COMPUTER MODELING AND DESIGN OPTIMIZATION OF VERTICAL STEAM CONDENSERS

by

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ABSTRACT

The objective of this study is to investigate Vertical Steam Condenser design optimization, in an effort to develop condensers with minimum weight and volume. Candidate vertical condenser tubes considered were of two types: (1) external axial flutes with smooth internal tubes, and (2) doubly enhanced tubes that include internal (coolant side) heat transfer augmentation. A computer program (VERTCON-2) was developed to be used as a preliminary design tool for selection of the optimal condenser that meets the specified geometrical, and operating parameter limits.

The computer model considers condensation in the presence of non-condensable gases. Also, a method was developed to evaluate thermal resistance of the condenser tube walls due to the axial flutes.

Thesis Supervisor: Warren M. Rohsenow

Title: Professor of Mechanical Engineering

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I. INTRODUCTION

A design optimization procedure for vertical steam condensers is established here to develop condensers with minimum weight and volume. The performance comparison of enhanced steam condensers conducted in Reference-3 suggests the following methodology that provides the basis for this optimization scheme.

The optimization procedure is outlined as follows:

1. Design for the following given condenser operating conditions:
   a. total steam condensed
   b. condenser operating pressure
   c. coolant inlet temperature
   d. coolant flow rate
2. This design is subject to the following specified parameter limits:
   a. maximum condenser length (L_{max})
   b. maximum pumping power (P_{max})
   c. coolant velocity range from a minimum velocity (V_{min}) to a maximum velocity (V_{max}).
3. For a given heat transfer surface minimum weight and volume will always be achieved for a maximum coolant flow velocity that results in:
   a. condenser length is equal to L_{max} for pumping power less than P_{max}, or
   b. pumping power equal to P_{max} for condenser length less than L_{max}.
4. Thus, the optimization routine proceeds as follows:
   a. select value for coolant velocity (V)
   b. calculate overall heat transfer coefficient (U) using the equations from Chapter II

c. calculate tube length and the rest of the condenser geometry to finally determine total condenser length (L_c)

d. check if pumping power (P) is less than P_{max} and $L_c = L_{max}$

e. if the above conditions for P and L_c are not met, then either change V or enhancement of heat transfer surface

5. a. if the calculated condenser length $L_c < L_{max}$ and $P < P_{max}$ change to less enhanced surface until $L_c = L_{max}$ or $P = P_{max}$

b. if $L_c > L_{max}$ and $P < P_{max}$ increase the surface enhancement until either $L_c = L_{max}$ or $P = P_{max}$

c. if $L_c > L_{max}$ and $P > P_{max}$ then decrease V until either $L_c = L_{max}$ and $P \leq P_{max}$ or $P = P_{max}$ and $L \leq L_{max}$

Further details of this optimization procedure can be found in Chapter V-Computer Modeling.

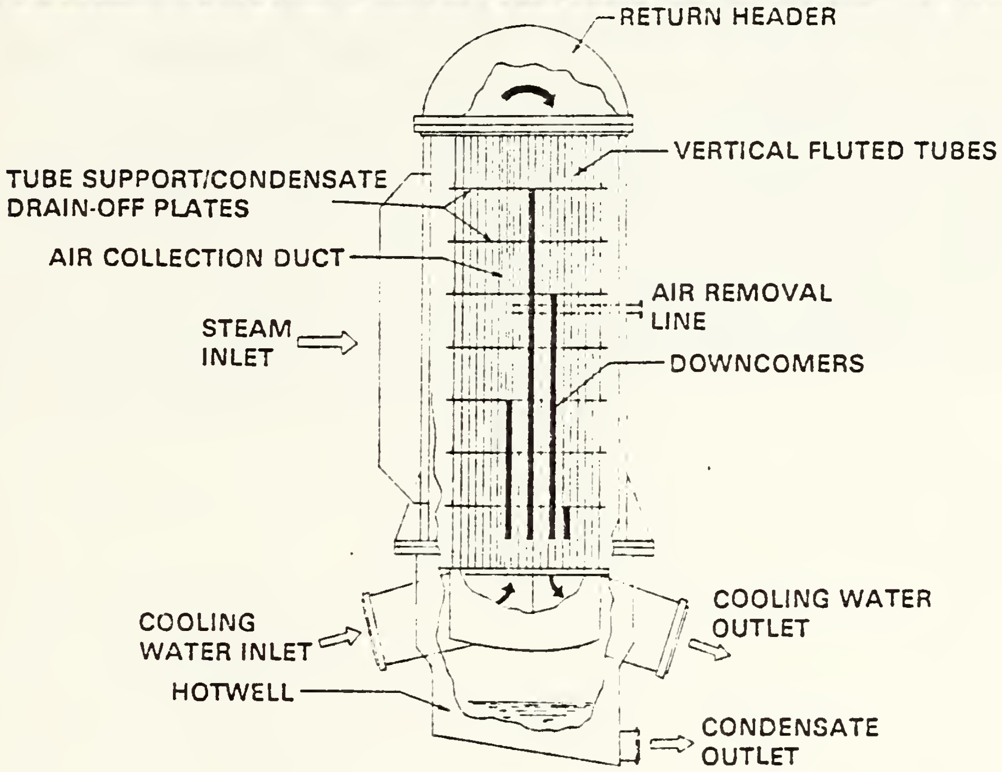
Vertical condenser tubes can provide great flexibility in this design optimization procedure due to the different possible combinations of internal and external enhancement configurations. Thus it is possible to "fine tune" the overall heat transfer coefficient to obtain the proper condenser length.

The procedure that has been outlined here is included in a preliminary design program VERTCON-2 along with the proposed vertical condenser arrangement shown in Figure-1.

Steam enters the condenser shell along a major portion of the tube length. Steam lanes distribute the steam around the tube bundle with radial steam inflow into the tube bundle towards the air cooler section and air removal duct. Steam condenses on the vertical tubes, and the condensate collects in the valley of the tube fluted surface flowing down the tube length. The tube support plates have a dual

FIGURE 1

VERTICAL ENHANCED TUBE CONDENSER



purpose by also serving as condensate drain-off plates. The condensate is stripped off the tubes and collected on the tube support plates, and then it is removed to the hotwell section via downcomer drainage tubes. Improved condensate control is afforded by this condensate drainage and removal scheme. The tube support/condensate drain-off plates divide the condenser up into sections, where the full depth of tubes in each section see relatively fresh steam. The tubes will see fresh steam through the depth of the tube bundle since the tubes are not subjected to condensate indundation effects because of effective condensate removal and vertical orientation of the tubes. A continuous duct that runs along the total tube length is utilized for air removal. Figure-1 shows a double flow coolant water circuit with the inlet/outlet header located in the bottom of the condenser. The condenser hotwell is located below and also encases the inlet/outlet header. This hotwell location adds to the total height of the condenser.

II. HEAT TRANSFER CALCULATIONS

Nomenclature

A	area (ft^2)
a	amplitude of the flute (ft)
Cp	specific heat (BTU/lbm °F)
D,d	diameter (ft)
e	helical ridge height (ft)
f	friction factor $\left[=(\Delta P/\rho)(D/L)(2g_c/V^2)\right]$
G	mass flux (lbm/ft^2hr)
g_c	gravitational constant ($lbm\ ft/lbf\ hr^2$)
h	heat transfer coefficient ($BTU/hr\ ft^2\ ^{\circ}F$)
h_{fg}	latent heat of vaporization (BTU/lbm)
K	thermal conductivity ($BTU/hr\ ft^2\ ^{\circ}F$)
L	length (ft)
l	lead of ridge (axial distance per 360° turn)(ft)
m	operand for friction factor equation
Nu	Nusselt number
P	pressure (lbf/ft^2) or (in-hg-abs)
p	pitch of ridging or flute (ft)
Pr	Prandtl number
Q	heat flow (BTU/hr)
r	operand for friction factor equation
Re	Reynolds number
Re_f	flooding Reynolds number for flute $\left[= 4W_f/\mu X_L\right]$**
R_f	thermal resistance of axial flute ($hrft^2\ ^{\circ}F/BTU$)
R_s	thermal resistance of scale ($hrft^2\ ^{\circ}F/BTU$)
R_w	thermal resistance of tube wall ($hrft^2\ ^{\circ}F/BTU$)
St	Stanton number
T	temperature (°F)
ΔT	$T_{sat}-T_W$ (°F)
ΔT_{lm}	mean overall temperature difference (°F)
U	overall heat transfer coefficient ($BTU/hrft^2\ ^{\circ}F$)

U^+	Dimensionless velocity $\left[= U/U^*\right]$ *
U^*	friction velocity $\left[= \sqrt{\tau_o g_c/\rho}\right]$ *
V	velocity (ft/hr) or (ft/sec)
W_f	flooding axial mass flow of condensate per flute (lbm/hr)
X_L	half-perimeter length of flute (ft)

Subscripts

b	fluid at the bulk temperature (°F)
c	condensate
f	fluid or flooding
i	inside, or inlet
n	nominal
o	outside, or outlet
s	increment, or section
sat	saturation
w	wall

Superscripts

+	dimensionless parameter

Greek Symbols

α	height of the condensate in the center of the flute (ft)
γ	fragment of $U_e^+ \left[= -2.5\ln(2e/d_i) + 3.75\right]$ *
λ	dimensionless group $= \frac{4\rho^2}{\mu^2} g_c \frac{(\alpha_o)^4}{X_L}$ **
μ	dynamic viscosity (lbm/hr ft)
ρ	density (lbm/ft^3)
σ	surface tension (lbf/ft)
Ω	non-dimensional group †
τ	apparent wall shear stress

* Reference (6)
** Reference (4)
† Reference (1)

II.A ANALYSIS SUMMARY

The rate of heat flow Q over an entire heat exchanger is related to the mean overall temperature difference ΔT_{lm} and the total heat transfer area A by the overall heat transfer coefficient U.

$$Q = UA\,\Delta T_{lm} \qquad \text{(II-1)}$$

From the analysis of externally ridged tubes it proves convenient to base the heat transfer coefficient U on the surface area of a smooth tube having an outside diameter D_n equal to the diameter measured over the mid-height of the external flutes (see Figure-2 for external flute geometry).

$$U_n = \frac{Q}{\pi D_n L}\,\Delta T_{lm} \qquad \text{(II-2a.)}$$

where: $$D_n = D_w + 2a \qquad \text{(II-2b.)}$$

The overall U depends on the resistances in series; between cooling water and the tube wall, within the tube wall, and between the tube wall and the working fluid. Fouling resistance on either side of the tube wall can be combined in one term R_s.

Tube wall resistance is considered to be the summation of an equivalent smooth tube resistance and flute resistance. Tube wall resistance for a smooth (non-enhanced) tube can be expressed as:

$$R_w = \frac{1}{h_w} = \frac{\ln(D_o/D_i)}{K}\frac{D_{ref}}{2} \qquad \text{(II-3)}$$

Where D_{ref} is the reference diameter on which the overall U is based on. For a fluted tube the overall U is based

on the nominal diameter D_n. As will be discussed in the next section , the flute resistance can be written:

$$R_{flute} = \frac{\ln\left[7.985(a)^{.7688}\left(\frac{a}{p}\right)^{-.0242} + 1\right] D_{ref}}{2k} \quad \text{(II-4)}$$

Therefore, the total tube wall resistance for the fluted tube can be written:

$$R_w = \frac{\ln\left[D_w/D_i\left(7.985(a)^{.7688}\left(\frac{a}{p}\right)^{-.0242} + 1\right)\right] D_n}{2k} \quad \text{(II-5)}$$

Therefore, the concept of resistances in series yields:

$$\frac{1}{U_n} = \frac{D_n/D_i}{h_f} + R_s + R_w + \frac{1}{h_c} \qquad \text{(II-6)}$$

The overall heat transfer coefficient U_n in Equation (II-6) does not consider non-condensable gas effects. Condensing in the presence of non-condensable gases will be taken up in Chapter V.

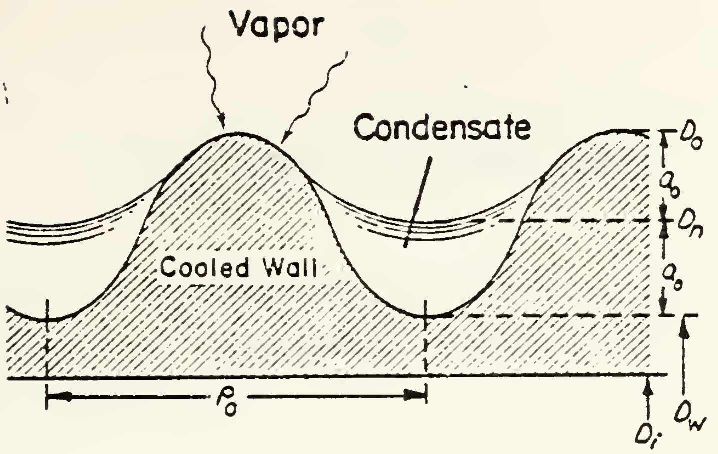


FIGURE 2
Cross Section of Vertical Fluted Condenser Surface

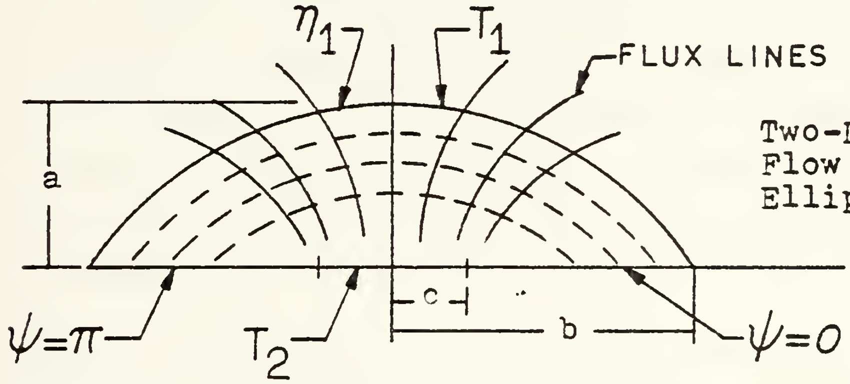


FIGURE 3a.
Two-Dimensional Heat Flow Through an Elliptical Boundary

η_i -Constant Temperature Line
η,ψ,Z -Coordinates
Z -Coordinate into Page

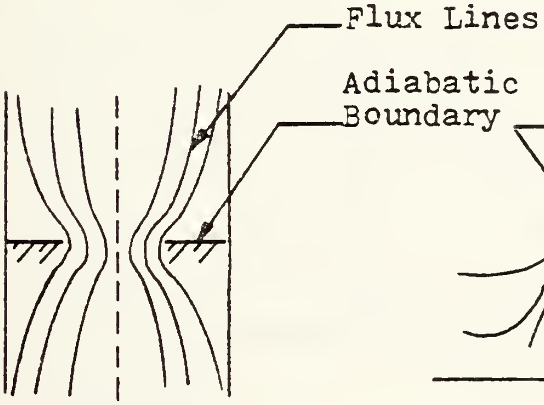


FIGURE 3b.
Heat Flow Restriction Model

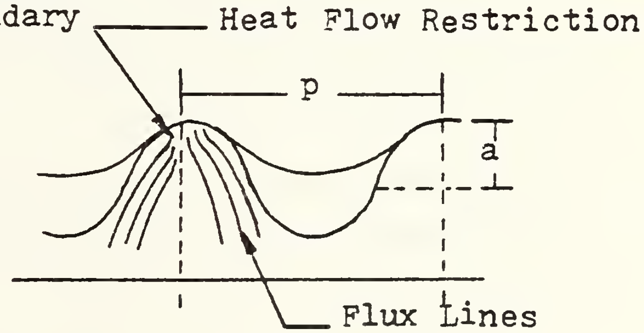


FIGURE 3c.
Application of Heat Flow Restriction Model to Axial Fluted Tubes

II.B TUBE WALL RESISTANCE

One method to evaluate tube wall resistance for fluted tubes would be to use Equation-3, where D_o and D_{ref} equal the nominal diameter D_n.

$$R_w = \frac{\ln(D_n/D_i)D_n}{2K} \qquad \text{(II-7)}$$

This formulation is not sensitive to flute geometry such as changes in amplitude-to-pitch (a/p)ratio. In comparing Equation-7 with finite element formulation differences of up to 30 percent in flute resistance are observed. This could amount to 10 percent difference when considering total tube wall resistance for tubes made with low conductivity materials such as titanium with thin tube wall dimensions.

What is proposed here is to model flute and condensate layer as a heat flow restriction, by describing two-dimensional heat flow through an elliptical boundary. (Figure-3a) This serves as an accurate model for a heat flow restriction as shown in Figure-3b.

The procedure is to apply this same model to describe the heat flow through the axial flutes (Figure-3c).

Some aspects of curvilinear coordinate systems (7) will be reviewed in order to develop nomenclature to be used in later equations.

Rectangular coordinates x,y,z, can be expressed in terms of new coordinates u_1, u_2, u_3 by the equations: (see Figure-4)

$$x = x(u_1, u_2, u_3)$$

$$y = y(u_1, u_2, u_3)$$

$$z = z(u_1, u_2, u_3)$$

The coordinates are chosen such that u_1, u_2, u_3, are orthogonal curvilinear coordinates. From examination of Figure-4 it can be seen that if a particle moves from point P in such a way that u_2 and u_3 are held constant and only u_1 varies, then a curve in space is generated.

Let $\bar{r}$ represent the position vector of a point P in space:

$$\bar{r} = x\hat{i} + y\hat{j} + z\hat{k}$$

where $\hat{i}$, $\hat{j}$, and $\hat{k}$ are unit vectors in the x, y and z directions respectively.

Then a tangent vector to u_1 curve at point P is given by:

$$\bar{U}_1 = \frac{\partial \bar{r}}{\partial u_1} = \frac{\partial \bar{r}}{\partial s_1} \frac{dS_1}{du_1} \qquad \text{(II-8)}$$

$S_1 =$ arc length along the u_1 curve

$\frac{\partial \bar{r}}{\partial s_1}$ is a unit vector

thus: $\bar{U}_1 = h_1 \bar{u}_1$

where $\bar{u}_1$ is the unit vector tangent to u_1 in the direction of increasing arc length,

thus: $h_1 = \frac{dS_1}{du_1}$ which is the length of $\bar{U}_1$

Similarly for the other coordinates:

$$\bar{U}_1 = h_1\bar{u}_1 \quad \bar{U}_2 = h_2\bar{u}_2 \quad \bar{U}_3 = h_3\bar{u}_3$$

$$h_1 = \frac{dS_i}{du_i} \quad i = 1,2,3 \quad dS_i = h_i du_i \quad i = 1,2,3 \qquad \text{(II-9)}$$

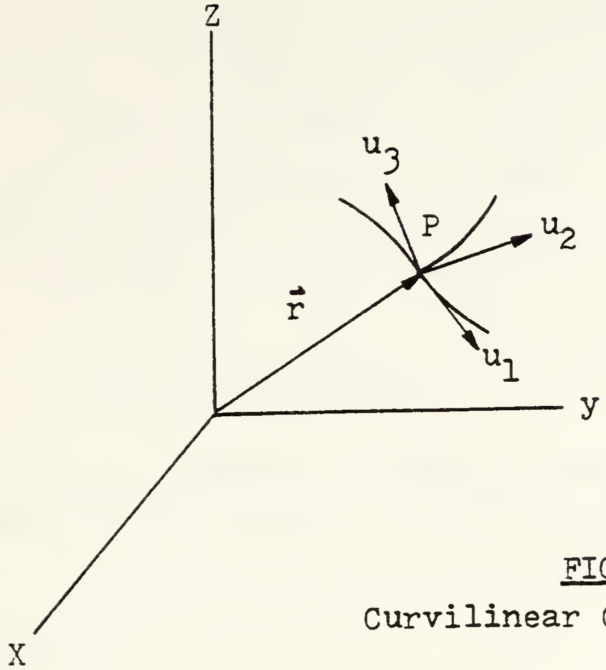


FIGURE 4
Curvilinear Coordinate System

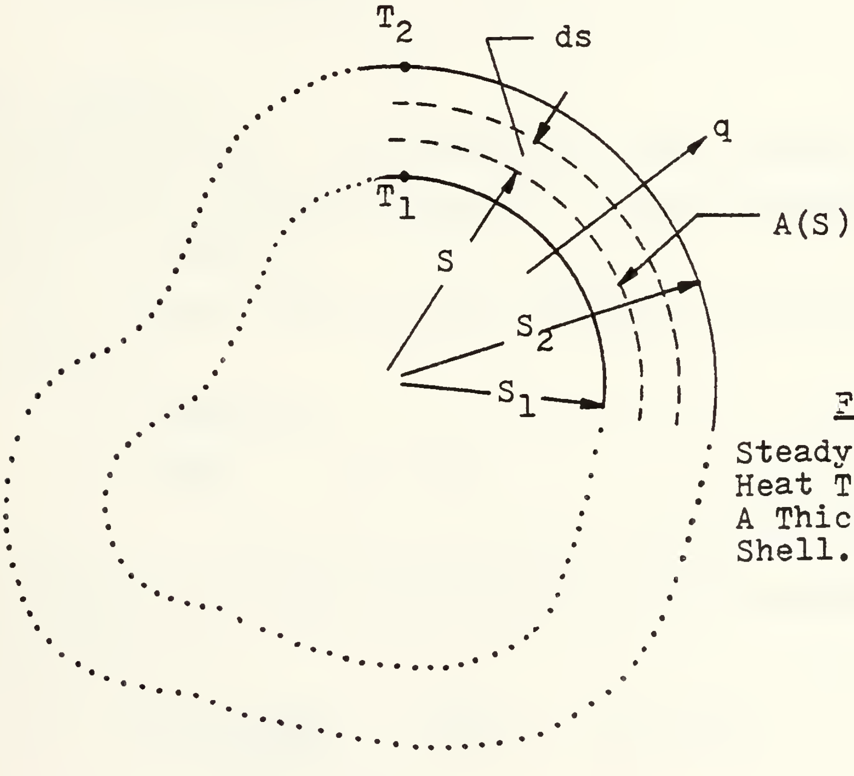


FIGURE 5
Steady One-Dimensional Heat Transfer Through A Thickwalled Cylindrical Shell.

The unit vectors $\bar{u}_1$, $\bar{u}_2$ and $\bar{u}_3$ form a right-handed orthogonal curvillinear coordinate system. The vectors $\bar{U}_1 du_1$, $\bar{U}_2 du_2$, and $\bar{U}_3 du_3$ are mutually perpendicular vectors having as their length the arc-length differentials dS_1, dS_2, and dS_3. Therefore, an element of volume can be determined from the following vector product:

$$dV = \bar{U}_1 du_1 \times \bar{U}_2 du_2 \cdot \bar{U}_3 du_3$$
$$dV = h_1 h_2 h_3 du_1 du_2 du_3 \qquad \text{(II-10)}$$

Also, a vector element of surface area dA, on the surface $u_1 =$ constant is given by the vector product:

$$dA_2 = h_2 h_3 du_2 du_3 \qquad \text{(II-11a)}$$

and similarly:

$$dA_2 = h_3 h_1 du_3 du_1 \qquad \text{(b)}$$

$$dA_3 = h_1 h_2 du_1 du_2 \qquad \text{(c)}$$

GENERAL PROBLEM

Consider steady one-dimensional heat transfer through a thick-walled cylindrical shell a section of which is displayed in Figure-5.

First assume rate of heat transfer is constant at every section.

$$Q = \text{constant}$$
$$Q = -KA\frac{dT}{ds} \qquad \text{(II-12)}$$

Where S is defined in Figure-5, integrating Equation (II-12) between inner and outer surfaces with the assumption K is constant yields:

$$Q = \frac{T_1 - T_2}{(1/K)\int_{s_1}^{s_2} ds/A(s)} = \frac{\Delta T}{R} \qquad \text{(II-13)}$$

where R is heat transfer resistance.

thus: $$R = \frac{\Delta T}{Q} = \frac{1}{K}\int \frac{ds}{dA(s)} \qquad \text{(II-14)}$$

Equation (II-14) is only applicable when heat flux is constant across the heat transfer area. In many geometries the distribution of heat flux across the heat transfer area may not be uniform, but equation (II-14) can be expanded for variable heat flux (16).

