUBE) SW(HR.F/BTU)                    NPU04940
C CTR2E - THERMAL RESIST EVAP(TUBE)SW FOULING(HR.F/BTU)             NPU04950
C CTR3E - THERMAL RESIST EVAP(TUBE) WALL THICK(HR.F/BTU)            NPU04960
C CTR5E - THERMAL RESIST EVAP(TUBE) FOR AMMONIA(HR.F/BTU)           NPU04970
C CTSLC - CONDENSER TUBE SHEET LABOR COSTS($)                       NPU02260
C CTSLE - EVAP TUBE SHEET LABOR($)                                  NPU05360
C CTSMC - CONDENSER TUBE SHEET MATL COSTS($)                        NPU02270
C CTSME - EVAP TUBE SHEET MATERIAL COST($)                          NPU05370
C CTURB - COST OF AMMONIA TURBINE($)                                NPU00610
C CTWI - CONDENSER TUBE SIDE WALL TEMPERATURE(DEG F)                NPU01450
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C CTW2 - CONDENSER SHELLSIDE WALL TEMPERATURE(DEG F)                    NPU01460
C CTW2C - CONVERSION OF CTW2 TO S.I. UNITS(DEG C)                       NPU02550
C CTWC - CONDENSER TUBE WELDING COST($)                                 NPU02370
C CTWE - EVAP TUBE WELDING COST($)                                      NPU05450
C CWINSE - EVAP WATER INLET,NOZZLES AND SUPPORTS COST($)                NPU05440
C CWINSC - COND WATER INLETS,NOZZLES & SUPPORTS COST($)                 NPU02360
C DBLKT - ABSOLUTE DIFFERENCE BETWEEN TBLKE & RTBLKE(DEG F)             NPU04900
C DBLKTC - DIFFERENCE BETWEEN REVISED AND ASSUMED TBULK TEMP            NPU01170
C DCOND - TOTAL PRESSURE LOSSES ACROSS CONDENSER(LBF/IN2)               NPU06430
C DCONDC - CONVERSION OF DCOND TO S.I. UNITS(KPA)                       NPU02560
C DCOREC - CONDENSER TUBING FRICTION LOSSES(LBF/IN2)                    NPU06420
C DCOREE - EVAPORATOR TUBING FRICTION LOSSES(LBF/IN2)                   NPU06080
C DDEC - PRESSURE LOSSES DUE TO SW DENSITY DIFF.(LBF/IN2)               NPU06310
C DDENC - LOSSES DUE TO SW DENSITY COMPRESSIBILITY(LBF/IN2)             NPU00990
C DDUCTC - SW COLD PIPE INLET/OUTLET DUCT LOSSES(LBF/IN2)               NPU06270
C DDUCTH - SW HOT PIPE INLET DUCT LOSSES(LBF/IN2)                       NPU03110
C DELB - RATIO OF DBLKT TO SCALB                                        NPU04920
C DELBC - RATIO OF DBLKTC WRT SCALBC                                    NPU01190
C DELC - RATIO OF DHNH3C WRT SCALC                                      NPU02090
C DELH - RATIO OF HNH3ER WITH RESPECT TO HNH3E                          NPU05300
C DELT3 - RATIO OF DTEMP3 WITH RESPECT TO SCALT                         NPU06000
C DELT4 - RATIO OF DTEMP4 AND SCALT1                                    NPU01880
C DELTAC - CONDENSING TEMPERATURE(DEG F)                                NPU01480
C DELTAE - EVAP DELTA T FOR BOILING(DEG F)                              NPU05020
C DELTCC - CONVERSION OF DELTAC TO S.I. UNITS(DEG C)                    NPU02570
C DELTEC - CONVERSION OF DELTAE TO S.I. UNITS(DEG C)                    NPU02580
C DELV - RATIO OF DVNH3ER WITH RESPECT TO SCALV                         NPU05960
C DEVAP - TOTAL PRESSURE LOSSES ACROSS EVAPORATOR(LBF/IN2)              NPU06090
C DEVAPC - CONVERSION OF DEVAP TO S.I. UNITS(KPA)                       NPU02590
C DF - GEN-TURB DOUBLE FLOW FACTOR                                      NPU00830
C DH5 - CONSTRAINT DIFFER FOR ENTHALPY AT STATE PT 5                    NPU00430
C DHEAD - LOSSES DUE TO SW DENSITY COMPRESSIBILITY(FT)                  NPU00980
C DHNH3C - ABSOLUTE DIFFERENCE BETWEEN HNH3CR AND HTNH3C                NPU02070
C DHTNH3 - ABSOLUTE DIFF BET HNH3ER AND HNH3E(BTU/HR.FT2.F)             NPU05290
C DICP - COLD PIPE INNER DIAMETER(FT)                                   NPU03490
C DICPC - CONVERSION OF DICP TO S.I. UNITS(M)                           NPU02600
C DIHP - HOT PIPE INNER DIAMETER(FT)                                    NPU03500
C DIHPC - CONVERSION OF DIHP TO S.I. UNITS(M)                           NPU02610
C DINH3 - AMMONIA CIRC PIPE INNER DIAMETER(FT)                          NPU03510
C DINH3C - CONVERSION OF DINH3 TO S.I. UNITS(M)                         NPU02620
C DINH3R - AMMONIA RE-FLUX PIPE INNER DIAMETER(FT)                      NPU03520
C DINHRC - CONVERSION OF DINH3R TO S.I. UNITS(M)                        NPU02630
C DPCORE - EVAPORATOR SHELLSIDE PRESSURE LOSSES(LBF/IN2)                NPU06590
C DPDM - DEMISTER PRESSURE DROP MULTIPLE                                NPU03870
C DPELEC - COND CORE PRESS DROP DUE TO ELEV(LBF/IN2)                    NPU00950
C DPELER - PRESSUR LOSS DUE TO RE-FLUX PIPING EZR(LBF/IN2)              NPU00130
C DPELEV - PRESSURE LOSS DUE TO AMMONIA PIPING EZ2(LBF/IN2)             NPU06620
C DPIPC - TOTAL SW COLD PIPE FRICTION LOSSES(LBF/IN2)/                  NPU06300
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C DPIPCC - CONVERSION OF DPIPC TO S.I.UNITS(KPA)                          NPU02640
C DPIPEC - SW COLD PIPE FRICTION LOSSES(LBF/IN2)                          NPU06290
C DPIPEH - SW HOT PIPE FRICTION LOSSES(LBF/IN2)                           NPU03360
C DPIPH - TOTAL SW HOT PIPE FRICTION LOSSES(LBF/IN2)                      NPU04680
C DPIPHC - CONVERSION OF DPIPH TO S.I. UNITS(KPA)                         NPU02650
C DPIPN - AMMONIA PCIRC PIPE PRESSURE LOSSES(LBF/IN2)                     NPU06630
C DPIPNC - CONVERSION OF DPIPN TO S.I. UNITS(KPA)                         NPU02660
C DPIPNR - RE-FLUX AMMONIA PIPE LOSSES(LBF/IN2)                           NPU00140
C DPMCC - CONVERSION OF DPMPCC TO S.I. UNITS(M)                           NPU02680
C DPMEC - CONVERSION OF DPMPEC TO S.I. UNITS(M)                           NPU02690
C DPMINC -MINOR INLET/OUTLET CONDENSER LOSSES(LBF/IN2)                    NPU06370
C DPMINE - MINOR INLET/OUTLET EVAPORATOR LOSSES(LBF/IN2)                  NPU05620
C DPMNCC - CONVERSION OF DPMPNC TO S.I. UNITS(M)                          NPU02670
C DPMPC - COND. OR SW COLD PIPE PUMP PRESSURE HEAD(LBF/IN2)               NPU06440
C DPMPCC - COND. OR SW COLD PIPE PUMP PRESSURE HEAD(LBF/IN2)              NPU06450
C DPMPE - EVAP. OR SW HOT PIPE PUMP PRESSURE HEAD(LBF/IN2)                NPU06110
C DPMPEC - EVAP. OF SW HOT PIPE PUMP PRESSURE HEAD(FT)                    NPU06120
C DPMPN - AMMONIA CIRC PUMP RESSURE HEAD(LBF/IN2)                         NPU06650
C DPMPNC - AMMONIA CIRC PUMP PRESSURE HEAD (FT)                           NPU06660
C DPMPNR - RE-FLUX AMMONIA PUMP HEAD(LBF/IN2)                             NPU00160
C DPNH3P - AMMONIA CIRCULATION PIPE FRICTION LOSSES(LBF/IN2)              NPU06580
C DPNHRP - RE-FLUX AMMONIA PIPE FRICTION LOSSES(LBF/IN2)                  NPU00110
C DPNRC - CONVERSION OF DPIPNR TO S.I. UNITS(KPA)                         NPU02700
C DPNRCC - CONVERSION OF DPPNRC TO S.I. UNITS(M)                          NPU02710
C DPPNRC - RE-FLUX AMMONIA PUMP HEAD(FT)                                  NPU00170
C DPTHER - PUMPED AMMONIA THERMO. PRESSURE HEAD(LBF/IN2)                  NPU06640
C DPTHRF - PUMPED RE-FLUX AMMONIA THOERMO. PHEAD(LBF/IN2)                 NPU00150
C DSDEM - MOISTURE SEPERATOR PRESSURE DROP(LBF/IN2)                       NPU05730
C DSDEMC - CONVERSION OF DSDEM TO S.I. UNITS(KPA)                         NPU02720
C DSEVAP - EVAPORATOR SHELLSIDE PRESSURE DROP(LBF/IN2)                    NPU05710
C DSEVPC - CONVERSION OF DSEVAP TO S.I. UNITS(KPA)                        NPU02730
C DTC - DRILLING TIME/TUBE SHEET THICK COND(MIN/IN)                       NPU02230
C DTE - EVAP DRILLING TIME/TUBE SHEET THICK(MIN/IN)                       NPU05340
C DTEMP1 - CONSTRAINT FOR SAT UPPER TEMP SYSTEM BOUND                     NPU04370
C DTEMP2 - CONSTRAINT FOR SAT LOWER TEMP SYSTEM BOUND                     NPU04380
C DTEMP3 - ABSOLUTE DIFF BETWEEN T3R AND T3(DEG F)                        NPU05980
C DTEMP4 - ABSOLUTE DIFF BETWEEN T1R AND T1(DEG F)                        NPU01860
C DTEMP5 - CONDENSER OUTLET TEMPERATURE BOUND(DEG F)
C DTLC - CONSTRAINT FOR SAT CONDENSER TUBE LENGTH                         NPU02220
C DVNH3E - ABSOLUTE DIFF BETWEEN VNH3ER AND VNH3E(FT/SEC)                 NPU05940
C E1 - EVAPORATOR TUBING COST/FT($/FT)                                    NPU04050
C E2 - EVAP TUBING MACHINING COST($)                                      NPU04060
C EAF - EVAPORATOR TUBE FRONTAL AREA(FT2)                                 NPU01540
C EAFF - EVAPORATOR FREE-FLOW AREA(FT2)                                   NPU01550
C EAREA - TUBE PROFILE AREA FOR EVAPORATOR TUBES(IN2)                     NPU04620
C EBASE - HALF HORIZ DIST BETWEEN TUBE,CENTER TO CENTER(IN)               NPU04560
C ECC - CONDENSER TUBE ABSOLUTE ROUGHNESS (FT)                            NPU03720
C ECE - EVAPORATOR TUBE ROUGHNESS FACTOR(FT)                              NPU03750
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C ECP - COLD PIPE ABSOLUTE ROUGHNESS(FT)                                NPU03740
C EDC - COND SHELLSIDE FLOW CHANNEL DIAMETER(FT)                        NPU01760
C EDE - SHELLSIDE NH3 FLOW CHANNEL DIAMETER(FT)                         NPU01570
C EELEV - TWO-PHASE EFFECT OF EVAP ELEVATION(LBF/IN2)                   NPU01630
C EEP - GENERATOR MECHANICAL & ELECTRICAL EFFICIENCY(PCT)               NPU03880
C EEPC - CONVERSION OF EEP(DECIMAL)                                     NPU00320
C EFF - EVAP SHELLSIDE FRICTION FACTOR                                  NPU05640
C EFFIC - CONDENSER FIN EFFICIENCY- INSIDE(DECIMAL)                     NPU01370
C EFFIE - EVAPORATOR TUBE FIN EFFICIENCY(DECIMAL)                       NPU04750
C EFFOC - CONDENSER FIN EFFICIENCY- OUTSIDE(DECIMAL)                    NPU01380
C EFFOE - EVAP OUTER TUBE FIN EFFICIENCY(DECIMAL)                       NPU04790
C EFRICT - TWO-PHASE EFFECT OF EVAP PIPE FRICTION (LBF/IN2)             NPU01600
C EFT - EVAP FILM TEMPERATURE(DEG F)                                    NPU05010
C EFTC - CONVERSION OF EFT TO S.I UNITS(DEG C)                          NPU02740
C EHP - ABSOLUTE ROUGHNESS FACTOR OF SW HOT PIPE                        NPU05690
C EHT - HALF VERT HEIGHT BETWEEN TUBE,CENTER TO CENTER(IN)              NPU04540
C ELBOW - 90 DEG ELBOW EQUIV. PIPE DIA(L/D)                             NPU03760
C ELECT - NET ELECTRICAL OUTPUT REQUIRED(MW)                            NPU03980
C ELMTD - EVAP LOG MEAN TEMPERATURE DIFFERENCE(DEG F)                   NPU05040
C ELMTDC - CONVERSION OF ELMTD TO S.I. UNITS(DEG C)                     NPU02750
C EMC - COLD PIPE SW PUMP MOTOR EFFICIENCY(PCT)                         NPU03890
C EMCC - CONVERSION OF EMC(DECIMAL)                                     NPU06480
C EME - HOT PIPE SW PUMP MOTOR EFFICIENCY(PCT)                          NPU03900
C EMEC - CONVERSION OF EME(DECIMAL)                                     NPU06150
C EMNH3 - AMMONIA CIRC PUMP MOTOR EFFICIENCY(PCT)                       NPU03910
C EMNH3C - CONVERSION OF EMNH3(DECIMAL)                                 NPU06700
C EMNHR - AMMONIA RE-FLUX PUMP MOTOR EFFICIENCY(PCT)                    NPU03920
C EMNHRC - CONVERSION OF EMNHR(DECIMAL)                                 NPU00220
C EMOM - TWO-PHASE EFFECT OF EVAP DROP MOMENTUM(LBF/IN2)                NPU01620
C ENH3P - AMMONIA PIPE ABSOLUTE ROUGHNESS(FT)                           NPU03770
C ENTU - NUMBER OF TRANSFER UNITS FOR EVAP                              NPU04860
C EPC - COLD PIPE SW PUMP MECHANICAL EFFICIENCY(PCT)                    NPU03930
C EPCC - CONVERSION OF EPC(DECIMAL)                                     NPU06460
C EPE - HOT PIPE SW PUMP MECHANICAL EFFICIENCY(PCT)                     NPU03940
C EPEC - CONVERSION OF EPE T(DECIMAL)                                   NPU06130
C EPLAT - VERTICAL HEIGHT BETWEEN TUBE,CENTER TO CENTER(IN)             NPU04610
C EPLONG - HORIZ DIST BETWEEN TUBE,CENTER TO CENTER(IN)                 NPU04600
C EPNH3 - AMMONIA CIRC PUMP MECHANICAL EFFICIENCY(PCT)                  NPU03950
C EPNH3C - CONVERSION OF EPNH3(DECIMAL)                                 NPU06670
C EPNHR - AMMONIA RE-FLUX PUMP MECHANICAL EFFICIENCY(PCT)               NPU03960
C EPNHRC - CONVERSION OF EPNHR(DECIMAL)                                 NPU00190
C EPR - EVAPORATOR TUBE PROFILE PITCH RATIO(IN)                         NPU03650
C EPSE - EVAPORATOR EFFECTIVENESS(DECIMAL)                              NPU04870
C ESPACE - EVAP VAPOR FLOW AREA TO MOISTURE SEPARATOR(FT2)              NPU04630
C ETAREA - TUBE PROFILE AREA PER TUBE(IN2)                              NPU04570
C ETRP - GENERATOR TURBINE MECHANICAL EFFICIENCY(PCT)                   NPU03970
C ETRPC - CONVERSION OF ETRP(DECIMAL)                                   NPU00340
C ETURB - INTERNAL TURBINE EFFICIENCY(DECIMAL)                          NPU00590
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C ETURBP - CONVERSION OF ETURB(PCT)                                   NPU00600
C ETW1 - EVAP TUBESIDE WALL TEMPERATURE(DEG F)                        NPU05000
C ETW2 - EVAP SHELLSIDE WALL TEMPERATURE(DEG F)                       NPU04990
C ETW2C - CONVERSION OF ETW2 TO S.I. UNITS(DEG C)                     NPU02760
C EZ1 - DATUM OF AMMONIA PIPING CONFIGURATION                         NPU06600
C EZ2 - ELEVATION OF AMMONIA PIPING(FT)                               NPU03780
C EZR - RE-FLUX AMMONIA PIPING CONFIGURATION HEIGHT(FT)               NPU00120
C FFC - FRICTION FACTOR OF CONDENSER TUBING                           NPU06400
C FFCP - FRICTION FACTOR OF SW COLD PIPE                              NPU06240
C FFE - FRICTION FACTOR OF EVAPORATOR PIPING                          NPU05650
C FFHP - FRICTION FACTOR OF SW HOT PIPE                               NPU02350
C FFNH3P - FRICTION FACTOR OF PIPED AMMONIA                           NPU06560
C FLNH3C - CONVERSION OF FLONH3 TO S.I. UNITS(KG/HR)                  NPU02770
C FLNHRC - CONVERSION OF FLONHR TO S.I. UNITS(KG/HR)                  NPU02780
C FLOCP - COLD PIPE SW FLOW RATE(LBM/HR)                              NPU06220
C FLOCPC - CONVERSION OF FLUCP TO S.I. UNITS(KG/HR)                   NPU02790
C FLOHP - HOT PIPE SW MASS FLOW RATE(LBM/HR)                          NPU04450
C FLOHPC - CONVERSION OF FLOHP TO S.I. UNITS(KG/HR)                   NPU02800
C FLONH3 - AMMONIA MASS FLOW RATE(LBM/HR)                             NPU05580
C FLONHR - RE-FLUX AMMONIA FLOW RA  E(LBM/HR)                         NPU00050
C FNH3RP - FRICTION FACTOR OF PIPED RE-FLUX AMMONIA                   NPU00090
C FPF - PRICING FACTOR FOR DOUBLE-FLOW GEN-TJRBINE                    NPU00820
C G - 32.174 FT/SEC2                                                  NPU03410
C GAL - 7.481 GAL/FT3                                                 NPU03430
C GC - 32.174 FT.LBM/LBF.SEC2                                         NPU03420
C GF - NH3 MASS VEL THRU EVAP FREE-FLOW AREA(LBM/FT2.SEC)             NPU05670
C H1 - ENTHALPY AT STATE PT 1(BTU/LBM)                                NPU05520
C H2 - ENTHALPY AT STATE PT 2(BTU/LBM)                                NPU05540
C H3 - ENTHALPY OF AMMONIA AT STATE PT 3(BTU/LBM)                     NPU05810
C H3A - ENTHALPY AT STATE PT 3A(BTU/LBM)                              NPU05550
C H3F - ENTHALPY OF SAT LIQ AMMONIA AT STATE PT 3(BTU/LBM)            NPU05770
C H3G - ENTHALPY SAT VAPOR AMMONIA AT STATE PT 3(BTU/LBM)             NPU05800
C H4 - ENTHALPY OF AMMONIA AT STATE PT 4(BTU/LBM)                     NPU05920
C H4F - ENTHALPY OF SAT LIQ AMMONIA AT STATE PT 4(BTU/LBM)            NPU05880
C H4G - ENTHALPY SAT VAPOR AMMONIA AT STATE PT 4(BTU/LBM)             NPU05910
C H5 - ENTHALPY AT STATE PT 5(BTU/LBM)                                NPU00400
C HDE - ENTHALPY MOISTURE SEPERATOR GRAVITY DRAIN(BTU/LBM)            NPU05840
