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A HEAT EXCHANGER OPTIMIZATION CODE

J. E. Mott, Consultant
The University of Tennessee

UNION CARBIDE CORPORATION
NUCLEAR DIVISION
OAK RIDGE GASEOUS DIFFUSION PLANT

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ABSTRACT

A Fortran IV program was developed to furnish an optimum design for a cross-flow gas cooler. It permits either single-phase or two-phase flow in any coolant passage and will accommodate multiple-pass configurations. The output of the program gives detailed physical parameters as well as cost information. A description of the heat transfer analysis, cost optimization, and design procedure, along with an example of the program's use, is presented in this report.

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A HEAT EXCHANGER OPTIMIZATION CODE

INTRODUCTION

The design of a gas cooler is a complex task which requires the examination and optimization of a wide variety of heat exchanger surfaces. Preliminary studies have shown that a poor choice of either heat transfer surface or design parameters can easily double the costs and reduce the effectiveness of the heat exchange equipment.

Where cost is the principal concern, many matrices must be examined to determine the best choice for a particular heat exchange process. Each type must be optimized for minimum cost operation, and these costs must be compared to determine the optimum core. This process is complicated by the lack of a general correlation to predict the heat transfer and pressure drop performance of the wide variety of heat transfer matrices commercially available. The project was undertaken to develop a tool for rapidly optimizing, evaluating, and comparing gas cooler designs. This report describes the analytical approach and the result of this effort.

SUMMARY

A Fortran IV program, which can be used on either the IBM 360 computer or on a remote terminal of Com-Share's SDS 940, has been developed to compute an optimum design for a cross-flow gas cooler. The program permits either single-phase or two-phase flow in any coolant passage and will accommodate multiple-pass configurations. Although it was designed for use with a plain plate-fin matrix, it can be used to provide approximate results for both tube and plate-fin and tube and circular-fin matrices.

The program, when furnished with gas and coolant inlet temperatures, outlet temperatures, mass flow rates, physical properties, and cost information, will perform one of the following functions:

1. Compute a heat exchanger design with a specified gas-side frontal area and a specified number of coolant passes,
2. Compute a heat exchanger design with a specified gas-side frontal area and an optimum (for minimum cost) number of coolant passes,
3. Compute a heat exchanger design with a specified number of coolant passes and an optimum gas-side frontal area, or
4. Compute a heat exchanger design with optimum gas-side frontal area and optimum number of coolant passes.

The output of the program gives detailed physical parameters as well as cost information. A description of the heat transfer analysis, cost optimization and design procedure, along with an example of the program's use, is presented in this report.

ANALYSIS

The design of the cross-flow heat exchanger is based on the first law of thermodynamics and on the conductance equation of heat transfer. These equations applied to the gas and to the coolant are as follows:

$$\dot{Q} = \dot{m}_g (i_{gi} - i_{go}) = \dot{m}_g c_{pg} (T_{gi} - T_{go}). \quad (1)$$

$$\dot{Q} = \dot{m}_c (i_{co} - i_{ci}). \quad (2)$$

$$= \dot{m}_c c_{pc} (T_{co} - T_{ci}) ; T_c \neq T_{sat}$$

$$= \dot{m}_c (x) (i_{fg}) ; T_c = T_{sat}.$$

$$\dot{Q} = U A F \Delta T_{lm} \quad (3)$$

where

$\dot{Q}$ = heat load, Btu/hr;

$\dot{m}_g$ = gas mass flow rate, lb/hr;

$\dot{m}_c$ = coolant mass flow rate, lb/hr;

i_g = gas enthalpy, Btu/lb;

i_c = coolant enthalpy, Btu/lb;

c_{pg} = specific heat of the gas, Btu/lb-°F;

c_{pc} = specific heat of the coolant, Btu/lb-°F;

x = coolant exit quality;

i_{fg} = coolant latent heat of vaporization, Btu/lb;

T_{gi} = gas inlet temperature, °F;

T_{go} = gas outlet temperature, °F;

T_{ci} = coolant inlet temperature, °F;

T_{co} = coolant outlet temperature, °F;

T_{sat} = coolant saturation temperature, °F;