Equation (II-14) can be rewritten:

$$Q = -K\iint_A \frac{dT}{ds_1}\, ds_2 ds_3 \qquad \text{(II-15)}$$

where:
$$ds_1 = h_1 du_1$$
$$ds_2 = h_2 du_2$$
$$ds_3 = h_3 du_3$$

thus: $$\frac{dT}{ds_1} = \frac{1}{h_1}\,\frac{dT}{du_1}$$

and substituting into Equation (II-15):

$$Q = -K\,\frac{dT}{du_1}\iint_A \frac{h_2 h_3}{h_1}\, du_2 du_3$$

thus:

$$R = \frac{\Delta T}{Q} = \frac{1}{K}\int_{(u_1)_I}^{(u_1)_{II}} \frac{du_1}{\iint \frac{h_2 h_3}{h_1}\, du_2 du_3} \qquad \text{(II-16)}$$

Equation (II-16) is applicable to variable heat flux distribution over the area of heat transfer. But this equation is restricted to one-dimensional, steady state analysis, where temperature is a function of u_1 only.

Equation (II-16) can be applied to the model of two dimensional heat flow through an elliptical boundary.

Referring to Figure-3 and Equation (II-16) the following expressions are obtained:

$\eta_i =$ constant temperature lines

η, ψ, z coordinates, where z coordinate is into page

$$h_i = \left| \frac{\partial \bar{r}}{\partial u_i} \right| \quad \text{where} \quad \bar{r} = x\hat{i} + y\hat{j} + z\hat{k}$$

thus:
$$h_i^{\,2} = \left(\frac{\partial x}{\partial u_i}\right)^2 + \left(\frac{\partial y}{\partial u_i}\right)^2 + \left(\frac{\partial z}{\partial u_i}\right)^2 \qquad \text{(II-17)}$$

in this case:

$$x = c \cosh\eta \cos\psi$$
$$y = c \sinh\eta \sin\psi$$
$$z = z$$

from Equation (II-17):

$$h_1 = h_2$$
$$h_3 = 1$$

substituting into Equation (II-16)

$$R = \frac{1}{K}\int_0^{\eta_1} \frac{d\eta}{\int_0^{\pi}\int_0^{L} d\psi dz}$$

and:
$$R = \frac{\eta_1}{K\pi L} \qquad \text{(II-18)}$$

$$x = c \cosh\eta \cos\psi$$
at $\psi = 0$
$$x = b = c \cosh\eta$$
and similarly:
$$y = c \sinh\eta \sin\psi$$
at $\psi = \pi/2$
$$y = a = c \sinh\eta$$
therefore:
$$\eta_1 = \sinh^{-1}\left(\frac{a}{c}\right)$$
and from a fundamental identity:
$$\eta_1 = \ln\left[\frac{a}{c} + \sqrt{\frac{a}{c}^2 + 1}\right] \qquad \text{(II-19)}$$

For axial fluted tubes Equations (II-18) and (II-19) will take the form:

$$R = \frac{\eta}{K\pi L} \qquad \text{(II-20a.)}$$

$$\eta = \ln\left[C\left(\frac{a}{p}\right) + \sqrt{C^2\left(\frac{a}{p}\right)^2 + 1}\right] \quad \text{(b)}$$

where C is defined as the flute resistance coefficient.

Equation (II-20) is in good agreement with actual physical situation:

If $a = o$ this implies $\eta = \ln(1) = o$, thus $R = o$

1. Therefore a flute of zero amplitude has zero thermal resistance.

2. As flute amplitude increases the flute resistance also increases.

Equation (II-20) can be simplified as follows:

1. Amplitude-to-pitch (a/p) ratios for fluted tubes are in the range of .1-to-.5.

2. The flute resistance coefficient (C) will have an approximate value of $C = .5$ thus,

$$\sqrt{C^2\left(\frac{a}{p}\right)^2+1} \approx 1$$

and equations (II-20) simplifies to:

$$R = \frac{\ln\left[C\left(\frac{a}{p}\right)+1\right]}{K\pi L} \quad \text{(II-21)}$$

To express thermal resistance in units of ($hrft^2$ F/BTU) multiply R from Equation (II-21) by a reference area (A_{ref}). This reference area should be the same reference area that the overall heat transfer coefficient is based on. thus:

$$R = \frac{A_{ref}(\Delta T)}{Q} \quad (hrft^2\ {}^{\circ}F/BTU) \quad \text{(II-22)}$$

$$R_f = \frac{\ln\left[C\left(\frac{a}{p}\right)+1\right]D_{ref}}{2K} \quad \text{(II-23)}$$

Values for flute resistance are calculated from a finite element formulation using the ADINAT (2) finite element program. The flute resistances obtained from ADINAT are substituted into Equation (II-23), and the values for flute resistance coefficient (C) are determined at different values of a/p, and for various tube dimensions. One result is that C is a function of both flute amplitude a, and a/p ratio. Thus C as a function of a and a/p was correlated using multiple non-linear regression (15) analysis.

Then a final formulation for flute thermal resistance was established in the following form:

$$R_f = \frac{\ln\left[b_o(a)^{b_1}\left(\frac{a}{p}\right)^{(1+b_2)}+1\right]D_{ref}}{2K} \quad \text{(II-24)}$$

where $C = b_o(a)^{b_1}\left(\frac{a}{p}\right)^{b_2}$

b_o, b_1, b_2 are correlation coefficients resulting from the multiple non-linear regression.

The convection heat transfer coefficient along the perimeter of the flute is required for input into ADINAT. The distribution of convection heat transfer coefficient was estimated by first calculating the condensate film thickness (δ) along the perimeter using a procedure established by BARNES (4). The convection heat transfer at a particular point along the flute perimeter can be approximated by:

$$h_c = \frac{K}{\delta} \quad (\mathrm{BTU/hrft^2\,°F})$$

where K is the thermal conductivity of the condensate, and δ is the condensate film thickness.

As stated before, the finite element results, along with regression analysis were used to determine the value of C shown below:

$$C = b_o(a)^{b_1}\left(\frac{a}{p}\right)^{b_2} \qquad \text{(II-25)}$$

$b_o = 7.985$

$b_1 = .7688$

$b_2 = {}^{-}1.0242$

Finally, equation (II-24) can be written:

$$R_f = \frac{\ln\left[7.985(a)^{.7688}\left(\frac{a}{p}\right)^{-.0242} + 1\right] D_{ref}}{2K} \qquad \text{(II-26)}$$

The final results of this analysis is depicted in Figure (6). Curve-1 represents flute thermal resistance as a function of a/p ratio for constant amplitude flutes. In Curve-2 both flute amplitude and a/p ratio are varying, but flute pitch is held constant.

Also, compared on this plot are the flute thermal resistance calculated from the finite element results, from Equation (II-7) based on nominal tube diameter, from Equation (II-26) based on heat flow restriction model. For ease of comparison thermal resistance have been normalized on the maximum range of thermal resistance considered, which is denoted R_f*. Figure-16 shows that the discrepancy in value between the finite element solution and result based on nominal tube diameter from Equation (II-7) increases with increasing flute amplitude and a/p ratio. Both Curve-1 and Curve-2 together cover a broad range of flute amplitude and a/p values to demonstrate the fit of Equation (II-26).

From the results that were depicted in Figure-6 indicate that Equation (II-26) accurately models the flute thermal resistance. This equation is based on finite element solution which must also be checked to ascertain whether it is a valid model. If the finite element model is developed methodically from a simple configuration where the results are known, to the more complicated model, then confidence can be placed on the results. In this case accurate specification of convection heat transfer coefficients was important to the accuracy of the finite element solution. Also, a simple check can be made to determine if the finite element grid has the proper number of elements to model the problem. This is done by checking the heat flux approaching a boundary node from different elements. If the grid is sufficient there should be no jump in the value of total heat flux across the boundary at a specified location. These general guidelines were followed in modeling tube wall resistance for axially fluted tubes.

Curve-1 - Constant a (a= .002 ft.)
Curve-2 - Constant p (p= .00625 ft.)
———————— Finite Element Solution
- - - - - - - - - Equation (II-26) Heat Flow Restriction Model
——— ——— ——— Equation (II-7) Average Flute Resistance

$R^* = 4.03 \times 10^{-4}$ ($hr ft^2 \, ^\circ F/BTU$)

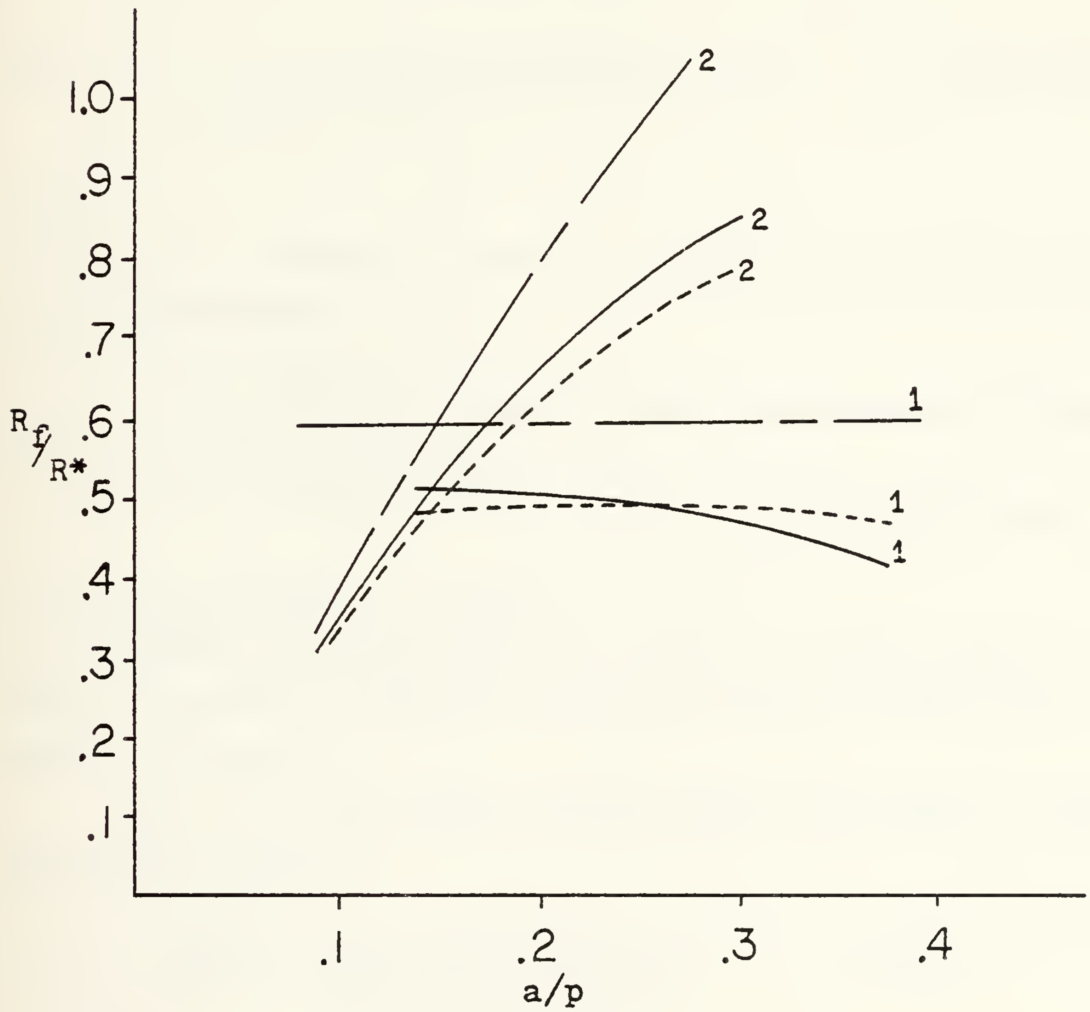


FIGURE 6

Comparison of Flute Thermal Resistance Calculations

II.C WATERSIDE HEAT TRANSFER COEFFICIENTS

1. Smooth Internal Tube

For the smooth internal tube the McAdams correlation was used for determination of the heat transfer coefficient for cooling water.

$$\left[\frac{hD}{K_b}\right] = 0.023\left[\frac{GD}{\mu_b}\right]^{0.8}\left[\frac{\mu Cp}{k}\right]^{0.4} \qquad \text{(II-27a.)}$$

or:

$$h_f = 0.023\,\frac{K_b}{D}\left[Re_D\right]^{0.8}\left[Pr\right]_b^{\;0.4} \qquad \text{(II-27b.)}$$

where:

(a) all fluid properties are evaluated at the bulk fluid temperature;

(b) 2300 Re_D 10^7 where;

Re_D= Reynolds number based upon hydraulic diameter

(c) 0.5 Pr_b 120 where;

Pr_b= Prandtl number based on bulk temperature

2. Helical Internal Ridging

Internal tube augmentation was also investigated, with the application of integral multiple helix ridging (13) as shown in Figure-7.

Friction factor data was correlated by means of the following equation:

$$\sqrt{\frac{f}{8}} = \frac{1}{2.46\ \ln\left[r+(7/Re)^m\right]} \qquad \text{(II-28)}$$

For the roughness of the helical internal ridge both r and m are treated as variables, and are tied in with tube geometry.

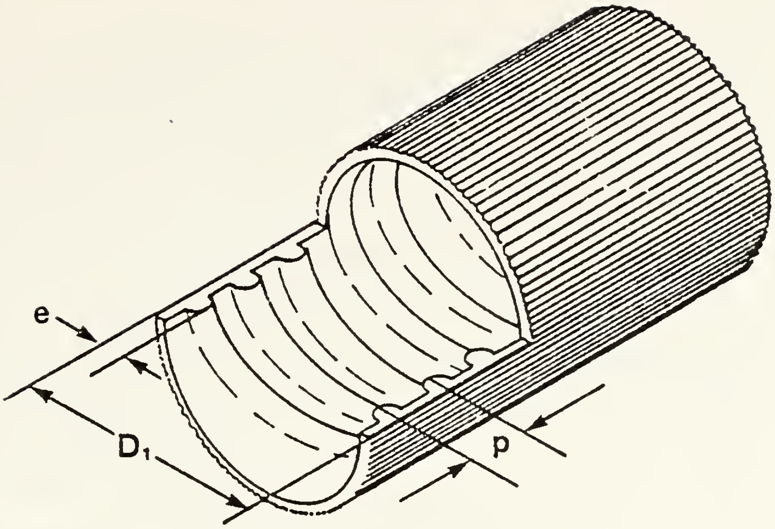
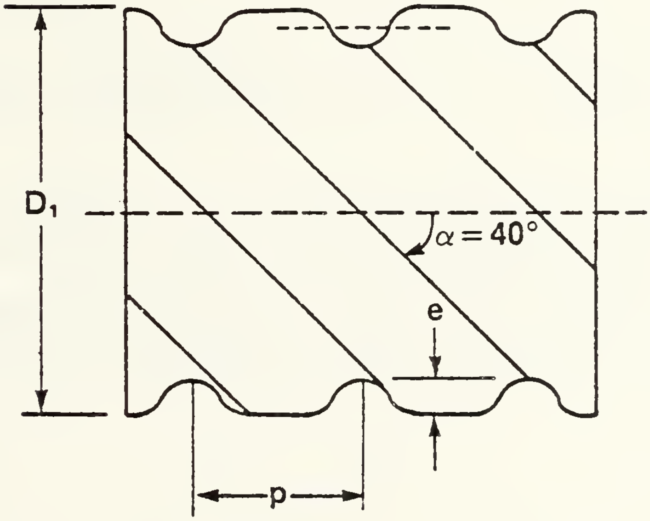


FIGURE 7

Internal Enhancement Geometry



Both m and r vary with the dimensionless parameter e/l, where e is the ridge height and l is the lead of the ridge. In reference (13) WITHERS has made a distinction based on the criterion p/d = 0.36 in correlating the friction behavior to the internal geometry. It has been proposed that a shift in flow behavior occurs at p/d = 0.36. For higher values of p/d a greater degree of swirling could occur, compared with cascading of flow that takes place if p/d is less than 0.36.

The heat transfer correlation equation developed from data for tubes of various configurations, and solved for the Stanton number becomes:

$$St = \frac{\sqrt{f/8}}{5.68(e/p)^{-1/8}\sqrt{Pr}\left[(e/d_i)Re\sqrt{f/8}\right]^{0.136} + \gamma} \qquad \text{(II-29a.)}$$

where: $\gamma = -\left[2.5 \ln(2e/d_i) + 3.75\right]$ (II-29b.)

Then, h_f becomes:

$$h_f = \frac{K_c Re\ Pr\ St}{d_i} \qquad \text{(II-30)}$$

These equations are applicable to Reynolds number range 10,000-120,000 and Prandtl number range 4-10.

See Appendix-B for tube data, and friction factor characteristics of multiple-helix internal ridged tubes.

II.D CONDENSATE HEAT TRANSFER COEFFICIENT

In reference-2 BARNES developed an equation for the average value of condensate heat transfer coefficient (h_c) for an externally fluted tube of length L. In this formulation h_c depends upon physical properties of the condensing fluid geometric factors, and mass flow rate of condensate in the flutes.

$$\bar{h}_c = .6027\left[\frac{h_{fg}W_f}{L\Delta T}\right]^{.0074} \frac{a^{.2307}(Nu_o\Omega^{\frac{1}{4}})^{.9226}}{p} \left[\frac{K^3\rho\sigma h_{fg}g_c}{\mu\Delta T}\right]^{.2307}$$

(II-31)

The non-dimensional group $Nu_o\Omega^{\frac{1}{4}}$ defined in Reference-2 is a function of flute amplitude-to-pitch (a/p) ratio. It should be noted in Equation-31 the parameter L is the length of the tube between condensate drain off plates.

The parameter W_f is the flooding axial mass flow of condensate per flute (LBM/hr) which is also a function of physical properties of the fluid, and tube geometry. In Reference-7, PANCHEL and BELL defined the following flooding Reynolds number for condensate flow in the axial flutes:

$$Re_f = \frac{4Wf}{\mu X_L} \qquad \text{(II-32)}$$

and an additional non-dimensional group:

$$\lambda = \frac{4\rho^2}{\mu^2}\, g_c \frac{(\alpha_o)^4}{X_L} \qquad \text{(II-33)}$$

where α_o = 2a in the case where the flute is flooded. Also, the following correlation was developed for λ_f based on the

half-perimeter length (X_L) of the flute:

$$\lambda_f = 36(a/p)\exp(3.33\ a/p)\ Re_f \quad \text{(II-34)}$$

Substituting equations (32)and 33)into equation (34) and solving for W_f yields:

$$W_f = \frac{8}{9}\ g_c \frac{\rho^2}{\mu}\ a^3 p\ \exp\left[-3.33\left(\frac{a}{p}\right)\right] \quad \text{(II-35)}$$

where W_f is now based on full perimeter length of the flute.

The non-dimensional group $Nu_o\Omega^{\frac{1}{4}}$ can also be defined by the following correlation:

$$Nu_o\Omega^{\frac{1}{4}} = b_o + \sum_{k=1}^{4} b_k(a/p)^k \qquad \text{(II-36)}$$

where b_o, b_k are the correlation coefficients:

$b_o = 3.661218$

$b_1 = 6.4526$

$b_2 = {}^-15.265$

$b_3 = 16.14543$

$b_4 = {}^-6.561166$

Equation (II-36) was obtained from multiple linear regression, using techniques of Reference-15.

II.E SELECTION OF EXTERNAL ENHANCEMENT GEOMETRY

The heat transfer performance of various vertical tubes with different levels of external enhancement were examined. This was done to provide guidance in the selection of external enhancement geometry. The basic parameters that need to be specified are:

1. Flute amplitude (a)
2. Flute amplitude-to-pitch (a/p) ratio,

and thus flute pitch (p) is also set.

3. The ratio of condensation rate-to-flooding condensation rate (W/W_f).

The following plots of h_c/h_c* versus a/p ratio (Figure-8) illustrate the relationship between these basic parameters. The variable h_c is the condensing heat transfer coefficient, which is normalized in the plots by the maximum value of h_c (denoted h_c*). The curves of Figure-8 denote lines of constant flute pitch (p) where curve-1 is for W/W_F= 0.1 and curve-2 is for W/W_F= 0.25. It can be observed that greater enhancement is achieved with larger values of a/p ratios, and smaller values of flute pitch. In Figure-9 a line of constant flute amplitude is also plotted at W/W_f=.1. Figure-9 shows that increased enhancement is more readily achieved from increasing a/p ratio by reducing flute pitch.