C HF5 - ENTHALPY OF LIQ AT STATE PT 5(BTU/LBM)                        NPU00440
C HF(P3) - SUBROUTINE FOR SAT LIQ AMMONIA(BTU/LBM)                    NPU05780
C HF(P4) - SUBROUTINE FOR SAT LIQ AMMONIA(BTU/LBM)                    NPU05890
C HF(PCOND) - SUBROUTINE FOR ENTHALPY(BTU/LBM)                        NPU05530
C HFSWC - PSEUDO HT COEFF COND SW FOULING(BTU/HR.FT2.F)               NPU02170
C HFSWCC - CONVERSION OF HFSWC TO S.I. UNITS(W/M2.C)                  NPU02820
C HFSWE - PSEUDO HT COEF FOR EVAP SW FOULING(BTU/HR.FT2.F)            NPU04820
C HFSWEC - CONVERSION OF HFSWE TO S.I. UNITS(W/M2.C)                  NPU02830
C HG5 - ENTHALPY OFB  VAPOR AT STATE PT 5(BTU/LBM)                    NPU00410
C HG(P3) - SUBROUTINE FOR SAT VAPOR AMMONIA(BTU/LBM)                  NPU05790
C HG(P4) - SUBROUTINE FOR SAT VAPOR AMMONIA(BTU/LBM)                  NPU05900
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C HG(PCOND) - SUBROJTINE FOR ENTHALPY OF A VAPOR(BTU/LBM)            NPU00420
C HG(PEVAP) - SUBROUTINE FOR ENTHALPY(BTU/LBM)                       NPU05570
C HNH3C - PSEUDO HT COEFF COND AMMONIA(BTU/HR.FT2.F)                  NPU02190
C HNH3CC - CONVERSION OF HNH3C TJ S.I. UNITS(W/M2.C)                  NPU02840
C HNH3E - PSEUDO HT COEF FOR EVAP AMMONIA(BTU/HR.FT2.F)               NPU04840
C HNH3EC - CONVERSION OF HNH3E TO S.I. UNITS(W/M2.C)                  NPU02850
C HNH3CR - REVISED AMMONIA HT COEFF COND.(BTU/HR.FT2.F)               NPU02040
C HNH3ER - REVISED EVAP SHELLSIDE HT COEF(BTU/HR.FT2.F)               NPU05160
C HPC - 550 FT.LBF/SEC.HP                                             NPU03440
C HPKI - INLET DUCT TO SW HOT PIPE RESISTANCE COEFFICIENT             NPU02810
C HPKE - OUTLET SW HOT PIPE RESISTANCE COEFFICIENT                    NPU03080
C HSWC - PSEUDO HT COEFF COND SW(BTU/HR.FT2.F)                        NPU02160
C HSWCC - CONVERSION OF HSWC TO S.I. UNITS(W/M2.C)                    NPU02860
C HSWE - PSEUDO HT COEF FOR EVAP SW(BTU/HR.FT2.F)                     NPU04810
C HSWEC - CONVERSION OF HSWE TO S.I. UNITS(W/M2.C)                    NPU02870
C HTDR - EVAP NH3 HT COEF DEVELOPING REGION(BTU/HR.FT2.F)             NPU05240
C HTFDR - EVAP NH3 HT COEF FULLY DEVELOPED REG(BTU/HR.FT2.F)          NPU05250
C HTFSWC - HT COEF FOR COND. SW FOULING(BTU/HR.FT2.F)                 NPU01400
C HTFSWE - EVAP SW FOULING COEF(BTU/HR.FT2.F)                         NPU04770
C HTNH3C - INITIAL AMMONIA HT COEFFICIENT-COND.(BTU/HR.FT2.F)         NPU01250
C HTNH3E - EVAP SHELLSIDE HT COEFFICIENT(BTU/HR.FT2.F)                NPU04420
C HTSWC - HT COEFFICIENT FOR CONDENSER SW(BTU/HR.FT2.F)               NPU01360
C HTSWE - EVAP TUBESIDE HT COEFFICIENT(BTU/HR.FT2.F)                  NPU04740
C HWC - PSEUDO HT COEFF COND WALL THICK.(BTU/HR.FT2.F)                NPU02180
C HWCC - CONVERSION OF HWC TO S.I. UNITS(W/M2.C)                      NPU02880
C HWE - PSEUDO HT COEF FOR EVAP WALL THICK(BTU/HR.FT2.F)              NPU04830
C HWEC - CONVERSION OF HWE TO S.I UNITS(W/M2.C)                       NPU02910
C MATL - TUBING MATERIAL SELECTION                                    NPU03830
C OBJ - OBJECTIVE FUNCTION TO MINIMIZE COST                           NPU02490
C P1 - PRESSURE OF SAT AMMONIA AT STATE PT 1(LBF/IN2)                 NPU01840
C P3 - PRESSURE OF AMMONIA AT STATE PT 3(LBF/IN2)                     NPU05760
C P3C - CONVERSION OF P3 TO S.I. UNITS(KPA)                           NPU02890
C P4 - PRESSURE OF AMMONIA AT STATE PT 4(LBF/IN2)                     NPU05740
C PARAL - PARASITIC PUMP REQUIREMENTS(MW)                             NPU06730
C PAVGD - AVG PRESSURE ACROSS MOISTURE SEPERATOR(LBF/IN2)             NPU05830
C PAVGE - AVERAGE EVAPORATOR PRESSURE(LBF/IN2)                        NPU00010
C PCOND - OPERATION PRESSURE OF THE CONDENSER(LBF/IN2)                NPU03530
C PCONDC - CONVERSION OF PCOND TO S.I. UNITS(KPA)                     NPU02900
C PEVAP - OPERATING PRESSURE OF THE EVAPORATOR(LBF/IN2)               NPU03540
C PEVAPC - CONVERSION OF PEVAP TO S.I. UNITS(KPA)                     NPU02920
C PIE - 3.141592654                                                   NPU03450
C PNH3E - PRANDLT NUMBER FOR EVAP SHELLSIDE FLOW                      NPU05170
C PNSWC - PRANDLT NUMBER FOR CONDENSER SW                             NPU01350
C PNSWE - PRANDLT NUMBER FOR EVAP TUBESIDE FLOW                       NPU04730
C PPP - PERCENT OF PARASITIC POWER(PCT)                               NPU02420
C PROF - TUBE PROFILE SELECTION                                       NPU03710
C PRSYST - CONSTRAINT FOR SAT SYSTEM PRESS RATIO                      NPU04360
C PWRCP - POWER COND. OR CSW COLD PIPE PUMP(HP)                       NPU06470
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C PWRCPM - CONVERSION OF PWRCP(MW)                              NPU06490
C PWREP - POWER EVAP SW PUMP(HP)                                NPU06140
C PWREPM - CONVERSION OF PWREP(MW)                              NPU06160
C PWRNP - POWER OF AMMONIA CIRC PUMP(HP)                        NPU06680
C PWRNPC - CONVERSION OF PWRNP(MW)
C PWRRM - POWER AMMONIA RE-FLUX PUMP MOTOR(MW)                  NPU00230
C PWRRP - POWER AMMONIA RE-FLUX PUMP(HP)                        NPU00200
C PWRTR - GEN-TURBINE OUTLET POWER(HP)                          NPU00860
C QC - AMOUNT OF HEAT REJECTION(BTU/LBM)                        NPU00670
C QCC - CONVERSION OF QC TO S.I. UNITS(MW)                      NPU02930
C QCPMP - DISCHARGE RATE OF COND SW PUMP(FT3/SEC)               NPU06500
C QCPMCC - CONVERSION OF QCPMPC TO S.I.UNITS(LIT/MIN)           NPU02950
C QCT - HEAT TRANSFERED PER TUBE(BTU/HR)                        NPU01440
C QE - AMOUNT OF EVAPORATOR HEAT ADDITION(BTU/HR)               NPU05030
C QEC - CONVERSION OF QE TO S.I. UNITS(MW)                      NPU02940
C QEPMCC - CONVERSION OF QEPMCC TO S.I. UNITS(LIT/MIN)          NPU02960
C QEPMP - DISCHARGE RATE OF EVAP SW PUMP(FT3/SEC)               NPU06170
C QET - HEAT TRANSFERRED PER EVAP TUBE(BTU/HR)                  NPU04980
C QIN - AMOUNT OF HEAT ADDED TO THE SYSTEM(MW)                  NPU02440
C QNPMCC - CONVERSION OF QNPMPC TO S.I.UNITS(LIT/MIN)           NPU02970
C QNPMP - DISCHARGE RATE OF AMMONIA CIRC PUMP(FT3/SEC)          NPU06710
C QRPMCC - CONVERSION OF QRPMPC TO S.I. UNITS(LIT/MIN)          NPU02980
C QRPMP - DISCHARGE RATE AMMONIA RE-FLUX PUMP(FT3/SEC)          NPU00240
C QRPMPC - DISCHARGE RATE AMMONIA RE-FLUX PUMP(GAL/MIN)         NPU00250
C RAVG - AVG SW DENSITY OVER A SPECIFIED DEPTH(LBM/FT3)         NPU00970
C RDEPTH - SW DENSITY F(DEPTH) (LBM/FT3)                        NPU00960
C REMAX REYNOLDS NUMBER FOR MAX EVAP SHELLSIDE FLOW             NPU05630
C REMAXC - REYNOLDS NO FOR MAX CONDENSER SHELLSIDE FLOW         NPU01900
C RFNH3(CFT) - SUBROUTINE FOR AMMONIA DENSITY(LBM/FT3)          NPU01970
C RFNH3(EFT) - SUBROUTINE FOR AMMONIA DENSITY(LBM/FT3)          NPU05510
C RFNH3(T1) - SUBROUTINE SAT LIQ AMMONIA DENSITY(LBM/FT3)       NPU05060
C RGNH3(PEVAP) - SUBROUTINE AMMONIA VAPOR DENSITY(LBM/FT3)      NPU05700
C RKCP - SW COLD PIPE RESISTANCE COEFFICIENT                    NPU06280
C RKE - EVAPORATOR OUTLET RESISTANCE COEFFICIENT                NPU05500
C RKHE - EVAPORATOR TUBING RESISTANCE COEFFICIENT               NPU06070
C RKHC - CONDENSER TUBING RESISTANCE COEFFICIENT                NPU06410
C RKHP - SW HOT PIPE RESISTANCE COEFFICIENT                     NPU03330
C RKI - EVAPORATOR INLET RESISTANCE COEFFICIENT                 NPU05480
C RKNH3P - PIPED AMMONIA RESISTANCE COEFFICIENT                 NPU06570
C RKNHRP - PIPED RE-FLUX AMMONIA RESISTANCE COEFFICIENT         NPU00100
C RHOSWC - DENSITY OF CONDENSER SW(LBM/FT3)                     NPU00730
C RHOSWE - EVAPORATOR SW DENSITY(LBM/FT3)
C RHOSW(TBLKC) - SUBROUTINE SW DENSITY(LBM/FT3)                 NPU00790
C RHOSW(TCIC) - SUBROUTINE FOR DENSITY OF SW(LBM/FT3)           NPU00740
C RHOSW(THIE) - SUBROUTINE SW DENSITY(LBM/FT3)                  NPU00890
C RHOSWD(TLCP) - SUBROUTINE SW DENSITY(DEPTH) (LBM/FT3)         NPU00880
C RHOSWR - REVISED CONDENSER SW DENSITY(LBM/FT3)                NPU01280
C RNH3CH - PSEUDO REYNOLDS NO.FOR HORIZ. CONDENSER              NPU02010
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C RNH3CV - PSEUDO REYNOLDS NO.FOR VERT. CONDENSER                      NPU02020
C RNH3EH - PSEUDO REYNOLDS NR FOR THIN FILM EVAPORATION                NPU05150
C RNH3EV - PSEUDO REYNOLDS NR FOR CONVECTIVE EVAPORATION               NPU05230
C RNH3P - REYNOLDS NUMBER OF PIPED AMMONIA FLOW                        NPU06550
C RNH3RP - REYNOLDS NUMBER OF RE-FLUX AMMONIA PIPE FLOW                NPU00080
C RNSWC - REYNOLDS NUMBER OF CONDENSER TUBESIDE FLOW                   NPU06390
C RNSWCP - REYNOLDS NUMBER OF COLD PIPE SW FLOW                        NPU06230
C RNSWE - REYNOLDS NUMBER OF EVAPORATOR TUBESIDE FLOW                  NPU05720
C RNSWHP - REYNOLDS NUMBER OF HOT PIPE SW FLOW                         NPU02310
C RONH3C - CONDENSER AMMONIA DENSITY(LBM/FT3)                          NPU01950
C RONH3E - DENSITY OF EVAP AMMONIA(LBM/FT3)                            NPU05490
C RONH3P - EVAP SHELLSIDE AMMONIA DENSITY(LBM/FT3)                     NPU05590
C RONH3R - RE-FLUX AMMONIA DENSITY(LBM/FT3)                            NPU00030
C RONH3S - NH3 DENSITY AT MOISTURE SEPARATOR INLET(LBM/FT3)            NPU01030
C RONH3T - CONDENSER SHELLSIDE VAPOR DENSITY(LBM/FT3)                  NPU01670
C RONH3V - EVAPORATOR AMMONIA VAPOR DENSITY(LBM/FT3)                   NPU05680
C ROSWCP - DENSITY OF COLD PIPE SW(LBM/FT3)                            NPU06190
C ROSWHP - HOT PIPE SW DENSITY(LBM/FT3)                                NPU04430
C RTBLKC - REVISED COND BULK TEMPERATURE(DEG F)
C RTBLKE - REVISED EVAP TBLKE TEMPERATURE(DEG F)                       NPU04890
C RTNCT - REVISED NJMBER OF CONDENSER TUBES                            NPU01040
C S4 - ENTROPY AT STATE PT 4(BTU/LBM.R)                                NPU00520
C S4F - ENTROPY OF LIQ AT STATE PT 4(BTU/LBM.R)                        NPU00480
C S4G - ENTROPY OF VAPOR AT STATE PT 4(BTU/LBM.R)                      NPU00490
C S5F - ENTROPY OF A LIQ AT STATE PT 5(BTU/LBM.R)                      NPU00550
C S5G - ENTROPY OF A VAPOR AT STATE PT 5(BTU/LBM.R)                    NPU00560
C SCALB - ABSOLUTE TBLKE TEMPERATURE(DEG F)                            NPU04910
C SCALBC - ABSOLUTE CONDENSER BULK TEMPERATURE(DEG F)                  NPU01180
C SCALC - ABSOLUTE COAMMONIA HT COEFF                                  NPU02080
C SCALH - ABSOLUTE VALUE OF HNH3ER(BTU/HR.FT2.F)                       NPU05330
C SCALT - ABSOLUTE VALUE OF TR3(DEG F)                                 NPU05990
C SCALT1 - ABSOLUTE VALUE OF T1R(DEG F)                                NPU01870
C SCALV - ABSOLUTE VALUE FOR VNH3ER(FT/SEC)                            NPU05950
C SF(T1) - SUBROUTINE FOR ENTROPY OF A LIQ(BTU/LBM.R)                  NPU00530
C SF(T4) - SUBROUTINE FOR ENTROPY OF A LIQ(BTU/LBM.R)                  NPU00500
C SF(T5) - SUBROUTINE FOR ENTROPY OF SAT LIQ NH3(BTU/LBM.R)            NPU01640
C SG(T1) - SUBROUTINE FOR ENTROPY OF A VAPOR(BTU/LBM.R)                NPU00540
C SG(T4) - SUBROUTINE FOR ENTROPY OF A VAPOR(BTU/LBM.R)                NPU00510
C SG(T5) - SUBROUTINE FOR ENTROPY OF SAT VAP NH3(BTU/LBM.R)            NPU01920
C SN - VERTICAL HEIGHT BETWEEN TUBES,CENTER TO CENTER(IN)              NPU04550
C SNC - HORIZONTAL DISTANCE BETWEEN CONDENSER TUBES(IN)                NPU01660
C SPE - VERTICAL DROP DISTANCE BETWEEN TUBES(IN)                       NPU04580
C SPEC - CONVERSION OF SPE(FT)                                         NPU04590
C SWFC - SALT WATER FOULING COEFFICIENT(HR.FT.F/BTU)                   NPU01510
C T1 - SAT LIQ AMMONIA TEMPERATURE AT STATE PT 1(DEG F)                NPU04240
C T1C - CONVERSION OF T1 TO S.I. UNITS(DEG C)                          NPU03020
C T1R - REVISED TEMPERATURE OF AMMONIA AT STATE PT 1(DEG F)            NPU01850
C T3 - AMMONIA VAPOR TEMPERATURE AT STATE PT 3(DEG F)                  NPU04280
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C T3A - SAT VAPOR AMMONIA TEMP AT STATE PT 3A(DEG F)                NPU04270
C T3AC - CONVERSION OF T3A TO S.I. UNITS(DEG C)                     NPU03030
C T3C - CONVERSION OF T3 TO S.I. UNITS(DEG C)                       NPU03040
C T3R - REVISED TEMPERATURE AT STATE PT 3(DEG F)                    NPU05970
C T4 - TEMPERATURE OF AMMONIA AT STATE PT 3(DEG F)                  NPU05820
C T4C - CONVERSION OF T4 TO S.I. UNITS(DEG C)                       NPU03050
C T5 - TEMPERATURE OF AMMONIA AT STATE PT 5(DEG F)                  NPU01530
C TAVGE - AVERAGE EVAPORATOR TEMPERATURE(DEG F)                     NPU00020
C TBLKC - CONDENSER SW BULK TEMPERATURE(DEG F)                      NPU06320
C TBLKCR - CONVERSION OF TBLKC(DEG R)                               NPU06330
C TBLKE - EVAPORATOR SW BULK TEMPERATURE(DEG F)                     NPU04460
C TBLKER - CONVERSION OF TBLKE(DEG R)                               NPU04690
C TCE - THERMODYNAMIC CYCLE EFFICIENCY(PCT)                         NPU02450
C TCIC - CONDENSER SW INLET TEMPERATURE(DEG F)                      NPU00710
C TCICC - CONVERSION OF TCIC TO S.I. UNITS(DEG C)                   NPU03060
C TCOC - CONDENSER SW OUTLET TEMPERATURE(DEG F)                     NPU00720
C TCOCC - CONVERSION OF TCOC TO S.I. UNITS(DEG C)                   NPU03070
C TC(TDOC) - SUBROUTINE TO COST COND TI TUBING/FT($/FT)             NPU04210
C TC(TDOE) - SUBROUTINE TO COST EVAP TI TUBING O/FT($/FT)           NPU04160
C TCC(TDOC) - SUBROUTINE TO COST COND TI TUBING/FT($/FT)            NPU04230
C TDIC - CONDENSER TUBE INNER DIAMETER(IN)                          NPU04290
C TDICC - CONVERSION OF TDIC(FT)                                    NPU01290
C TDIE - EVAPORATOR TUBE INNER DIAMETER(IN)                         NPU04310
C TDIEC - CONVERSION OF TDIE(FT)                                    NPU04330
C TDOC - CONDENSER TUBE OUTTER DIAMETER(IN)                         NPU03560
C TDOCC - CONVERSION OF TDOC(FT)                                    NPU04340
C TDOCCC - CONVERSION OF TDOC TO S.I. UNITS(MM)                     NPU03090
C TDOE - EVAPORATOR TUBE OUTTER DIAMETER(IN)                        NPU03570
C TDOEC - CONVERSION OF TDOE(FT)                                    NPU04350
C TDOECC - CONVERSION OF TDOE TO S.I. UNITS(MM)                     NPU03100
C TEC(TDOE) - SUBROUTINE TO COST ENHANCED EVAP TI TUBE($/FT)        NPU04180
C TENS - SURFACE TENSION OF AMMONIA(LB/FT)                          NPU05270
C TE(TDOC) - SUBROUTINE TO SIZE ENHANCED COND TI TUBING(IN)         NPU04220
C TE(TDOE) - SUBROUTINE TO SIZE ENHANCED EVAP TI TUBING(IN)         NPU04170
C THTAC - TOTAL CONDENSER HT SURFACE AREA(FT2)                      NPU02200
C THTACC - CONVERSION OF THTAC TO S.I. UNITS(M2)                    NPU03140
C THTAE - TOTAL EVAP HEAT TRANSFER AREA(FT3)                        NPU04530
C THTAEC - CONVERSION OF THTAE TO S.I. UNITS(M2)                    NPU03150
C THTAER - REVISED EVAP HEAT TRANSFER AREA(FT2)                     NPU05310
C THIE - EVAPORATOR INLET SW TEMPERATURE(DEG F)                     NPU03580
C THIEC - CONVERSION OF THIE TO S.I. UNITS(DEG C)                   NPU03120
C THOE - EVAP SW OUTLET TEMP(DEG F)                                 NPU04880
C THOEC - CONVERSION OF THOE TO S.I. UNITS(DEG C)                   NPU03130
C TKNH3C - COND AMMONIA THERMAL CONDUCTIVITY(BTU/HR.FT.F)           NPU01980
C TKNH3(CFT) - SUBROUTINE FOR NH3 THERMAL CONDUCTIVITY              NPU01990
C TKNH3E - THERMAL COND OF AMMONIA IN EVAP(BTU/HR.FT.F)             NPU06030