ΔT_{lm} = log mean temperature difference,

$$= \frac{(T_{gi} - T_{ci}) - (T_{go} - T_{co})}{\ln\left(\frac{T_{gi} - T_{ci}}{T_{go} - T_{co}}\right)}; \quad (4)$$

F = counterflow correction factor;

UA = heat exchanger conductance,

$$= \frac{1}{\frac{1}{\eta_g h_g A_g} + R_m + \frac{1}{\eta_c h_c A_c}} \quad (5)$$

where

η = fin efficiency,

h = convective heat transfer coefficient,

A = surface area, and

R_m = metal thermal resistance;

and the subscript c refers to the coolant side, while the subscript g refers to the gas side.

COMPUTATION OF INDIVIDUAL PARAMETERS

With gas and coolant inlet and outlet temperatures, mass flow rates, and physical properties specified, the only unknown variables in equations 1 through 3 are UA and F . The counterflow correction factor, F , can be determined once the number of passes are specified. The conductance is therefore fixed, and the heat exchanger is thus optimized with constant conductance. Before describing the design procedure, the methods for obtaining each of the individual parameters used in equations 1 through 3 will be described.

Counterflow Correction Factor F

The counterflow correction factor must be calculated for a cross flow heat exchanger with one gas pass and a variable number of coolant passes. Factor F is defined as the ratio of the effective temperature difference in a cross flow heat exchanger to that in a counterflow heat exchanger with the same terminal temperatures. It is a function of heat exchanger geometry, number of passes, degree of mixing of each of the two fluids, and terminal temperatures.

The computation of F has been made with the assumption that the fins prevent *mixing* in the gas stream. Figure 1 shows cross sections of a plain plate-fin and a tube and plate-fin heat exchanger, along with corresponding temperature profiles. From this figure we see that the gas is essentially unmixed, i.e., $T_a \neq T_b$, but that the coolant is completely mixed, due to turbulence, over any flow channel cross section.

This is also true for the tube-and-fin exchanger provided there is only one row of tubes per pass. When more than one row of tubes is used per pass, this treatment, although no longer strictly valid, still yields a good approximation for F.

Stevens, Fernandez, and Woolf* have presented relations for the effectiveness of one-, two-, and three-pass cross-flow heat exchangers. For the flow arrangement described above, they are as follows:

$$E_c = 1 - \exp(-Z_g K) \quad (\text{one pass}), \quad (6a)$$

$$E_c = 1 - \frac{1}{[K/2 + (1 - K/2) \exp(2Z_g K)]} \quad (\text{two passes}), \text{ and} \quad (6b)$$

$$E_c = 1 - \frac{1}{(1 - K/2)^2 \exp(3Z_g K) + [K(1 - K/4) - K^2 Z_g (1 - K/2)] \exp(Z_g K)} \quad (6c)$$

(three passes)

where

$$E_c = \frac{T_{co} - T_{ci}}{T_{gi} - T_{ci}}$$

*Stevens, R. A., Fernandez, J., and Woolf, J. R., "Mean Temperature Difference in One-, Two-, and Three-Pass Cross-Flow Heat Exchangers," *ASME Transactions*, 11, pp. 287-293, January 1957.

Figure 1
TEMPERATURE DISTRIBUTION IN CROSS-FLOW HEAT EXCHANGER MATRICES

$$Z_g = \frac{T_{co} - T_{ci}}{T_{gi} - T_{go}},$$

$$K = 1 - \exp(-NTU),$$

$$NTU = \frac{U a_g}{\dot{m}_g c_{pg}}, \text{ and}$$

$$a_g = \text{area per pass} = A_g/n$$

Since F is defined by $F = Q/UA\Delta T_{lm}$, it can be written as

$$F = \frac{1}{n \times NTU(1 - Z_g)} \ln\left(\frac{1 - E_c}{1 - E_c/Z_g}\right) \quad (7)$$

For this design, all the terminal temperatures are known, and one must use a trial-and-error procedure to obtain the value of NTU used to evaluate F .

When more than three passes are used, it is assumed that the exchanger is essentially counterflow, and F is set equal to one. F is also equal to one when the coolant changes phase throughout a pass.