Increased steamside enhancement cannot be achieved without paying a cost. The cost in this case is increased tube wall resistance. In Figure-10 normalized flute resistance for constant amplitude flutes (curve-1, a= .0125") and constant pitch flutes (curve-2, p= .025") are plotted along with heat transfer data for W/W_F= .1. Figure-11 is a similar plot except curve-1 is for a= .01, and curve-2 is for p= .050. Flute thermal resistance was calculated using

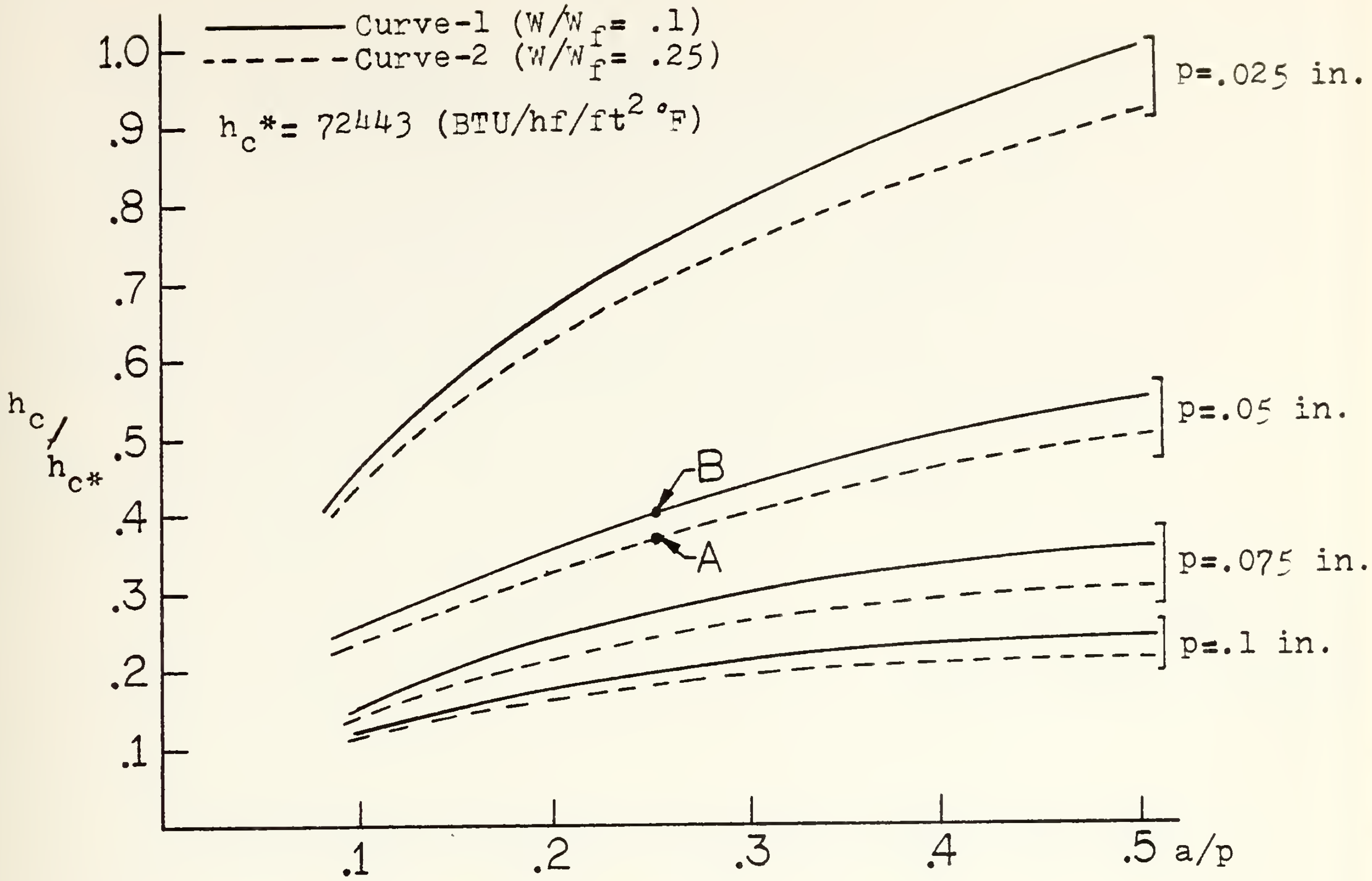


FIGURE 8 - Constant Flute Pitch Curves

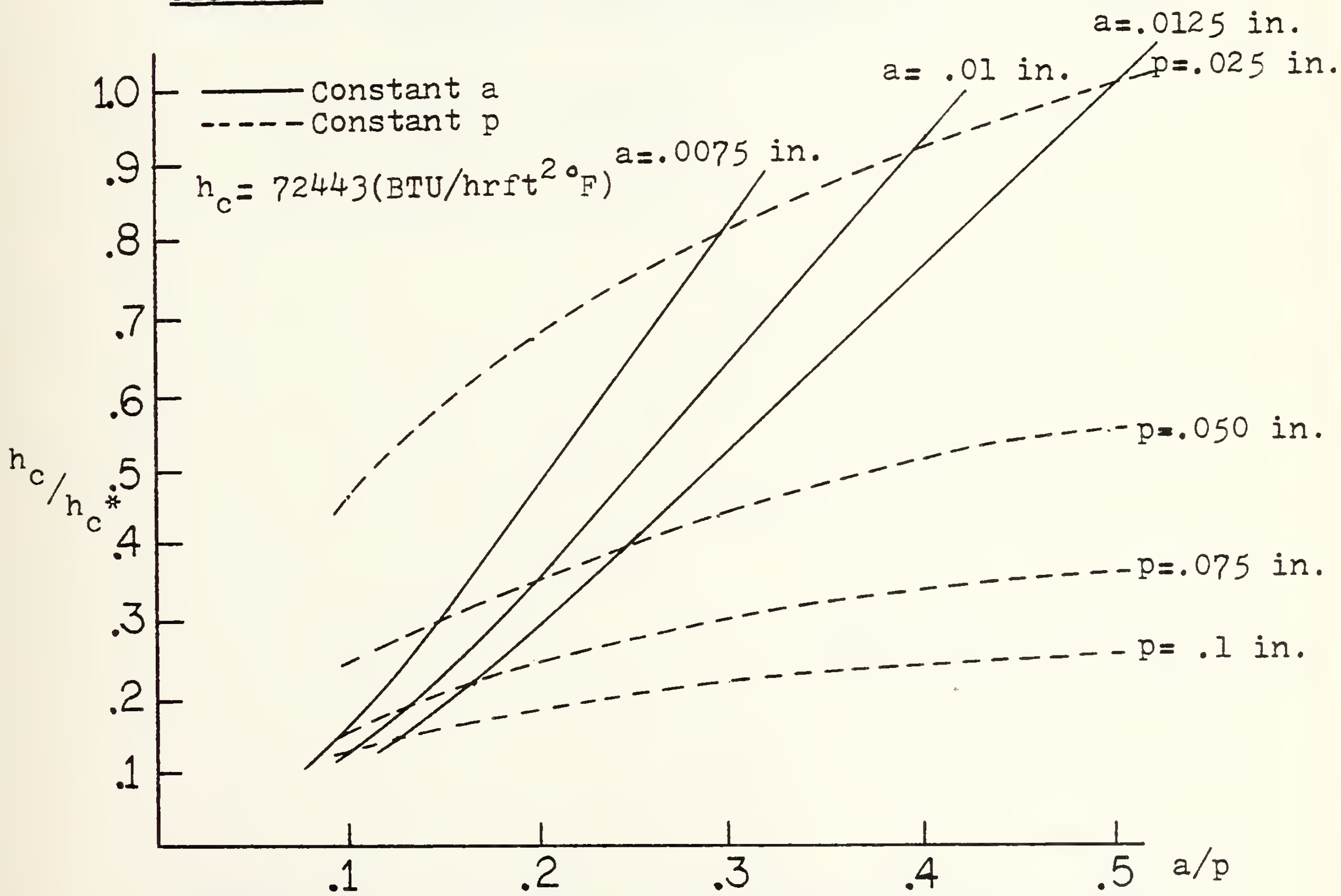


FIGURE 9 - Constant Flute Amplitude and Pitch Curves at $W/W_F = .1$

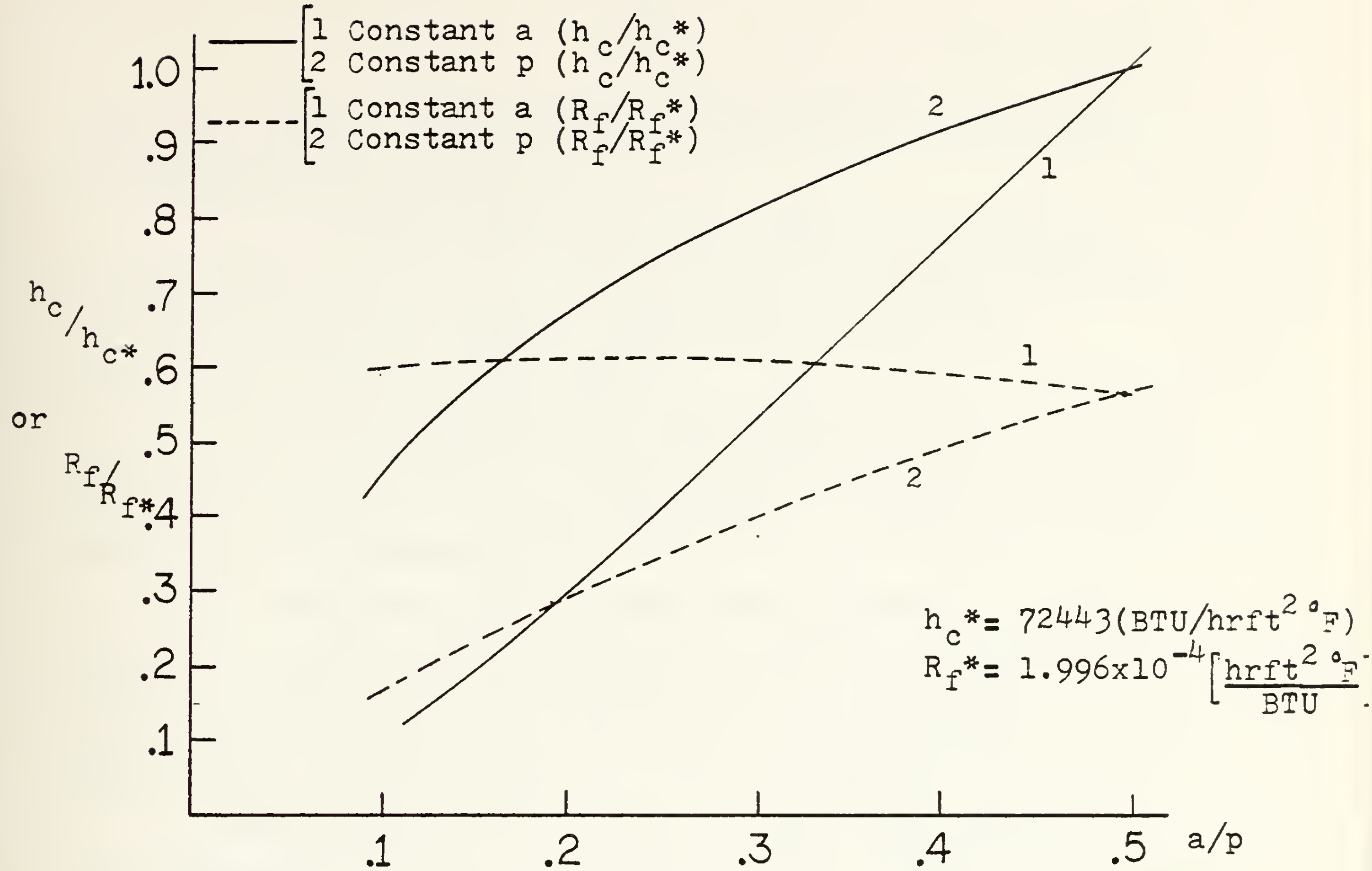


FIGURE 10 - Constant Amplitude (a=.0125in.) and Constant Pitch (p= .025in.) Curves

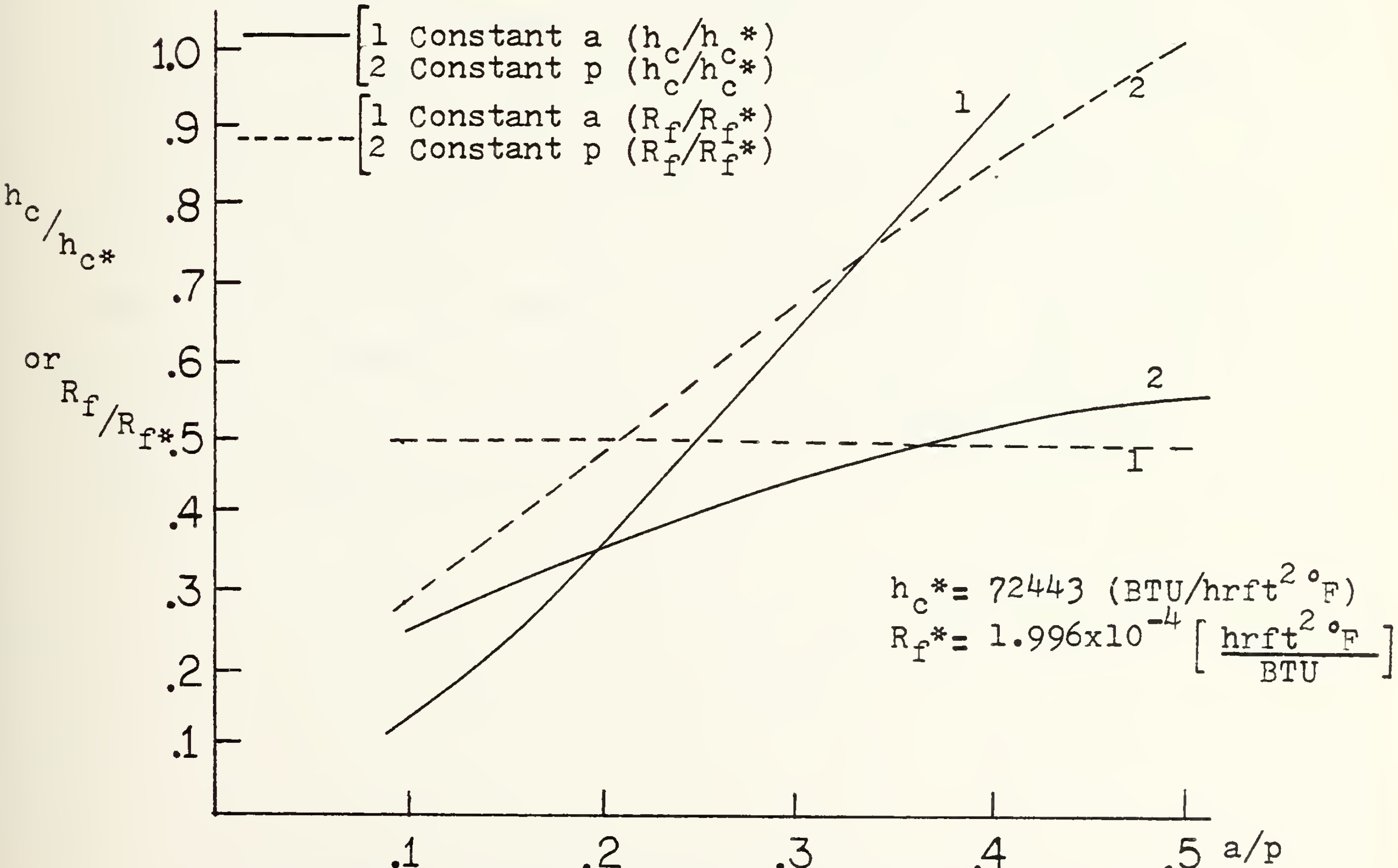


FIGURE 11 - Constant Amplitude (a=.01in.) and Constant Pitch (p=.050 in.) Curves

Equation (II-26). From Figures 10 and 11, the following conclusions can be inferred:

1. When comparing on the basis of flute pitch at the same a/p ratio, it can be seen that flute resistance is larger and heat transfer enhancement is less for increased values of flute pitch.

2. When comparing on the basis of flute amplitude at the same value of p, it can be seen that flute resistance is larger and heat transfer enhancement is greater for increased values of flute amplitude.

This indicates that the best tube geometry is small amplitude flutes with larger a/p ratios, and thus smaller values of flute pitch in comparison with other tubes. Thus the benefits of increased heat transfer enhancement are obtained with minimum increase in flute thermal resistance.

As demonstrated by Figure-8, greater heat transfer enhancement is achieved at smaller values of W/W_F, but there is a lower limit in practical condenser design for W/W_F. Reduced values of condensate drainage plate spacing are required to achieve smaller values of W/W_F. Close drainage plate spacing will complicate condenser fabrication and maintenance. Therefore acceptable values of condensate drainage plate spacing should be used as criteria for specifying W/W_F in a practical condenser design. This criteria is used in the program VERTCON-2 to specify W/W_F. This effect of W/W_F on condensate drainage plate spacing can also be observed in Figure-8, where the drainage plate spacing at Point A (W/W_F= .25) is equal to 1.632 ft. while at Point B (W/W_F= .1) the spacing is 0.622 ft.

III. CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES

Nomenclature

A	Flow area (m^2), (ft^2)
a	Constant defined in Equation (III-6)
B	Term defined by Equation (III-14a)
C_{pf}	Specific heat of cooling water $(J/kg\,^\circ K)$, $(BTU/lbm\,^\circ F)$
D_o	External diameter of tube (m), (ft)
D_n	Nominal diameter of axial fluted (m), (ft) tube
D_p	Coefficient of vapor diffusion in gas (kg/m s)/Pa, $(Lbm/ft\ s)/Lbf/ft^2$
G_c	Vapor mass velocity approaching tube row (kg/m^2s), (Lbm/ft^2s)
g	Gravitational acceleration (m/s^2), (ft/s^2)
H_c	Specific enthalpy of vapor (J/kg), (BTU/Lb)
h_{fg}	Specific latent heat (J/kg), (BTU/Lb)
H_v	Specific enthalpy of vapor (J/kg), (BTU/Lb)
L	Length of condenser (m), (ft)
N	Number of tubes per row
P_a	Partial pressure of air in bulk of mixture (P_a), (lb/ft^2)
P_m	Pressure of mixture (P_a), (lb/ft^2)
P_s	Partial pressure of vapor at condensate surface (P_a), (lb/ft^2)
P_v	Partial pressure of vapor in bulk of mixture (P_a), (lb/ft^2)
q	Heat flux (W/m^2), $(BTU/hrft^2)$
T_f	Temperature of coolant $(^\circ K)$, $(^\circ R, ^\circ F)$
T_{LM}	Mean temperature of coolant in tube $(^\circ K)$, $(^\circ R, ^\circ F)$
T_s	Temperature of condensate surface $(^\circ K)$, $(^\circ R, ^\circ F)$
T_v	Temperature of vapor $(^\circ K)$, $(^\circ R, ^\circ F)$
T_w	Temperature of outer tube wall $(^\circ K)$, $(^\circ R, ^\circ F)$
U_n	Heat transfer coefficient for convection and tube resistances as defined by Equation (II-6) $(BTU/hrft^2\,^\circ F)$
U_o	Overall heat transfer coefficient between vapor and coolant $(W/m^2\,^\circ K)$, $(BTU/hrft^2\,^\circ F)$
V_v	Specific volume of vapor (m^3/kg), (ft^3/lb)

W_a	Flowrate of air (kg/s), (lb/s)
W_c	Rate of condensation per row (kg/s), (lb/sec)
W_v	Flowrate of cooling fluid (kg/s), (lb/s)
W_v	Flowrate of vapor (kg/s), (lb/s)
W_s	Condenser steam flow (Kg/hr), (Lb/hr)

Subscripts

N	Subscript indicating nominal tube diameter
v	Subscript indicating vapor
1	Subscript indicating conditions at coolant inlet
2	Subscript indicating conditions at coolant outlet

Greek Symbols

ε_v	Volume ratio of air to mixture
ε_w	Weight ratio of air to mixture
ε_{Wo}	Weight ratio of air to mixture at outlet of air cooler
μ_a	Absolute viscosity of air (n/m s)
μ_m	Absolute viscosity of mixture (n/m s)
μ_v	Absolute viscosity of vapor (n/m s)
ξ	Term defined by Equation (III-)
ψ	Coefficient of mass transfer $(kg/m^2 s)/P_a$

Dimensionless Groups

Nu_D	Nusselt number defined by Equation (III-6)
Re_m	Reynolds number of vapor gas mixture

HEAT AND MASS TRANSFER EQUATIONS

A significant decrease in condensing heat transfer coefficient (h_c) can result from the presence of very small amounts of non-condensable gases. The motion of the vapor with entrained gases moving towards a heat transfer surface, results in an accumulation of non-condensable gases. A vapor partial pressure gradient is required for the vapor to be able to diffuse through the gas blanket. This results in a liquid-vapor interface temperature below the temperature of the main vapor gas mixture. The heat and mass transfer equations developed by CHISHOLM (6) were applied in the VERTCON-2 condenser design program in predicting the condensation of steam in the presence of non-condensable gases. Diffusion coefficients obtained from experiments were used in this analysis by CHISHOLM (6).

The motion of the vapor towards the condenser surface is described by:

$$G_c = \psi(P_v - P_s) \qquad \text{(III-1)}$$

The rate of heat transfer per unit area is given by:

$$q = G_c \; (H_v - H_c) \qquad \text{(III-2)}$$

and, if its assumed that only laten heat is removed, then:

$$q = G_c h_{fg} \qquad \text{(III-3)}$$

From Clapeyron's approximate equations assuming that $P_v - P_s$ is small, then:

$$P_v - P_s = (T_v - T_s) \frac{1}{(T_v) V_v} \qquad \text{(III-4)}$$

By combining equations III-1, III-2, and III-4 results in the governing heat transfer equation within the vapor-gas mixture shown below:

$$q = \frac{\psi h_{fg}{}^2}{(T_v)V_v}(T_v - T_s) \qquad \text{(III-5)}$$

The coefficient of mass transfer is evaluated from:

$$Nu_D = aRe_m{}^{0.5}\xi^{-1/3}\varepsilon_v{}^{-0.6} \qquad \text{(III-6)}$$

where:

$$Nu_D = \frac{\psi D_o}{D_p} \qquad \text{(III-7)}$$

$$Re_m = \frac{W_v + W_a}{A} \quad \frac{D_o}{\mu_m} \qquad \text{(III-8)}$$

$$\xi = \frac{P_v - P_s}{P_m} \qquad \text{(III-9)}$$

For $Re_m > 350$

a = 0.52 for first tube row
= 0.67 for second tube row, and
= 0.82 for third and later tube rows

For $Re_m < 350$

a = 0.52 for all rows

The absolute viscosity of the mixture is determined from:

$$\mu_m = \left[(1-\varepsilon_v)\mu_v + 1.6.\varepsilon_v\mu_a\right]/(1+0.61\varepsilon_v) \qquad \text{(III-10)}$$

The coefficient of steam in air is obtained from:

$$D_p = \frac{2.93 \times 10^{-9}}{P_m} \left[\frac{T_v}{273}\right]^{0.8} \qquad \text{(III-11)}$$

(kg/m s)/Pa

The ratio of air volume to total mixture volume is obtained from:

$$\varepsilon_v = P_a/P_m \qquad \text{(III-12)}$$

and the partial pressures are evaluated from the equation:

$$\frac{P_m}{P_v} = 1 + 0.622 \, \frac{\varepsilon_W}{1-\varepsilon_W} \qquad \text{(III-13)}$$

Combining equations III-5, III-6, III-7, and III-9 yields:

$$q = B(T_v - T_s)^{2/3} \qquad \text{(III-14a.)}$$

where:

$$B = \frac{D_p}{D_o} \, a \, \frac{Re_m^{0.5}}{\varepsilon_v^{0.6}} \, P_m^{1/3} \left[\frac{1}{T_v V_v}\right]^{2/3} h_{fg}^{5/3} \qquad \text{III-14b.)}$$

The heat transfer coefficient between the condensate and the cooling water is evaluated from:

$$q = U(T_s - T_f) \qquad \text{(III-15)}$$

where U is the overall heat transfer coefficient defined in equation (II- 6).

Combining equations III-14 and III-15 yields:

$$\left[\frac{q}{B}\right]^{3/2} + \left[\frac{q}{U}\right] = T_v - T_f \qquad \text{(III-16)}$$

or:

$$q = (T_v - T_f) \Big/ \left[\frac{q^{\frac{1}{2}}}{B^{3/2}} + \frac{1}{U}\right] \qquad \text{(III-17)}$$

For given values of B, and U and temperatures T_v and T_f this equation can be solved for q. Equation (III-17) will converge for any positive initial value of q. For the first tube row and first iteration assume q is zero and the coolant temperature is the inlet value, and for the next row and first iteration assume q and mean temperature of the coolant T_{LM} from the previous row.

In evaluating U the condensate properties are first evaluated at the vapor temperature. After obtaining q on this basis, the overall heat-transfer coefficient between vapor and coolant is calculated from:

$$U_o = q/(T_v - T_L) \qquad \text{(III-18)}$$

Using the overall heat transfer coefficient and log mean temperature difference for the coolant, Equation (III-18) can be written:

$$q = U_o \frac{(T_{f2} - T_{f1})}{\ln\left[\dfrac{T_v - T_{f1}}{T_v - T_{f2}}\right]} \qquad \text{(III-19)}$$

substituting: $Q = W_f C_f (T_{f2} - T_{f1})$ (III-20)

results in:

$$\frac{T_v - T_{f1}}{T_v - T_{f2}} = \exp(-U_o \pi D_N LN/W_f C_{pf} \qquad \text{(III-21)}$$

and the mean temperature of the coolant is evaluated from:

$$T_{LM} = T_v - (T_{f2} - T_{f1}) W_f C_f / U_o \pi D_N LN \qquad \text{(III-22)}$$

For successive iterations T_{LM} is used as the cooling fluid temperature (T_f) in Equation (III-17). At this point

the outer tube wall temperature and surface temperature of the condensate can be calculated, so that a more accurate value of U can be determined. This procedure is repeated with more accurate values of U, B, and T_{LM} until successive values of q are within specified tolerances.