C TKNH3(EFT) - SUBROUTINE THERMAL COND AMMONIA(BTU/HR.FT.F)         NPU05470
C TKSWC - CONDENSER SW THERMAL CONDUCTIVITY(BTU/HR.FT.F)            NPU01320
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C TKSWE - EVAP SW THERMAL CONDUCTIVITY(BTU/HR.FT.F)                    NPU04490
C TKSW(TBLKC) - SUBROUTINE FOR COND. THERMAL CONDUCT.                   NPU01340
C TKW - TUBE MATL THERMAL CONDUCTIVITY(BTU/HR.FT.F)                     NPU03840
C TKWC - CONVERSION OF TKW TO S.I. UNITS(W/M.C)                         NPU03160
C TLC - CONDENSER TUBE LENGTH(FT)                                       NPU03590
C TLCC - CONVERSION OF TLC TO S.I. UNITS(M)                             NPU03170
C TLCP - COLD PIPE LENGTH(FT)                                           NPU03790
C TLCPC - CONVERSION OF TLCP TO S.I. UNITS(M)                           NPU03180
C TLCR - REVISED CONDENSER TUBE LENGTH(FT)                              NPU02210
C TLCRC - CONVERSION OF TLCR TO S.I. UNITS(M)                           NPU02990
C TLD - TUBE LENGTH FOR DEVELOPING CONVECTIVE FLOW(FT)                  NPU05200
C TLE - EVAPORATOR TUBE LENGTH(FT)                                      NPU03600
C TLEC - CONVERSION OF TLE TO S.I. UNITS(M)                             NPU03190
C TLER - REVISED EVAP TUBE LENGTH(FT)                                   NPU05320
C TLERC - CONVERSION OF TLER TO S.I. UNITS(M)                           NPU03000
C TLFD - TUBE LENGTH FOR FULLY DEVELOPED CONVECTIVE FLOW(FT)            NPU05210
C TLHP - HOT PIPE LENGTH(FT)                                            NPU03800
C TLHPC - CONVERSION OF TLHP TO S.I. UNITS(M)                           NPU03200
C TLNH3P - AMMONIA CIRC PIPE LENGTH(FT)                                 NPU03810
C TLNHPC - CONVERSION OF TLNH3P TO S.I. UNITS(M)                        NPU03210
C TLNHRP - AMMONIA RE-FLUX PIPE LENGTH(FT)                              NPU03820
C TLNRPC - CONVERSION OF TLNHRP TO S.I. UNITS(M)                        NPU03010
C TNCT - TOTAL NUMBER OF CONDENSER TUBES                                NPU01000
C TNET - TOTAL NUMBER OF EVAPORATOR TUBES                               NPU04510
C TP(TDOC) - SUBROUTINE TO SIZE COND TI TUBE WALL THICK(IN)             NPU04200
C TP(TDOE) - SUBROUTINE TO SIZE EVAP TI TUBE WALL THICK(IN)             NPU04150
C TR1C - THERMAL RESISTANCE FOR SW(HR.FT2.F/BTU)                        NPU02110
C TR2C - THERMAL RESISTANCE FOR SW FOULING(HR.FT2.F/BTU)                NPU02120
C TR3C - THERMAL RESISTANCE FOR WALL THICK(HR.FT2.F/BTU)                NPU02130
C TR5C - THERMAL RESISTANCE FOR AMMONIA(HR.FT2.F/BTU)                   NPU02140
C TR1E - THERMAL RESISTANCE FOR EVAP SW(HR.FT2.F/BTU)                   NPU04760
C TR2E - THERMAL RESISTANCE EVAP SW FOULING(HR.FT2.F/BTU)               NPU05100
C TR3E - THERMAL RESISTANCE EVAP WALL THICK(HR.FT2.F/BTU)               NPU04780
C TR5E - THERMAL RESISTANCE FOR EVAP AMMONIA(HR.FT2.F/BTU)              NPU04800
C TRNE1 - EVAPORATOR TRANSITION REYNOLDS NUMBER                         NPU05130
C TRNE2 - EVAPORATOR TRANSITION REYNOLDS NUMBER                         NPU05220
C TRNE3 - EVAPORATOR TRANSITION REYNOLDS NUMBER                         NPU05260
C TSAT(PCOND) - SUBRAUTINE FOR SAT AMMONIA TEMP(DEG F)                  NPU04250
C TSAT(PEVAP) - SUBROUTINE FOR SAT AMMONIA TEMP(DEG F)                  NPU04260
C TSDE - EVAPORATOR TUBE SHEET DIAMETER(FT)                             NPU04640
C TSDEC - CONVERSION OF TSDE TO S.I. UNITS(M)                           NPU03230
C TSDCC - CONVERSION OF TSDC TO S.I. UNITS(M)                           NPU03220
C TTC - CONDENSER TUBE WALL THICKNESS(IN)                               NPU04190
C TTCC - CONVERSION OF TTC TO S.I. UNITS(MM)                            NPU03240
C TTE - EVAPORATOR TUBE WALL THICKNESS(IN)                              NPU04040
C TTEC - CONVERSION OF TTE TO S.I. UNITS(MM)                            NPU03250
C TTSC - THICKNESS OF TUBE SHEET COND(IN)                               NPU02240
C TTSE - THICKNESS OF EVAP TUBE SHEET(IN)                               NPU05350
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C TYPEC CONDENSER TYPE SELECTION                                      NPU03680
C TYPEE - EVAPORATOR TYPE  SELECTION                                  NPU03670
C UAC - CONDENSER TUBE CONDUCTANCE(BTU/LBM.F)                         NPU01200
C UC - OVERALL CONDENSER HT COEFFICIENT(BTU/HR.FT2.F)                 NPU02150
C UCC - CONVERSION OF UC TO S.I. UNITS(W/M2.C)                        NPU03260
C UE - EVAP OVERALL HT COEF(OUTER)(BTU/HR.FT2.F)                      NPU04850
C UEC - CONVERSION OF UE TO S.I. UNITS(W/M2.C)                        NPU03270
C VAPC - SPECIFIC VOL COND SHELLSIDE SAT NH3 VAP(FT3/LBM)             NPU01790
C VAPE - SPECIFIC VOL SAT NH3 VAPOR EVAP SHELLSIDE(FT3/LBM)           NPU01610
C VAVGC - AVG SPECIFIC VOF  COND SHELLSIDE NH3 FLOW(FT3/LBM)          NPU01800
C VAVGE - AVERAGE NH3 SPECIFIC VOL EVAP SHELLSIDE(FT3/LBM)            NPU01590
C VF1 - SPECIFIC VOLUME AT STATE PT 1(FT3/LBM)                        NPU05050
C VFR - SPECIFIC VOLUME OF PUMPED RE-FLUX AMMONIA(FT3/LBM)            NPU00260
C VHEAD - MOISTURE SEPARATOR INLET VEL HEAD(LBF/IN2)                  NPU01020
C VISSW - SUBROUTINE SW VISCOSITY(LBM/FT.HR)                          NPU00770
C VISSWC - CONDENSER SW VISCOSITY(LBM/HR.FT)                          NPU01260
C VISSWE - EVAP SW VISCOSITY(LBM/FT.HR)                               NPU04700
C VISSW(TBLKC) - SUBROUTINE SW VISCOSITY(LBM/FT.HR)                   NPU00800
C VISSW(TBLKE) - SUBROUTINE SW VISCOSITY(LBM/FT.HR)                   NPU04710
C VISSW(THIE) - SUBROUTINE SW VISCOSITY(LBM/FT.HR)                    NPU00780
C VISWCP - VISCOSITY OF COLD PIPE SW(LBM/HR.FT)                       NPU06210
C VISWHP - VISCOSITY OF HOT PIPE SW(LBM/HR.FT)                        NPU02250
C VLIQC - SPECIFIC VOL COND SHELLSIDE SAT NH3 LIQ(FT3/LBM)            NPU01780
C VLIQE - SPECIFIC VOL SAT NH3 LIQ  EVAP SHELLSIDE(FT3/LBM)           NPU01580
C VMAXC - MAXIMUM CONDENSER SHELLSIDE NH3 VELOCITY(FT/SEC)            NPU01710
C VMAXE - MAX VELOCITY OF EVAP SHELLSIDE AMMONIA(FT/SEC)              NPU05610
C VNH3C - CONDENSER SHELLSIDE AMMONIA VELOCITY(FT/SEC)                NPU01680
C VNH3E - AVG VEL OF EVAP SHELLSIDE AMMONIA(FT/SEC)                   NPU05600
C VNH3ER - REVISED AMMONIA SHELLSIDE VELOCITY(FT/SEC)                 NPU05930
C VNH3P - VELOCITY OF PIPED AMMONIA(FT/SEC)                           NPU06540
C VNH3RP - RE-FLUX AMMONIA VELOCITY(FT/SEC)                           NPU00060
C VNH3S - NH3 VELOCITY AT MOISTURE SEPARATOR INLET(FT/SEC)            NPU01010
C VSNH3C - CONDENSER AMMONIA VISCOSITY(LBM/HR.FT)                     NPU01490
C VSNH3(CFT) - SUBROUTINE FOR AMMONIA VISCOSITY(LBM/HR.FT)            NPU01940
C VSNH3E - VISCOSITY OF AMMONIA IN EVAP(LBM/HR.FT)                    NPU06020
C VSNH3(EFT) - SUBROUTINE FOR VISCOSITY AMMONIA(LBM/HR.FT)            NPU06040
C VSNH3P - VISCOSITY OF PIPED AMMONIA(LBM/HR.FT)                      NPU06520
C VSNHRP - RE-FLUX AMMONIA VISCOSITY(LBM/HR.FT)                       NPU00040
C VSWC - CONDENSER SW FLOW VELOCITY(FT/SEC)                           NPU03610
C VSWCC - CONVERSION OF VSWC TO S.I. UNITS(M/SEC)                     NPU03280
C VSWCP - COLD PIPE SW FLOW VELOCITY(FT/SEC)                          NPU03620
C VSWCPC - CONVERSION OF VSWCP TO S.I. UNITS(M/SEC)                   NPU03300
C VSWE - EVAPORATOR SW FLOW VELOCITY(FT/SEC)                          NPU03630
C VSWEC - CONVERSION OF VSWE TO S.I. UNITS(M/SEC)                     NPU03290
C VSWHP - HOT PIPE SW FLOW VELOCITY(FT/SEC)                           NPU03640
C VSWHPC - CONVERSION OF VSWHP TO S.I. UNITS(M/SEC)                   NPU03310
C WC - SHELLSIDE NH3 FLOW RATE PER TUBE(LBM/HR)                       NPU02000
C WE - AVG MASS FLOW RATE PER EVAP TUBE(LBM/HR)                       NPU05140
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C WEA - AMMONIA FLOW RATE PER UNIT AXIAL LENGTH(LBM/HR.FT)          NPU05190
C WELECG - GROSS ELECTRICAL LOADING(MW)                              NPU00380
C WELECT - NET ELECT OUTPUT AS A F(EFFIC)(MW)                        NPU00360
C WELOSS - MEASURED LOSSES IN EFFICIENCY(MW)                         NPU00370
C WNET - NET WORK IN THERMODYNAMIC CYCLE(MW)                         NPU02430
C WPNH3 - THERMODYNAMIC AMMONIA CIRC PUMP WORK(BTU/LBM)              NPU05080
C WPSNH3 - ISENTROPIC AMMONIA CIRC PUMP WORK(BTU/LBM)                NPU05070
C X3A - QUALITY OF AMMONIA AT STATE PT 3A(DECIMAL)                   NPU01560
C X3 - QUALITY OF AMMONIA AT STATE PT 3(DECIMAL)                     NPU05750
C X3P - QUALITY OF AMMONIA AT STATE PT 3(PERCENT)                    NPU05860
C X4 - QUALITY OF AMMONIA AT STATE PT 4(DECIMAL)                     NPU05850
C X4P - QUALITY OF AMMONIA AT STATE PT 4(PERCENT)                    NPU05870
C X5 - QUALITY AT STATE PT 5(DECIMAL)                                NPU00450
C X5P - CONVERSION X5(PCT)                                           NPU00460
C X5S - QUALITY AT STATE PT 5S(DECIMAL)                              NPU00570
C X5SP - CONVERSION OF X5S(PCT)                                      NPU00580
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C THESIS PRESENTATION:MODEL ANALYSIS OF THE CLOSED-CYCLE
C OCEAN THERMAL ENERGY CONVERSION(OTEC)POWER SYSTEM
C BY LCDR RAYMOND C. SCHAUBEL,USN
      SUBROUTINE ANALIZ (ICALC)
      COMMON /GLOBCM/ DICP,DIHP,DINH3,DINH3R,PEVAP,PCOND,TDOC,TDOE,TLE,T
     1LC,VSWC,VSWCP,VSWE,VSWHP,EPR,CPR,PRSYST,DTEMP1,DTEMP2,DTEMP5,DH5,D
     2X5,DTLC,ETURBP,TSDE,TSDC,OBJ
      IF (ICALC.GT.1.) GO TO 170
C
C CONSTANTS
C
      BTUC=778.2
      CONC=35
      CONH=35
      CMW=746.E-06
      G=32.174
      GC=32.174
      GAL=7.481
      HPC=550.
      PIE=3.141592654
C
C SALT WATER TEMPERATURE INPUTS(DEG F)
C
      TCIC=40.
      THIE=80.
C
C INITIALIZED DESIGN VARIABLES
C
      DICP=19.6
      DIHP=19.3
      DINH3=2.0
      DINH3R=2.0
      PCOND=89.
      PEVAP=130.
      TDOC=1.
      TDOE=1.
      TLC=56.5
      TLE=40.
      VSWC=6.
      VSWCP=5.5
      VSWE=6.
      VSWHP=4.5
      EPR=1.5
      CPR=1.5
C
C EVAPORATOR/CONDENSER TRIAL SELECTIONS
C     TYPEE=1. HORIZONTAL EVAPORATOR OWENS CORRELATION
C     TYPEE=2. HORIZONTAL EVAPORATOR NON-BOILING CORRELATION
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C     TYPEE=3. HORIZONTAL EVAPORATOR BOILING CORRELATION
C     TYPEC=1. HORIZONTAL CONDENSER
C     TYPEC=2. VERTICAL CONDENSER
C
      TYPEC=1.
      TYPEE=1.
C
C ENHANCEMENT SELECTION
C
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.
      EHFE=1.
C
C TUBE FIN EFFICIENCY
C
      EFFIE=1.0
      EFFOE=1.0
      EFFOC=1.0
      EFFIC=1.0
C
C TUBE PROFILE SELECTION
C     PROF=1. STAGGERED EQUI-LATERAL
C     PROF=2. IN-LINE EQUI-SIDED
      PROF=1.
C
C SYSTEM PIPE GEOMETRY AND MATERIAL
C
      ECC=150.E-06
      ECP=500.E-06
      ECE=150.E-06
      EHP=500.E-06
      ELBOW=30.
      ENH3P=500.E-06
      TLCP=3000.
      TLHP=300.
      TLNH3P=150.
      TLNHRP=50.
C
C TUBING MATERIAL SELECTION
C     TMATL=1. ALUMINUM
C     TMATL=2. TITANIUM
C
      TMATL=1.
      TKW=77.
```

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C
C PROJECTED INFLATION RATE FOR AL TUBE REINSTALLATION(PCT)
C
      AINT=10.
C
C EVAP/MOISTURE SEPARATOR QUALITY REQUIREMENTS(PCT)
C
      X3P=92.
      X4P=99.5
C
C SALT WATER FOULING COEFFICIENT(HR.FT.F/BTU)
C
      SWFC=.00025
C
C PUMP AND GEN-TURB EFFICIENCIES
C
      EEP=96.6
      EMC=98.
      EME=98.
      EMNH3=98.
      EMNHR=98.
      EPC=85.
      EPE=85.
      EPNH3=75.
      EPNHR=75.
      ETRP=99.8
C
C NET OUTPUT POWER REQUIRED(MW)
C
      ELECT=30.
C
C INPUT DATA SI UNIT CONVERSION
C
      DICPC=.3048*DICP
      DIHPC=.3048*DIHP
      DINH3C=.3048*DINH3
      DINHRC=.3048*DINH3R
      PCONDC=6.89476E-03*PCOND
      PEVAPC=6.89476E-03*PEVAP
      TCICC=5.*(TCIC-32.)/9.
      TDOCCC=25.4*TDOC
      TDOECC=25.4*TDOE
      THIEC=5.*(THIE-32.)/9.
      TKWC=1.73707*TKW
      TLCC=.3048*TLC
      TLEC=.3048*TLE
      TLHPC=.3048*TLHP
      TLNHPC=.3048*TLNH3P
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      TLNRPC=.3048*TLNHRP
      TLCPC=.3048*TLCP
      VSWCC=.3048*VSWC
      VSWEC=.3048*VSWE
      VSWCPC=.3048*VSWCP
      VSWHPC=.3048*VSWHP
C
C SUMMARY OF INPUT DATA
C
      WRITE (6,1150)
      IF (TYPEE.EQ.2.) GO TO 10
      WRITE (6,1160)
      GO TO 20
10    CONTINUE
      WRITE (6,1170)
20    CONTINUE
      WRITE (6,1180) TDOE,TDOECC
      WRITE (6,1190) TLE,TLEC
      WRITE (6,1200) VSWE,VSWEC
      WRITE (6,1210) PEVAP,PEVAPC
      IF (TMATL.EQ.2.) GO TO 30
      WRITE (6,1220) TKW,TKWC
      GO TO 40
30    CONTINUE
      WRITE (6,1230) TKW,TKWC
40    CONTINUE
      IF (PROF.EQ.2.) GO TO 50
      WRITE (6,1240)
      GO TO 60
50    CONTINUE
      WRITE (6,1250)
60    CONTINUE
      WRITE (6,1260) EPR
      IF (EHFE.EQ.2.) GO TO 70
      WRITE (6,1270)
      GO TO 80
70    CONTINUE
      WRITE (6,1280)
80    CONTINUE
      IF (TYPEC.EQ.2.) GO TO 90
      WRITE (6,1290)
      GO TO 100
90    CONTINUE
      WRITE (6,1300)
100   CONTINUE
      WRITE (6,1310) TDOC,TDOCCC
      WRITE (6,1320) TLC,TLCC
      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) TLHP,TLHPC
      WRITE (6,1450) VSWHP,VSWHPC
      WRITE (6,1460) THIE,THIEC
      WRITE (6,1470) CONH
      WRITE (6,1480)
      WRITE (6,1490) DICP,DICPC
      WRITE (6,1500) TLCP,TLCPC
      WRITE (6,1510) VSWCP,VSWCPC
      WRITE (6,1520) TCIC,TCICC
      WRITE (6,1530) CONC
      WRITE (6,1540)
      WRITE (6,1550) DINH3,DINH3C
      WRITE (6,1560) TLNH3P,TLNHPC
      WRITE (6,1570)
      WRITE (6,1580) DINH3R,DINHRC
      WRITE (6,1590) TLNHRP,TLNRPC
      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
C
C **** EXECUTION OF INPUT DATA ****
C
C TUBE WALL THICKNESS(INCH) AND TUBE COST PER FT($/FT)
C     ALUMINUM TUBING
C
      IF (TMAIL.GT.1.5) GO TO 210
      IF (EHFE.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
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=ACC(TDOC)
      C2=0.0
      GO TO 250
C
C     TITANIUM TUBING
C
210   CONTINUE
      IF (EHFE.EQ.2.) GO TO 220
      TTE=TP(TDOE)
      E1=TC(TDOE)
      E2=1.21
      GO TO 230
220   CONTINUE
      TTE=TE(TDOE)
      E1=TEC(TDOE)
      E2=1.21
230   CONTINUE
      IF (EHFC.EQ.2.) GO TO 240
      TTC=TP(TDOC)
      C1=TC(TDOC)
```

```
      C2=1.21
      GO TO 250
240   CONTINUE
      TTC=TE(TDOC)
      C1=TCC(TDOC)
      C2=1.21
250   CONTINUE
C
C DESIGN VARIABLE DIMENSION CONVERSION
C
      P1=PCOND
      H1=HF(PCOND)
      T1=TSAT(PCOND)
      T3A=TSAT(PEVAP)
      T3=TSAT(PEVAP)
      T5=TSAT(PCOND)
      TDIC=TDOC-2.*TTC
      TDICC=TDIC/12.
      TDIE=TDOE-2.*TTE
      TDIEC=TDIE/12.
      TDOCC=TDOC/12.
      TDOEC=TDOE/12.
C
C CONSTRAINT FOR SAT SYSTEM PRESSURE RATIO
C
      PRSYST=PEVAP/PCOND
C
C CONSTRAINT FOR SAT UPPER TEMP BOUND
C
      DTEMP1=THIE-T3A
C
C CONSTRAINT FOR SAT LOWER TEMP BOUND
C
      DTEMP2=T1-TCIC
C
C CONSTRAINT FOR CONDENSER OUTLET TEMPERATURE BOUND
C
      DTEMP5=T1-TCOC
C
C
C                   *************************
C                   *                       *
C                   *                       *
C                   *   EVAPORATOR SECTION  *
C                   *                       *
C                   *                       *
C                   *************************
C
```

```
C
C INITIALLY ASSUME AN EVAP TUBE LENGTH(FT)
C ASSUME A HT COEF FOR EVAP SHELLSIDE
      HTNH3E=1000.
C
C HOT PIPE SW MASS FLOW RATE(LBM/HR)
C
      ROSWHP=RHOSW(THIE)
      FLOHP=3600.*ROSWHP*PIE*DIHP**2*VSWHP/4.
C
C EVAP SW DENSITY(INITIALLY ASSUME TBULK=THIE)
C
      TBLKE=THIE
260   CONTINUE
      RHOSWE=RHOSW(TBLKE)
      TKSWE=TKSW(TBLKE)
C
C TOTAL NUMBER OF EVAP TUBES
C
      TNET=4.*FLOHP/(3600.*RHOSWE*PIE*TDIEC**2*VSWE)
C
C TUBE SHEET DIAMETER(FT)
C
C     TUBE PROFILE - STAGGERED
C
      IF (PROF.EQ.2.) GO TO 270
      EHT=EPR*TDOE*0.5
      EBASE=EPR*TDOE*0.866
      ETAREA=EHT*EBASE*2.
      SN=2.*EHT
      SPE=2.*EBASE-TDOE
      SPEC=SPE/12.
      GO TO 280
270   CONTINUE
C
C     TUBE PROFILE - IN-LINE
C
      EPLONG=EPR*TDOE
      SN=EPLONG
      EPLAT=EPR*TDOE
      ETAREA=EPLONG*EPLAT
      SPE=EPR*TDOE-TDOE
      SPEC=SPE/12.
280   CONTINUE
      EAREA=ETAREA*TNET
      TSDE=((4.*EAREA/PIE)**0.5)/12.
C
C TOTAL HT AREA OF EVAP(FT2 OR M2)
```

```
C
      THTAE=TNET*PIE*TDOEC*TLE
C
C CMIN FOR EVAP(BTU/HR.F OR W/C)
C
      CPSWE=CPSW(TBLKE)
      CMINE=FLOHP*CPSWE
C
C OVERALL HT COEF(BTU/HR.FT2.F OR W/M2.C)
C
      TBLKER=TBLKE+459.69
C
C     REYNOLDS NUMBER FOR EVAP TUBESIDE
C
      VISSWE=VISSW(TBLKE)
      RNSWE=3600.*RHOSWE*VSWE*TDIEC/VISSWE
C
C     PRANDTL NUMBER FOR EVAP TUBESIDE
C
      PNSWE=CPSWE*VISSWE/TKSWE
C
C     HT COEF FOR EVAP TUBESIDE
C
      IF (RNSWE.GT.2300.) GO TO 290
C
C     LAMINAR FLOW USING SIEDER-TATE CORRELATION
C
      HTSWE=1.86*TKSWE*(RNSWE*PNSWE)**.3333*(TDIEC/TLE)**.3333/TDIEC
      GO TO 300
290   CONTINUE
C
C     TURBULENT FLOW USING DITUS-BOELTER CORRELATION
C     (INITIALLY ASSUME TBLKE=THIE)
C
      HTSWE=.023*TKSWE*RNSWE**.8*PNSWE**.4/TDIEC
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=1./SWFC
      TR2E=TDOEC/(EFFIE*HTFSWE*TDIEC)
C
C     THERMAL RESISTANCE FOR WALL THICKNESS(HR.FT2.F/BTU)
C
```

```
      TR3E=TDOEC*ALOG(TDOE/TDIE)/(2.*TKW)
C
C     THERMAL RESISTANCE FOR NH3 FOULING(HR.FT2.F/BTU)
C     CONSIDERED NEGLIGIBLE
C
C
C     THERMAL RESISTANCE FOR NH3
C
      EFFOE=1.
      TR5E=1./(EFFOE*HTNH3E)
C
C PSEUDO HT COEF FOR SW(BTU/HR.FT2.; OR W/M2.C)
C
      HSWE=1./TR1E
C
C PSEUDO HT COEF FOR SW FOULING(BTU/HR.FT2.F OR W/M2.C)
C
      HFSWE=1./TR2E
C
C PSEUDO HT COEF FOR WALL THICK(BTU/HR.FT2.F OR W/M2.C)
C
      HWE=1./TR3E
C
C PSEUDO HT COEF FOR AMMONIA(BTU/HR.FT2.F OR W/M2.C)
C
      HNH3E=1./TR5E
C
C     OVERALL HT COEF CALCULATION-OUTTER
C     SURFACE(BTU/HR.FT2.F OR W/M2.C)
C
      UE=1./(TR1E+TR2E+TR3E+TR5E)
C
C NUMBER OF TRANSFER UNITS FOR EVAP(NTU)
C
      ENTU=UE*THTAE/CMINE
C
C EVAP EFFECTIVENESS(EPSILON)
C
      EPSE=1.-EXP(-ENTU)
C
C EVAP SW OUTLET TEMP(F OR C)
C
      THOE=THIE-(THIE-T3A)*(1.-EXP(-ENTU))
C
C REVISED SW AVG BULK TEMP(F)
C
      RTBLKE=(THOE+THIE)/2.
C
```

```
C TEST FOR A SAT TBULK TEMPERATURE
C
      DBLKT=ABS(TBLKE-RTBLKE)
      SCALB=ABS(TBLKE)
      IF (SCALB.LT.0.1) SCALB=0.1
      DELB=DBLKT/SCALB
      IF (DELB.LT.0.001) GO TO 310
      TBLKE=RTBLKE
      GO TO 260
310   CONTINUE
      TBLKE=RTBLKE
C
C FILM TEMP FOR PROPERTY EVALUATION
C INITIALLY ASSUME T3(IDEAL)=T3(ACTUAL)
C
C     THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE(UA)
C     (BTU/HR.F)
C
      AO=PIE*TDOEC*TLE
C
C     THERMAL RESISTANCE SW(HR.F/BTU)
C
      CTR1E=TR1E/AO
C
C     THERMAL RESISTANCE SW FOULING(HR.F/BTU)
C
      CTR2E=TR2E/AO
C
C     THERMAL RESISTANCE WALL THICKNESS(HR.F/BTU)
C
      CTR3E=TR3E/AO
C
C     THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)
C     NEGLIGIBLE
C
C
C     THERMAL RESISTANCE NH3(HR.F/BTU)
C
      CTR5E=TR5E/AO
C
C     HEAT TRANSFERED PER TUBE(BTU/HR)
C
      QET=(TBLKE-T3)/(CTR1E+CTR2E+CTR3E+CTR5E)
C
C     SHELLSIDE WALL TEMPERATURE(F)
C
      ETW2=TBLKE-QET*(CTR1E+CTR2E+CTR3E)
C
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```
C     TUBESIDE WALL TEMPERATURE(F)
C
      ETW1=TBLKE-QET*(CTR1E+CTR2E)
C
C     EVAP FILM TEMP CALCULATION(F)
C
      EFT=(ETW2+T3)/2.
C
C DELTA T TEMPERATURE(F)
C
      DELTAE=ETW2-T3
C
C AMOUNT OF HEAT ADDITION(BTU/HR OR W)
C
      QE=CMINE*(THIE-THOE)
C
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*THTAE)
C
C ISENTROPIC NH3 PUMP WORK(BTU/LBM)
C INITIALLY ASSUME P1(IDEAL)=P1(ACTUAL)
C
      VF1=1./RFNH3(T1)
      WPSNH3=VF1*(PEVAP-P1)*144./BTUC
C
C THERMODYNAMIC NH3 PUMP WORK(BTU/LBM)
C
      EPNH3C=EPNH3/100.
      WPNH3=WPSNH3/EPNH3C
C
C WORKING FLUID PROPERTIES
C
      CPNH3E=CPNH3(EFT)
      VSNH3E=VSNH3(EFT)
      TKNH3E=TKNH3(EFT)
      RONH3E=RFNH3(EFT)
C
C ENTHALPY AT STATE PT 2(BTU/LBM)
C
      H2=H1+WPNH3
C
C ENTHALPY AT STATE PT 3A(BTU/LBM)
C
      H3A=HF(PEVAP)
C
C INITIALIZE NH3 MASS FLOW RATE(LBM/HR)
```

```
C
      H3=HG(PEVAP)
      FLONH3=QE/(H3-H2)
  320 CONTINUE
C
C PRESS DROP EVAP SHELLSIDE(LBF/IN2)
C
C ASSUME VISCOSITY(TWALL)=VISCOSITY(TBULK)
C     MAX VELOCITY THRU MIN-FLOW AREA F(TUBE PROFILE)
C
      RONH3P=RFNH3(T1)
      VNH3E=4.*FLONH3/(3600.*RONH3P*PIE*DINH3**2)
      VMAXE=VNH3E*(SN/(SN-TDOE))
C
C     REYNOLDS NO. FOR MAX SHELLSIDE FLOW
C
      REMAX=3600.*RONH3E*VMAXE*TDOE/(12.*VSNH3E)
C
C     EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
C
      IF (PROF.EQ.2.) GO TO 330
      EFF=(0.25+0.118/((SN-TDOE)/TDOE)**1.08)*REMAX**(-.16)
      GO TO 340
  330 CONTINUE
      EFF=(0.44+(0.08*SN/TDOE)/((SN-TDOE)/TDOE)**(0.43+1.13*TDOE/SN))*RE
     1MAX**(-.15)
  340 CONTINUE
C
C     MASS VELOCITY FOR MIN FREE-FLOW AREA(LBM/FT2.SEC)
C
      EAF=TSDE*TLE
      EAFF=EAF*((SN-TDOE)/SN)
      EGF=FLONH3/(3600.*EAFF)
C
C     CALCULATION OF PRESS DROP EVAP SHELLSIDE(LBF/IN2)
C     USING THE HOMOGENEOUS TWO-PHASE MODEL
C
      X3A=1.0
      EDE=(EPR*TDOE-TDOE)/12.
      VLIQE=1./RFNH3(T3A)
      VAPE=1./RGNH3(PEVAP)
      VAVGE=VLIQE*(1.+X3A*(VAPE-VLIQE)/VLIQE)
      EFRICT=(EFF*EGF**2*VAVGE*TSDE)/(144.*EDE*2.*GC)
      EMOM=(EGF**2*VAVGE)/(144.*GC)
      EELEV=(EGF*TSDE)/(144.*GC)
      DSEVAP=EFRICT+EMOM+EELEV
C
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*TLE
      VNH3S=FLONH3/(3600.*RONH3S*ESPACE)
      VHEAD=RONH3S*VNH3S**2/(2.*GC*144.)
      DSDEM=20.*VHEAD
C
C MOISTURE SEPARATOR DISCHARGE DRAIN ENTHALPY(BTU/LBM)
C
      P4=P3-DSDEM
      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.)*HDE)
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=DVNH3E/SCALV
      IF (DELV.LT.0.001) GO TO 350
      VNH3E=VNH3ER
      GO TO 320
```

```
350   CONTINUE
      VNH3E=VNH3ER
C
C REVISED TEMP AT STATE PT 3(DEG F)
C
      T3R=TSAT(P3)
C
C TEST FOR SAT T3(DEG 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
C PRANDTL NUMBER
C
      PNH3E=CPNH3E*VSNH3E/TKNH3E
C
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
C HORIZONTAL NON-BOILING USING OWENS CORRELATION
C
C     TRANSITION REYNOLDS NUMBER
C
      TRNE1=1680.*(CPNH3E*VSNH3E/TKNH3E)**(-1.5)
C
C     REYNOLDS NUMBER(PSEUDO)
C
      WE=FLONH3/TNET
      RNH3EH=4.*WE/(TLE*VSNH3E)
      IF (RNH3EH.GT.TRNE1) GO TO 370
C
C     LAMINAR FLOW USING OWENS CORRELATION
C
      HNH3ER=2.2*(SPEC/TDOEC)**.1*(VSNH3E**2/(3600.**2*G*RONH3E**2*TKNH3
     1E**3))**(-.3333)*RNH3EH**(-.3333)
      GO TO 420
370   CONTINUE
```

```
C
C     TURBULENT FLOW USING OWENS CORRELATION
C
      HNH3ER=.185*(SPEC/TDOEC)**.1*(VSNH3E**2/(3600.**2*G*RONH3E**2*TKNH
     13E**3))**(-.3333)*PNH3E**.5
      GO TO 420
380   CONTINUE
C
C IDENTIFY PARAMETERS FOR TYPEE=2 OR 3
C
C     FLOW RATE AVERAGED PER TUBE
C
      WE=FLONH3/TNET
C
C     FLOW RATE PER UNIT AXIAL LENGTH OF TUBE
C
      WEA=WE/(PIE*TDOEC/2.)
C
C     TUBE LENGTH FOR DEVELOPING FLOW(FT)
C
      TLD=WEA**(4./3.)/(4.*PIE*TKNH3E/CPNH3E)*(3.*VSNH3E/(3600.**2*RONH3
     1E**2*G))**.3333
C
C     TUBE LENGTH FOR FULLY DEVELOPED FLOW(FT)
C
      TLFD=TNET**.5*PIE*TDOEC/2.
      IF (TYPEE.GT.2.) GO TO 400
C
C HORIZONTAL NON-BOILING LORENZ AND YUNG CORRELATIONS
C
C     TRANSITION REYNOLDS NUMBER FOR CORRELATIONS
C
      TRNE2=5800.*(CPNH3E*VSNH3E/TKNH3E)**(-1.06)
C
C     REYNOLDS NUMBER(PSEUDO-VERTICAL)
C
      RNH3EV=4.*WE/(PIE*TDOEC*VSNH3E)
      IF (RNH3EV.GT.TRNE2) GO TO 390
C
C     LAMINAR FLOW USING LORENZ AND YUNG CORRELATIONS
C
C     CONVECTION IN DEVELOPING REGION
C
      HTDR=3.*CPNH3E*WEA/TLFD
C
C     CONVECTION IN FULLY DEVELOPED REGION
C
      HTFDR=.821*(VSNH3E**2/(3600.**2*RONH3E**2*TKNH3E**3*G))**(-.3333)*
```

```
     1RNH3EV**(-.22)*(1.-TLD/TLFD)
C
C     LAMINAR FLOW CALCULATIONS
C
      HNH3ER=HTDR+HTFDR
      GO TO 420
390   CONTINUE
C     TURBULENT FLOW USING LORENZ AND YUNG CORRELATIONS
C
C     CONVECTION IN DEVELOPING REGION
C
      HTDR=3.*CPNH3E*WEA/TLFD
C
C     CONVECTION IN FULLY DEVELOPED REGION
C
      HTFDR=3.8E-03*(VSNH3E**2/(3600.**2*RONH3E**2*TKNH3E**3*G))**(-.333
     13)*RNH3EV**.4*PNH3E**.65
C
C     TURBULENT FLOW CALCULATION
C
      HNH3ER=HTDR+HTFDR
      GO TO 420
400   CONTINUE
C
C HORIZONTAL BOILING USING LORENZ AND YUNG CORRELATIONS
C
C     TRANSITION REYNOLDS NUMBER FOR CORRELATIONS
C
      TRNE3=5800.*(CPNH3E*VSNH3E/TKNH3E)**(-1.06)
C
C     REYNOLDS NUMBER(PSEUDO-VERTICAL)
C
      RNH3EV=4.*WE/(PIE*TDOCC*VSNH3E)
      IF (RNH3EV.GT.TRNE3) GO TO 410
C
C     LAMINAR FLOW USING LORENZ AND YUNG CORRELATIONS
C
C     CONVECTION IN DEVELOPING REGION
C
      HTDR=3.*CPNH3E*WEA/TLFD
C
C     CONVECTION IN FULLY DEVELOPED REGION
C
      HTFDR=.821*(VSNH3E**2/(3600.**2*RONH3E**2*TKNH3E**3*G))**(-.3333)*
     1RNH3EV**(-.22)*(1.-TLD/TLFD)
C
C     BOILING USING LORENZ AND YUNG CORRELATION
C
```

```
      HFNH3E=HF(P3)
      HGNH3E=HG(P3)
      TENS=1.6038998E-03
      CSF=.0154
      HTB=VSNH3E*(HGNH3E-HFNH3E)/(CSF**3*(TENS/RONH3E)**.5)*(CPNH3E/((HG
     1NH3E-HFNH3E)*PNH3E))**3*DELTAE**2
C
C     LAMINAR FLOW CORRELATION
C
      HNH3ER=HTB+HTDR+HTFDR
      GO TO 420
410   CONTINUE
C
C     TURBULENT FLOW USING LORENZ AND YUNG CORRELATIONS
C
C     CONVECTION IN DEVELOPING REGION
C
      HTDR=3.*CPNH3E*WEA/TLFD
C
C     CONVECTION IN FULLY DEVELOPED REGION
C
      HTFDR=3.8E-03*(VSNH3E**2/(3600.**2*RONH3E**2*TKNH3E**3*G))**(-.333
     13)*RNH3EV**.4*PNH3E**.65
C
C     BOILING USING LORENZ AND YUNG CORRELATION
C
      HGNH3E=HG(P3)
      HFNH3E=HF(P3)
      TENS=1.6038998E-03
      CSF=.0154
      HTB=VSNH3E*(HGNH3E-HFNH3E)/(CSF**3*(TENS/RONH3E)**.5)*(CPNH3E/((HG
     1NH3E-HFNH3E)*PNH3E))**3*DELTAE**2
C
C     TURBULENT FLOW CALCULATIONS
C
      HNH3ER=HTB+HTDR+HTFDR
420   CONTINUE
C
C TEST FOR SAT HTNH3E
C
      DHTNH3=ABS(HNH3ER-HTNH3E)
      SCALH=ABS(HNH3ER)
      IF (SCALH.LT.0.1) SCALH=0.1
      DELH=DHTNH3/SCALH
      IF (DELH.LT.0.001) GO TO 430
      HTNH3E=HNH3ER
      GO TO 260
430   CONTINUE
```

```
      HTNH3E=HNH3ER
C
C REVISED EVAP HEAT TRANSFER AREA(FT2)
C
      THTAER=ENTU*CMINE/UE
C
C REVISED EVAP TUBE LENGTH(FT)
C
      TLER=THTAER/(PIE*TDOEC*TNET)
C
C TEST FOR SAT TUBE LENGTH
C
C COST CF EVAPORATOR UNIT($)
      IF (TSDE.GT.35.) GO TO 480
C
C EVAPORATOR TUBE SHEET DIAMETER(10-35)FT
C
C     DRILLING TIME/TUBE SHELL THICK(MIN/IN)
C
      DTE=0.66*(TDOE-.5)
C
C     THICKNESS OF TUBE SHEET(IN)
C
      TTSE=0.56*TSDE**0.68
C
C     TUBE SHEET LABOR COST($)
C
      CTSLE=156695.*(TNET/9630.)*(DTE/0.66)*(TTSE/4.)