Then, the total condensation for a tube row is calculated from:

$$W_c = q \pi D_N L N / h_{fg} \qquad \text{(III-23)}$$

Standard design methods use either the BEAMA (17) code in the UK or the HEI (18) code in the US. Both codes recommend that the air ejector should be sized to reduce the vapor gas mixture leaving the condenser to 7.5 °F below the saturation temperature corresponding to condenser pressure. As an example a vacuum of 29 in.Hg corresponds to a ratio of non-condensable gas to mixture flowrate (ε_{wo}) to the ejector of:

$$\varepsilon_{wo} = \frac{W_A}{W_V + W_A} = 0.31$$

Figure- 12 can be used to determine the flowrate of non-condensable gases (W_A) to be used in the sizing of the air ejector. In this study the BEAMA code was correlated using multiple linear regression (15), resulting in:

$$W_A = 10.0 + .1823\text{x}10^{-3} W_s - .975\text{x}10^{-10} W_s^{\,2} \qquad \text{(III-24)}$$

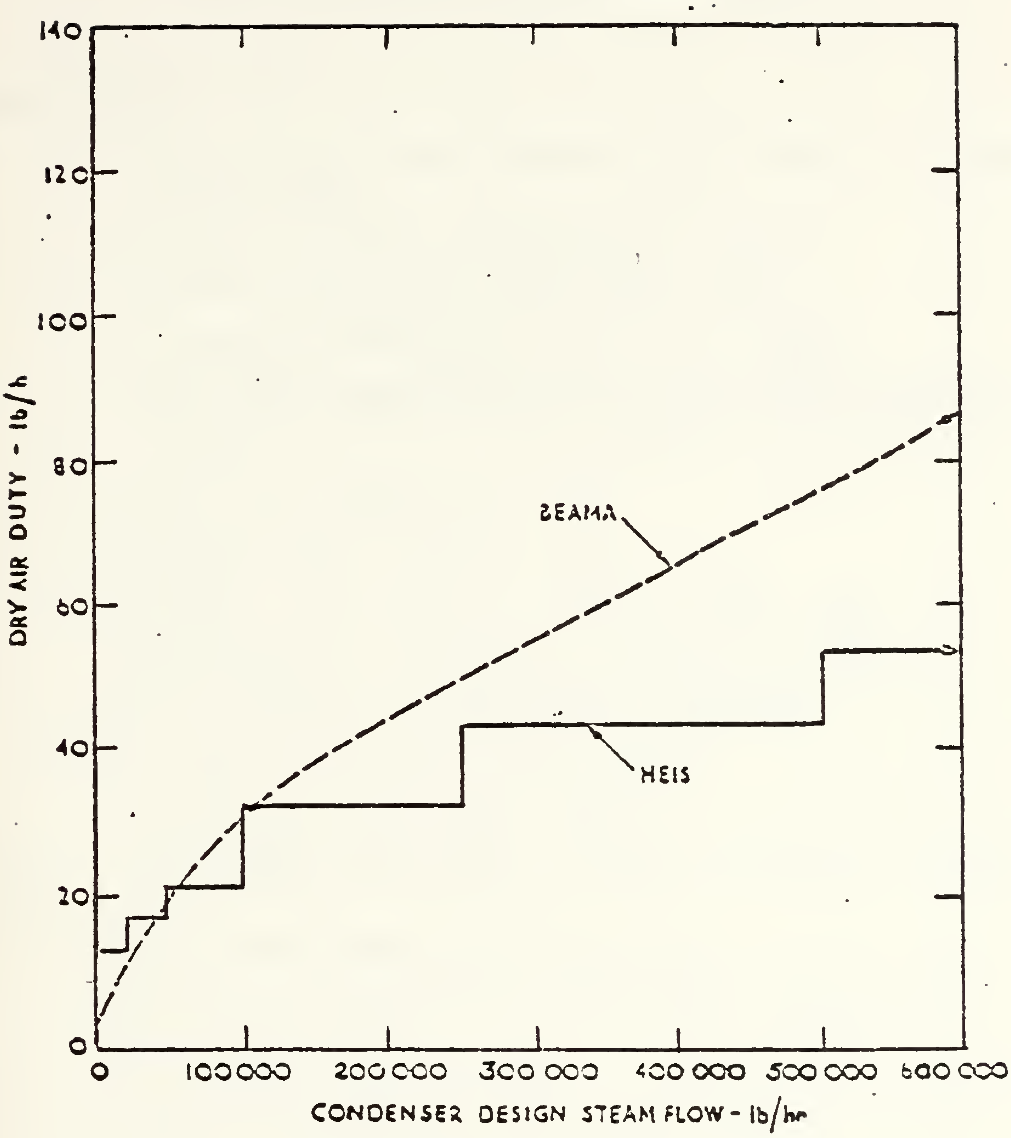


FIGURE 12

Recommended Air Extraction Capacity

IV. PRESSURE DROP CALCULATIONS

Nomenclature

A_f	Minimum flow area between outer tubes of a tube bank (ft^2)
A	Constant defined by Equation (IV-1)
B	Constant defined by Equation (IV-1)
C	Conversion constant (lb/ft^2 to in.Hg)
D_e	Equivalent diameter of flow area (the ratio of flow area to wetted perimeter multplied by four) (ft)
D_i	Inside tube diameter (ft)
D_o	Outside tube diameter (ft)
e	Internal tube helical ridge height (ft)
E	Dimensionless relative roughness = ε/D_i
g	Gravitational acceleration (ft/sec^2)
L	Perimeter of tube bundle (ft)
L_t	Condenser tube length (ft)
L_1, L_2 L_3, L_4	Flow length along tube bundle perimeter defined in Figure-13 (ft)
N_e	Equivalent number of tube rows
p	Pitch of internal tube helical ridging (ft)
P	Tube pitch (ft)
P_t	Total pressure (in.Hg)
P_s	Specified condenser pressure (static)(in.Hg)
ΔP_b	Pressure loss through tube bank (in.Hg)
ΔP_{bm}	Mean penetration pressure loss (in.Hg)
ΔP_{dm}	Mean total pressure loss around tube bundle (in.Hg)
ΔP_{en}	Entrance pressure loss (in.Hg)
$\Delta P_2, \Delta P_3$ ΔP_4	Pressure loss along flow length L_2, L_3, L_4 respectively (in.Hg)
ΔP_{WI}	Waterbox inlet loss (ft.of water)

ΔP_{WO}	Waterbox outlet loss (ft. of water)
ΔP_{TE}	Tube end loss (combined inlet and outlet), (ft. of water)
V_f	Flow velocity of cooling water (ft/sec)
V_s	Flow velocity of steam (ft/sec)

Greek Symbols

ε	Tube surface roughness (ft)
ρ	Steam density (lb/ft^3)
μ	Steam viscosity (lb/hr-ft)

Dimensionless Groups

f	Friction factor
f^1	Friction factor defined by Equation (IV-19)
f_{en}	Friction factor for internally enhanced tubes
Re	Reynolds number

IV.A TUBESIDE PRESSURE DROP

The total tubeside pressure drop is calculated from the summation of the following pressure drop components:

1. pressure loss in condenser tubes
2. tube end pressure loss (inlet and outlet)
3. waterbox pressure loss (inlet and outlet)

The pressure loss for smooth internal vertical tubes is evaluated using a friction factor relationship developed by CHURCHILL (19) as shown in Equation (IV-1) which is applicable for any flow regime as a function of Reynolds number and relative roughness of the tube.

$$f = 8.0\left[(8/Re)^{12} + 1.0/(A+B)^{3/2}\right]^{0.0833} \qquad \text{(IV-1)}$$

where:

$A = \left[2.457 \ln(1.0/(7/Pe)^{08} + 0.27E)\right]^{16}$

$B = (37{,}530/Re)^{16}$

$=$ surface roughness

$= 5\text{x}10^{-6}$ft. recommended value (20) for smooth tubing

$E = \epsilon/D_i$ dimensionless relative roughness where D_i is the inside tube diameter (ft.)

Pressure loss for multiple-helix internal ridged tubes is calculated using friction factor (f) correlations evaluated from Equation (II-7) resulting from manufacturer's test data obtained from Reference-13. The friction factor data was correlated with f_{en} as a function of Reynolds number, and the m & r operands with the following result:

$$f_{en} = 8\left[-\frac{1}{2.46\ \ln\left[r+(7/Re)^m\right]}\right]^2 \qquad \text{(IV-2)}$$

Tube properties and friction factor characteristics for multiple-helix internal ridged tubes can be found in Appendix-B.

Tube end and waterbox pressure losses for single and double pass condensers are calculated using pressure loss correlations evaluated from multiple non-linear regression using HEI standards (18). Pressure drop data from the HEI code can be found in Appendix-B. The resulting correlations are shown here:

Single Pass Condensers

Waterbox Inlet Loss:

$$P_{WI} = .01422V_f^{2.04} \qquad \text{(IV-3)}$$

Waterbox outlet Loss:

$$P_{WO} = .00237V_f^{2.33} \qquad \text{(IV-4)}$$

Tube End Loss (Combined Inlet & Outlet):

$$P_{TE} = .0178V_f^{2} \qquad \text{(IV-5)}$$

Double Pass Condensers

Waterbox Inlet Loss:

$$P_{WI} = .01422V_f^{2.04} \qquad \text{(IV-6)}$$

Waterbox Outlet Loss:

$$P_{WO} = .00635V_f^{2.1} \qquad \text{(IV-7)}$$

Tube End Loss (Combined Inlet & Outlet):

$$P_{TE} = .0356V_f^{2} \qquad \text{(IV-8)}$$

IV.B SHELLSIDE PRESSURE DROP

Vapor condensing temperature is a function of vapor pressure, which follows a specific saturation temperature-pressure relationship. With the pressure drops encountered in the steam flow from the steam inlet to the air cooler section through the tube bundle, there is a resulting reduction in condensing temperature in the direction of flow. This temperature reduction will affect the rate of heat transfer in different sections of the condenser.

In Reference-6 HARRINGTON has noted that the condenser performance standards of the HEI (18) are based on static pressure, but that the condenser tube more closely senses a temperature related to the total or stagnation pressure. The method for evaluating the steam pressure distribution established by HARRINGTON along with the assumption that condenser performance is based on stagnation pressure was used in this study. Also, this can be considered as a one-dimensional design method where the vapor flow distribution is represented by a single-flow path. The flow path is considered as cross flow, radially inward from the tube bundle perimeter to the air cooler section of the condenser.

The pressure loss incurred by the steam flow can be considered as two components:

(1) steam distribution loss for steam flow from the inlet to around the tube bundle

(2) steam penetration loss for steam flow through the tube bundle from the tube bundle perimeter to the air cooler section of the condenser.

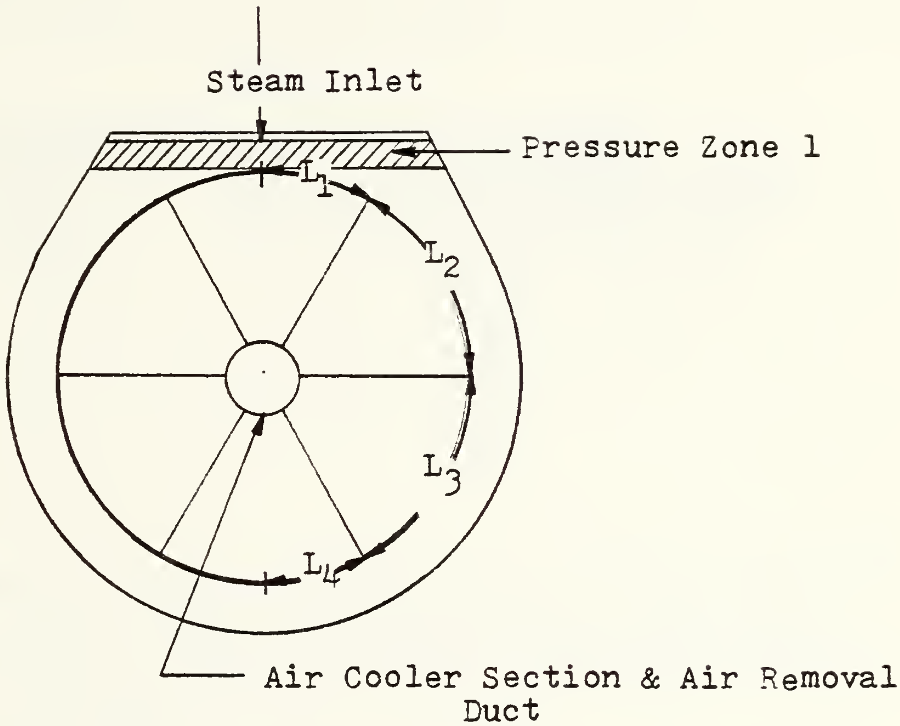


FIGURE 13

Steam Distribution Pressure Zones

Steam Distribution Loss

Refer to Figure-13for the following discussion. The total or stagnation pressure is calculated for pressure zone-1 and is evaluated from:

$$P_t = P_s + \frac{C \rho V_s^2}{2g} \qquad \text{(IV-9)}$$

The steam velocity required in Equation (IV-9) and for the entrance loss calculation is assumed to correspond to the recommended maximum main steam lane entrance velocities as shown in Table- 1 .

Table- 1

Recommended Steam Design Velocities

Condenser Design Pressure (in.Hg)	Recommended Maximum Main Steam Lane Entrance Velocity, fps
1	500
2	400
3	300
4	250
5	200

The entrance flow area will have to be sized accordingly with respect to this entrance flow velocity. All pressure losses around the tube bundle perimeter are based on the total pressure in zone-1.

The next step is to calculate the entrance pressure loss (L_1) from pressure zone-1 to the main steam inlet lane as follows:

$$\Delta P_{en} = 0.05 \frac{C \rho V_s^2}{2g} \qquad \text{(IV-10)}$$

The next step is to calculate pressure loss due to friction along L_2. Calculate the steam velocity from the

flow area in this zone. It assumed the net steam flow is reduced proportionately along the flow length L_1, L_2, L_3, and L_4.

Reynolds number can then be evaluated from:

$$Re = \frac{\rho D_e V_s}{\mu} \qquad \text{(IV-11)}$$

where D_e is the equivalent diameter of flow area, which is the ratio of flow area to wetted perimeter multiplied by four.

Finally the pressure loss due to friction along L_2 is evaluated using the Tanning equation as follows:

$$P_{L2} = \frac{4C\rho f L_2 V_s^{\,2}}{D_e 2g} \qquad \text{(IV-12)}$$

The friction factor f is obtained from Figure-14 , or by evaluating the following correlation obtained by using multiple non-linear regression analysis (15), for an assumed roughness value.

$$f = .0436\ Re^{-.186} \qquad \text{(IV-13)}$$

This procedure is repeated to determine the frictional pressure loss for L_3 and L_4. This calculation scheme can be conducted with a greater number of smaller length segments to obtain a more accurate pressure distribution around the tube bundle.

The mean distribution pressure loss around the tube bundle can then be calculated as follows:

$$P_{dn} = \frac{L_1\ P_{en} + L_2\ P_2 + L_3\ P_3 + L_4\ P_4}{L_1 + L_2 + L_3 + L_4} \qquad \text{(IV-14)}$$

This mean total pressure identifies the condensing conditions at the tube bundle perimeter.

Steam Penetration Loss in Tube Bank

Calculation of Pressure loss through a tube bank is accomplished by replacing the actual tube bank with an equivalent rectangular bank that has the same perimeter and number of tubes. The equivalent number of rows in the equivalent tube bank depth is given by:

$$N_e = \frac{NP}{L} \qquad \text{(IV-15)}$$

Next calculate the velocity through the minimum flow area between the tubes. The flow area is obtained from:

$$A_f = \frac{L}{P}(P - D_o)L_T \qquad \text{(IV-16)}$$

The steam velocity entering the tube bank can be calculated using this value for flow area (A). Then Reynolds number is evaluated from:

$$Re = \frac{\rho D_o V_s}{\mu} \qquad \text{(IV-17)}$$

The terms (ρ & μ) in Equation (IV-17) are calculated at the temperature corresponding to the mean pressure at the tube bank entrance.

The tube bank pressure loss equation, which is modified for uniformly decreasing mass flow (21)

$$P_b = 4f^1\left(0.5 + \frac{Ne}{3}\right)\frac{C\, V_s^2}{2g} \qquad \text{(IV-18)}$$

The friction factor f^1 is obtained from Figure-15, or by evaluating the following correlation obtained by applying multiple non-linear regression analysis (15).

$$f^{1} = 65.5Re^{-.702}(P/D_o)^{-4.257} \quad (10 \leq Re \leq 1000) \qquad \text{(IV-19a.)}$$

$$f^{1} = .591Re^{-.136}(P/D_o)^{-1.137} \quad (1000 < Re \leq 1.0x10^{5}) \qquad \text{(IV-19b)}$$

HARRINGTON (6) has suggested that the following equation be used to determine the mean tube bank pressure loss:

$$P_{bm} = 4f^{1}(0.5 + \frac{N_e}{4})\frac{C\rho V_s^{2}}{2g} \qquad \text{(IV-20)}$$

The mean condensing pressure in the tube bank is then evaluated from:

$$P_{bm} = P_t - \Delta P_{dm} - \Delta P_{bm} \qquad \text{(IV-21)}$$

All of the heat transfer calculations are based on the mean condensing pressure (P_{bm}) in the tube bank.

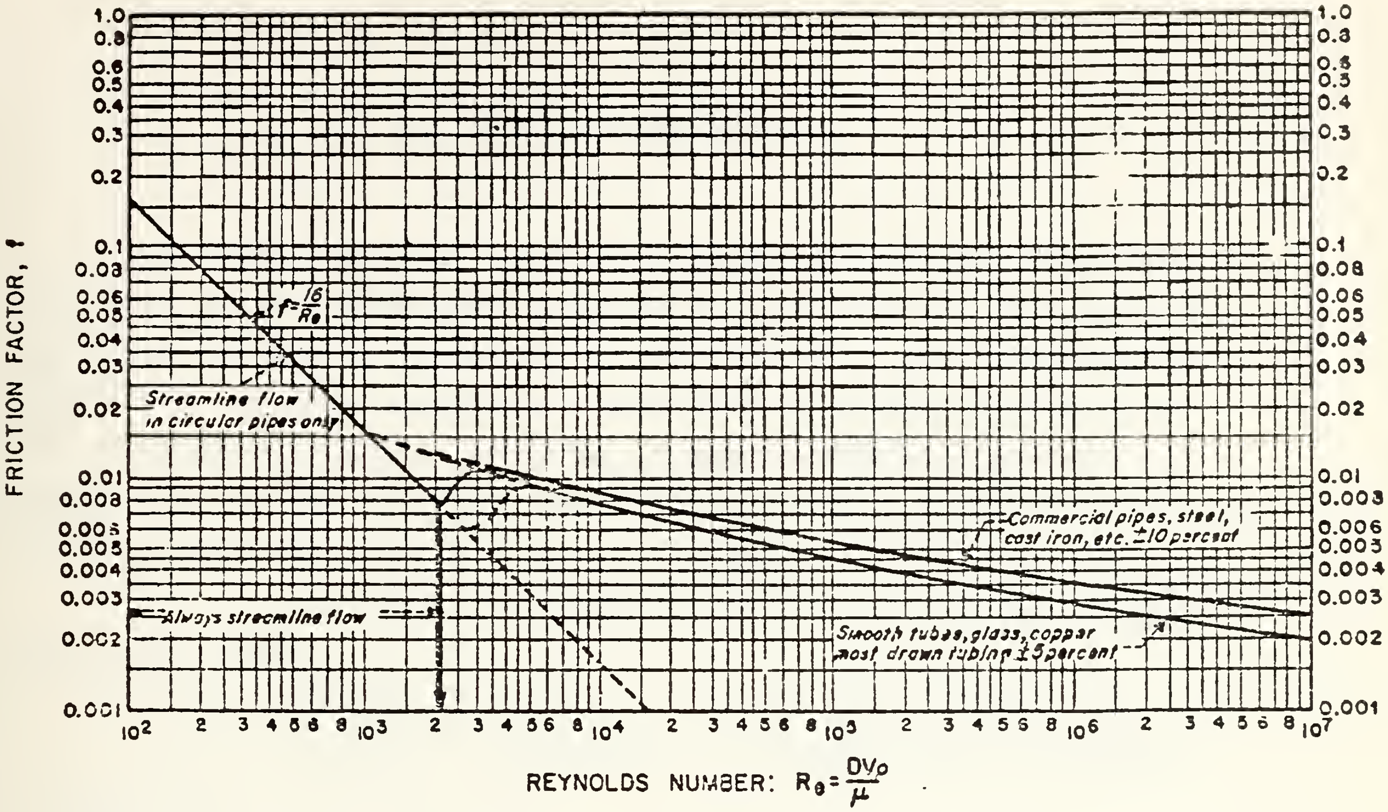


FIGURE 14

Friction Factor for Steam Distribution Losses

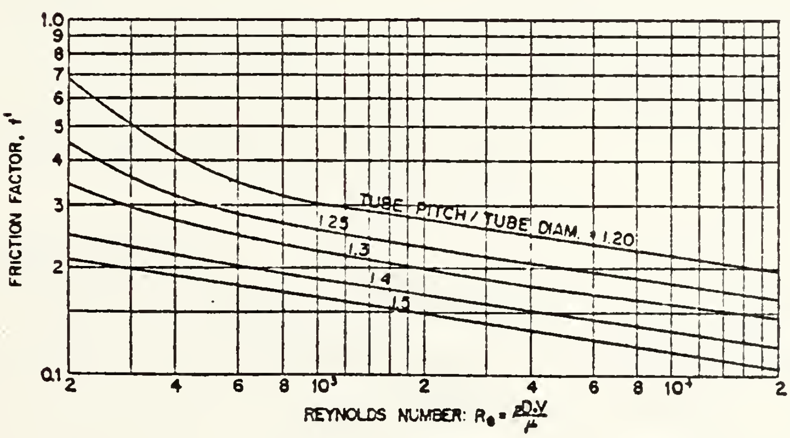


FIGURE 15

Friction Factor for Steam Penetration (Tube Bank) Losses

V. COMPUTER MODELING

The basic procedure for heat enchanger sizing that was employed in the VERTCON-1 program (3) is used in the Vertical Condenser Sizing Program No. 2 (VERTCON-2). In addition the VERTCON-2 program considers condensation in the presence of non-condensable gases, and the program optimizes the condenser design for minimum weight and volume. Other features of the VERTCON-2 program are as follows:

1. The option of smooth internal tubes or internally enhanced tubes.
2. Design of single or double pass condensers.
3. Option of submarine, or surface ship condenser design.
4. Two basic configurations are available for double pass condensers:
   (a) Conventional return header design
   (b) "U-Tube" construction
5. Diagnostic messages used to warn when condenser drainage plate spacing is beyond specifications based on heat transfer performance and mechanical design.
6. Computation of recommended values for condenser drainage plate spacing and maximum main steam lane entrance velocity when they are not specified.

Program Discription

The program VERTCON-2 is described here, and diagrammatically in the simplified program flow chart (Figure-16). The program is based on the FORTRAN-77 standard, and is listed in Appendix-A. Also included in Appendix-A are sample program input and output files.