C
C     TUBE SHEET MATERIAL COST($)
C
      CTSME=189.486*TSDE**2.3
      IF (TMATL.EQ.1.) GO TO 440
C
C     TUBE MATERIAL COST($)
C
      CTME=(E1*TLE+E2)*TNET*(TDOE/1.5)
C
C     TUBE INSTALLATION COST($)
C
      CTIE=34.*TNET*TDOE**0.7
      GO TO 450
440   CONTINUE
C
C     TUBF MATERIAL COST($)
C
      CTME=(E1*TLE+E2)*TNET*(TDOE/1.5)*(1.+(1.+AINT/100.)**10+(1.+AINT/1
     100.)**20)
```

AD-A098 567 NAVAL POSTGRADUATE SCHOOL MONTEREY CA F/G 20/13
OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM.(U)
DEC 80 R C SCHAUBEL

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3 of 3

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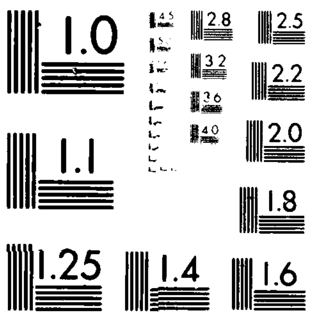

MICROCOPY RESOLUTION TEST CHART

NATIONAL BUREAU OF STANDARDS 1963 A

```
C
C      TUBE INSTALLATION COST($)
C
       CTIE=34.*TNET*TDOE**0.7*(1.+(1.+AINT/100.)**10+(1.+AINT/100.)**20)
450    CONTINUE
C
C      HEAT EXCH SHELL COST($)
C
       CHSE=177265.*(TLE+6.)/31.*(TSDE/18.)**2
C
C      NH3 DIST PLATE AND BAFFLES($)
C
       CDPBE=93865.75*(TNET/9630.)*(DTE/0.66)*(TSDE/18.)**2
C
C      BUSTLE,FLANGES,CHANNELS,AND FLOW PLATES COST($)
C
       CBFCFE=308550.*(TSDE/18.)**2
C
C      HEAT EXCH HEADS COST($)
C
       CHE=53240.*(TSDE/18.)**3
C
C      WATER INLET,NOZZLES AND SUPPORTS COST($)
C
       CWINSE=220310.75*(TSDE/18.)**2
C
C      TUBE WELDING COST($)
C
       CTWE=0.0
       IF (TMATL.EQ.1.) GO TO 470
       IF (TNET.GT.36000) GO TO 460
       CTWE=14.73*TNET**1.03*(TDOE/1.5)**0.7
       GO TO 470
460    CONTINUE
       CTWE=0.8797*TNET**1.3*(TDOE/1.5)**0.7
470    CONTINUE
       CEVAP=(CTSLE+CTSME+CTME+CTIE+CHSE+CBFCFE+CHE+CWINSE+CTWE+CDPBE)
       GO TO 550
480    CONTINUE
C
C EVAPORATOR TUBE SHEET DIAMETER(35-50)FT
C
C      DRILLING TIME/TUBE SHELL THICK(MIN/IN)
C
       DTE=0.66*(TDOE-.5)
C
C      THICKNESS OF TUBE SHEET(IN)
C
```

```
      TTSE=0.56*TSDE**0.68
C
C     TUBE SHEET MATERIAL AND LABOR COSTS($)
C
      IF (TMATL.EQ.1.) GO TO 490
      CTSLE=55.189*TNET**0.791*TSDE**0.68*DTE
      CTSME=29.566*TSDE**2.014*TTSE
      GO TO 500
490   CONTINUE
      CTSLE=73.81*TNET**0.791*TSDE**0.68*DTE
      CTSME=354.3*TSDE**1.61*TTSE
500   CONTINUE
      IF (TMATL.EQ.1.) GO TO 510
C
C     TUBE MATERIAL COST($)
C
      CTME=(E1*TLE+E2)*TNET*(TDOE/1.5)
C
C     TUBE INSTALLATION COST($)
C
      CTIE=36.542*TNET*TDOE**0.7
      GO TO 520
510   CONTINUE
C
C     TUBE MATERIAL COST($)
C
      CTME=(E1*TLE+C2)*TNET*(TDOE/1.5)*(1.+(1.+AINT/100.)**10+(1.+AINT/1
     100.)**20)
C
C     TUBE INSTALLATION COST($)
C
      CTIE=36.542*TNET*TDOE**0.7*(1.+(1.+AINT/100.)**10+(1.+AINT/100.)**
     120)
520   CONTINUE
C
C     HEAT EXCH SHELL COST($)
C
      CHSE=12.544*(TLE+6.)*TSDE**2.06
C
C     NH3 DIST PLATE AND BAFFLES COST($)
C
      CDPBE=158.099*TSDE**1.82+72.419*TNET**0.873*DTE
C
C     BUSTLE,FLANGE,CHANNELS AND FLOW PLATE COSTS($)
C
      CBFCFE=472.977*TSDE**2.12
C
C     HEAT EXCH HEAD COST($)
```

```
C
      CHE=1725.31*TSDE**1.45
C
C     WATER INLET,NOZZLES AND SUPPORTS COST($)
C
      CWINSE=7445.297*TSDE**1.1
C
C     TUBE WELDING COSTS($)
C
      CTWE=0.0
      IF (TMATL.EQ.1.) GO TO 540
      IF (TNET.GT.36000) GO TO 530
      CTWE=14.73*TNET**1.03*(TDOE/1.5)**0.7
      GO TO 540
530   CONTINUE
      CTWE=0.8797*TNET**1.3*(TDOE/1.5)**0.7
540   CONTINUE
      CEVAP=(CTSLE+CTSME+CTME+CTIE+CHSE+CDPBE+CBFCFE+CHE+CWINSE+CTWE)
550   CONTINUE
C
C
C              *****************************
C              *                           *
C              *                           *
C              *   PARASITIC LOSS SECTION  *
C              *                           *
C              *                           *
C              *****************************
C
C
C EVAPORATOR SALT WATER PUMP OR HOT PIPE PUMP
C DELTA P EVAP SW PUMP
      ROSWHP=RHOSW(THIE)
      THIER=THIE+459.69
      VISWHP=VISSW(THIE)
C
C DELTA P SW HOT PIPE USING DARCEY-WEISBACH CORRELATION
C (LBF/IN2)
C
C     REYNOLDS NUMBER FOR HOT PIPE FLOW
C
      RNSWHP=3600.*ROSWHP*VSWHP*DIHP/VISWHP
C
C     FRICTION FACTOR FOR LAMINAR FLOW
C
      IF (RNSWHP.GT.2300.) GO TO 560
      FFHP=64./RNSWHP
      GO TO 570
```

```
560   CONTINUE
C
C     FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
      FFHP=1.325/(ALOG(EHP/(3.7*DIHP)+5.74/RNSWHP**.9))**2
570   CONTINUE
C
C     INTAKE DUCT LOSSES(ASSUME K=1.5 FOR INLET SCREEN AND
C     ENTERS PLENUM PRIOR TO EVAP WHERE AREA IS ABRUPTLY
C     CHANGED TO 2 TIMES THE PIPE DIAMETER(LBF/IN2)
C
      HPKI=1.5
      HPKE=(1.-(DIHP/(2.*DIHP))**2)**2
      DDUCTH=(HPKI+HPKE)*ROSWHP*VSWHP**2/(2.*GC*144.)
C
C     PIPE FRICTION LOSSES(LBF/IN2)
C
      RKHP=FFHP*TLHP/DIHP
      DPIPEH=RKHP*ROSWHP*VSWHP**2/(2.*GC*144.)
C
C     OUTLET DUCT LOSSES(ASSUME NO OUTLET PIPING)
C
C
C     DELTA P PIPE LOSS CALCULATION(LBF/IN2)
C
      DPIPH=DDUCTH+DPIPEH
C
C DELTA P EVAPORATOR(ASSUME NO OUTLET PIPING) USING DARCEY-
C WEISBACH CORRELATION(LBF/IN2)
C
C     MINOR ENTRY/EXIT LOSSES(ASSUME KI=.05 WELL ROUNDED
C     TUBE ENTRANCE AND KE=1. EXPANSION TO AN INFINITE
C     RESEVOIR)(LBF/IN2)
C
      RKI=.05
      RKE=1.
      DPMINE=(RKI+RKE)*RHOSWE*VSWE**2/(2.*GC*144.)
C
C     DELTA P EVAPORATOR CORE(LBF/IN2)
C
C     REYNOLDS NUMBER FOR EVAP(DETERMINED IN EVAP SECTION)
C
C     FRICTION FACTOR LAMINAR FLOW
C
      IF (RNSWE.GT.2300.) GO TO 580
      FFE=64./RNSWE
      GO TO 590
580   CONTINUE
```

```
C                                                                       A1105
C     FRICTION FACTOR TURBULENT FLOW USING STREETER CORR                A1106
C                                                                       A1107
      FFE=1.325/(ALOG(ECE/(3.7*TDIEC)+5.74/RNSWE**.9))**2               A1108
590   CONTINUE                                                          A1109
C                                                                       A1110
C     DELTA P EVAP CORE CALCULATED(LBF/IN2)                             A1111
C                                                                       A1112
      RKHE=FFE*TLE/TDIEC                                                A1113
      DCOREE=RKHE*RHOSWE*VSWE**2/(2.*GC*144.)                           A1114
C                                                                       A1115
C     DELTA P EVAPORATOR CALCULATED(LBF/IN2)                            A1116
C                                                                       A1117
      DEVAP=DPMINE+DCOREE                                               A1118
C                                                                       A1119
C     DELTA P EVAP SW PUMP CALCULATION(LBF/IN2 OR FT)                   A1120
C                                                                       A1121
      DPMPE=DDUCTH+DPIPEH+DPMINE+DCOREE                                 A1122
      DPMPEC=144.*DPMPE*GC/(ROSWHP*G)                                   A1123
C                                                                       A1124
C POWER EVAP SW PUMP(HP)                                                A1125
C                                                                       A1126
      EPEC=EPE/100.                                                     A1127
      PWREP=FLOHP*DPMPEC*G/(EPEC*HPC*3600.*GC)                          A1128
C                                                                       A1129
C POWER EVAP SW PUMP MOTOR(MW)                                          A1130
C                                                                       A1131
      EMEC=EME/100.                                                     A1132
      PWREPM=PWREP*CMW/EMEC                                             A1133
C                                                                       A1134
C DISCHARGE RATE EVAP SW PUMP(FT3/SEC OR GAL/MIN)                       A1135
C                                                                       A1136
      QEPMP=PIE*DIHP**2*VSWHP/4.                                        A1137
      QEPMPC=QEPMP*60.*GAL                                              A1138
C                                                                       A1139
C COST OF EVAP SW PUMP($)                                               A1140
C                                                                       A1141
      CESWP=((QEPMPC/1000.)*.75+50.)*1.21E+03                           A1142
C                                                                       A1143
C CONDENSER SALT WATER PUMP OR COLD PIPE PUMP                           A1144
C                                                                       A1145
C DELTA P CONDENSER SW PUMP(ASSUME TSW(IN)=TSW(OUT),V1=V2)              A1146
      ROSWCP=RHOSW(TCIC)                                                A1147
      TCICR=TCIC+459.69                                                 A1148
      VISWCP=VISSW(TCIC)                                                A1149
      FLOCP=3600.*ROSWCP*PIE*DICP**2*VSWCP/4.                           A1150
C                                                                       A1151
C DELTA P SW COLD PIPE(ASSUME V1=V2,TSW(IN)=TSW(OUT) USING              A1152
```

```
C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C
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
       FFCP=64./RNSWCP
       GO TO 610
600    CONTINUE
C
C      FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
       FFCP=1.325/(ALOG(ECP/(3.7*DICP)+5.74/RNSWCP**.9))**2
610    CONTINUE
C
C      MINOR ENTRY/EXIT LOSSES(ASSUME KI=.05 WELL ROUNDED
C      PIPE ENTRANCE AND ENTERS PLENUM PRIOR TO CONDENSER
C      WHERE AREA IS ABRUPTLY CHANGED TO 2 TIMES PIPE
C      DIAMETER)(LBF/IN2)
C
       CPKI=.05
       CPKE=(1.-(DICP/(2.*DICP))**2)**2
       DDUCTC=(CPKI+CPKE)*ROSWCP*VSWCP**2/(2.*GC*144.)
C
C      PIPE FRICTION LOSSES(LBF/IN2)
C
       RKCP=FFCP*(TLCP/DICP+ELBOW)
       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)
       RAVG=(RDEPTH+64.184)/2.
       DHEAD=((RDEPTH-64.184)/RDEPTH)*TLCP/2.
       DDENC=G*RAVG*DHEAD/(144.*GC)
C
C DELTA P CONDENSER(ASSUME NO OUTLET PIPING) USING
C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C
C INITIALLY ASSUME TBLK=TSW(IN)
       TBLKC=TCIC
```

```
620   CONTINUE
      TBLKCR=TBLKC+459.69
      RHOSWC=RHOSW(TBLKC)
C
C     MINOR ENTRY/EXIT LOSSES(ASSUME KI=.05 WELL ROUNDED
C     TUBE ENTRANCE AND KE=1. EXPANSION TO AN INFINITE
C     RESEVOIR(LBF/IN2)
C
      RKI=.05
      RKE=1.
      DPMINC=(RKI+RKE)*RHOSWC*VSWC**2/(2.*GC*144.)
C
C     DELTA P CONDENSER CORE(LBF/IN2)
C
C
C     REYNOLDS NUMBER FOR COND TUBESIDE
C
      VISSWC=VISSW(TBLKC)
      RNSWC=3600.*RHOSWC*VSWC*TDICC/VISSWC
C
C     FRICTION FACTOR LAMINAR FLOW
C
      IF (RNSWC.GT.2300.) GO TO 630
      FFC=64./RNSWC
      GO TO 640
630   CONTINUE
      FFC=1.325/(ALOG(ECC/(3.7*TDICC)+5.74/RNSWC**.9))**2
640   CONTINUE
C
C     DELTA P COND CORE CALCULATIO(LBF/IN2)
C
      RKHC=FFC*TLC/TDICC
      DCOREC=RKHC*RHOSWC*VSWC**2/(2.*GC*144.)
C
C     DELTA P CONDENSER DUE TO CORE ELEVATION(LBF/IN2)
C
      IF (TYPEC.GT.1.) GO TO 650
      DPELEC=0.
      GO TO 660
650   CONTINUE
C
C     VERTICAL TUBED CONDENSER SW INLET ENTERS AT THE TOP
C     OF THE HEAT EXCHANGER
C
      DPELEC=0.
660   CONTINUE
C
C     DELTA P CONDENSER CALCULATION(LBF/IN2)
```

```
C
      DCOND=DPMINC+DCOREC+DPELEC
C
C     DELTA P COND SW PUMP CALCULATION(LBF/IN2 OR FT)
C
      DPMPC=DDUCTC+DDENC+DPMINC+DCOREC+DPIPEC
      DPMPCC=144.*DPMPC*GC/(ROSWCP*G)
C
C POWER COND SW PUMP(HP)
C
      EPCC=EPC/100.
      PWRCP=FLOCP*DPMPCC*G/(EPCC*HPC*3600.*GC)
C
C POWER COND SW PUMP MOTOR(MW)
C
      EMCC=EMC/100.
      PWRCPM=PWRCP*CMW/EMCC
C
C DISCHARGE RATE OF COND SW PUMP(FT3/SEC OR GAL/MIN)
C
      QCPMP=PIE*DICP**2*VSWCP/4.
      QCPMPC=QCPMP*60.*GAL
C
C COST OF COND SW PUMP($)
C
      CCSWP=((QCPMPC/1000.)*0.75+50.)*1.21E+03
C
C AMMONIA CIRCULATION PUMP
C
C DELTA P NH3 CIRC PUMP
      VSNH3P=VSNH3(T1)
      RONH3P=RFNH3(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
      VNH3P=4.*FLONH3/(3600.*RONH3P*PIE*DINH3**2)
C
C     REYNOLDS NUMBER FOR NH3 PIPE FLOW
C
      RNH3P=3600.*RONH3P*VNH3P*DINH3/VSNH3P
C
C     FRICTION FACTOR LAMINAR FLOW
C
      IF (RNH3P.GT.2300.) GO TO 670
      FFNH3P=64./RNH3P
```

```
      GO TO 680
670   CONTINUE
C
C     FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
      FFNH3P=1.325/(ALOG(ENH3P/(3.7*DINH3)+5.74/RNH3P**.9))**2
680   CONTINUE
C
C     PIPE FRICTION LOSSES(LBF/IN2)
C
      RKNH3P=FFNH3P*(TLNH3P/DINH3+4.*ELBOW)
      DPNH3=RKNH3P*RONH3P*VNH3P**2/(2.*GC*144.)
C
C DELTA P CORE EVAPORATOR
C
      DPCORE=DSEVAP+DSDEM
C
C DELTA P DUE TO PIPING ELEVATION(LBF/IN2)
C
      EZ1=0.
      EZ2=TSDE+25
      DPELEV=RONH3P*G*(EZ2-EZ1)/(GC*144.)
C
C
C     DELTA P PIPE LOSS CALCULATION(LBF/IN2)
      DPIPN=DPNH3+DPELEV
C
C DELTA P THERMODYNAMICALLY(LBF/IN2)
C
      DPTHER=PEVAP-PCOND
C
C DELTA P NH3 CIRC PUMP(LBF/IN2 OR FT)
C
      DPMPN=DPNH3+DPCORE+DPELEV+DPTHER
      DPMPNC=144.*DPMPN*GC/(RONH3P*G)
C
C POWER NH3 CIRC PUMP(HP)
C
      EPNH3C=EPNH3/100.
      PWRNP=FLONH3*DPMPNC*G/(EPNH3C*HPC*3600.*GC)
C
C POWER NH3 CIRC PUMP MOTOR(MW)
C
      EMNH3C=EMNH3/100.
      PWRNPM=PWRNP*CMW/EMNH3C
C
C DISCHARGE RATE NH3 CIRC PUMP(FT3/SEC OR GAL/MIN)
C
```

```
      QNPMP=PIE*DINH3**2*VNH3P/4.
      QNPMPC=QNPMP*60.*GAL
C
C COST OF NH3 CIRC PUMP($)
C
      VF1=1./RFNH3(T1)
      CNH3P=(FLONH3*VF1/80100.)**0.64*1.21E+05
C
C EVAPORATOR RE-FLUX PUMP
C ASSUME RE-FLUX MASS FLOW RATE=0.3 X FLONH3
C
C DELTA P NH3 RE-FLUX PIPING(ASSUME V1=V2)
C USING DARCEY-WEISBACH CORRELATION(LBF/IN2)
      PAVGE=(PEVAP+P3)/2.
      TAVGE=TSAT(PAVGE)
      RONH3R=RFNH3(TAVGE)
      VSNHRP=VSNH3(TAVGE)
C
C     NH3 FLOW VELOCITY(FT/SEC)
C
      FLONHR=0.3*FLONH3
      VNH3RP=4.*FLONHR/(3600.*RONH3R*PIE*DINH3R**2)
C
C     REYNOLDS NO FOR NH3 RE-FLUX PIPE FLOW
C
      RNH3RP=3600.*RONH3R*VNH3RP*DINH3R/VSNHRP
C
C     FRICTION FACTOR LAMINAR FLOW
C
      IF (RNH3RP.GT.2300.) GO TO 690
      FNH3RP=64./RNH3RP
      GO TO 700
690   CONTINUE
C
C     FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
      FNH3RP=1.325/(ALOG(ENH3P/(3.7*DINH3R)+5.74/RNH3RP**0.9))**2
700   CONTINUE
C
C     PIPE FRICTION LOSSES(LBF/IN2)
C
      RKNHRP=FNH3RP*(TLNHRP/DINH3R+4.*ELBOW)
      DPNHRP=RKNHRP*RONH3R*VNH3RP**2/(2.*GC*144.)
C
C DELTA P DUE TO PIPING ELEVATION(LBF/IN2)
C
      EZR=TSDE+10
      DPELER=RONH3R*G*(EZR-EZ1)/(GC*144.)
```

```
C
C DELTA P PIPE LOSSES(LBF/IN2)
C
      DPIPNR=DPNHRP+DPELER
C
C DELTA P NH3 RE-FLUX THERMODYNAMICALLY(LBF/IN2)
C USING A SINGLE PHASE PRESSURE MODEL
C
      RONHEV=RGNH3(PEVAP)
      DPTHRF=2.*EFF*EGF**2*TNET**0.5/(144.*RONHEV*GC)
C
C DELTA P NH3 RE-FLUX PUMP(LBF/IN OR FT)
C
      DPMPNR=DPNHRP+DPELER+DPTHRF
      DPPNRC=144.*DPMPNR*GC/(RONH3R*G)
C
C POWER NH3 RE-FLUX PUMP(HP)
C
      EPNHRC=EPNHR/100.
      PWRRP=FLONHR*DPPNRC*G/(EPNHRC*HPC*3600.*GC)
C
C POWER NH3 RE-FLUX PUMP MOTOR(MW)
C
      EMNHRC=EMNHR/100.
      PWRRM=PWRRP*CMW/EMNHRC
C
C DISCHARGE RATE NH3 RE-FLUX PUMP(FT3/SEC OR GAL/MIN)
C
      QRPMP=PIE*DINH3R**2*VNH3RP/4.
      QRPMPC=QRPMP*60.*GAL
C
C COST OF NH3 RE-FLUX PUMP($)
C
      VFR=1./RFNH3(TAVGE)
      CNH3RP=(FLONHR*VFR/80100.)**0.64*1.21E+05
C
C PARASITIC PUMP LOSSES(MW)
C
      PARAL=PWREPM+PWRCPM+PWRNPM+PWRRM
C
C
C        ******************************************
C        *                                        *
C        *                                        *
C        *   TURBINE AND ELECTRICAL POWER SECTION *
C        *                                        *
C        *                                        *
C        ******************************************
```

```
C
C GROSS ELECTRICAL LOAD(MW)
C
C     NET ELECTRICAL OUTPUT DESIRED(ELECT-
C     PROVIDED IN INITIAL PARAMETERS)(MW)
C
C     ELECTRICAL LOADING AS EFFECTED BY EFFICIENCY(MW)
C
      EEPC=EEP/100.
      ETRPC=ETRP/100.
      WELECT=ELECT/(EEPC*ETRPC)
      WELOSS=WELECT-ELECT
C
C     GROSS ELECT LOADING INCL PARASITIC LOSSES(MW)
C
      WELECG=WELECT+PARAL
C
C POWER GENERATOR-TURBINE(HP)
C
      PWRTR=1341.*WELECG
C
C TURBINE EFFICIENCY REQUIREMENT(PCT)
C
C     ENTHALPY AT STATE PT 5(BTU/LBM)
C
      H5=H4-(3412.2E+03*WELECG/FLONH3)
C
C     CONSTRAINT FOR SAT STATE PT 5
C
      H5G=HG(PCOND)
      DH5=H5G-H5
C
C     QUALITY OF NH3 AT STATE PT5(PCT)
C
      H5F=HF(PCOND)
      X5=(H5-H5F)/(H5G-H5F)
      X5P=X5*100.
C
C     QUALITY OF NH3 EXHAUST AT STATE PT 5S(PCT)
C
      S4F=SF(T4)
      S4G=SG(T4)
      S4=S4F+X4*(S4G-S4F)
      S5F=SF(T5)
      S5G=SG(T5)
      X5S=(S4-S5F)/(S5G-S5F)
      X5SP=X5S*100.
C
```

```
C      CONSTRAINT FOR A SAT QUALITY AT STATE PT 5S
C
       DX5=X5-X5S
C
C      ENTHALPY AT STATE PT 5S
C
       H5S=H5F+X5S*(H5G-H5F)
C
C      TURBINE EFFICIENCY CALCULATION(PCT)
C
       ETURB=(H4-H5)/(H4-H5S)
       ETURBP=ETURB*100.
C
C COST OF NH3 TURBINE- GENERATOR($)
C
       FPF=1.447
       DF=2.0
       CTURB=(0.375+(WELECG*1000.)/(136000*DF))*FPF*2.42E+06
       CGEN=(WELECG*.023+.3)*1.21E+06
       CELECT=CGEN+CTURB
C
C
C                 ************************
C                 *                      *
C                 *                      *
C                 *   CONDENSER SECTION  *
C                 *                      *
C                 *                      *
C                 ************************
C
C AMOUNT OF HEAT REJECTION(BTU/HR OR MW)
C
       QC=FLONH3*(H5-H1)
C
C CMIN FOR CONDENSER(BTU/HR.F)
C
       CPSWC=CPSW(TBLKC)
       CMINC=FLOCP*CPSWC
C
C COND SW OUTLET TEMP(F OR C)
C
       TCOC=TCIC+QC/CMINC
C
C TOTAL NUMBER OF CONDENSER TUBES
C
       RHOSWC=RHOSW(TBLKC)
       TNCT=4.*FLOCP/(3600.*RHOSWC*PIE*TDICC**2*VSWC)
C
```

```
C TUBE SHEET DIAMETER(FT)                                              A1537
C                                                                      A1538
C     TUBE PROFILE - STAGGERED                                         A1539
C                                                                      A1540
      IF (PROF.EQ.2.) GO TO 710                                        A1541
      CHT=CPR*TDOC*0.5                                                 A1542
      CBASE=CPR*TDOC*0.866                                             A1543
      CTAREA=CHT*CBASE*2.                                              A1544
      SNC=2.*CHT                                                       A1545
      GO TO 720                                                        A1546
710   CONTINUE                                                         A1547
C                                                                      A1548
C     TUBE PROFILE - IN-LINE                                           A1549
C                                                                      A1550
      CPLONG=CPR*TDOC                                                  A1551
      CPLAT=CPR*TDOC                                                   A1552
      CTAREA=CPLONG*CPLAT                                              A1553
720   CONTINUE                                                         A1554
      CAREA=CTAREA*TNCT                                                A1555
      TSDC=((4.*CAREA/PIE)**0.5)/12.                                   A1556
C                                                                      A1557
C LOG MEAN TEMP DIFFERENCE  OF CONDENSER(F OR C)                       A1558
C                                                                      A1559
      CLMTD=((T1-TCOC)-(T1-TCIC))/ALOG((T1-TCOC)/(T1-TCIC))            A1560
C                                                                      A1561
C COND SW AVG BULK TEMP(F)                                             A1562
C                                                                      A1563
      RTBLKC=(TCOC+TCIC)/2.                                            A1564
C                                                                      A1565
C TEST FOR SAT TBLKC                                                   A1566
C                                                                      A1567
      DBLKTC=ABS(TBLKC-RTBLKC)                                         A1568
      SCALBC=ABS(TBLKC)                                                A1569
      IF (SCALBC.LT.0.1) SCALBC=0.1                                    A1570
      DELBC=DBLKTC/SCALBC                                              A1571
      IF (DELBC.LT.0.001) GO TO 730                                    A1572
      TBLKC=RTBLKC                                                     A1573
      GO TO 620                                                        A1574
730   CONTINUE                                                         A1575
      TBLKC=RTBLKC                                                     A1576
      TBLKCR=TBLKC+459.69                                              A1577
C                                                                      A1578
C CONDENSER CONDUCTANCE(BTU/HR.F)                                      A1579
C                                                                      A1580
      UAC=QC/CLMTD                                                     A1581
C                                                                      A1582
C NUMBER OF TRANSFER UNITS FOR COND(NTU)                               A1583
C                                                                      A1584
```

```
      CNTU=UAC/CMINC
C
C COND EFFECTIVENESS(EPSILON)
C
      EPSC=1.-EXP(-CNTU)
C
C INITIALLY ASSUME A TLC
C
C OVERALL HEAT TRANSFER COEFFICIENT(BTU/HR.FT2.F OR W/M2)
C     INITIALLY ASSUME HTNH3
C
      HTNH3C=1000.
740   CONTINUE
C
C     REYNOLDS NUMBER FOR COND TUBESIDE
C     (PROPERTIES EVAL AT TBULK)
C
      VISSWC=VISSW(TBLKC)
      RHOSWR=RHOSW(TBLKC)
      RNSWC=3600.*RHOSWR*VSWC*TDICC/VISSWC
C
C     PRANDTL NUMBER FOR COND TUBESIDE
C
      CPSWC=CPSW(TBLKC)
      TKSWC=TKSW(TBLKC)
      PNSWC=CPSWC*VISSWC/TKSWC
C
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
C     LAMINAR FLOW USING SEIDER-TATE CORRELATION(ASSUME
C     VISCOSITY(TBULK)=VISCOSITY(TWALL)
C
      HTSWC=1.86*TKSWC*((RNSWC*PNSWC)**.3333)*(TDICC/TLC)**.3333/TDICC
      GO TO 760
750   CONTINUE
C
C     TURBULENT FLOW USING DITTUS-BOELTER CORRELATION
C
      HTSWC=.023*TKSWC*RNSWC**.8*PNSWC**.3/TDICC
760   CONTINUE
C
C FILM TEMP FOR PROPERTY EVALUATION
C
C     THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE
C     UA - OUTSIDE(BTU/HR.F)
```

```
C
C     THERMAL RESISTANCE SW(HR.F/BTU)
C
      CTR1C=1/(EFFIC*HTSWC*PIE*TDICC*TLC)
C
C     THERMAL RESISTANCE FOR SW FOULING(HR.F/BTU)
C
      HTFSWC=1./SWFC
      CTR2C=1./(EFFIC*HTFSWC*PIE*TDICC*TLC)
C
C     THERMAL RESISTANCE FOR WALL THICKNESS(HR.F/BTU)
C
      CTR3C=ALOG(TDOCC/TDICC)/(2.*PIE*TKW*TLC)
C
C     THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)
C     (CONSIDERED NEGLIGIBLE)
C
C
C     THERMAL RESISTANCE FOR NH3(HR.F/BTU)
C
      CTR5C=1./(EFFOC*HTNH3C*PIE*TDOCC*TLC)
C
C     HEAT TRANSFERED PER TUBE(BTU/HR)
C
      QCT=(T1-TBLKC)/(CTR1C+CTR2C+CTR3C+CTR5C)
C
C     TUBE SIDE WALL TEMP(F)
C
      CTW1=TBLKC+QCT*(CTR1C+CTR2C)
C
C     SHELLSIDE WALL TEMP(F)
C
      CTW2=TBLKC+QCT*(CTR1C+CTR2C+CTR3C)
C
C     COND FILM TEMP CALCULATION(F)
C
      CFT=(CTW2+T1)/2.
C
C CONDENSER DELTA T TEMP(F)
C
      DELTAC=T1-CTW2
C
C COND SHELLSIDE HEAT TRANSFER COEF(BTU/HR.FT2.F OR W/M2)
C
      VSNH3C=VSNH3(CFT)
      RONH3C=RFNH3(CFT)
      TKNH3C=TKNH3(CFT)
C
```

```
C      REYNOLDS NUMBER(PSEUDO-HORIZ COND)
C
       WC=FLONH3/TNCT
       RNH3CH=2.*WC/(TLC*VSNH3C)
C
C      REYNOLDS NUMBER(PSEUDO-VERT COND) REF:MCADANS
C
       RNH3CV=4.*WC/(PIE*TDOCC*VSNH3C)
C
C CONDENSER TYPE IDENTICATION
C TYPEC=1 HORIZONTAL CONDENSER
C TYPEC=2 VERTICAL CONDENSER
C
       IF (TYPEC.GT.1.) GO TO 780
C
C      HORIZONTAL TYPE CONDENSER
C
       IF (RNH3CH.GT.2100.) GO TO 770
C
C      LAMINAR FLOW USING NUSSELT CORRELATION
C
       HNH3CR=.95*((3600.**2*TKNH3C**3*RONH3C**2*G*TLC)/(VSNH3C*WC))**.33
      133
       GO TO 800
770    CONTINUE
C
C      TURBULENT FLOW USING NUSSELT CORR MOD BY MCADAMS
C
       HNH3CR=1.045*((3600.**2*TKNH3C**3*RONH3C**2*G*TLC)/(VSNH3C*WC))**.
      13333
       GO TO 800
780    CONTINUE
C
C      VERTICAL TYPE CONDENSER
C
       IF (RNH3CV.GT.1800.) GO TO 790
C
C      LAMINAR FLOW USING NUSSELT CORR MOD BY MCADAMS
C
       CORRF=1.28
       HNH3CR=CORRF*1.47*(VSNH3C**2/(3600.**2*TKNH3C**3*RONH3C**2*G))**(-
      1.3333)*RNH3CV**(-.3333)
       GO TO 800
790    CONTINUE
C
C      TURBULENT FLOW USING KIRKBRIDE CORRELATION
C
       HNH3CR=.0077*(VSNH3C**2/(3600.**2*TKNH3C**3*RONH3C**2*G))**(-.3333
```

```
     1)*KNH3CV**.4
800   CONTINUE
C
C TEST FOR SAT HTNH3C
C
      DHNH3C=ABS(HNH3CR-HTNH3C)
      SCALC=ABS(HNH3CR)
      IF (SCALC.LT.0.1) SCALC=0.1
      DELC=DHNH3C/SCALC
      IF (DELC.LT.0.001) GO TO 810
      HTNH3C=HNH3CR
      GO TO 740
810   CONTINUE
      HTNH3C=HNH3CR
C
C PRESSURE DROP ACROSS CONDENSER SHELLSIDE(LBF/IN2)USING
C TWO-PHASE MODEL(HOMOGENEOUS)
C
C MAX VELOCITY THRU MINIMUM- FLOW AREA F(TUBE PROFILE)
      RONH3T=RGNH3(PCOND)
      VNH3C=4.*FLONH3/(3600.*RONH3T*PIE*TSDC**2)
      VMAXC=VNH3C*(SNC/(SNC-TDOC))
C
C REYNOLDS NUMBER FOR MAXIMUM SHELLSIDE FLOW
C
      REMAXC=3600.*RONH3C*VMAXC*TDOC/(12.*VSNH3C)
C
C EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
C
      IF (PROF.EQ.2.) GO TO 820
      CFF=(0.25+0.118)/((SNC-TDOC)/TDOC)**1.08*REMAXC**(-.16)
      GO TO 830
820   CONTINUE
      CFF=(0.44+(0.08*SNC/TDOC)/((SNC-TDOC)/TDOC)**(0.43+1.13*TDOC/SNC))
     1*REMAXC**(-.15)
830   CONTINUE
C
C MASS VELOCITY FOR MINIMUM FREE-FLOW AREA(LBM/FT2.SEC)
C
      IF (TYPEC.GT.1.) GO TO 840
      CAF=TSDC*TLC
      CL=TSDC
      GO TO 850
840   CONTINUE
      BAND=TLC
      CAF=PIE*TSDC*BAND
      CL=TLC
850   CONTINUE
```

```
      CAFF=CAF*((SNC-TDOC)/SNC)
      CGF=FLONH3/(3600.*CAFF)
C
C CALCULATION OF CONDENSER SHELLSIDE PRESSURE DROP(LBF/IN2)
C USING THE HOMOGENEOUS TWO-PHASE MODEL
C
      EDC=(CPR*TDOC-TDOC)/12.
      VLIQC=1./RFNH3(T5)
      VAPC=1./RGNH3(PCOND)
      VAVGC=VLIQC*(1.+X5*(VAPC-VLIQC)/VLIQC)
      CFRICT=(CFF*CGF**2*VAVGC*CL)/(144.*EDC*2.*GC)
      CMOM=(CGF**2*VAVGC)/(144.*GC)
      CELEV=(CGF*CL)/(144.*GC)
      DSCOND=CFRICT+CMOM+CELEV
C
C PROPERTIES AT STATE PT 1
C
      P1=PCOND-DSCOND
      H1=HF(P1)
      T1R=TSAT(P1)
C
C TEST FOR SAT T1(DEG F)
C
      DTEMP4=ABS(T1R-T1)
      SCALT1=ABS(T1R)
      IF (SCALT1.LT.0.1) SCALT1=0.1
      DELT4=DTEMP4/SCALT1
      IF (DELT4.LT.0.001) GO TO 860
      T1=T1R
      GO TO 260
860   CONTINUE
      T1=T1R
C
C THERMAL RESISTANCES FOR OVERALL HEAT TRANSFER COEF - U
C OUTSIDE(HR.FT2.F/BTU)
C
C     OUTSIDE TUBE SURFACE AREA(FT2)
C
      CAO=PIE*TDOCC*TLC
C
C     THERMAL RESISTANCE FOR SW(HR.FT2.F/BTU)
C
      TR1C=CAO*CTR1C
C
C     THERMAL RESISTANCE FOR SW FOULING(HR.FT2.F/BTU)
C
      TR2C=CAO*CTR2C
C
```

```
C     THERMAL RESISTANCE FOR WALL THICKNESS(HR.FT2.F/BTU)
C
      TR3C=CAO*CTR3C
C
C     THERMAL RESISTANCE FOR NH3 FOULING(HR.FT2.F/BTU)
C     (CONSIDERED NEGLIGIBLE)
C
C     THERMAL RESISTANCE FOR NH3(HR.FT2.F/BTU)
C
      TR5C=CAO*CTR5C
C
C OVERALL HEAT TRANSFER COEF U - OUTSIDE(BTU/HR.FT2.F
C OR W/M2)
C
      UC=1./(TR1C+TR2C+TR3C+TR5C)
C
C PSEUDO HT COEF FOR SW(BTU/HR.FTI.F OR W/M2.C)
C
      HSWC=1./TR1C
C
C PSEUDO HT COEF FOR SW FOULING(BTU/HR.FT2.F OR W/M2.C)
C
      HFSWC=1./TR2C
C
C PSEUDO HT COEF FOR WALL THICK(BTU/HR.FT2.F OF W/M2.C)
C
      HWC=1./TR3C
C
C PSEUDO HT COEF FOR NH3(BTU/HR.FT2.F OR W/M2.C)
C
      HNH3C=1./TR5C
C
C TOTAL CONDENSER HEAT TRANSFER AREA(FT2 OR M2)
C
      THTAC=CNTU*CMINC/UC
C
C REVISED CONDENSER TUBE LENGTH(FT)
C
      TLCR=THTAC/(PIE*TDOCC*TNCT)
C
C CONSTRAINT FOR A SAT TUBE LENGTH
C
      DTLC=TLC-TLCR
C
C COST OF CONDENSER UNIT($)
C
      IF (TSDC.GT.35.) GO TO 910
C
```

```
C CONDENSER TUBE SHEET DIAMETER(10-35)FT
C
C     DRILLING TIME/TUBE SHELL THICK(MIN/IN)
C
      DTC=0.66*(TDOC-.5)
C
C     THICKNESS OF TUBE SHEET(IN)
C
      TTSC=0.56*TSDC**0.68
C
C     TUBE SHEET LABOR COSTS($)
C
      CTSLC=156695.*(TNCT/9630.)*(DTC/0.66)*(TTSC/4.0)
C
C     TUBE SHEET MATERIAL COST($)
C
      CTSMC=189.486*TSDC**2.3
      IF (TMATL.EQ.1.) GO TO 870
C
C     TUBE MATERIAL COST($)
C
      CTMC=(C1*TLC+C2)*TNCT*(TDOC/1.5)
C
C     TUBE INSTALLATION COST($)
C
      CTIC=34.*TNCT*TDOC**0.7
      GO TO 880
870   CONTINUE
C
C     TUBE MATERIAL COST($)
C
      CTMC=(C1*TLC+C2)*TNCT*(TDOC/1.5)*(1.+(1.+AINT/100.)**10+(1.+AINT/1
     100.)**20)
C
C     TUBE INSTALLATION COST($)
C
      CTIC=34.*TNCT*TDOC**0.7*(1.+(1.+AINT/100.)**10+(1.+AINT/100.)**20)
880   CONTINUE
C
C     HEAT EXCH SHELL COST($)
C
      CHSC=177265.*(TLC+6.)/31.*(TSDC/18.)**2
C
C     NH3 DIST PLATE AND BAFFLES COST($)
C
      CDPBC=1.539E-02*DTC*TNCT*TSDC**2
C
C     BUSTLE,FLANGES,CHANNELS AND FLOW PLATES COST($)
```

```
C
      CBFCFC=1185.286*TSDC**2
C
C     HEAT EXCH HEAD COST($)
C
      CHC=53240.*(TSDC/18.)**3
C
C     WATER INLETS,NOZZLES AND SUPPORTS COST($)
C
      CWINSC=10106.475*TSDC
C
C     TUBE WELDING COST($)
C
      CTWC=0.0
      IF (TMATL.EQ.1.) GO TO 900
      IF (TNCT.GT.36000.) GO TO 890
      CTWC=14.73*TNCT**1.03*(TDOC/1.5)**0.7
      GO TO 900
890   CONTINUE
      CTWC=0.8797*TNCT**1.3*(TDOC/1.5)**0.7
900   CONTINUE
      CCOND=(CTSLC+CTSMC+CTMC+CTIC+CHSC+CDPBC+CBFCFC+CHC+CWINSC+CTWC)
      GO TO 980
910   CONTINUE
C
C CONDENSER TUBE SHEET DIAMETER(35-50)FT
C
C     DRILLING TIME/TUBE SHELL THICKNESS(MIN/IN)
C
      DTC=0.66*(TDOC-0.5)
C
C     THICKNESS OF TUBE SHEET(IN)
C
      TTSC=0.56*TSDC**0.68
C
C     TUBE SHEET MATERIAL AND LABOR COST($)
C
      IF (TMATL.EQ.1.) GO TO 920
      CTSLC=55.189*TNCT**0.791*TSDC**0.68*DTC
      CTSMC=29.566*TSDC**2.014*TTSC
      GO TO 930
920   CONTINUE
      CTSLC=73.81*TNCT**0.791*TSDC**0.68*DTC
      CTSMC=354.3*TSDC**1.61*TTSC
930   CONTINUE
      IF (TMATL.EQ.1.) GO TO 940
C
C     TUBE MATERIAL COST($)
```

```
C
      CTMC=(C1*TLC+C2)*TNCT*(TDOC/1.5)
C
C     TUBE INSTALLATION COST($)
C
      CTIC=36.542*TNCT*TDOC**0.7
      GO TO 950
940   CONTINUE
C
C     TUBE MATERIAL COST($)
C
      CTMC=(C1*TLC+C2)*TNCT*(TDOC/1.5)*(1.+(1.+AINT/100.)**10+(1.+AINT/1
     100.)**20)
C
C     TUBE INSTALLATION COST($)
C
      CTIC=36.542*TNCT*TDOC**0.7*(1.+(1.+AINT/100.)**10+(1.+AINT/100.)**
     120)
950   CONTINUE
C
C     HEAT EXCH SHELL COST($)
C
      CHSC=12.544*(TLC+6.)*TSDC**2.06
C
C     NH3 DIST PLATE AND BAFFLES COST($)
C
      CDPBC=9.8252*TNCT**0.978*DTC
C
C     BUSTLE,FLANGES,CHANNELS AND FLOW PLATES COST($)
C
      CBFCFC=383.824*TSDC**2.184
C
C     HEAT EXCH HEAD COST($)
C
      CHC=938.62*TSDC**1.43
C
C     WATER INLET,NOZZLES AND SUPPORTERS COST($)
C
      CWINSC=7453.6*TSDC**1.056
C
C     TUBE WELDING COST($)
C
      CTWC=0.0
      IF (TMATL.EQ.1.) GO TO 970
      IF (TNCT.GT.36000.) GO TO 960
      CTWC=14.73*TNCT**1.03*(TDOC/1.5)**0.7
      GO TO 970
960   CONTINUE
```

```
      CTWC=0.8797*TNCT**1.3*(TDOC/1.5)**0.7
970   CONTINUE
      CCOND=(CTSLC+CTSMC+CTMC+CTIC+CHSC+CDPBC+CBFCFC+CHC+CWINSC+CTWC)
980   CONTINUE
C
C
C              ********************************
C              *                              *
C              *   SYSTEM EVALUATION SECTION  *
C              *                              *
C              *                              *
C              ********************************
C
C
C PERCENT OF PARASITIC POWER
      PPP=(PARAL/WELECG)*100.
C
C THERMODYNAMIC CYCLE EFFICIENCY(PCT)
C
      WNET=WELECG-PARAL
      QIN=QE/(3412.2E+03)
      TCE=(WNET/QIN)*100.