The program input consists of input data files, and

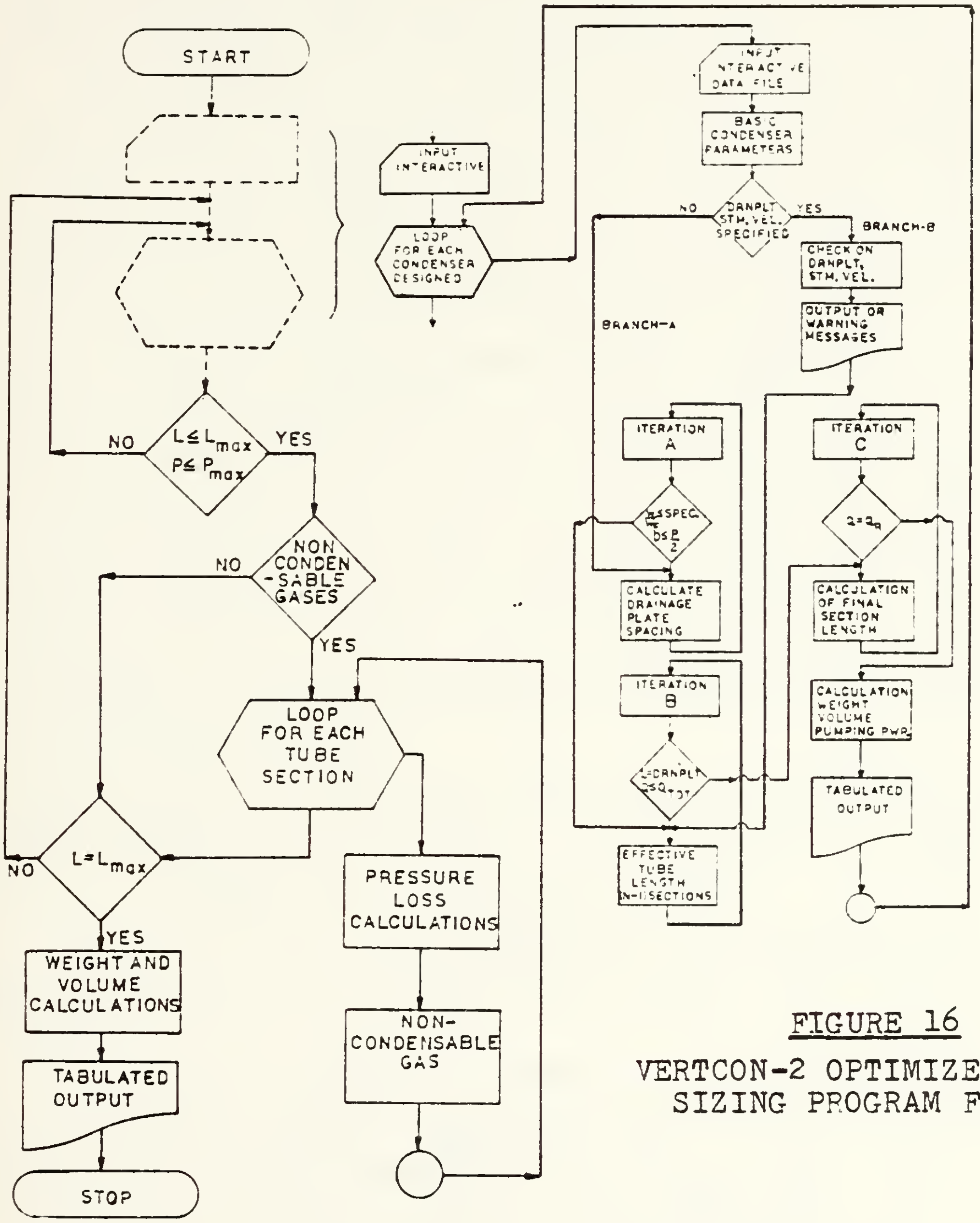


FIGURE 16

VERTCON-2 OPTIMIZED CONDENSER SIZING PROGRAM FLOWCHART

interactive input entered at the terminal. The input data file consists of tube geometry information for the various candidate tubes to be considered, and specification of condenser geometry, materials and operating conditions. The design features are selected interactively at the terminal. See Appendix-A for further details on data input.

The program is set up for sizing just one condenser or a series of condensers. After inputing data the program prompts at the terminal for selection of design features. If condenser drainage plate spacing and maximum main steam lane entrance velocity are not specified the program calculates recommended values (Branch-A and Iteration-A, Figure-16). The determination of these values are based on heat transfer performance and mechanical design to provide proper tube support. If these values are specified then the program checks if condenser drainage plate spacing is beyond recommended values (Branch-B, Figure-16). If the values specified exceed those dictated by proper design, then diagnostic warning messages are printed at the terminal and in the program output.

After input of interactive data at the terminal, the program conducts the selection of flow velocity and tube geometry to optimize the design as follows:

1. For the first candidate tube geometry the condenser is sized at the specified minimum flow velocity ($V = V_{min}$).

2.(a) If condenser length (L) is less than specified maximum condenser length (L_{max}) and pumping power is less than specified maximum pumping power (P_{max}) then Step 1 is repeated for a larger value of flow velocity.

2.(b) This process is repeated until $L = L_{max}$ or $P = P_{max}$.

3. If at V_{min}:L L_{max} then this candidate tube is rejected because of insufficient enhancement.

4. Steps 1 through 3 are repeated for all candidate tubes.

The condenser sizing in Steps 1 through 4 is evaluated as follows:

1. In iteration B and C (Figure-16) the total effective tube length for the condenser is calculated by adding N tube sections of length equal to the drainage plate spacing until a proper heat balance is obtained. Iteration B adds up N-1 of these sections of equal length. Iteration C then calculates the final section length, which may have a length less than the specified drainage plate spacing.

2. After the total effective tube length is determined the condenser geometry, weight, volume, and system power are calculated for each condenser.

It should be noted that the influence of non-condensable gases has not been considered yet, and a multiplicative factor (F_{NC}) is simply used to account for non-condensable gas effects.

At this point the minimum condenser sizing for each tube configuration are compared to select the condenser configuration with overall minimum weight and volume.

If the option is selected, the program will now consider the effects of non-condensable gases. Using the tube bundle and shell geometry of the selected condenser configuration, the mean pressure around the tube bundle is determined. With the conditions identified at the tube bundle perimeter the program steps through the condenser tube bundle row by row to calculate the effective tube length at local conditions including non-condensable gas effects.

If the condenser length calculated from the consideration of non-condensable gases differs from the original calculated value, then a new value for the multiplicative factor F_{NC} is selected and the complete process including tube and flow velocity selection is repeated until these length values

are equal. By conducting the calculations in this manner the non-condensable gas effects will be evaluated a minimum number of times, thus significantly reducing computer solution time. There is a limit to the number (N_I) of these iterations that will be completed by the program. If after N_I iterations there has not been proper convergence, then warning messages are displayed on the terminal screen and printed in the program output. This indicates that one of the specified condenser parameters is out of limits for the particular condenser design. Non-convergence would most likely be due to factors that influence steam-side heat transfer performance. One culprit could possibly be too small a value for tube pitch-to-diameter ratio, that would cause excessive steam side pressure drop.

The tabulated program output displays for each condenser the following information:

(1) All input data and design features selected.

(2) If applicable, diagnostic warning messages.

(3) Detailed weight breakdown of condenser components.

(4) Total condenser volume, center of gravity, and total weight in dry and wet conditions, and system pumping power.

APPENDIX A

DATA INPUT

See Table-3 for a sample input data file

INPUT DATA FILE

1. Assigned Logical Operator - 20 to specify input device. See lines 47-52 of progress listing for further details.
2. Free Format
3. Variable names starting with I, J, K, L, M, or N are integer variables and others are real variables, except for specific cases as noted.

LINE 1

Enter: SUB or SUR

SUB - Submarine Condenser Design

SUR - Surface Ship Condenser Design

LINE 2

Enter: NUMBT,LMAX, PMAX, VMIN, VMAX

NUMBT- Number of Tube configurations to be tested

LMAX - Maximum condenser length (ft)

PMAX - Maximum pumping power (hp)

VMIN - Minimum coolant flow velocity (ft/sec)

VMAX - Maximum coolant flow velocity (ft/sec)

Note: LMAX is a real variable

LINE 3

Enter: TCI, TSAT, QSTM, STMLD, PSAT PD

TCI - Inlet Coolant Temperature (F)

TSAT - Steam Inlet Saturation Temperature (F)

QSTM - Quality of Inlet Steam to Condenser

STMLD - Steam Condenser (lb/hr)

TABLE-2: SAMPLE INPUT DATA FILE
..4 TUBE TEST CASE..

```
SUB
4,24.34,275.,8.,15.
66.1,143.89,.85,257000.,6.5,1.35
2.,6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
1,.555,.625,.01,.245,0.,0.
48.,0.,0.,9.6,.00033
2,.555,.625,.01,.245,.0204,.191
48.,.59,.00197,9.6,.00033
2,.555,.625,.01,.245,.0125,.4750
48.,.762,0.0,9.6,.00033
2,.555,.625,.01,.245,.024,.0949
48.,.58,.0075,9.6,.00033
```

PSAT - Condenser Operating Pressure (in.Hg) Absolute

PD - Tube pitch-to-diameter ratio

LINE 4

Enter: TIS, TOTS, TSHL, TTSP, THW, THDR, ITSD

TIS - Thickness of internal tube sheet (in.)

TOTS - Thickness Outer Tube Sheet (in)

TSHL - Thickness of condenser shell (in.)

TTSP - Thickness tube support plate (in.)

THW - Thickness of hot well plate (in.)

THDR - Thickness of Headers (in.)

ITSD - Internal tube sheet density (lb/Ft^3)(real variable)

Note: ITSD is a real variable.

LINE 5

Enter: OTSD, SHLD, TSPD, HWD, HDRD, TBD, ETUBE, TCOVER, COVERD

OTSD - Outer tube sheet density (lb/ft^3)

SHLD - Shell density (lb/ft^3)

TSPD - Tube support plate density (lb/ft^3)

HWD - Hotwell plate density (lb/ft^3)

HDRD - Header density (lb/ft^3)

TBD - Tube bundle density (lb/ft^3)

ETUBE - Modulus of elasticity of tube material (lb/in^2)

TCOVER - Thickness of U-tube header cover (in.)

COVERD - Density of U-tube header cover (in.)

LINE 6

Enter: NTYPE, DI, DW, AE, AEPE, EI, PI

NTYPE - Tube type: a. for smooth internal tube
b. for doubly enhanced tubes

DI - Internal tube diameter (in.)

DW - Tube wall diameter (in.)

AE - External flute amplitude (in.)

AEPE - Amplitude-to-pitch ratio of external ridging

EI - Internal helix ridge height (in.)

PI - Pitch of Internal Helix (in.)

Note: For smooth internal tube (NTYPE-1) enter ØØ for EI, PI. Values of EI and PI are obtained from tube data in Appendix-B.

LINE 7

Enter: FLUTE, MI, RI, KW, RSCALE

FLUTE - Number of flutes per tube

MI - Operand for friction factor, helical internal ridging (real variable)

RI - Operand for Friction Factor, Helical internal ridging

KW - Conductivity of tube wall (BTU/hr-ft- F)

RSCALE - Fouling resistance (hr-ft- F/BTU)

Note: MI is a real variable.

To consider more tube configurations, repeat lines 6 and 7 for each additional tube to be included in the analysis.

INTERACTIVE DATA INPUT

Enter data at the terminal as follows:

When the program commences, the following message appears on the terminal screen:

```
** WELCOME TO PROGRAM: VERTCON, VERSION-2 **
         ENTER PROGRAM RUN NUMBER
```

Enter Run number, which can be an integer from 0-to-999.

Next at the terminal appears:

```
ENTER THE NUMBER OF CONDENSERS TO BE SIZED
```

Enter the number of condensers to be sized, which is also an integer from 1-to-999.

<u>Next at the terminal appears</u>:

```
ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CONDENSER
```

Enter the proper integer number 1 or 2 for single or double pass condenser.

<u>Next at the terminal appears</u>:

```
TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR DOUBLE
PASS CONDENSERS:
(1)  CONVENTIONAL RETURN HEADER DESIGN
(2)  "U-TUBE" CONSTRUCTION
ENTER 1 OR 2 FOR CONFIGURATION SELECTION
```

Enter the proper integer number 1 or 2 for configuration selection.

<u>Next at the terminal appears</u>:

```
ENTER:  (1)  COOLANT FLOW RATE (GPM)
```

Enter as real number with units of gallons per minute.

<u>Next at the terminal appears</u>:

```
DO YOU WANT TO INCLUDE IN THIS ANALYSIS THE EFFECTS
OF CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE
GASES    YES OR NO
```

Enter YES or Y, NO or N for proper choice.

<ins>Next at the terminal appears</ins>:

DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE
ENTRANCE VELOCITY (FT/SEC)?

YES OR NO

Enter YES or Y, NO or N for proper choice.

If yes is entered, <ins>Next at the terminal appears</ins>:

ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)

Enter this as a real number.

If no is entered, <ins>Next at the terminal appears</ins>:

PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN
STEAM LANE ENTRANCE VELOCITY USING CONDENSER
OPERATING PRESSURE AS SELECTION CRITERIA.

<ins>Next at the terminal appears</ins>:

DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE
PLATE SPACING (FT.)?

YES OR NO

Enter YES or Y, NO or N for proper choice.

If yes is entered, <ins>Next at the terminal appears</ins>:

ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)

Enter this as a real number.

If no was entered, Next at the terminal appears:

PROGRAM WILL SELECT RECOMMENDED CONDENSATE DRAINAGE PLATE SPACING

Note: At this point, entry of Interactive Data is completed for the first condenser. The program now completes the sizing for the first condenser and will prompt the terminal for further interactive data for any additional condensers.

```
. . . TABLE-3: SAMPLE OUTPUT DATA FILE . . .

** PROGRAM: VERTCON,VERSION-2 **


PROGRAM RUN NUMBER(  1)


**** VARIABLE LIST ****

AE= EXTERNAL FLUTE AMPLITUDE (IN.)   .  MI= OPERAND FOR FRICTION FACTOR;
AEPE= AMPLITUDE-TO-PITCH RATIO OF    .      HELICAL INTERNAL RIDGING
    EXTERNAL RIDGING                 .  NTYPE= TUBE TYPE: 1-FOR SMOOTH IN-
ATB= AREA OF TUBE BUNDLE (FT**2)     .      TERNAL TUBE; AND 2-FOR DOUBLY
ATS= AREA OF TUBE SHEET (FT**2)      .      ENHANCED TUBES
DI= INTERNAL TUBE DIAMETER (IN.)     .  PI= PITCH OF INTERNAL HELIX(IN.)
DPTHDR= HEADER DEPTH (FT.)           .  PSAT= CONDENSER OPERATING PRESSURE
DRYCG= HEIGHT OF CENTER OF GRAVITY   .      (IN.HG)ABSOLUTE
    ABOVE CONDENSER BOTTOM AT DRY    .  QSTM= QUALITY OF INLET STEAM TO
    WEIGHT (FT.)                     .      CONDENSER
DTB= DIAMETER OF TUBE BUNDLE (FT.)   .  RI= OPERAND FOR FRICTION FACTOR,
DTS= DIAMETER OF TUBE SHEET(FT.)     .      HELICAL INTERNAL RIDGING
DW= TUBE WALL DIAMETER (IN.)         .  RSCALE= FOULING RESISTANCE
EI= OPERAND FOR FRICTION FACTOR      .      (HR-FT**2-DEG.F/BTU)
    HELICAL INTERNAL RIDGING         .  STMLD= STEAM LOAD (LBM/HR)
FE= OPERAND FOR CONDENSING HEAT      .  TSAT= SATURATION TEMP. (DEG.F)
    TRANSFER COEFFICIENT             .  VTB= VOLUME OF TUBE BUNDLE (FT**3)
KW= CONDUCTIVITY OF TUBE WALL        .  WEXP= EXPANSION JOINT WEIGHT (LB.)
```

(BTU/HR-FT-DEG.F)
LANE= STEAM LANE BREADTH (FT.)
LHW= LENGTH OF HOTWELL (FT.)
LTOT= TOTAL TUBE LENGTH (FT.)
TCI= INLET COOLANT TEMP. (DEG.F)

WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
WETCG= HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM AT WET WEIGHT (FT.)

```
                    *****************************
                       CONDENSER NUMBER(  1)
                    *****************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HEADER DESIGN.
. . DOUBLY ENHANCED TUBES . .
. . TUBE NUMBER(  4) SELECTED . .
                         **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.22 (FT/SEC) . .
DI = 0.555     DW = 0.625     AE = 0.010     AEPE = 0.2450    NTYPE =  2
EI = 0.0240    PI = 0.0949    FE =  4.54     MI = 0.580       RI = 0.0075
KW =   9.60    TCI =  66.10   PSAT = 6.50    QSTM = 0.85      TSAT =143.89
STMLD =  257000.0             RSCALE = 0.00033

              **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.06 (FT.)

. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2050.0

. . EFFECTIVE TUBE LENGTH:              14.39 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:       6096. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:     41488. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:     994. (BTU/HR-FT**2-DEG.F)

LTOT = 16.24      LANE =   1.24      DTB =   3.68     ATB =  10.66
VTB = 153.39      DTS =   6.16       ATS =  29.81


           **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =    440.1   WMISC = 11432.2    LHW =   5.02     DPTHDR =  3.08
```