C
C
C             *********************************
C             *                               *
C             *                               *
C             *  OBJECTIVE FUNCTION SECTION   *
C             *                               *
C             *                               *
C             *********************************
C
C
      OBJ=CEVAP+CCOND+CELECT+CESWP+CCSWP+CNH3P
      CPKW=OBJ/(ELECT*1000.)
      IF (ICALC.LT.3.) RETURN
C
C
C INPUT/OUTPUT DATA SI UNIT CONVERSION
      CFTC=5.*(CFT-32.)/9.
      CLMTDC=5.*CLMTD/9.
      CTW2C=5.*(CTW2-32.)/9.
      DCONDC=6.89476*DCOND
      DEVAPC=6.89476*DEVAP
      DELTEC=5.*DELTAE/9.
      DELTCC=5.*DELTAC/9.
      DICPC=.3048*DICP
```

```
DIHPC=.3048*DIHP                                              A2065
DINH3C=.3048*DINH3                                            A2066
DINHRC=.3048*DINH3R                                           A2067
DPIPCC=6.89476*DPIPC                                          A2068
DPIPHC=6.89476*DPIPH                                          A2069
DPIPNC=6.89476*DPIPN                                          A2070
DPMNCC=.3048*DPMPNC                                           A2071
DPMCC=.3048*DPMPCC                                            A2072
DPMEC=.3048*DPMPEC                                            A2073
DPNRC=6.89476*DPIPNR                                          A2074
DPNRCC=.3048*DPPNRC                                           A2075
DSDEMC=6.89476*DSDEM                                          A2076
DSEVPC=6.89476*DSEVAP                                         A2077
DSCONC=6.89476*DSCOND                                         A2078
EFTC=5.*(EFT-32.)/9.                                          A2079
ETW2C=5.*(ETW2-32.)/9.                                        A2080
ELMTDC=5.*ELMTD/9.                                            A2081
FLNH3C=.45359*FLONH3                                          A2082
FLNHRC=.45359*FLONHR                                          A2083
FLOCPC=.45359*FLOCP                                           A2084
FLOHPC=.45359*FLOHP                                           A2085
HFSWCC=5.6782*HFSWC                                           A2086
HFSWEC=5.6782*HFSWE                                           A2087
HNH3CC=5.6782*HNH3C                                           A2088
HNH3EC=5.6782*HNH3E                                           A2089
HSWCC=5.6782*HSWC                                             A2090
HSWEC=5.6782*HSWE                                             A2091
HWCC=5.6782*HWC                                               A2092
P3C=6.89476*P3                                                A2093
PCONDC=6.89476*PCOND                                          A2094
HWEC=5.6782*HWE                                               A2095
PEVAPC=6.89476*PEVAP                                          A2096
QCC=QC/3412.2E+03                                             A2097
QEC=QE/(3412.2E+03)                                           A2098
QCPMCC=3.785*QCPMPC                                           A2099
QEPMCC=3.785*QEPMPC                                           A2100
QNPMCC=3.785*QNPMPC                                           A2101
QRPMCC=3.785*QRPMPC                                           A2102
TLCRC=.3048*TLCR                                              A2103
TLERC=.3048*TLER                                              A2104
TLNRPC=.3048*TLNHRP                                           A2105
T1C=5.*(T1-32.)/9.                                            A2106
T3AC=5.*(T3A-32.)/9.                                          A2107
T3C=5.*(T3-32.)/9.                                            A2108
T5C=5.*(T5-32.)/9.                                            A2109
T4C=5.*(T4-32.)/9.                                            A2110
TCICC=5.*(TCIC-32.)/9.                                        A2111
TCOCC=5.*(TCOC-32.)/9.                                        A2112
```

```
      TDOCCC=25.4*TDOC
      TDOECC=25.4*TDOE
      THIEC=5.*(THIE-32.)/9.
      THOEC=5.*(THOE-32.)/9.
      THTACC=9.2903E-02*THTAC
      THTAEC=9.2903E-02*THTAE
      TKWC=1.73707*TKW
      TLCC=.3048*TLC
      TLCPC=.3048*TLCP
      TLEC=.3048*TLE
      TLHPC=.3048*TLHP
      TLNHPC=.3048*TLNH3P
      TSDCC=0.3048*TSDC
      TSDEC=0.3048*TSDE
      TTCC=25.4*TTC
      TTEC=25.4*TTE
      UCC=5.6782*UC
      UEC=5.6782*UE
      VSWCC=.3048*VSWC
      VSWEC=.3048*VSWE
      VSWCPC=.3048*VSWCP
      VSWHPC=.3048*VSWHP
C
C SUMMARY OF INPUT/OUTPUT DATA
C
      WRITE (6,1720)
      IF (TYPEE.EQ.2.) GO TO 990
      WRITE (6,1730)
      GO TO 1000
990   CONTINUE
      WRITE (6,1740)
1000  CONTINUE
      WRITE (6,1750) QE,QEC
      WRITE (6,1760) FLOHP,FLOHPC
      WRITE (6,1770) THIE,THIEC
      WRITE (6,1780) THOE,THOEC
      WRITE (6,1790) FLONH3,FLNH3C
      WRITE (6,1800) PEVAP,PEVAPC
      WRITE (6,1810) T3A,T3AC
      WRITE (6,1820) T3,T3C
      WRITE (6,1830) X3P
      WRITE (6,1840) DSEVAP,DSEVPC
      WRITE (6,1850)
      WRITE (6,1860) TDOE,TDOECC
      WRITE (6,1870) TTE,TTEC
      WRITE (6,1880) TLER,TLERC
      IF (TMATL.EQ.2.) GO TO 1010
      WRITE (6,1890)
```

```
      GO TO 1020
1010  CONTINUE
      WRITE (6,1900)
1020  CONTINUE
      IF (PROF.EQ.2.) GO TO 1030
      WRITE (6,1910)
      GO TO 1040
1030  CONTINUE
      WRITE (6,1920)
1040  CONTINUE
      WRITE (6,1930) EPR
      IF (EHFE.EQ.2.) GO TO 1050
      WRITE (6,1940)
      GO TO 1060
1050  CONTINUE
      WRITE (6,1950)
1060  CONTINUE
      WRITE (6,1960) VSWE,VSWEC
      WRITE (6,1970) ETW2,ETW2C
      WRITE (6,1980) EFT,EFTC
      WRITE (6,1990) DELTAE,DELTEC
      WRITE (6,2000) ELMTD,ELMTDC
      WRITE (6,2010) EPSE
      WRITE (6,2020) ENTU
      WRITE (6,2030) UE,UEC
      WRITE (6,2040) HSWE,HSWEC
      WRITE (6,2050) HFSWE,HFSWEC
      WRITE (6,2060) HWE,HWEC
      WRITE (6,2070) HNH3E,HNH3EC
      WRITE (6,2080) THTAE,THTAEC
      WRITE (6,2090) TSDE,TSDEC
      WRITE (6,2100) TNET
      WRITE (6,2110) DEVAP,DEVAPC
      WRITE (6,2120)
      WRITE (6,2130) P3,P3C
      WRITE (6,2140) T4,T4C
      WRITE (6,2150) X4P
      WRITE (6,2160) DSDEM,DSDEMC
      IF (TYPEC.EQ.2.) GO TO 1070
      WRITE (6,2170)
      GO TO 1080
1070  CONTINUE
      WRITE (6,2180)
1080  CONTINUE
      WRITE (6,2190) QC,QCC
      WRITE (6,2200) FLOCP,FLOCPC
      WRITE (6,2210) TCIC,TCICC
      WRITE (6,2220) TCOC,TCOCC
```

```
      WRITE (6,2230) FLONH3,FLNH3C                                      A2209
      WRITE (6,2240) PCOND,PCONDC                                       A2210
      WRITE (6,2250) T5,T5C                                             A2211
      WRITE (6,2260) T1,T1C                                             A2212
      WRITE (6,2270) DSCOND,DSCONC                                      A2213
      WRITE (6,2280)                                                    A2214
      WRITE (6,2290) TDOC,TDOCCC                                        A2215
      WRITE (6,2300) TTC,TTCC                                           A2216
      WRITE (6,2310) TLCR,TLCRC                                         A2217
      IF (TMATL.EQ.2.) GO TO 1090                                       A2218
      WRITE (6,2320)                                                    A2219
      GO TO 1100                                                        A2220
 1090 CONTINUE                                                          A2221
      WRITE (6,2330)                                                    A2222
 1100 CONTINUE                                                          A2223
      IF (PROF.EQ.2.) GO TO 1110                                        A2224
      WRITE (6,2340)                                                    A2225
      GO TO 1120                                                        A2226
 1110 CONTINUE                                                          A2227
      WRITE (6,2350)                                                    A2228
 1120 CONTINUE                                                          A2229
      WRITE (6,2360) CPR                                                A2230
      IF (EHFC.EQ.2.) GO TO 1130                                        A2231
      WRITE (6,2370)                                                    A2232
      GO TO 1140                                                        A2233
 1130 CONTINUE                                                          A2234
      WRITE (6,2380)                                                    A2235
 1140 CONTINUE                                                          A2236
      WRITE (6,2390) VSWC,VSWCC                                         A2237
      WRITE (6,2400) CTW2,CTW2C                                         A2238
      WRITE (6,2410) CFT,CFTC                                           A2239
      WRITE (6,2420) DELTAC,DELTCC                                      A2240
      WRITE (6,2430) CLMTD,CLMTDC                                       A2241
      WRITE (6,2440) EPSC                                               A2242
      WRITE (6,2450) CNTU                                               A2243
      WRITE (6,2460) UC,UCC                                             A2244
      WRITE (6,2470) HSWC,HSWCC                                         A2245
      WRITE (6,2480) HFSWC,HFSWCC                                       A2246
      WRITE (6,2490) HWC,HWCC                                           A2247
      WRITE (6,2500) HNH3C,HNH3CC                                       A2248
      WRITE (6,2510) THTAC,THTACC                                       A2249
      WRITE (6,2520) TSDC,TSDCC                                         A2250
      WRITE (6,2530) TNCT                                               A2251
      WRITE (6,2540) DCOND,DCONDC                                       A2252
      WRITE (6,2550)                                                    A2253
      WRITE (6,2560) DIHP,DIHPC                                         A2254
      WRITE (6,2570) TLHP,TLHPC                                         A2255
      WRITE (6,2580) VSWHP,VSWHPC                                       A2256
```

```
      WRITE (6,2590) FLOHP,FLOHPC                                      A2257
      WRITE (6,2600) THIE,THIEC                                        A2258
      WRITE (6,2610) CONH                                              A2259
      WRITE (6,2620) DPIPH,DPIPHC                                      A2260
      WRITE (6,2630)                                                   A2261
      WRITE (6,2640) DICP,DICPC                                        A2262
      WRITE (6,2650) TLCP,TLCPC                                        A2263
      WRITE (6,2660) VSWCP,VSWCPC                                      A2264
      WRITE (6,2670) FLOCP,FLOCPC                                      A2265
      WRITE (6,2680) TCIC,TCICC                                        A2266
      WRITE (6,2690) CONC                                              A2267
      WRITE (6,2700) DPIPC,DPIPCC                                      A2268
      WRITE (6,2710)                                                   A2269
      WRITE (6,2720) DINH3,DINH3C                                      A2270
      WRITE (6,2730) TLNH3P,TLNHPC                                     A2271
      WRITE (6,2740) FLONH3,FLNH3C                                     A2272
      WRITE (6,2750) DPIPN,DPIPNC                                      A2273
      WRITE (6,2760)                                                   A2274
      WRITE (6,2770) DINH3R,DINHRC                                     A2275
      WRITE (6,2780) TLNHRP,TLNRPC                                     A2276
      WRITE (6,2790) FLONHR,FLNHRC                                     A2277
      WRITE (6,2800) DPIPNR,DPNRC                                      A2278
      WRITE (6,2810)                                                   A2279
      WRITE (6,2820)                                                   A2280
      WRITE (6,2830) DPMPEC,DPMEC                                      A2281
      WRITE (6,2840) QEPMPC,QEPMCC                                     A2282
      WRITE (6,2850) EPE,EME                                           A2283
      WRITE (6,2860)                                                   A2284
      WRITE (6,2870) DPMPCC,DPMPC                                      A2285
      WRITE (6,2880) QCPMPC,QCPMCC                                     A2286
      WRITE (6,2890) EPC,EMC                                           A2287
      WRITE (6,2900)                                                   A2288
      WRITE (6,2910) DPMPNC,DPMNCC                                     A2289
      WRITE (6,2920) QNPMPC,QNPMCC                                     A2290
      WRITE (6,2930) EPNH3,EMNH3                                       A2291
      WRITE (6,2940)                                                   A2292
      WRITE (6,2950) DPPNRC,DPNRCC                                     A2293
      WRITE (6,2960) QRPMPC,QRPMCC                                     A2294
      WRITE (6,2970) EPNHR,EMNHR                                       A2295
      WRITE (6,2980)                                                   A2296
      WRITE (6,2990) EEP                                               A2297
      WRITE (6,3000) ETRP                                              A2298
      WRITE (6,3010) ETURBP                                            A2299
      WRITE (6,3020) X5P                                               A2300
      WRITE (6,3030)                                                   A2301
      WRITE (6,3040) PWRTR,WELECG                                      A2302
      WRITE (6,3050) WELOSS                                            A2303
      WRITE (6,3060) PWREP,PWREPM                                      A2304
```

```
      WRITE (6,3070) PWRCP,PWRCPM
      WRITE (6,3080) PWRNP,PWRNPM
      WRITE (6,3090) PWRRP,PWRRM
      WRITE (6,3100)
      WRITE (6,3110) ELECT
      WRITE (6,3120) PPP
      WRITE (6,3130) TCE
      WRITE (6,3140)
      WRITE (6,3150) CEVAP
      WRITE (6,3160) CCOND
      WRITE (6,3170) CTURB
      WRITE (6,3180) CGEN
      WRITE (6,3190) CESWP
      WRITE (6,3200) CCSWP
      WRITE (6,3210) CNH3P
      WRITE (6,3220) CNH3RP
      WRITE (6,3230)
      WRITE (6,3240) OBJ
      WRITE (6,3250) CPKW
      RETURN
C
1150  FORMAT (1H1,19X,21HINITIAL DESIGN VALUES)
1160  FORMAT (1H0,2X,23HEVAPORATOR - HORIZONTAL)
1170  FORMAT (1H0,2X,21HEVAPORATOR - VERTICAL)
1180  FORMAT (1H0,4X,9HTUBE O.D.,7X,F8.3,4H(IN),12X,F8.3,4H(MM))
1190  FORMAT (1H0,4X,11HTUBE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))
1200  FORMAT (1H0,4X,11HSW TUBE VEL,5X,F8.3,6H(FT/S),10X,F8.3,5H(M/S))
1210  FORMAT (1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),7X,F8.3,5H(MPA
     1))
1220  FORMAT (1H0,4X,24HTUBE MATERIAL - ALUMINUM,//,6X,15HTHERMAL COND(K
     1),F8.3,13H(BTU/HR.FT.F),4X,F8.3,7H(W/M.C))
1230  FORMAT (1H0,4X,24HTUBE MATERIAL - TITANIUM,//,6X,15HTHERMAL COND(K
     1),F8.3,13H(BTU/HR.FT.F),4X,F8.3,7H(W/M.C))
1240  FORMAT (1H0,4X,37HTUBE PROFILE - STAGGERED EQUI-LATERAL)
1250  FORMAT (1H0,4X,33HTUBE PROFILE - IN-LINE EQUI-SIDED)
1260  FORMAT (1H0,6X,11HPITCH RATIO,6X,F5.2)
1270  FORMAT (1H0,4X,24HENHANCEMENT - PLAIN TUBE)
1280  FORMAT (1H0,4X,28HENHANCEMENT - LINDE-PROMOTER)
1290  FORMAT (1H0,2X,22HCONDENSER - HORIZONTAL)
1300  FORMAT (1H0,2X,20HCONDENSER - VERTICAL)
1310  FORMAT (1H0,4X,9HTUBE O.D.,7X,F8.3,4H(IN),12X,F8.3,4H(MM))
1320  FORMAT (1H0,4X,11HTUBE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))
1330  FORMAT (1H0,4X,11HSW TUBE VEL,5X,F8.3,6H(FT/S),10X,F8.3,5H(M/S))
1340  FORMAT (1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),7X,F8.3,5H(MPA
     1))
1350  FORMAT (1H0,4X,24HTUBE MATERIAL - ALUMINUM,//,6X,15HTHERMAL COND(K
     1),F8.3,13H(BTU/HR.FT.F),3X,F8.3,7H(W/M.C))
1360  FORMAT (1H0,4X,24HTUBE MATERIAL - TITANIUM,//,6X,15HTHERMAL COND(K
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     1),F8.3,13H(BTU/HR.FT.F),3X,F8.3,7H(W/M.C))                       A2353
1370  FORMAT (1H0,4X,37HTUBE PROFILE - STAGGERED EQUI-LATERAL)           A2354
1380  FORMAT (1H0,4X,33HTUBE PROFILE - IN-LINE EQUI-SIDED)               A2355
1390  FORMAT (1H0,6X,11HPITCH RATIO,6X,F5.2)                             A2356
1400  FORMAT (1H0,4X,24HENHANCEMENT - PLAIN TUBE)                        A2357
1410  FORMAT (1H0,4X,28HENHANCEMENT - LINDE-PROMOTER)                    A2358
1420  FORMAT (1H0,2X,19HSALT WATER HOT PIPE)                             A2359
1430  FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),12X,F8.3,3H(M))          A2360
1440  FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))       A2361
1450  FORMAT (1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),10X,F8.3,5H(M/S))   A2362
1460  FORMAT (1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),9X,F8.3,7H(DEG C A2363
     1))                                                                 A2364
1470  FORMAT (1H0,4X,12HSW SALINITY ,7X,F5.0,1X,5H0/000)                 A2365
1480  FORMAT (1H0,2X,20HSALT WATER COLD PIPE)                            A2366
1490  FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),12X,F8.3,3H(M))          A2367
1500  FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))       A2368
1510  FORMAT (1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),10X,F8.3,5H(M/S))   A2369
1520  FORMAT (1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),9X,F8.3,7H(DEG C A2370
     1))                                                                 A2371
1530  FORMAT (1H0,4X,12HSW SALINITY ,7X,F5.0,1X,5H0/000)                 A2372
1540  FORMAT (1H0,2X,17HAMMONIA CIRC PIPE)                               A2373
1550  FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),12X,F8.3,3H(M))          A2374
1560  FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))       A2375
1570  FORMAT (1H0,2X,20HAMMONIA RE-FLUX PIPE)                            A2376
1580  FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),12X,F8.3,3H(M))          A2377
1590  FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),12X,F8.3,3H(M))       A2378
1600  FORMAT (1H0,2X,29HPUMP AND GEN-TURB PERFORMANCE)                   A2379
1610  FORMAT (1H0,4X,12HEVAP SW PUMP)                                    A2380
1620  FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR, A2381
     11X,F5.2,5H(PCT))                                                   A2382
1630  FORMAT (1H0,4X,12HCOND SW PUMP)                                    A2383
1640  FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR, A2384
     11X,F5.2,5H(PCT))                                                   A2385
1650  FORMAT (1H0,4X,17HAMMONIA CIRC PUMP)                               A2386
1660  FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR, A2387
     11X,F5.2,5H(PCT))                                                   A2388
1670  FORMAT (1H0,21HGEN-TURB EFFICIENCIES)                              A2389
1680  FORMAT (1H0,6X,14HGEN MECH&ELECT,3X,F5.2,5H(PCT))                  A2390
1690  FORMAT (1H0,6X,9HTURB MECH,8X,F5.2,5H(PCT))                        A2391
1700  FORMAT (1H0,2X,18HPOWER REQUIREMENTS)                              A2392
1710  FORMAT (1H0,4X,16HNET POWER OUTPUT,10X,F8.3,4H(MW))                A2393
1720  FORMAT (1H1,19HOPTIMIZATION VALUES)                                A2394
1730  FORMAT (1H0,2X,23HEVAPORATOR - HORIZONTAL)                         A2395
1740  FORMAT (1H0,2X,21HEVAPORATOR - VERTICAL)                           A2396
1750  FORMAT (1H0,4X,9HHT ABSORB,1X,F14.1,8H(BTU/HR),7X,F8.3,4H(MW))     A2397
1760  FORMAT (1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))   A2398
1770  FORMAT (1H0,6X,10HSW TEMP IN,4X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))  A2399
1780  FORMAT (1H0,6X,11HSW TEMP OUT,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C)) A2400
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 1790 FORMAT (1H0,4X,8HNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))     A2401
 1800 FORMAT (1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA   A2402
     1))                                                                   A2403
 1810 FORMAT (1H0,6X,13HEVAP SAT TEMP,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C   A2404
     1))                                                                   A2405
 1820 FORMAT (1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))   A2406
 1830 FORMAT (1H0,6X,14HOUTLET QUALITY,4X,F5.2,5H(PCT))                    A2407
 1840 FORMAT (1H0,4X,14HNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP   A2408
     1A))                                                                  A2409
 1850 FORMAT (1H0,4X,20HTUBE CHARACTERISTICS)                              A2410