	MATERIAL DENSITY(LB/FT**3)	THICKNESS (IN.)	WEIGHT (LB.)
OUTER TUBE SHEET-----	558.0	6.0000	13879.
INNER TUBE SHEET-----	488.0	2.0000	4046.
TUBE SUPPORT PLATE---	488.0	0.5625	2040.
TUBE BUNDLE----------	282.0	*.****	5531.
CONDENSER SHELL------	488.0	0.6250	8059.
HOTWELL--------------	488.0	0.5625	3569.
WATERBOX-------------	558.0	2.6400	8163.

```
. . TOTAL CONDENSER DRY WEIGHT=   57160.9 (LB.)
. . TOTAL CONDENSER WET WEIGHT=   65432.0 (LB.)
                                    29.21 (TON)

. . TOTAL CONDENSER HEIGHT=  24.34 (FT.)
. . OUTER SHELL DIAMETER=     6.27 (FT.)
. . ENCLOSED BOX VOLUME=    1023.6 (FT**3)
    DRYCG =  12.28    WETCG =  11.69

. . CONDENSER FRICTIONAL HEAD LOSS=  35.8 (FT.)
. . TOTAL SYSTEM HEAD LOSS=          88.8 (FT.)
. . TOTAL SYSTEM PUMPING POWER=     246.8 (HP)
```

```
C*********************
C  PROGRAM: VERTCON2
C*********************
C PROGRAM "VERTCON2" IS A PRELIMINARY DESIGN ROUTINE USED TO OPTIMIZE
C VERTICAL TUBE STEAM SHIP/SUBMARINE PROPULSION CONDENSERS TO PRODUCE
C CONDENSER DESIGNS WITH MINIMUM WEIGHT AND VOLUME. THIS PROGRAM CAN
C ALSO  CONSIDER CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES.
C THE PROGRAM TAKES INITIAL INPUT DATA BOTH INTERACTIVELY AND FROM DATA
C FILES. THE PROGRAM OUTPUT CONSISTS OF TABULATED VALUES FOR TOTAL &
C COMPONENT CONDENSER WEIGHTS, VOLUMES, AND PUMPING POWER.
C
C TWO BASIC VERTICAL TUBE CONFIGURATIONS ARE AVAILABLE WITH THIS PROGRAM
C AND THEY ARE: (1) DOUBLY ENHANCED TUBES, OR (2) SMOOTH INTERNAL TUBES
C WITH EXTERNAL ENHANCEMENT ONLY. SEVERAL GEOMETRIC VARIATIONS WITHIN
C THESE TWO BASIC CONFIGURATIONS CAN BE INVESTIGATED WITH "VERTCON1".
C
C ALSO, TWO BASIC CONSTRUCTION CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE-
C PASS" CONDENSERS: (1) CONVENTIONAL RETURN HEADER DESIGN, AND
C                   (2) "U-TUBE" CONSTRUCTION
C
C THE VALUES OF CONDENSATE DRAINAGE PLATE SPACING (FT.), AND MAXIMUM
C MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC) MAY BE SPECIFIED BY THE
C USER INTERACTIVELY WHEN RUNNING THIS PROGRAM. IF THESE VALUES ARE
C NOT SPECIFIED, THEN THE PROGRAM WILL AUTOMATICALLY DETERMINE
C RECOMMENDED VALUES.
C
      DIMENSION DATA(500,5), OUTPUT(600,6),HWAVG(100),HCAVG(100),
     1 UAVG(100),TLT(100),XLTOT(100),XVM(100),TDATA(100,13),EFF(100)
      REAL ITSD,KSAT,KSW,KW,L1,LANE,LHW,LTOT,MI,MU,NT,NU,LMIN,LMAX,
     1 LMIN1,LMAX1,LMIN2,LMAX2,LPLUS,MULT1,MULT2,MULT3,MULT4,MULTP,MF
      INTEGER STM,CONFIG,CTYPE,DRAIN,GAS,OPT
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA NOE/'NO  '/,NO/'N   '/,ITYPE/'SUB '/
      DATA PIE/3.1415927/
```

```
C
C * * * STATEMENT FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * *
C
      NU(TCS)= .122181 - .1481615E-02 * TCS + .5516445E-05 * TCS**2
      RE(VM)=    VM * DI * 300. / NU(TCS)
      PRSW(TCS)= .0014369 * TCS**2 - .3134 * TCS + 21.89
      KSW(TCS)=  -.000003086 * TCS**2 + .001 * TCS + .291
C NU=   KINEMATIC VISCOSITY OF COOLING WATER (FT**2/HR)
C RE=   REYNOLDS NUMBER COOLING WATER
C PRSW= PRANDTL NUMBER COOLING WATER
C KSW=  CONDUCTIVITY OF COOLING WATER (BTU/HR-FT-DEG.F)
C
      FF(ARG2)= -1. / (2.46 * LOG(ARG2**MI + RI))
      ST(ARG3,ARG4)= FF(ARG2) / (5.68 * ARG3**(-.125) *SQRT(PRSW(TCS))
     1 * ARG4**.136 + GAMMA)
C FF= FRICTION FACTOR [SQRT(F/8)]
C ST= STANTON NUMBER
C
      KIN= 20
      KOUT= 21
      KSCR= 6
C KIN,KOUT, AND KSCR ARE OPERATORS THAT SPECIFY INPUT & OUTPUT DEVICES.
C IN THIS CASE KIN & KOUT SPECIFY INPUT & OUTPUT DATA FILES RESPECTIVELY
C AND KSCR SPECIFIES INPUT & OUTPUT AT COMPUTER TERMINAL.
C
C * * * * DATA(INTERACTIVE & FILE)-INPUT/OUTPUT * * * *
C
      WRITE(KSCR,100)
 100  FORMAT(1X,T14,'* * WELCOME TO PROGRAM: VERTCON,VERSION-2 * *'//T
     12,'ENTER PROGRAM RUN NUMBER')
      READ(KSCR,*) NUMBER
      WRITE(KOUT,102) NUMBER
 102  FORMAT(/////T21,'** PROGRAM: VERTCON,VERSION-2 **'////T25,'PROGR
     1AM RUN NUMBER(',I3,')'/////)
C
      WRITE(KOUT,104)
```

```
104   FORMAT(1X,T26,'**** VARIABLE LIST ****'//T2,'AE= EXTERNAL FLUTE A
     1MPLITUDE (IN.)',T38,'.',T41,'MI= OPERAND FOR FRICTION FACTOR;'/T2,
     1'AEPE= AMPLITUDE-TO-PITCH RATIO OF',T38,'.',T45,'HELICAL INTERNAL
     1RIDGING'/T6,'EXTERNAL RIDGING',T38,'.',T41,'NTYPE= TUBE TYPE: 1-FO
     1R SMOOTH IN-'/T2,'ATB= AREA OF TUBE BUNDLE (FT**2)',T38,'.',T45,'T
     1ERNAL TUBE; AND 2-FOR DOUBLY'/T2,'ATS= AREA OF TUBE SHEET (FT**2)'
     1,T38,'.',T45,'ENHANCED TUBES'/T2,'DI= INTERNAL TUBE DIAMETER (IN.)
     1',T38,'.',T41,'PI= PITCH OF INTERNAL HELIX(IN.)'/T2,'DPTHDR= HEADE
     1R DEPTH (FT.)',T38,'.',T41,'PSAT= CONDENSER OPERATING PRESSURE'/T2
     1,'DRYCG= HEIGHT OF CENTER OF GRAVITY',T38,'.',T45,'(IN.HG)ABSOLUTE
     1'/T6,'ABOVE CONDENSER BOTTOM AT DRY',T38,'.',T41,'QSTM= QUALITY OF
     1 INLET STEAM TO'/T6,'WEIGHT (FT.)',T38,'.',T45,'CONDENSER'/T2,'DTB
     1= DIAMETER OF TUBE BUNDLE (FT.)',T38,'.',T41,'RI= OPERAND FOR FRIC
     1TION FACTOR,'/T2,'DTS= DIAMETER OF TUBE SHEET(FT.)',T38,'.',T45,'H
     1ELICAL INTERNAL RIDGING')
        WRITE(KOUT,106)
106     FORMAT(1X,T2,'DW= TUBE WALL DIAMETER (IN.)',T38,'.',T41,'RSCALE=
     1 FOULING RESISTANCE'/T2,'EI= OPERAND FOR FRICTION FACTOR',T38,'.',
     1T45,'(HR-FT**2-DEG.F/BTU)'/T6,'HELICAL INTERNAL RIDGING',T38,'.',T
     141,'STMLD= STEAM LOAD (LBM/HR)'/T2,'FE= OPERAND FOR CONDENSING HEA
     1T',T38,'.',T41,'TSAT= SATURATION TEMP. (DEG.F)'/T6,'TRANSFER COEFF
     1ICIENT',T38,'.',T41,'VTB= VOLUME OF TUBE BUNDLE (FT**3)'/T2,'KW= C
     1ONDUCTIVITY OF TUBE WALL',T38,'.',T41,'WEXP= EXPANSION JOINT WEIGH
     1T (LB.)'/T6,'(BTU/HR-FT-DEG.F)',T38,'.',T41,'WMISC= WEIGHT OF MISC
     1ELLANEOUS'/T2,'LANE= STEAM LANE BREADTH (FT.)',T38,'.',T45,'COMPON
     1ENTS (LB.)'/T2,'LHW= LENGTH OF HOTWELL (FT.)',T38,'.',T41,'WETCG=
     1HEIGHT OF CENTER OF GRAVITY'/T2,'LTOT= TOTAL TUBE LENGTH (FT.)',T3
     18,'.',T45,'ABOVE CONDENSER BOTTOM AT WET'/T2,'TCI= INLET COOLANT T
     1EMP. (DEG.F)',T38,'.',T45,'WEIGHT (FT.)'/)
C
        WRITE(KSCR,108)
108     FORMAT(1X,'ENTER THE NUMBER OF CONDENSERS TO BE SIZED')
        READ(KSCR,*) NCOND
C
        DO 10 N= 1, NCOND
C
```

```
      CONFIG= 3
      WRITE(KOUT,110)
 110  FORMAT('1',T21,'*****************************')
      WRITE(KOUT,112) N
      WRITE(KSCR,112) N
 112  FORMAT(1X,T25,'CONDENSER NUMBER(',I3,')')
      WRITE(KOUT,114)
 114  FORMAT(1X,T21,'*****************************')
C
      READ(KIN,116) CTYPE
 116  FORMAT(A4)
      READ(KIN,*) NUMBT,LMAX,PMAX,VMIN,VMAX
      READ(KIN,*) TCI,TSAT,QSTM,STMLD,PSAT,PD
      READ(KIN,*) TIS,TOTS,TSHL,TTSP,THW,THDR,ITSD
      READ(KIN,*) OTSD,SHLD,TSPD,HWD,HDRD,TBD,ETUBE,TCOVER,COVERD
C
      DO 15 KT= 1, NUMBT
      READ(KIN,*) (TDATA(KT,J),J=1,7)
      READ(KIN,*) (TDATA(KT,J),J=8,12)
  15  CONTINUE
C
      IF(CTYPE .EQ. ITYPE) THEN
      INDEX1= 1
      WRITE(KOUT,118)
 118  FORMAT(1X,'. . CONDENSER TYPE: SUBMARINE DESIGN')
      ELSE
      INDEX1= 2
      WRITE(KOUT,120)
 120  FORMAT(1X,'. . CONDENSER TYPE: SURFACE SHIP DESIGN')
      END IF
C
      WRITE(KSCR,122)
 122  FORMAT(1X,'ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CON
     1DENSER')
      READ(KSCR,*) NPASS
      IF(NPASS .LE. 1) THEN
```

```
      WRITE(KOUT,124)
124   FORMAT(1X,'. . SINGLE PASS CONDENSER . .')
      ELSE
      END IF
C
      IF(NPASS .GT. 1) THEN
      WRITE(KSCR,126)
126   FORMAT(1X,'TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE PA
     1SS CONDENSERS: '/T5,'(1) CONVENTIONAL RETURN HEADER DESIGN, AND'/T
     15,'(2) "U-TUBE" CONSTRUCTION'/T2,'ENTER 1 OR 2 FOR CONFIGURATION S
     1ELECTION')
      READ(KSCR,*) CONFIG
      ELSE
      END IF
      IF(CONFIG .GT. 2) THEN
      GO TO 900
      ELSE IF(CONFIG .LE. 1) THEN
      WRITE(KOUT,128)
128   FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HE
     1ADER DESIGN.')
      ELSE
      WRITE(KOUT,130)
130  FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH "U-TUBE" CONSTRUCTION.')
      END IF
900   CONTINUE
C
      WRITE(KSCR,132)
132   FORMAT(1X,'ENTER: (1) COOLANT FLOW RATE (GPM)')
      READ(KSCR,*) GPM
C
      WRITE(KSCR,134)
134   FORMAT(1X,'DO YOU WANT TO INCLUDE IN THIS ANALYSIS THE EFFECTS O
     1F CONDENSATION IN THE'/T2,' PRESENCE OF NON-CONDENSABLE GASES?'/T6
     2,'YES OR NO')
      READ(KSCR,136) GAS
136   FORMAT(A4)
```

```
      IF((GAS .EQ. NO) .OR. (GAS .EQ. NOE)) THEN
      NON= 2
      ELSE
      NON= 1
      END IF
C
      WRITE(KSCR,138)
 138  FORMAT(1X,'DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE ENTRAN
     1CE VELOCITY (FT/SEC)?'/T6,'YES OR NO')
      READ(KSCR,140) STM
 140  FORMAT(A4)
      IF((STM .EQ. NO) .OR. (STM .EQ. NOE)) THEN
      INDEX2= 2
      WRITE(KSCR,142)
 142  FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN STEAM LA
     1NE ENTRANCE VELOCITY USING CONDENSER OPERATING PRESSURE AS SELECTI
     1ON CRITERIA.'//)
      ELSE
      INDEX2= 1
      WRITE(KSCR,146)
 146  FORMAT(1X,'ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/S
     1EC)')
      READ(KSCR,*) STMVEL
      END IF
C
      IF(INDEX2 .GT. 1) GO TO 911
      GO TO 912
 911  CONTINUE
      IF(PSAT .LT. 1.) THEN
      STMVEL= 500.
      ELSE IF(PSAT .GT. 5.) THEN
      STMVEL= 200.
      ELSE IF(PSAT .LE. 3.) THEN
      VEL= 600. - 100. * PSAT
      STMVEL= ANINT(VEL)
      ELSE IF(PSAT .GT. 3.) THEN
```

```
      VEL= 450. - 50. * PSAT
      STMVEL= ANINT(VEL)
      ELSE
      END IF
C STMVEL= MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)
 912  CONTINUE
C
      WRITE(KSCR,148)
 148  FORMAT(1X,'DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE PLATE SPAC
     1ING (FT.)?'/T6,'YES OF NO')
      READ(KSCR,150) DRAIN
 150  FORMAT(A4)
      IF((DRAIN .EQ. NO) .OR. (DRAIN .EQ. NOE)) THEN
      INDEX3= 2
      WRITE(KSCR,152)
 152  FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED CONDENSATE DRAINAGE P
     1LATE SPACING.'//)
      ELSE
      INDEX3= 1
      WRITE (KSCR,154)
 154  FORMAT(1X,'ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)')
      READ(KSCR,*) DRNPLT
      END IF
C
C * * * * FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * * *
C
      HFG=   1069.185 - 26.72568 * PSAT + 6.659357 * PSAT**2 -
     1       .9288262 * PSAT**3 + .05023536 * PSAT**4
      DENSC= PSAT**2 * .008829 - PSAT * .21843 + 62.347
      KSAT=  (PSAT**2 * (-.05904) + PSAT * 3.0435 + 356.6) * .001
      MU=    5.045709 - .0463856 * TSAT + .1242855E-03 * TSAT**2
      SIGMA= .0057445 - .7136956E-05 * TSAT - .64168E-08 * TSAT**2
      SPVOL= 1207.892 - 756.2292 * PSAT + 209.3 * PSAT**2 -
     1       25.8823 * PSAT**3 + 1.16875 * PSAT**4
C HFG=   LATENT HEAT OF CONDENSATION (BTU/LBM)
C DENSC= DENSITY OF CONDENSATE (LBM/FT**3)
```

```
C KSAT=  CONDUCTIVITY OF CONDENSATE (BTU/HR-FT-DEG.F)
C MU=    ABSOLUTE VISCOSITY OF CONDENSATE (LBM/HR-FT)
C SIGMA= SURFACE TENSION OF CONDENSATE (LBF/FT)
C SPVOL= SPECIFIC VOLUME OF STEAM AT SAT. CONDITIONS (FT**3/LBM)
C
        MF= 1.025
C
        DO 20 NLOOP= 1, 25
C
        DO 25 KT= 1, NUMBT
C
        NTYPE=  TDATA(KT,1)
        DI=     TDATA(KT,2)
        DW=     TDATA(KT,3)
        AE=     TDATA(KT,4)
        AEPE=   TDATA(KT,5)
        EI=     TDATA(KT,6)
        PI=     TDATA(KT,7)
        FLUTE=  TDATA(KT,8)
        MI=     TDATA(KT,9)
        RI=     TDATA(KT,10)
        KW=     TDATA(KT,11)
        RSCALE= TDATA(KT,12)
        WF= 17875.5 / MU * DENSC**2 * AE**3 * AE/AEPE * EXP(-3.33 *AEPE)
C WF= FLOODING CONDENSATE FLOW RATE PER FLUTE (LB/HR)
C
        FE= 3.661218 + 6.4526 * AEPE - 15.265 * AEPE**2 + 16.14543
     1      * AEPE**3 - 6.56116 * AEPE**4
        TDATA(KT,13)= FE
C FE= OPERAND FOR CONDENSING HEAT TRANSFER COEFFICIENT
C
        IF(AE .LE. .015) THEN
        SPEC1= 3.54272 - 522.1174 * AE + .1928462E+05 * AE**2
        ELSE IF (AE .LE. .025) THEN
        SPEC1= .3110063 - 25.2006 * AE + 520.0139 * AE **2
        ELSE
```

```
      SPEC1= .0303163 - 1.345982 * AE + 14.93315 * AE**2
      END IF
C
      NOTE= 0
      DO 30 LOOP1= 1, 250
      IF(NOTE .GT. 0) GO TO 922
      IF(LOOP1 .EQ. 1) THEN
      VM= VMIN / 1.5
      ELSE IF(LOOP1 .EQ. 2) THEN
      VM= VMAX * 1.5
      VDIFF= VMAX * 1.5 - VMIN / 1.5
      ELSE IF(LOOP1 .EQ. 3) THEN
      VM= (VMAX * 1.5 + VMIN / 1.5) / 2.
      ELSE
      END IF
C
C * * * * CALCULATION OF BASIC CONDENSER DATA * * * *
C
 922  QT= STMLD * QSTM * HFG
      AW= GPM * .0022283 / VM
      WC= AW * VM * 230400.
C QT= CONDENSER HEAT LOAD (BTU/HR)
C AW= TOTAL CROSS-SECTIONAL AREA REQUIRED FOR COOLANT FLOW (FT**2)
C WC= COOLING WATER FLOW RATE (LB/HR)
C
      TOUT=  TCI + QT / (.95 * WC)
      ARG1=  DI / 24.
      NT=    AW / (PIE * ARG1**2)
      NT=    ANINT(NT)
      TNT=   NT * NPASS
      DN=    DW + AE * 2.
      OD=    DW + AE * 4.
      DTEMP= (TSAT - TCI) / (TSAT - TOUT)
      ODTLM= (TOUT - TCI) / LOG(DTEMP)
      OTCS=   (TOUT + TCI) / 2.
      CNU= .122181 - .148162E-02 * OTCS + .5516445E-05 * OTCS**2
```

```
C TOUT=  OUTLET COOLANT TEMPERATURE (DEG.F)
C NT=    NUMBER OF TUBES PER CONDENSER PASS
C TNT=   TOTAL NUMBER OF TUBES
C DN=    NOMINAL TUBE DIAMETER (IN.)
C OD=    OUTSIDE TUBE DIAMETER (IN.)
C ODTLM= CONDENSER LOG MEAN TEMPERATURE DIFFERENCE (DEG.-F)
C
        CALL GEOM(OD,TNT)
C
C * * * * DETERMINATION OF CONDENSATE DRAINAGE PLATE SPACING * * * *
C
        MACH=  STMVEL / SQRT(3672.76 * (TSAT + 459.67))
        CDRAG= .75 * MACH + .975
        PITCH= PD * OD
C MACH=  MACH NUMBER OF STEAM INLET FLOW TO CONDENSER
C CDRAG= DRAG COEFFICIENT FOR STEAM INLET FLOW OVER CONDENSER TUBES
C PD=    TUBE PITCH-TO-DIAMETER RATIO
C PITCH= TUBE PITCH (IN.)
C
        IF(INDEX3 .LE. 1) GO TO 934
        DRNPLT= 2.5
C
        DO 35 N1= 1, 60
C
        WDRAG= CDRAG * OD * DRNPLT * STMVEL**2 / (24. * SPVOL)
C WDRAG= DRAG FORCE LOADING ON TUBES DUE TO STEAM VELOCITY(LBF.)
        DELT= 5.
C "DRNPLT=2.5" & "DELT=5." ARE BOTH ASSUMED VALUES TO INITIATE THE
C FOLLOWING ITERATIVE CALCULATIONS.
C
        IF(N1 .EQ. 1) THEN
        TMIN1= TCI + .15
        T2=   TMIN1
        ELSE IF(N1.EQ.2)THEN
        TMAX1=  TCI + 20.
        T2=    TMAX1
```

```
      TDIFF1= TMAX1- TMIN1
      ELSE IF(N1 .EQ. 3) THEN
      T2= TCI + TDIFF1 / 2.
      ELSE
      END IF
C
      DTS= T2 - TCI
      TCS= TCI + DTS / 2.
      ARG7= (TSAT - TCI) / (TSAT - T2)
      DTLM= DTS / LOG(ARG7)
C DTLM= LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C THIS ANALYSIS EXAMINES A SAMPLE TUBE SECTION OF LENGTH= DRNPLT TO
C DETERMINE RECOMMENDED VALUES FOR CONDENSATE DRAINAGE PLATE SPACING.
C THIS TUBE SECTION IS LOCATED IN THE COOLING WATER INLET SECTION OF THE
C CONDENSER, WHERE THE CONDENSATION RATE/PER UNIT TUBE LENGTH WILL BE AT
C A MAXIMUM.
C DRNPLT= CONDENSATE DRAINAGE PLATE SPACING (FT.)
C DELT=   SAT. TEMP. - TUBE WALL TEMP. (DEG.F)
C T2=     TUBE SECTION C.W. OUTLET TEMP. (DEG.F)
C DTS=    C.W. TEMP. RISE ACROSS TUBE SECTION (DEG.F)
C TCS=    AVGERAGE C.W. TEMP. FOR EACH SECTION (DEG.F)
C DTLM=   LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C
      IF(NTYPE .LE. 1) THEN
      HW= .023 * KSW(TCS) / DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR SMOOTH INTERNAL TUBE
C     (BTU/HR-FT**2-DEG.F)
      ELSE
      ARG2= 7. / RE(VM)
      GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
      ARG3=  EI / PI
      ARG4=  EI / DI * RE(VM) * FF(ARG2)
      HW=    12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR DOUBLY ENHANCED TUBES
C     (BTU/HR-FT**2-DEG.F)
      END IF
```

```
C
        DO 40 N3= 1, 10
C
        ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
        ARG6= KSAT**3 * DENSC*SIGMA*QSTM*HFG * 4.17*10.**8/(MU*DELT)
        ARG8= AE / 12.
        HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1         * ARG6**.2307
C HCOND= HEAT TRANSFER COEFFICIENT CONDENSATE (BTU/HR-FT**2-DEG.F)
C
        US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DW/DI *(7.985 *
     1    (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN /(24. * KW))
C US= OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-DEG.F)
C
        QS=   .95 * WC * DTS
        DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
C QS= HEAT LOAD FOR TUBE SECTION (BTU/HR)
  40    CONTINUE
C
        L1= QS * 12. / (NT * PIE * DN * US * DTLM)
        WCWF= QS / (NT * QSTM* HFG * WF * FLUTE)
        DEFL= .26526*WDRAG*DRNPLT**3 / (ETUBE*(OD**4 - DI**4))
C DEFL= MAXIMUM TUBE DEFLECTION (IN.)
C
        IF(N1 .EQ.1) THEN
        LMIN1= L1
        WMIN1= WCWF
        GO TO 933
        ELSE IF(N1 .EQ. 2) THEN
        LMAX1=   L1
        WMAX1=   WCWF
        DIFFL1=  LMAX1 - LMIN1
        WDIFF1=  WMAX1 - WMIN1
        GO TO 933
        ELSE
        END IF
```

```
      IF(ABS(DRNPLT - L1) .GT. .005) THEN
      MULT2= (DRNPLT - L1) / DIFFL1
      STEP1= MULT2 * TDIFF1
      T2=    T2 + STEP1
      GO TO 933
      ELSE IF(ABS(SPEC1 - WCWF) .GT. .001) THEN
      MULT3= (SPEC1 - WCWF) / WDIFF1
      STEP2= MULT3 * DIFFL1
      DRNPLT= DRNPLT + STEP2
      GO TO 933
      ELSE IF(DEFL .GT. PITCH/2.) THEN
      DRNPLT= DRNPLT - .05
      ELSE
      END IF
      IF((ABS(DRNPLT-L1) .LT. .005) .AND. (ABS(SPEC1-WCWF) .LT. .001)
     1   .AND. (DEFL .LT. PITCH/2.)) GO TO 934
933   CONTINUE
 35   CONTINUE
934   CONTINUE
      DEFL= 1375. * WDRAG * DRNPLT**3 / (ETUBE * (OD**4 - DI**4))
      IF(DEFL .GT. PITCH/2.) THEN
      WRITE(KOUT,156) DEFL
      WRITE(KSCR,156) DEFL
156   FORMAT(1X,'* * PROGRAM WARNING NR-3: SPECIFIED CONDENSATE DRAINA
     1GE PLATE SPACING IS BEYOND TUBE SUPPORT LIMIT * *'/T2,'. . MAXIMUM
     1 TUBE DEFLECTION= ',F6.3,' (IN.)')
      ELSE
      END IF
C * * * * CALCULATION OF EFFECTIVE TUBE LENGTH * * * *
C
      SUML=  0.0
      SUMQ=  0.0
      AVGHW= 0.0
      AVGHC= 0.0
      AVGU=  0.0
      T1= TCI
```

```
C
      DO 45 N4= 1, 100
C
      DELT= 5.
C
      DO 50 N5= 1, 25
C
      IF(N5 .EQ. 1) THEN
      TMIN= T1 + .15
      T2=   TMIN
      ELSE IF(N5 .EQ. 2) THEN
      TMAX=  T1 + 20.
      T2=    TMAX
      TDIFF= TMAX - TMIN
      ELSE IF(N5 .EQ. 3) THEN
      T2=    T1 + TDIFF / 2.
      ELSE
      END IF
C
      DTS=  T2 - T1
      TCS=  T1 + DTS / 2.
      ARG7= (TSAT - T1) / (TSAT - T2)
      DTLM= DTS / LOG(ARG7)
C
      IF(NTYPE .LE. 1) THEN
      HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
      ELSE
      ARG2= 7. / RE(VM)
      GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
      ARG3= EI / PI
      ARG4= EI / DI * RE(VM) * FF(ARG2)
      HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
      END IF
C
      DO 55 N6= 1, 10
C
```

```
      ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
      ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
      ARG8= AE / 12.
      HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1       * ARG6**.2307
C
      US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND +LOG(DW/DI * (7.985 *
     1  (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN / (24. * KW))
      QS= .95 * WC * DTS
      DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
   55 CONTINUE
C
      L1= QS * 12. / (NT * PIE * DN * US * DTLM)
C
      IF(N5 .EQ. 1) THEN
      LMIN2= L1
      GO TO 944
      ELSE IF(N5 .EQ. 2) THEN
      LMAX2=  L1
      DIFFL2= LMAX2 - LMIN2
      GO TO 944
      ELSE
      END IF
      IF(ABS(DRNPLT - L1) .GT. .005) THEN
      MULT4= (DRNPLT - L1) / DIFFL2
      STEP4= MULT4 * TDIFF
      T2= T2 + STEP4
      ELSE
      END IF
      IF(ABS(DRNPLT - L1) .LT. .005) GO TO 945
C
  944 CONTINUE
   50 CONTINUE
C
  945 WCWF= QS / (NT * QSTM * HFG * WF * FLUTE)
      IF(WCWF .GT. SPEC1 + .005) THEN
```

```
      WRITE(KOUT,158) N4
      WRITE(KSCR,158) N4
 158  FORMAT(1X,'* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION
     1 LIMITS - SECTION (',I3,') * * ')
      ELSE
      END IF
C
      DATA(N4,1)= QS
      DATA(N4,2)= HW
      DATA(N4,3)= HCOND
      DATA(N4,4)= US
      DATA(N4,5)= DRNPLT
C * * CALCULATION OF FINAL SECTION LENGTH * *
      SUML=  DATA(N4,5) + SUML
      SUMQ=  DATA(N4,1) + SUMQ
      AVGHW= DATA(N4,2) + AVGHW
      AVGHC= DATA(N4,3) + AVGHC
      AVGU=  DATA(N4,4) + AVGU
      DELT= 5.
      QFS= QT - SUMQ
      TCS= (TOUT + T2)/2.
C
      IF(NTYPE .LE. 1) THEN
      HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
      ELSE
      ARG2= 7./ RE(VM)
      ARG4= EI / DI * RE(VM) * FF(ARG2)
      HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
      END IF
C
      TL= 2.0
      DO 60 N7= 1, 10
      ARG5=  QSTM * HFG * WF / (TL * DELT)
      ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
      HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1       * ARG6**.2307
```

```
      US= 1. /(DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DW/DI * (7.985 *
     1   (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN/ (24. * KW))
      ARG7= (TSAT - T2) / (TSAT - TOUT)
      DTLM= (TOUT - T2) / LOG(ARG7)
      QP=   QFS * 12. / (PIE * DN * NT)
      TL=   QP / (US * DTLM)
      DELT= QP / (TL * HCOND)
   60 CONTINUE
C
      IF(TL .LE. DRNPLT) THEN
      NS = N4 + 1
      SUML= SUML + TL
      AVGHW= AVGHW / NS
      AVGHC= AVGHC / NS
      AVGU=  AVGU  / NS
      GO TO 950
      ELSE
      T1= T2
      END IF
   45 CONTINUE