 1860 FORMAT (1H0,6X,10HOUTTER DIA,4X,F8.3,4H(IN),11X,F8.3,4H(MM))         A2411
 1870 FORMAT (1H0,6X,10HWALL THICK,4X,F8.3,4H(IN),11X,F8.3,4H(MM))         A2412
 1880 FORMAT (1H0,6X,6HLENGTH,8X,F8.3,4H(FT),11X,F8.3,3H(M))               A2413
 1890 FORMAT (1H0,6X,19HMATERIAL - ALUMINUM)                               A2414
 1900 FORMAT (1H0,6X,19HMATERIAL - TITANIUM)                               A2415
 1910 FORMAT (1H0,6X,37HTUBE PROFILE - STAGGERED EQUI-LATERAL)             A2416
 1920 FORMAT (1H0,6X,33HTUBE PROFILE - IN-LINE EQUI-SIDED)                 A2417
 1930 FORMAT (1H0,8X,11HPITCH RATIO,4X,F5.2)                               A2418
 1940 FORMAT (1H0,6X,24HENHANCEMENT - PLAIN TUBE)                          A2419
 1950 FORMAT (1H0,6X,28HENHANCEMENT - LINDE-PROMOTER)                      A2420
 1960 FORMAT (1H0,6X,11HSW VELOCITY,3X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))      A2421
 1970 FORMAT (1H0,4X,17HT WALL(SHELLSIDE),F7.3,7H(DEG F),8X,F8.3,7H(DEG    A2422
     1C))                                                                  A2423
 1980 FORMAT (1H0,4X,9HFILM TEMP,7X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))      A2424
 1990 FORMAT (1H0,4X,15HDELTA T BOILING,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG   A2425
     1 C))                                                                 A2426
 2000 FORMAT (1H0,4X,8HL.M.T.D.,8X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))       A2427
 2010 FORMAT (1H0,4X,18HEVAP EFFECTIVENESS,7X,F8.3)                        A2428
 2020 FORMAT (1H0,4X,20HNR OF TRANSFER UNITS,5X,F8.3)                      A2429
 2030 FORMAT (1H0,4X,11HOVL HT COEF,5X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H   A2430
     1(W/M2.C))                                                            A2431
 2040 FORMAT (1H0,6X,8HH(WATER),6X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M   A2432
     12.C))                                                                A2433
 2050 FORMAT (1H0,6X,10HH(FOULING),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(   A2434
     1W/M2.C))                                                             A2435
 2060 FORMAT (1H0,6X,8HH(METAL),6X,F8.2,14H(BTU/HR.FT2.F),F9.2,8H(W/M2.C   A2436
     1))                                                                   A2437
 2070 FORMAT (1H0,6X,10HH(AMMONIA),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(   A2438
     1W/M2.C))                                                             A2439
 2080 FORMAT (1H0,4X,10HHT SURFACE,2X,F12.2,5H(FT2),6X,F12.2,4H(M2))       A2440
 2090 FORMAT (1H0,4X,14HTUBE SHEET DIA,2X,F8.3,4H(FT),11X,F8.3,3H(M))      A2441
 2100 FORMAT (1H0,6X,15HTOT NR OF TUBES,7X,F12.0)                          A2442
 2110 FORMAT (1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA   A2443
     1))                                                                   A2444
 2120 FORMAT (1H0,2X,36HMOISTURE SEPARATOR-INSIDE EVAP SHELL)              A2445
 2130 FORMAT (1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA   A2446
     1))                                                                   A2447
 2140 FORMAT (1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))   A2448
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2150  FORMAT (1H0,6X,14HOUTLET QUALITY,3X,F5.2,5H(PCT))                A2449
2160  FORMAT (1H0,6X,14HNH3 PRESS DROP,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA))  A2450
2170  FORMAT (1H1,2X,22HCONDENSER - HORIZONTAL)                        A2451
2180  FORMAT (1H0,2X,20HCONDENSER - VERTICAL)                          A2452
2190  FORMAT (1H0,4X,9HHT REJECT,1X,F14.1,8H(BTU/HR),7X,F8.3,4H(MW))   A2453
2200  FORMAT (1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))  A2454
2210  FORMAT (1H0,6X,10HSW TEMP IN,4X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))  A2455
2220  FORMAT (1H0,6X,11HSW TEMP OUT,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))  A2456
2230  FORMAT (1H0,4X,8HNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))  A2457
2240  FORMAT (1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA  A2458
     1))                                                               A2459
2250  FORMAT (1H0,6X,13HCOND SAT TEMP,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C  A2460
     1))                                                               A2461
2260  FORMAT (1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))  A2462
2270  FORMAT (1H0,4X,14HNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP  A2463
     1A))                                                              A2464
2280  FORMAT (1H0,4X,20HTUBE CHARACTERISTICS)                          A2465
2290  FORMAT (1H0,6X,10HOUTTER DIA,4X,F8.3,4H(IN),11X,F8.3,4H(MM))     A2466
2300  FORMAT (1H0,6X,10HWALL THICK,4X,F8.3,4H(IN),11X,F8.3,4H(MM))     A2467
2310  FORMAT (1H0,6X,6HLENGTH,8X,F8.3,4H(FT),11X,F8.3,3H(M))           A2468
2320  FORMAT (1H0,6X,19HMATERIAL - ALUMINUM)                           A2469
2330  FORMAT (1H0,6X,19HMATERIAL - TITANIUM)                           A2470
2340  FORMAT (1H0,6X,37HTUBE PROFILE - STAGGERED EQUI-LATERAL)         A2471
2350  FORMAT (1H0,6X,33HTUBE PROFILE - IN-LINE EQUI-SIDED)             A2472
2360  FORMAT (1H0,8X,11HPITCH RATIO,4X,F5.2)                           A2473
2370  FORMAT (1H0,6X,24HENHANCEMENT - PLAIN TUBE)                      A2474
2380  FORMAT (1H0,6X,28HENHANCEMENT - LINDE-PROMOTER)                  A2475
2390  FORMAT (1H0,6X,11HSW VELOCITY,3X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))  A2476
2400  FORMAT (1H0,4X,17HT WALL(SHELLSIDE),F7.3,7H(DEG F),8X,F8.3,7H(DEG  A2477
     1C))                                                              A2478
2410  FORMAT (1H0,4X,9HFILM TEMP,7X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))  A2479
2420  FORMAT (1H0,4X,12HDELTA T COND,4X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C)  A2480
     1)                                                                A2481
2430  FORMAT (1H0,4X,8HL.M.T.D.,8X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))   A2482
2440  FORMAT (1H0,4X,18HCOND EFFECTIVENESS,7X,F8.3)                    A2483
2450  FORMAT (1H0,4X,20HNR OF TRANSFER UNITS,5X,F8.3)                  A2484
2460  FORMAT (1H0,4X,11HOVL HT COEF,5X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H  A2485
     1(W/M2.C))                                                        A2486
2470  FORMAT (1H0,6X,8HH(WATER),6X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M  A2487
     12.C))                                                            A2488
2480  FORMAT (1H0,6X,10HH(FOULING),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(  A2489
     1W/M2.C))                                                         A2490
2490  FORMAT (1H0,6X,8HH(METAL),6X,F8.2,14H(BTU/HR.FT2.F),F9.2,8H(W/M2.C  A2491
     1))                                                               A2492
2500  FORMAT (1H0,6X,10HH(AMMONIA),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(  A2493
     1W/M2.C))                                                         A2494
2510  FORMAT (1H0,4X,10HHT SURFACE,2X,F12.2,5H(FT2),6X,F12.2,4H(M2))   A2495
2520  FORMAT (1H0,4X,14HTUBE SHEET DIA,2X,F8.3,4H(FT),11X,F8.3,3H(M))  A2496
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2530   FORMAT (1H0,6X,15HTOT NR OF TUBES,7X,F12.0)                              A2497
2540   FORMAT (1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA      A2498
      1))                                                                     A2499
2550   FORMAT (1H1,2X,19HSALT WATER HOT PIPE)                                  A2500
2560   FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M))               A2501
2570   FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M))            A2502
2580   FORMAT (1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))         A2503
2590   FORMAT (1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))        A2504
2600   FORMAT (1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C     A2505
      1))                                                                     A2506
2610   FORMAT (1H0,4X,12HSW SALINITY ,7X,F5.0,1X,5H0/000)                      A2507
2620   FORMAT (1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA      A2508
      1))                                                                     A2509
2630   FORMAT (1H0,2X,20HSALT WATER COLD PIPE)                                 A2510
2640   FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M))               A2511
2650   FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M))            A2512
2660   FORMAT (1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))         A2513
2670   FORMAT (1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))        A2514
2680   FORMAT (1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C     A2515
      1))                                                                     A2516
2690   FORMAT (1H0,4X,12HSW SALINITY ,7X,F5.0,1X,5H0/000)                      A2517
2700   FORMAT (1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA      A2518
      1))                                                                     A2519
2710   FORMAT (1H0,2X,17HAMMONIA CIRC PIPE)                                    A2520
2720   FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M))               A2521
2730   FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M))            A2522
2740   FORMAT (1H0,4X,8HNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))       A2523
2750   FORMAT (1H0,4X,14HNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP      A2524
      1A))                                                                    A2525
2760   FORMAT (1H0,2X,25HAMMONIA RE-FLUX CIRC PIPE)                            A2526
2770   FORMAT (1H0,4X,9HPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M))               A2527
2780   FORMAT (1H0,4X,11HPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M))            A2528
2790   FORMAT (1H0,4X,8HNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))       A2529
2800   FORMAT (1H0,4X,14HNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP      A2530
      1A))                                                                    A2531
2810   FORMAT (1H1,2X,29HPUMP AND GEN-TURB PERFORMANCE)                        A2532
2820   FORMAT (1H0,4X,12HEVAP SW PUMP)                                         A2533
2830   FORMAT (1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))             A2534
2840   FORMAT (1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN     A2535
      1))                                                                     A2536
2850   FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,     A2537
      11X,F5.2,5H(PCT))                                                       A2538
2860   FORMAT (1H0,4X,12HCOND SW PUMP)                                         A2539
2870   FORMAT (1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))             A2540
2880   FORMAT (1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN     A2541
      1))                                                                     A2542
2890   FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,     A2543
      11X,F5.2,5H(PCT))                                                       A2544
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2900  FORMAT (1H0,4X,17HAMMONIA CIRC PUMP)
2910  FORMAT (1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))
2920  FORMAT (1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN
     1))
2930  FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,
     11X,F5.2,5H(PCT))
2940  FORMAT (1H0,4X,20HAMMONIA RE-FLUX PUMP)
2950  FORMAT (1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))
2960  FORMAT (1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN
     1))
2970  FORMAT (1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,
     11X,F5.2,5H(PCT))
2980  FORMAT (1H0,4X,21HGEN-TURB EFFICIENCIES)
2990  FORMAT (1H0,6X,14HGEN MECH&ELECT,4X,F5.2,5H(PCT))
3000  FORMAT (1H0,6X,9HTURB MECH,9X,F5.2,5H(PCT))
3010  FORMAT (1H0,6X,13HTURB INTERNAL,5X,F5.2,5H(PCT))
3020  FORMAT (1H0,4X,19HTURB OUTLET QUALITY,1X,F5.2,5H(PCT))
3030  FORMAT (1H1,2X,18HPOWER REQUIREMENTS)
3040  FORMAT (1H0,4X,14HTURB-GEN GROSS,F10.3,4H(HP),12X,F8.3,4H(MW))
3050  FORMAT (1H0,6X,17HEFFICIENCY LOSSES,21X,F8.3,4H(MW))
3060  FORMAT (1H0,4X,12HEVAP SW PUMP,4X,F8.3,4H(HP),12X,F8.3,4H(MW))
3070  FORMAT (1H0,4X,12HCOND SW PUMP,4X,F8.3,4H(HP),12X,F8.3,4H(MW))
3080  FORMAT (1H0,4X,13HNH3 CIRC PUMP,3X,F8.3,4H(HP),12X,F8.3,4H(MW))
3090  FORMAT (1H0,4X,16HNH3 RE-FLUX PUMP,F8.3,4H(HP),12X,F8.3,4H(MW))
3100  FORMAT (1H0,42X,10H----------)
3110  FORMAT (1H0,10X,16HNET POWER OUTPUT,18X,F8.3,4H(MW))
3120  FORMAT (1H0,2X,23HPERCENT PARASITIC POWER,10X,F5.2,5H(PCT))
3130  FORMAT (1H0,2X,29HTHEMODYNAMIC CYCLE EFFICIENCY,4X,F5.2,5H(PCT))
3140  FORMAT (1H0,2X,18HCOST OF COMPONENTS)
3150  FORMAT (1H0,4X,10HEVAPORATOR,14X,F12.2,9H(DOLLARS))
3160  FORMAT (1H0,4X,9HCONDENSER,15X,F12.2,9H(DOLLARS))
3170  FORMAT (1H0,4X,11HGEN-TURBINE,13X,F12.2,9H(DOLLARS))
3180  FORMAT (1H0,4X,9HGENERATOR,15X,F12.2,9H(DOLLARS))
3190  FORMAT (1H0,4X,12HEVAP SW PUMP,12X,F12.2,9H(DOLLARS))
3200  FORMAT (1H0,4X,12HCOND SW PUMP,12X,F12.2,9H(DOLLARS))
3210  FORMAT (1H0,4X,13HNH3 CIRC PUMP,11X,F12.2,9H(DOLLARS))
3220  FORMAT (1H0,4X,16HNH3 RE-FLUX PUMP,8X,F12.2,9H(DOLLARS))
3230  FORMAT (1H0,28X,12H------------)
3240  FORMAT (1H0,10X,12HOPTIMUM COST,6X,F12.2,9H(DOLLARS))
3250  FORMAT (1H0,2X,22HCOST PER NET KW OUTPUT,4X,F12.2,17H(DOLLARS)
     1     )
      END
C FUNCTION SUBROUTINES FOR SALT WATER AND AMMONIA
C     SALT WATER DENSITY(LBM/FT3)
C
      FUNCTION RHOSW (T)
      RHOSW=63.90152+.017512*T-5.684E-04*T**2+6.336E-06*T**3-2.7949E-08*
     1T**4
```

```
      RETURN                                                            B   7
      END                                                               B   8-
C     SALT WATER DENSITY F(DEPTH) (LBM/FT3)                             C   1
C                                                                       C   2
      FUNCTION RHOSWD (Z)                                               C   3
      RHOSWD=64.184049+9.8878E-05*Z                                     C   4
      RETURN                                                            C   5
      END                                                               C   6-
C     SALT WATER SPECIFIC HEAT(BTU/LBM.F)                               D   1
C                                                                       D   2
      FUNCTION CPSW (T)                                                 D   3
      CPSW=0.947314+3.8183E-04*T-8.023E-06*T**2+8.468E-08*T**3-3.261E-10D   4
     1*T**4                                                             D   5
      RETURN                                                            D   6
      END                                                               D   7-
C     SALT WATER VISCOSITY(LBM/HR.FT)                                   E   1
C                                                                       E   2
      FUNCTION VISSW (T)                                                E   3
      VISSW=8.806895-.1988746*T+2.567299E-03*T**2-1.804352E-05*T**3+5.26E   4
     18E-08*T**4                                                        E   5
      RETURN                                                            E   6
      END                                                               E   7-
C     SALT WATER THERMAL CONDUCTIVITY(BTU/HR.FT.F)                      F   1
C                                                                       F   2
      FUNCTION TKSW (T)                                                 F   3
      TKSW=0.2787615+2.62894E-03*T-5.2014E-05*T**2+5.6873E-07*T**3-2.330F   4
     191E-09*T**4                                                       F   5
      RETURN                                                            F   6
      END                                                               F   7-
C     AMMONIA SAT TEMP(F)                                               G   1
C                                                                       G   2
      FUNCTION TSAT (P)                                                 G   3
      TSAT=-28.505313+1.2445557*P-.0049090885*P**2+.000009203093*P**3   G   4
      RETURN                                                            G   5
      END                                                               G   6-
C     ENTHALPY OF SAT AMMONIA VAPOR(BTU/LBM)                            H   1
C                                                                       H   2
      FUNCTION HG (P)                                                   H   3
      HG=603.58813+.3893361*P-.002002459*P**2+4.01727E-06*P**3          H   4
      RETURN                                                            H   5
      END                                                               H   6-
C     ENTHALPY OF SAT AMMONIA LIQUID(BTU/LBM)                           I   1
C                                                                       I   2
      FUNCTION HF (P)                                                   I   3
      HF=9.739724+1.3956686*P-.005461178*P**2+1.0190106E-05*P**3        I   4
      RETURN                                                            I   5
      END                                                               I   6-
C     VISCOSITY OF SAT AMMONIA LIQUID(LBM/HR.FT)                        J   1
```

```
C
      FUNCTION VSNH3 (T)
      VSNH3=.55203968261-.003405489426*T+1.01587304E-05*T**2-1.5046298E-
     108*T**3
      RETURN
      END
C     SPECIFIC HEAT OF SAT AMMONIA LIQUID(BTU/LBM.F)
C
      FUNCTION CPNH3 (T)
      CPNH3=1.0829682542+.00041058198*T+1.7658742E-06*T**2+1.6203699E-08
     1*T**3
      RETURN
      END
C     THERMAL CONDUCTIVITY OF SAT AMMONIA LIQ(BTU/HR.FT.F)
C
      FUNCTION TKNH3 (T)
      TKNH3=.33517857143-.000720535715*T-2.2321429E-07*T**2
      RETURN
      END
C     ENTROPY OF SAT AMMONIA LIQUID(BTU/LBM.R)
C
      FUNCTION SF (T)
      SF=.09771914+.002334056*T-1.66882E-06*T**2+2.1649E-09*T**3
      RETURN
      END
C     ENTROPY OF SAT AMMONIA VAPOR(BTU/LBM.R)
C
      FUNCTION SG (T)
      SG=1.33473105-.001973293*T+4.0808E-06*T**2-7.5887E-09*T**3
      RETURN
      END
C     DENSITY OF SAT AMMONIA LIQ(LBM/FT3)
C
      FUNCTION RFNH3 (T)
      RFNH3=41.415499-.0490092*T+.000039001*T**2-5.2019E-07*T**3
      RETURN
      END
C     DENSITY OF SAT AMMONIA VAPOR(LBM/FT3)
C
      FUNCTION RGNH3 (P)
      RGNH3=.01242535+.0032785773*P-2.52513E-07*P**2+8.9285E-10*P**3
      RETURN
      END
C     TITANIUM TUBE WALL THICKNESS - PLAIN(INCH)
C
      FUNCTION TP (D)
      TP=.007966159+.04023 7327*D-.035418004*D**2+.015039976*D**3-.002418
     12547*D**4
```

```
      RETURN
      END
C     TITANIUN TUBE COST/FT - PLAIN($/FT)
C
      FUNCTION TC (D)
      TC=3.43889-8.9432814*D+10.9182422*D**2-4.68667*D**3+.72236862*D**4
      RETURN
      END
C     TITANIUM TUBE WALL THICKNESS - LINDE-PROMOTER(INCH)
C
      FUNCTION TE (D)
      TE=.04163054-.03507642*D+.0239336*D**2-.00178528*D**3-.000241899*D
     1**4
      RETURN
      END
C     EVAP TITANIUM TUBE COST/FT - LINDE-PROMOTER($/FT)
C
      FUNCTION TEC (D)
      TEC=1.51007568-1.001975385*D+1.723921769*D**2
      RETURN
      END
C     COND TITANIUM TUBE COST/FT - LINDE-PROMOTER($/FT)
C
      FUNCTION TCC (D)
      TCC=-1.8368255813+3.7220155039*D
      RETURN
      END
C     ALUMINUM TUBE WALL THICKNESS - PLAIN(INCH)
C
      FUNCTION AP (D)
      AP=0.065
      RETURN
      END
C     ALUMINUM TUBE COST/FT - PLAIN($/FT)
C
      FUNCTION AC (D)
      AC=-.09288381+1.2279558*D-.9161318*D**2+.51160016*D**3-.09134991*D
     1**4
      RETURN
      END
C     ALUMINUM TUBE WALL THICK - LINDE-PROMOTER(INCH)
C
      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)
C
```

```
      FUNCTION AEC (D)                                                    Y   3
      AEC=2.157473-5.206014*D+7.336678*D**2-3.656495*D**3+.7204888*D**4   Y   4
      RETURN                                                              Y   5
      END                                                                 Y   6-
C     COND ALUMINUM TUBE COST/FT - LINDE-PROMOTER($/FT)                   Z   1
C                                                                         Z   2
      FUNCTION ACC (D)                                                    Z   3
      ACC=1.799865-4.543195*D+6.556578*D**2-3.305876*D**3+.6622916*D**4   Z   4
      RETURN                                                              Z   5
      END                                                                 Z   6-
```

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