  950 HWAVG(KT)= AVGHW
      HCAVG(KT)= AVGHC
      UAVG(KT)=  AVGU
C
C * * CONDENSER WEIGHT & VOLUME CALCULATIONS * *
C
      EFF(KT)= SUML / NPASS
      TLT(KT)= SUML / NPASS * MF
C
      IF(INDEX1 .LE. 1) THEN
      LTOT= (9.125 + TIS) / 6. + TLT(KT)
      ELSE
      LTOT= TIS / 6. + TLT(KT)
      END IF
C
      LANE= 1.22E-04 * STMLD * SPVOL / (TLT(KT) * STMVEL)
```

```
C TLT = TOTAL EFFECTIVE TUBE LENGTH (FT.),(TLT=TOTL)
C LTOT= TOTAL TUBE LENGTH (FT.)
C LANE= STEAM LANE BREADTH (FT.)
C
        ATB=   PIE * TOTRAD**2
        DTB=   SQRT(4. * ATB / PIE)
        ARG10= (DTB + 2. * LANE) / 2.
        ATS=   PIE * ARG10**2
        DTS=   DTB + 2 * LANE
        VTB=   TLT(KT) * ATB
C ATB= AREA OF TUBE BUNDLE (FT**2)
C DTB= DIAMETER OF TUBE BUNDLE (FT.)
C ATS= AREA OF TUBE SHEET (FT**2)
C DTS= DIAMETER OF TUBE SHEET (FT.)
C VTB= VOLUME OF TUBE BUNDLE (FT**3)
C
        DSHL=   DTS + TSHL / 6.
        DPTHDR= DTS / 2.
        VHW=    STMLD/3690.-.0417*ATB*NS/NPASS-.2618*TLT(KT)
        DHW=    DSHL + 1
        LHW=    VHW / (.7854 * DHW**2) + DPTHDR + .5
        CONDH=  LHW + DPTHDR + LTOT
C DSHL=   OUTER DIAMETER OF SHELL (FT.)
C DPTHDR= HEADER DEPTH (FT.)
C LHW=    LENGTH OF HOTWELL (FT.)
C VHW=    VOLUME OF HOTWELL (FT**3)
C CONDH=  TOTAL CONDENSER HEIGHT (FT.)
C
        CALL DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1              TOTHL,SYSL,PMPWR)
        IF(LOOP1 .EQ. 1) THEN
        LMIN= CONDH
        PMIN= PMPWR
        GO TO 955
        ELSE IF(LOOP1 .EQ. 2) THEN
        LPLUS= CONDH
```

```
      PPLUS= PMPWR
      DIFFL= LPLUS - LMIN
      DIFFP= PPLUS - PMIN
      GO TO 955
      ELSE
      END IF
C
      IF(NOTE .GT. 0) GO TO 964
      IF(ABS(LMAX - CONDH) .GT. .005) THEN
      MULT1= (LMAX - CONDH) / DIFFL
      CHANGE= MULT1 * VDIFF
      VM= VM + CHANGE
      GO TO 955
      ELSE
      END IF
964   IF(VM .LT. VMIN) THEN
      GO TO 956
      ELSE IF(VM .GT. VMAX) THEN
      VM= .99 * VM
      NOTE= 1
      ELSE IF(PMPWR .GT. PMAX) THEN
      MULTP= (PMAX - PMPWR) / DIFFP
      PSTEP= MULTP * VDIFF
      VM=    VM + PSTEP
      NOTE= 1
      ELSE
      END IF
      IF(NOTE .GT. 0) THEN
      GO TO 965
      ELSE
      GO TO 966
      END IF
965   IF(ABS(PMAX - PMPWR) .LE. .005) GO TO 956
      GO TO 955
966   IF((ABS(LMAX - CONDH) .LE. .005) .AND. (PMPWR .LE. PMAX))
     1    GO TO 956
```

```
 955    CONTINUE
        IF(LMIN .GT. LMAX) THEN
        GO TO 956
        ELSE
        END IF
   30   CONTINUE
C
 956    CONTINUE
        XLTOT(KT)= CONDH
        XVM(KT)=   VM
        NOTE= 0
        WRITE(KSCR,160) KT
 160    FORMAT(T2,'* * TUBE NO.(',I3,') TESTED * *')
   25   CONTINUE
C
        XCOMP= 0.0
C
        DO 65 KT1= 1, NUMBT
        IF(XVM(KT1) .GT. XCOMP) THEN
        XCOMP= XVM(KT1)
        OPT= KT1
        ELSE
        XCOMP= XCOMP
        END IF
  65    CONTINUE
C
        AVGHW=  HWAVG(OPT)
        AVGHC=  HCAVG(OPT)
        AVGU=   UAVG(OPT)
        EFFL=   EFF(OPT)
        TOTL=   TLT(OPT)
        CONDH=  XLTOT(OPT)
        VM=     XVM(OPT)
        NTYPE=  TDATA(OPT,1)
        DI=     TDATA(OPT,2)
        DW=     TDATA(OPT,3)
```

```
      AE=     TDATA(OPT,4)
      AEPE=   TDATA(OPT,5)
      EI=     TDATA(OPT,6)
      PI=     TDATA(OPT,7)
      FLUTE=  TDATA(OPT,8)
      MI=     TDATA(OPT,9)
      RI=     TDATA(OPT,10)
      KW=     TDATA(OPT,11)
      RSCALE= TDATA(OPT,12)
      FE=     TDATA(OPT,13)
C
      WRITE(KSCR,162)OPT,VM
 162  FORMAT(/T2,'* * TUBE NO.(',I3,') SELECTED AT V= ',F5.2,
     1       ' (FT/SEC) * *'//)
C
      IF(NON .LE. 1) GO TO 977
      GO TO 978
 977  CALL NONCOND(ODTLM,DN,OD,TNT,NPASS,TOTNC)
      TOTL= TOTNC
      WRITE(KSCR,164)
 164  FORMAT(T2,'* * NON-CONDENSABLE GAS ANALYSIS * *'/)
C
 978  IF(INDEX1 .LE. 1) THEN
      LTOT= (9.125 + TIS) / 6. + TOTL
      ELSE
      LTOT= TIS / 6. + TOTL
      END IF
      LANE=  1.22E-04 * STMLD * SPVOL / (TOTL * STMVEL)
      ATB=   PIE * TOTRAD**2
      DTB=   SQRT(4. * ATB / PIE)
      ARG10= (DTB + 2. * LANE) / 2.
      ATS=   PIE * ARG10**2
      DTS=   DTB + 2 * LANE
      VTB=   TOTL * ATB
C
      ARG12= DTS - 2 * LANE + OD / 6.
```

```
      WP1=   .785398 * (ARG12**2 - OD**2 * TNT * .00694)
      WP2=   .785398 * (DTS**2 - OD**2 *TNT* .00694)
      WOTS=  OTSD * TOTS / 6. * WP2
      WITS=  ITSD * TIS / 6. * WP2
      WTSP=  NS / NPASS * TSPD * TTSP/12. * WP1
      WTB=   .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT
      DSHL=  DTS + TSHL / 6.
      DPTHDR= DTS / 2.
      WSHL=  .785398 * SHLD * LTOT * (DSHL**2 -DTS**2)
C WOTS= WEIGHT OF OUTER TUBE SHEETS (LB.)
C WITS= WEIGHT OF INNER TUBE SHEETS (LB.)
C WTSP= WEIGHT OF TUBE SUPPORT PLATES (LB.)
C WTB=  WEIGHT OF TUBE BUNDLE (LB.)
C WSHL= WEIGHT OF SHELL (LB.)
C DSHL= OUTER DIAMETER OF SHELL (FT.)
C DPTHDR= HEADER DEPTH (FT.)
C
      WEXP= .39167 * DTB * TSHL * SHLD
      WHDR= .4083 * DTB**2 * THDR * HDRD
      VHW=  STMLD / 3690. - .0417 * ATB * NS/NPASS - .2618 * TOTL
      DHW=  DSHL + 1.
      LHW=  VHW / (.7854 * DHW**2) + DPTHDR + .5
      WHW=  .2618 * (DHW**2 / 4. + DHW * LHW) * THW * HWD
C LHW=  LENGTH OF HOTWELL (FT.)
C VHW=  VOLUME OF HOTWELL (FT.)
C WEXP= WEIGHT OF EXPANSION JOINT (LB.)
C WHDR= WEIGHT OF HEADER (LB.)
C WHW=  WEIGHT OF HOTWELL (LB.)
C
      IF(CONFIG .GT. 2) THEN
      GO TO 988
      ELSE
      END IF
      IF(CONFIG .GT. 1) THEN
      WCOVER= .20415 * DTB**2 * TCOVER * COVERD
      WHDR= WHDR/2. + WCOVER
```

```
      WUTUBE= .005454 * (DN**2 - DI**2) * DTB * TBD * TNT
      WTB=  .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT + WUTUBE
      ELSE
      END IF
 988  CONTINUE
      WT=    WOTS + WITS + WTSP + WTB + WSHL + WEXP + WHDR + WHW
      WMISC= WT * .25
      WDRY=  WT * 1.25
      WLIQ=  .005454 * DI**2 *NT*LTOT + 33.51 * DSHL**3 +STMLD/226935.
      WET=   WDRY + WLIQ
      TONS=  WET / 2240.
C WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
C WDRY=  TOTAL DRY WEIGHT OF CONDENSER (LB.)
C WLIQ=  TOTAL LIQUID WEIGHT (LB.)
C WET=   TOTAL WET WEIGHT OF CONDENSER (TONS)
C
      IF(NON .LT. 2) THEN
      CONDHC=  LHW + DPTHDR + LTOT
      CONDL=   CONDHC
      ELSE
      CONDH=   LHW + DPTHDR + LTOT
      CONDL=   CONDH
      END IF
C
      BOXVOL=  DSHL**2 * (LTOT + DPTHDR) + DHW**2 * LHW
C CONDH=  CONDENSER HEIGHT (FT.)
C BOXVOL= CONDENSER ENCLOSED BOX VOLUME (FT**3)
C
      ARG14= (LHW + LTOT/2.) * (WOTS + WITS + WTSP + WSHL + WEXP)
     1       + (LHW/2.) * WHW + CONDL/2. * WMISC
      IF(CONFIG .GT. 1) THEN
      DRYCG= (ARG14 +(LHW - DPTHDR/2.)*WDHDR/2. + (LHW+LTOT+DPTHDR/2.)
     1 * (WCOVER + WUTUBE) + (LHW + LTOT/2.) * WTB) / WDRY
      ELSE
      DRYCG= (ARG14 + (LHW + LTOT/2.) * (WTB + WHDR)) / WDRY
      END IF
```

```
          ARG15= STMLD / 3690. * 61.5
          ARG16= WLIQ - ARG15
          WETCG= (DRYCG * WDRY + ARG16 * (LHW + LTOT/2.) + ARG15 * VHW /
     1           25.1328) / WET
C DRYCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT DRY WEIGHT (FT.)
C WETCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT WET WEIGHT (FT.)
C
          CALL DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1                TOTHL,SYSL,PMPWR)
C
          A= N
          OUTPUT(N,1)= A
          OUTPUT(N,2)= DSHL
          OUTPUT(N,3)= CONDL
          OUTPUT(N,4)= BOXVOL
          OUTPUT(N,5)= WET
          OUTPUT(N,6)= PMPWR
C
C * * * * CALCULATION RESULTS - OUTPUT(DATA FILE) * * * *
C
          IF(NTYPE .LE. 1) THEN
          WRITE(KOUT,166)
 166      FORMAT(T2,'. . SMOOTH INTERNAL TUBES . .')
          ELSE
          WRITE(KOUT,168)
 168      FORMAT(T2,'. . DOUBLY ENHANCED TUBES . .')
          END IF
C
          WRITE(KOUT,170) OPT
 170      FORMAT(T2,'. . TUBE NUMBER(',I3,') SELECTED . .')
          IF(NON .LE. 1) THEN
          WRITE(KOUT,172)
 172      FORMAT(T2,'. . NON-CONDENSABLE GAS ANALYSIS . .')
          ELSE
```

```
      END IF
      WRITE(KOUT,174)
174   FORMAT(T2,T27,'**** DATA INPUT ****')
      WRITE(KOUT,176) GPM,VM
176   FORMAT(T2,'. . FLOW RATE: ',F6.0,' (GPM)  . . FLOW VELOCITY: ',
     1        F5.2,' (FT/SEC) . .')
      WRITE(KOUT,178) DI,DW,AE,AEPE,NTYPE,EI,PI,FE,MI,RI,KW,TCI,PSAT,
     1                QSTM,TSAT,STMLD,RSCALE
178   FORMAT(T2,'DI = ',F5.3,T16,'DW = ',F5.3,T30,'AE = ',F5.3,T44,'AE
     1PE = ',F6.4,T61,'NTYPE = ',I2/T2,'EI = ',F6.4,T16,'PI = ',F6.4,T30
     1,'FE = ',F5.2,T44,'MI = ',F5.3,T61,'RI = ',F6.4/T2,'KW = ',F6.2,T1
     16,'TCI = ',F6.2,T30,'PSAT = ',F4.2,T44,'QSTM = ',F4.2,T61,'TSAT =
     1',F6.2/T2,'STMLD = ',F9.1,T30,'RSCALE = ',F7.5/)
      WRITE (KOUT,180)
180   FORMAT(T17,'**** DATA OUTPUT - CONDENSER SIZING ROUTINE ****')
      WRITE(KOUT,182) STMVEL
182   FORMAT(T2,'. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: ',F4.0
     1,' (FT/SEC)')
      WRITE(KOUT,184) DRNPLT
184   FORMAT(T2,'. . CONDENSATE DRAINAGE PLATE SPACING: ',F5.2,
     1           ' (FT.)'/)
      WRITE(KOUT,186) QT,TOUT,TNT
186   FORMAT(1X,'. . CONDENSER HEAT LOAD:   ',E12.5,' (BTU/HR)'/T2,'.
     1. OUTLET COOLANT TEMP:    ',F6.2,' (DEG.F)'/T2,'. . TOTAL NUMBER O
     1F TUBES: ',F7.1/)
      WRITE(KOUT,188) TOTL,AVGHW,AVGHC,AVGU
188   FORMAT(1X,'. . EFFECTIVE TUBE LENGTH:',T39,F5.2,' (FT.)'/T2,'. .
     1 AVG. HEAT TRNFR. COEFF. C.W.:',T38,F6.0,' (BTU/HR-FT**2-DEG.F)'/T
     12,'. . AVG. HEAT TRNFR. COEFF. COND.:',T38,F6.0,' (BTU/HR-FT**2-DE
     1G.F)'/T2,'. . AVG. OVERALL HEAT TRNFR. COEFF.:',T38,F6.0,' (BTU/HR
     1-FT**2-DEG.F)'/)
      WRITE(KOUT,190) LTOT,LANE,DTB,ATB,VTB,DTS,ATS,WEXP,WMISC,LHW,
     1                DPTHDR
190   FORMAT(1X,'LTOT = ',F5.2,T19,'LANE = ',F5.2,T37,'DTB = ',F6.2,T5
     13,'ATB = ',F6.2/T2,'VTB = ',F6.2,T19,'DTS = ',F5.2,T37,'ATS = ',F6
     1.2///T12,'**** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****'//T
```

```
     12,'WEXP = ',F7.1,T19,'WMISC = ',F7.1,T37,'LHW = ',F5.2,T53,'DPTHDR
     1 = ',F5.2///T30,'MATERIAL',T47,'THICKNESS',T64,'WEIGHT'/T26,'DENSI
     1TY(LB/FT**3)',T49,'(IN.)',T65,'(LB.)')
        IF(INDEX1 .GT. 1) THEN
        WRITE(KOUT,192) ITSD,TIS,WITS
 192    FORMAT(T2,'TUBE SHEET-----------',T31,F5.1,T49,F6.4,T64,F6.0)
        ELSE
        WRITE(KOUT,194) OTSD,TOTS,WOTS,ITSD,TIS,WITS
 194    FORMAT(T2,'OUTER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1            'INNER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0)
        END IF
        WRITE(KOUT,198) TSPD,TTSP,WTSP,TBD,WTB,SHLD,TSHL,WSHL,HWD,THW,
     1                  WHW,HDRD,THDR,WHDR,WDRY
 198    FORMAT(T2,'TUBE SUPPORT PLATE---',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1        'TUBE BUNDLE----------',T31,F5.1,T49,'*.****',T64,F6.0/T2,
     1            'CONDENSER SHELL------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1            'HOTWELL--------------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1            'WATERBOX-------------',T31,F5.1,T49,F6.4,T64,F6.0//T2
     1           ,'. . TOTAL CONDENSER DRY WEIGHT=  ',F8.1,' (LB.)')
        WRITE(KOUT,200) WET,TONS,CONDL,DSHL,BOXVOL,DRYCG,WETCG,TOTHL,
     1  SYSL,PMPWR
 200    FORMAT(1X,'. . TOTAL CONDENSER WET WEIGHT=  ',F8.1,' (LB.)'/T35,
     1F6.2,' (TON)'//T2,'. . TOTAL CONDENSER HEIGHT= ',F6.2,' (FT.)'/T2,
     1'. . OUTER SHELL DIAMETER=   ',F6.2,' (FT.)'/T2,'. . ENCLOSED BOX
     1VOLUME=   ',F7.1,' (FT**3)'/T6,'DRYCG = ',F6.2,T24,'WETCG = ',F6.2
     1//T2,'. . CONDENSER FRICTIONAL HEAD LOSS= ',F5.1,' (FT.)'/T2,'. .
     1TOTAL SYSTEM HEAD LOSS=',T38,F5.1,' (FT.)'/T2,'. . TOTAL SYSTEM PU
     1MPING POWER=',T38,F5.1,' (HP)'///)
C
        IF(NON .LE. 1) GO TO 998
        GO TO 999
 998    IF(CONDHC .GT. CONDH) THEN
        MF= MF + .01
        ELSE IF(CONDHC .LT. CONDH) THEN
        MF= MF - .01
        ELSE
```

```
      END IF
      IF((ABS(CONDHC-CONDH) .LE. .005*CONDH) .AND. (CONDHC .LE.
     1     CONDH)) GO TO 999
C
      WRITE(KSCR,202) NLOOP
 202  FORMAT(T2,'* * NON-CONDENSABLE GAS ANALYSIS: ITERATION NR:('
     1          ,I3,') * *')
C
  20  CONTINUE
 999  CONTINUE
  10  CONTINUE
C
      WRITE(KOUT,204)
 204  FORMAT('1',T26,'**** DATA SUMMARY ****'//T4,'CONDENSER',T16,'SHE
     1LL',T27,'TOTAL',T36,'ENCLOSED',T48,'TOTAL WET',T61,'TOTAL SYSTEM'/
     1T5,'NUMBER',T15,'DIAMETER',T26,'HEIGHT',T35,'BOX VOLUME',T49,'WEIG
     1HT',T60,'PUMPING POWER'/T16,'(FT.)',T27,'(FT.)',T36,'(FT**3)',T50,
     1'(LB.)',T64,'(HP)')
C
      DO 70 N8= 1, NCOND
      WRITE(KOUT,206)(OUTPUT(N8,J),J=1,6)
 206  FORMAT(T6,F3.0,T15,F6.2,T26,F6.2,T36,F7.1,T49,F7.0,T63,F5.1)
  70  CONTINUE
      STOP
      END
C
      SUBROUTINE DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1                    TOTHL,SYSL,PMPWR)
C SUBROUTINE DELTAP CALCULATES THE COOLING WATER SIDE PRESSURE DROPS,
C AND EVALUATES THE CONDENSER AND TOTAL SYSTEM PUMPING POWER
      REAL LTOT
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      RE= VM * DI * 300. / CNU
      IF(NTYPE .EQ. 2) THEN
      FFT= 8. * (-1. / (2.46 * LOG((7/RE)**MI + RI)))**2
```

```
      ELSE
      ARG1= 1. / (7/RE)**.8 + 1.62E-5 / DI
      A=    (2.457 * LOG(ARG1))**16
      B=    (37530 / RE)**16
      FFT=  8. * ((8. / RE)**12 + 1. / (A + B)**1.5)**.0833
      END IF
      IF(CONFIG .EQ. 2) THEN
      FFT= FFT * (RE *(DI / (12 * DTB))**2)**.05
      PWO= .00237 * VM**2.33
      PTE= .0178  * VM**2
      ELSE IF(NPASS .EQ. 1)THEN
      PWO= .00237 * VM**2.33
      PTE= .0178  * VM**2
      ELSE
      PWO= .00635 * VM**2.1
      PTE= .0356  * VM**2
      END IF
      HT= FFT * LTOT * NPASS * VM**2 / (DI * 5.353)
      PWI= .01422 * VM**2.04
C
      HLP= 79.7 - .00338 * GPM
C HT=  FRICTIONAL HEAD LOSS FOR TUBES (FT.)
C HWB= WATERBOX INLET & OUTLET LOSSES (FT.)
C HE=  TUBE END LOSSES (FT.)
C HLP= SEAWATER CIRCULATING LOOP PIPING LOSSES(FT.)
C
      TOTHL= HT + PWO + PTE + PWI
      SYSL=  TOTHL + HLP
      PMPWR= .000352 * GPM * SYSL
C TOTHL= TOTAL FRICTIONAL HEAD LOSS FOR CONDENSER (FT.)
C SYSL=TOTAL FRICTIONAL HEAD LOSS OF CONDENSER AND SYSTEM PIPING(FT.)
C PMPWR= TOTAL SYSTEM PUMPING POWER (HP)
C
      RETURN
      END
C
```

```
      SUBROUTINE GEOM(OD,TNT)
C SUBROUTINE: RADIUS DETERMINES THE TUBE SHEET GEOMETRY
      DIMENSION RAD(250),NTUB(250)
      REAL NTUBES,NTUB
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA PIE/3.1415927/,VOIDID/.83333/
C VOIDID= CENTRAL VOID DIAMETER FOR AIR REMOVAL DUCT (FT.)
      PITCH= PD * OD / 12.
C PITCH= TUBE PITCH (FT.)
      RSPACE= PITCH * .8660254
C RSPACE= RADIAL TUBE ROW SPACING FROM TUBE CENTERLINE FOR 60 DEG.
C         PITCH(FT.)
      RAD(1)= VOIDID / 2.
C RAD= RADIUS OF DESIGNATED TUBE ROW
      NTUB(1)= PIE * VOIDID / PITCH
C NTUB= NUMBER OF TUBES PER DESIGNATED TUBE ROW
      SUMTUB= NTUB(1)
      NROW= 1
C
      DO 10 I= 2, 250
      ITOT= I
      RAD(I)= RAD(1) + NROW * RSPACE
      NTUB(I)= 2 * PIE * RAD(I) / PITCH
      SUMTUB= SUMTUB + NTUB(I)
      REMAIN= TNT - SUMTUB
      IF(REMAIN) 200,300,100
 100  NROW= NROW + 1
  10  CONTINUE
 200  PART= TNT - SUMTUB + NTUB(ITOT)
      NTUB(ITOT)= PART
 300  TOTRAD= RAD(ITOT)
C
      NROW= ITOT
      DO 20 J=1, ITOT
      RADIUS(J)= RAD(NROW)
```

```
      NTUBES(J)= NTUB(NROW)
      NROW= NROW - 1
   20 CONTINUE
C
      RETURN
      END
C
      SUBROUTINE NONCOND(ODTLM,DN,OD,TNT,NPASS,TOTNC)
C SUBROUTINE NONCOND: (1) CALCULATES THE PRESSURE DISTRIBUTION AROUND
C THE TUBE BUNDLE PERIPHERY, (2) CALCULATES THE MEAN TOTAL PRESSURE
C AROUND THE TUBE BUNDLE PERIPHERY, AND (3) CALCULATES TUBE LENGTH IN
C THE PRESENCE OF NON-CONDENSABLE GASES.
      DIMENSION WSTM(250),LTUBE(250)
      REAL LREF,L1,LANE,LTUBE,MIXVISC,NTUBES,LSEC
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1      TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA PIE/3.1415927/
      HFG(PCOND)= 1069.185 - 26.72568 * PCOND + 6.659357 * PCOND**2 -
     1            .9288262 * PCOND**3 + .05023536 * PCOND**4
      DENSTM(PCOND)= 1 / (1346.986 - 982.9167 * PCOND + 326.8442 *
     1       PCOND**2 - 49.5767 * PCOND**3 + 2.781686 * PCOND**4)
      TCOND(PCOND)= 46.66471 + 39.7291 * PCOND - 7.869679 * PCOND**2
     1       + .8627231 * PCOND**3 - .03663604 * PCOND**4
      STMVISC(PCOND)= 4.87475E-5 * TCOND(PCOND) + .02191
      AIRVISC(PCOND)= 6.0E-5 * TCOND(PCOND) + .04
      RE(PCOND)= DENSTM(PCOND) * DE * V * 3600 / STMVISC(PCOND)
      F(PCOND)= .0436 / RE(PCOND)**.186
C
C FIRST THE STAGNATION OR TOTAL PRESSURE AT ZONE 1 IS CALCULATED. ALL
C PRESSURE LOSSES AROUND THE TUBE BUNDLE PERIPHERY ARE BASED ON THE
C PRESSURE IN ZONE 1.
C
      PCOND= PSAT
      PT= PSAT + .00021977 * DENSTM(PCOND) / QSTM * STM**2
C PT=     TOTAL PRESSURE (IN. HG) ABSOLUTE
C PSAT=   SPECIFIED CONDENSER PRESSURE (STATIC,IN. HG) ABSOLUTE
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```
C STM=    MAXIMUM STEAM LANE ENTRANCE VELOCITY (FT/SEC)
C DENSTM= STEAM DENSITY (LB/FT**3)
        DPEN= 1.09885E-5 * DENSTM(PCOND) / QSTM * STM**2
C DPEN= ENTRANCE PRESSURE LOSS (IN.HG)
        DE= 2 * TOTL * LANE / (LANE + TOTL)
C DE= EQUIVALENT DIAMETER OF FLOW AREA(THE RATIO OF FLOW AREA TO
C     WETTED PERIMETER MULTIPLIED BY 4) (FT.)
C
        PL1= PT - DPEN
C PL1= PRESSURE OF CONDENSER ALONG SECTOR NR.1 (IN.HG) ABSOLUTE
        PCOND= PL1
        LREF= TOTRAD * PIE / 6.
        L1=   LREF * 2.
C
        DO 10 J= 1, 3
        CONST1= 9.2593E-5 * QT / (TOTL * LANE)
        V2=     CONST1 / (HFG(PCOND) * DENSTM(PCOND))
C V2= STEAM LANE VELOCITY AT CONDENSER SECTOR NR.- 2
        V= V2
        DP1= 8.7908E-4 * DENSTM(PCOND) / QSTM * F(PCOND)*L1*V2**2/DE
        PCOND= PL1 - DP1
        PL2=   PCOND
  10    CONTINUE
C
        DO 20 J= 1, 3
        V3= .5 * CONST1 / (HFG(PCOND) * DENSTM(PCOND))
        V=  V3
        DP2= 8.7908E-4 * DENSTM(PCOND) / QSTM * F(PCOND)*L1*V3**2/DE
        PCOND= PL2 - DP2
        PL3=   PCOND
  20    CONTINUE
        DO 30 J= 1, 3
        V4= .25 * CONST1 / (HFG(PCOND) * DENSTM(PCOND))
        V=  V4
        DP3= 8.7908E-4 * DENSTM(PCOND) / QSTM*F(PCOND)*LREF*V4**2/DE
        PCOND= PL3 - DP3
```

```
         PL4=    PCOND
   30    CONTINUE
         PDM= (LREF * (PL1 + PL4) + L1 * (PL2 + PL3)) / (3 * L1)
         PCOND= PDM
         TPDM= TCOND(PCOND)
C
C * * * CALCULATION OF TUBE LENGTH IN THE PRESENCE OF NON- CONDENSABLE
C                            GASES. * * *
         FAIR= 10. + .1823E-3 * STMLD - .975E-10 * STMLD**2
C FAIR= AIR FLOW (LB/HR)
         WAWV= FAIR / STMLD
C WAWV= RATIO OF WEIGHT OF AIR-TO-WEIGHT OF STEAM AT CONDENSER INLET
C
         FLUX=  QT / (TNT * PIE * DN * EFFL)
         QTUBE= QT / TNT
         TFLUX= FLUX / TNT
C QTUBE= AVERAGE TUBE HEAT LOAD (BTU/HR)
C FLUX=  CONDENSER HEAT FLUX (BTU/HR-FT**2)
C TFLUX= AVERAGE TUBE HEAT FLUX (BTU/HR-FT**2)
C TNT=   TOTAL NUMBER OF TUBES
C
         PCOND=  PSAT
         REMAIN= QT / (QSTM * HFG(PCOND))
         WAIR=   REMAIN * WAWV
C
         DO 75 NROW= 1, ITOT
         QROW= QTUBE * NTUBES(NROW)
         WSTM(NROW)= REMAIN
C WSTM= STEAM FLOW ENTERING SPECIFIED TUBE ROW (LB/HR)
C WAIR= AIR FLOW ENTERING SPECIFIED TUBE ROW (LB/HR)
         PCOND= PDM
         PROW= PCOND
         DP= 9.427E-6 / (PCOND * 70.73) * TCOND(PCOND)**.8
C DP= COEFFICIENT OF VAPOR DIFFUSION OF STEAM IN AIR
         EW= WAIR / (WAIR + WSTM(NROW))
C EW= WEIGHT RATIO OF AIR-TO-MIXTURE
```

```
      PMPV= 1. + 0.622 * EW / (1. - EW)
C PMPV= PARTIAL PRESSURE MIXTURE-TO-PARTIAL PRESSURE VAPOR
      EV= 1. - 1. / PMPV
C EV= PA/PM: PARTIAL PRESSURE OF AIR IN BULK OF MIXTURE
      MIXVISC= ((1-EV) * STMVISC(PCOND) + 1.61*EV*AIRVISC(PCOND))
     1          / (1. + .61 * EV)
C MIXVISC= ABSOLUTE VISCOSITY OF MIXTURE (LBM/HR-FT)
      PERIM= RADIUS(NROW) * PIE * 2
      A= (PERIM - NTUBES(NROW) * OD / 12.) * TOTL
C A= FLOW AREA FOR SPECIFIED TUBE ROW (FT**2)
      REM= (WSTM(NROW) + WAIR) * OD / (12. * A * MIXVISC)
C REM= REYNOLDS NUMBEROF VAPOR-GAS MIXTURE
      IF(REM .GT. 350) THEN
      AFACT= .82
      ELSE
      AFACT= .52
      END IF
      BFACT= 18745.3  * DP*AFACT*REM**.5 *(PCOND * 70.73)**.3333
     1 * HFG(PCOND)**1.66667 / (OD * EV**.6) * (DENSTM(PCOND) /
     2 ((459.67 + TCOND(PCOND)) * QSTM))**.66667
      FLX= TFLUX * NTUBES(NROW)
      QFLUX= FLX
C
      DO 80 J= 1, 5
      DTNEW= TCOND(PCOND) - TSAT + ODTLM
      QFLUX= DTNEW / (QFLUX**.5 / BFACT**1.5 + 1 / AVGU)
   80 CONTINUE
      UNEW= 1 / (QFLUX**.5 / BFACT**1.5 + 1 / AVGU)
      LTUBE(NROW)= QROW * 12. / (PIE * DN * NTUBES(NROW) * UNEW
     1            * DTNEW)
      PCOND= PSAT
      REMAIN= (QT - QROW) / (QSTM * HFG(PCOND))
C
      IF(NROW .EQ. 1) THEN
      QZ= QT
      ELSE
```

```
      QT= QT - QROW
      END IF
C
      IF(NROW .EQ. 1) THEN
      PRESS= PDM
      ELSE
      END IF
C
      PCOND= PRESS
      VROW=  REMAIN / (DENSTM(PCOND) / QSTM * A * 3600.)
      RET=   150. * DENSTM(PCOND)/QSTM * OD * VROW/STMVISC(PCOND)
C
      IF(RET .GT. 1000.) THEN
      FT= .591 * RET**-.136 * PD**-1.137
      ELSE
      FT= 65.5 * RET**-.702 * PD**-4.257
      END IF
C
      DPT=   7.32567E-5 * FT * DENSTM(PCOND) / QSTM * VROW**2
      PRESS= PRESS - DPT
   75 CONTINUE
      SUML= 0.0
C
      DO 90 J= 1, ITOT
      LSEC= LTUBE(J) * NTUBES(J)
      SUML= SUML + LSEC
   90 CONTINUE
C
      TOTNC= SUML / TNT
      QT= QZ
      RETURN
      END
```

APPENDIX B

CONTENTS

TABLE 4- Properties of Multiple-Helix Internal Ridged Tubes

Symbol	Tube no.	Nominal diameter, in	Root diameter d_r, in	Fin count, fins/in	Outside area, ft^2/ft	ID (max) d_i, in	No. of starts
△	12	$\frac{3}{4}$	0.623	(Stripped)	0.163	0.573	5
▷	30	$\frac{3}{4}$	0.627	26.4	0.640	0.575	6
▽	22	$\frac{3}{4}$	0.625	26	0.640	0.569	6
◁	27	$\frac{3}{4}$	0.622	26.5	0.640	0.572	6
▲	28	$\frac{3}{4}$	0.624	26	0.640	0.573	5
▶	31	$\frac{3}{4}$	0.624	26.6	0.640	0.576	6
▼	29	$\frac{3}{4}$	0.625	26.1	0.640	0.575	5
◀	38	$\frac{3}{4}$	0.633	38.0	0.830	0.572	8
◇	37	$\frac{3}{4}$	0.624	38.5	0.901	0.574	10
	40	$\frac{3}{4}$	0.627	27.3	0.689	0.561	10
○	41	$\frac{3}{4}$	0.628	38.1	0.852	0.572	12
◁	9	$\frac{3}{4}$	0.628	38.5	0.901	0.575	10
□	21	$\frac{7}{8}$	0.740	26	0.640	0.684	6
■	19	$\frac{7}{8}$	0.745	(Stripped)	0.195	0.692	6
	44	$\frac{3}{4}$	0.627	41.0		0.574	6
	43	$\frac{3}{4}$	0.628	41.0		0.573	10
○	42	$\frac{3}{4}$	0.626	41.0	0.901	0.573	10
	46	$\frac{3}{4}$	0.627	41.2		0.573	10
	45	$\frac{3}{4}$	0.627	41.2		0.577	10
○	13	1	0.883	26	0.841	0.820	6
◉	32	1	0.877	26	0.871	0.825	6
◓	25	1	0.880	26	0.841	0.816	6
◔	24	1	0.878	26	0.841	0.814	6
◒	23	1	0.880	26	0.841	0.816	6
◐	26	1	0.864	27.3	0.841	0.815	6

Symbol	Internal ridging: Height e, in	Internal ridging: Pitch p, in	Internal aspect ratios: e/d_i	Internal aspect ratios: e/p	Internal aspect ratios: p/d_i	ID of envelope tube, in
△	0.0125	0.475	0.0218	0.0263	0.829	1.000
▷	0.0162	0.279	0.0282	0.0581	0.485	1.000
▽	0.0165	0.320	0.0288	0.0516	0.558	1.000
◁	0.017	0.385	0.0297	0.0442	0.673	1.000
▲	0.019	0.475	0.0332	0.0400	0.829	1.000
▶	0.0198	0.287	0.0344	0.0690	0.498	1.000
▼	0.0207	0.469	0.0360	0.0441	0.816	1.000
◀	0.018	0.212	0.0315	0.0849	0.371	1.000
◇	0.0200	0.166	0.0348	0.1205	0.289	1.000
	0.0175	0.170	0.0312	0.1029	0.303	1.000
○	0.0215	0.138	0.0376	0.1558	0.241	1.000
◁	0.0204	0.191	0.0355	0.1070	0.332	1.000
□	0.017	0.391	0.0249	0.0435	0.572	1.2348
■	0.017	0.285	0.0246	0.0596	0.412	1.000
	0.021	0.207	0.0366	0.101	0.361	–
	0.021	0.124	0.0366	0.169	0.216	–
○	0.024	0.0949	0.0419	0.253	0.166	1.000
	0.021	0.094	0.0367	0.223	0.164	–
	0.015	0.094	0.0260	0.1596	0.163	–
○	0.0178	0.333	0.0217	0.0535	0.406	1.5936
◉	0.0193	0.335	0.0234	0.0576	0.406	1.5936
◓	0.0205	0.340	0.0253	0.0603	0.420	1.5936
◔	0.0205	0.330	0.0252	0.0621	0.405	1.5936
◒	0.0205	0.338	0.0251	0.0607	0.414	1.5936
◐	0.021	0.340	0.0258	0.0618	0.417	1.5936

TABLE 5 Friction Factor Characteristics of Multiple-Helix Internal Ridged Tubes

Tube no.	m	r
12	0.762	0
30	0.64	−0.00039
22	0.697	−0.00017
27	0.72	−0.00026
28	0.72	−0.00028
31	0.61	−0.00095
29	0.68	−0.00064
38	0.61	−0.00109
37	0.54	−0.00392
40	0.627	−0.00080
41	0.57	−0.00259
9	0.59	−0.00197
21	0.70	−0.00014
19	0.626	0
44	0.53	−0.00180
43	0.55	+0.00017
42	0.58	+0.00750
46	0.54	+0.00275
45	0.52	−0.00159
13	0.63	+0.00024
32	0.70	+0.00090
25	0.645	+0.00032
24	0.64	−0.00035
23	0.637	+0.00028
26	0.64	+0.00018

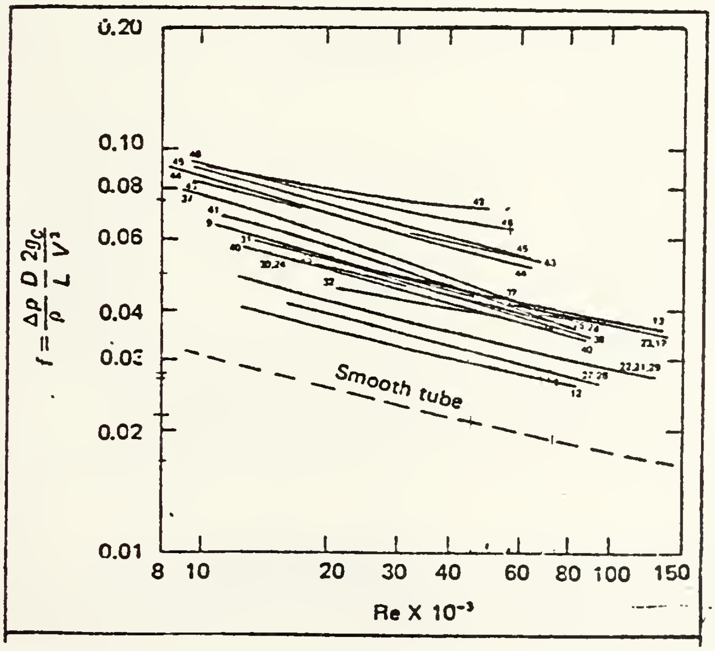


FIGURE 17 Friction Factor Curves for Multiple-Helix Internal Ridged Tubes

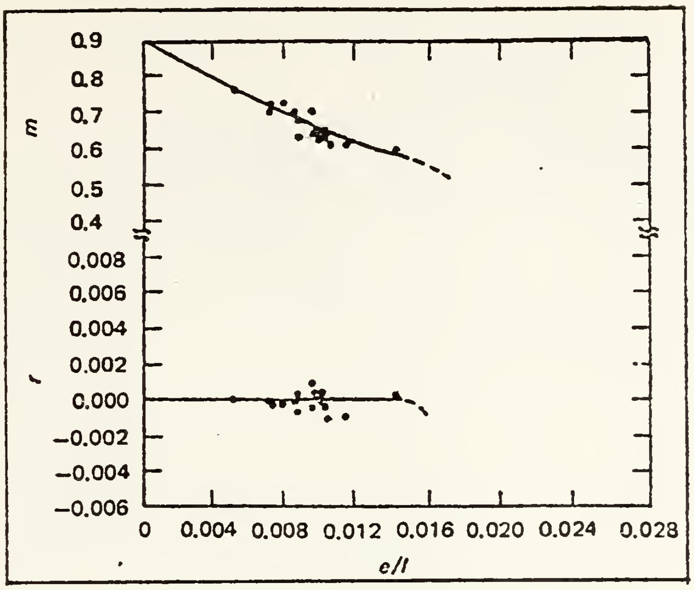


FIGURE 18a
Operands for friction factor equation versus geometric aspect ratio e/l for multiple-helix internal ridging with $p/d_i \geq 0.36$

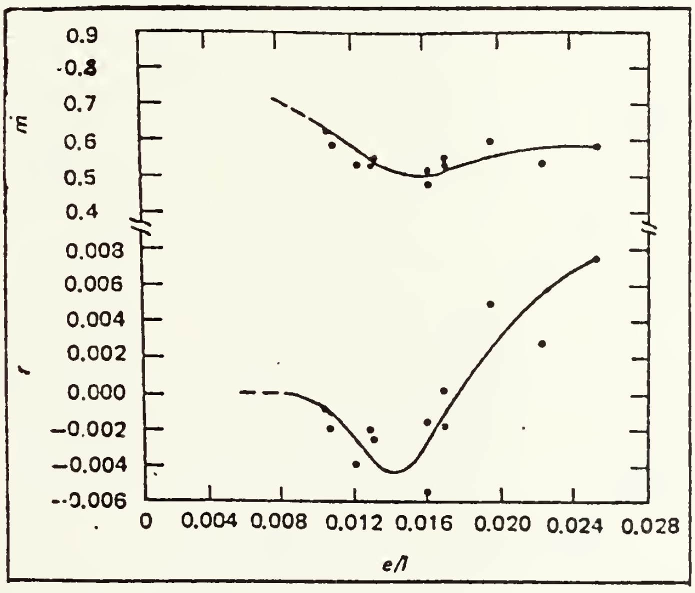


FIGURE 18b
Operand for friction factor equation versus geometric aspect ratio e/l for multiple-helix internal ridging with $p/d_i < 0.36$

FIGURE-19

WATERBOX AND TUBE END LOSSES SINGLE PASS CONDENSERS

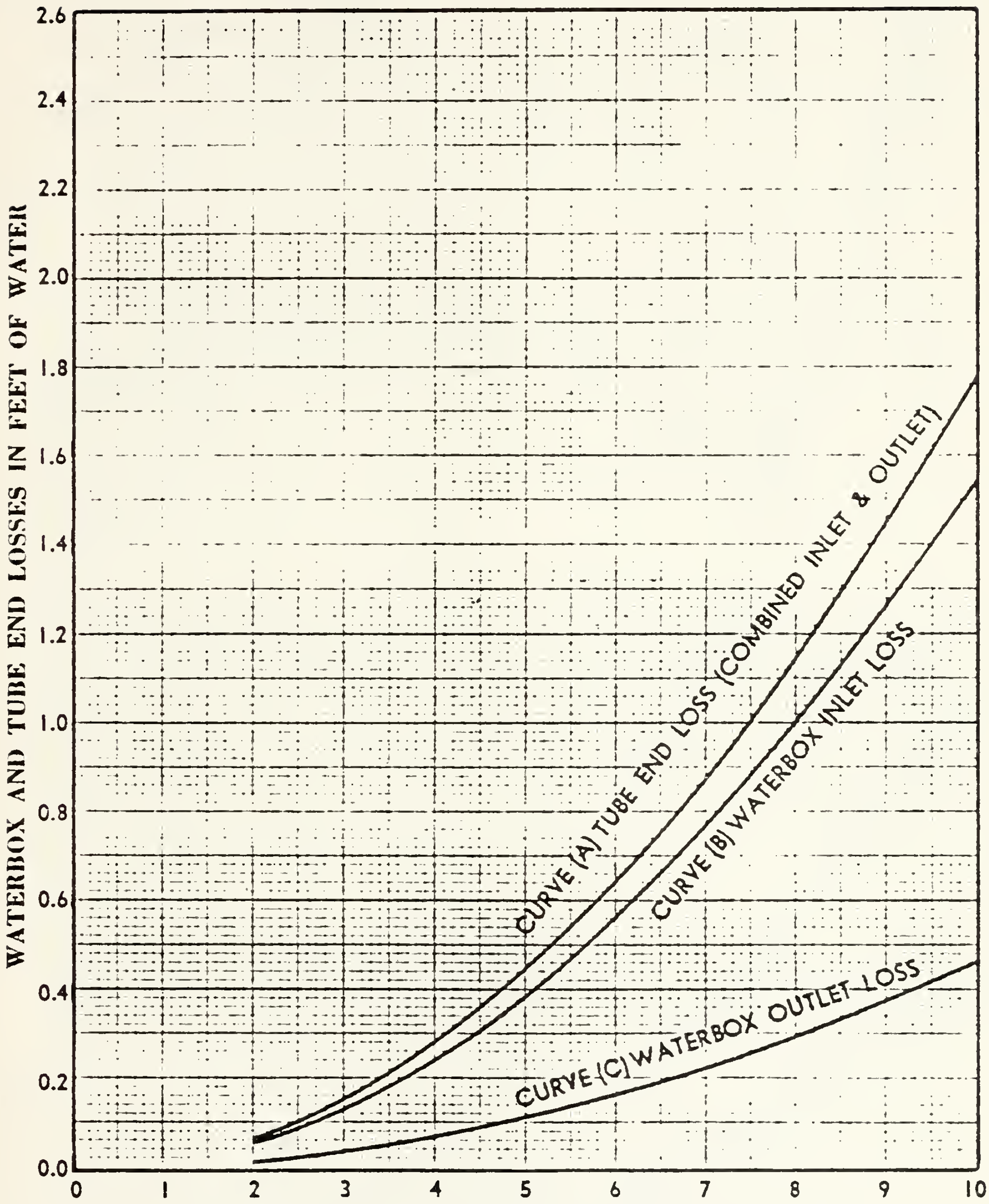
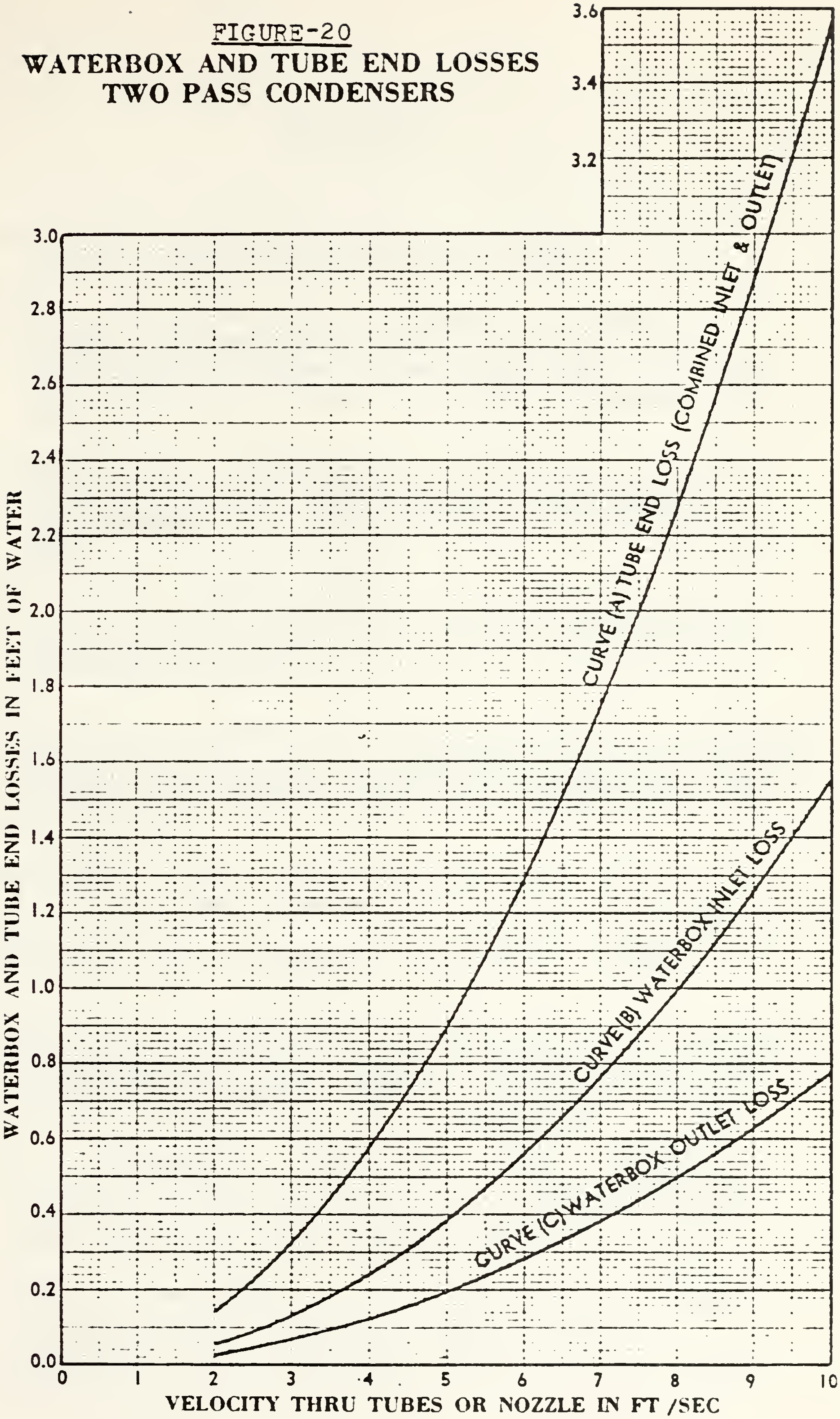


FIGURE-20

WATERBOX AND TUBE END LOSSES TWO PASS CONDENSERS



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