Fin Efficiency

The efficiency of a fin is defined as the ratio of the rate of heat actually transferred by the fin to the rate of heat transferred by the fin when the entire fin surface is held at the root temperature.

The efficiency of a circular fin with root radius r_o , fin radius r_f , thickness δ , and thermal conductivity k is given by*

$$\eta_f = \frac{2r_o[I_1(mr_f) K_1(mr_o)/K_1(mr_f) - I_1(mr_o)]}{m(r_f^2 - r_o^2)[I_1(mr_f) K_o(mr_o)/K_1(mr_f) + I_o(mr_o)]} \quad (8)$$

*See appendix A for derivation.

where $m = 2h/k\delta$ and I and K are the modified Bessel functions of the first and second kind. Equation 8 is used not only for the circular fins shown in figure 2a, but also to approximate the fin efficiency of the tube and plate-fin matrix, shown in figure 2b, by introduction of a fictitious fin radius defined by

$$\pi r_f^2 = S_L S_T.$$

The efficiency of the fins in the plate-fin arrangement, shown in figure 2c, is given by the following equations:

$$\eta_f = \frac{k_f \delta m}{h(b + p)} \left[\frac{H + \tanh(mb/2)}{1 + H \times \tanh(mb/2)} \right], \quad (9)$$

where

$$H = \frac{k_s}{k_f} \frac{s}{\delta} \frac{n}{m} \tanh\left(\frac{np}{2}\right),$$

$$m = \sqrt{2h/k_f \delta},$$

$$n = \sqrt{2h/k_s s},$$

k_f = fin thermal conductivity, and

k_s = splitter thermal conductivity.

Since this fin efficiency is used only on the finned surface, the values given by equations 8 and 9 are modified for use in equation 5 as follows

$$\eta A_{\text{total}} = A_{\text{total}} - A_{\text{fin}} + \eta_f \cdot A_{\text{fin}}.$$

Gas-Side Heat Transfer Coefficients and Friction Factors

Since there is no known relation which correlates the gas-side heat transfer coefficient and friction factor data for a variety of heat transfer matrices, the following approach was taken. The literature was scanned for heat exchange configurations with published experimental values of the Colburn j factor ($j = h/Gc_p PR^{2/3}$) and Fanning friction factor ($f = \tau/\rho v^2/2$) as a function of Reynolds number. Those which appeared to have promise were collected, their physical dimensions characterized (hydraulic diameter, volume/surface area ratio, and flow area/frontal area ratio) and the experimental j and f values were fitted to curves of the form

Figure 2a
TUBE AND CIRCULAR-FIN MATRIX

Figure 2b
TUBE AND PLATE-FIN MATRIX

Figure 2c
PLAIN PLATE-FIN MATRIX

$$j = \beta \text{Re}^{-q}$$

$$R_{\min} \leq \text{Re} \leq R_{\max} \quad (10)$$

$$f = \alpha \text{Re}^{-\ell}$$

These parameters were then stored and used to provide heat transfer coefficients and friction factors when needed.

Gas-side expansion and contraction pressure losses were neglected in the tube-and-fin cores, since each tube row consists of a contraction and expansion, and the frictional behavior of the first and last rows of tubes is not significantly different from interior rows. The entrance and exit losses are thus already accounted for in the friction factor data. For the plate-fin matrices, the expansion and contraction coefficients k_e and k_c given by Kays and London* were used to compute these pressure losses.

Single-Phase Coolant Heat Transfer Coefficients and Friction Factors

On the coolant side, the heat transfer coefficient was obtained from the Colburn equation:

$$\frac{h_i}{c_{pc} G_c} = 0.023 \left(\frac{G_c D_H}{\mu_c} \right)^{-0.2} \text{PR}_c^{-2/3}, \quad (11)$$

where G_c , μ_c , and PR_c are the mass velocity, viscosity, and Prandtl number of the coolant and D_H is the hydraulic diameter. No attempt was made to account for temperature-dependent properties, since most of the temperature drop occurs on the gas side. Also, there were no corrections made to account for an entrance region effect, since L/D is large for the coolant passages in most gas coolers. Smooth tube values of the Fanning friction factor were used, and these data fit the equation

$$f = 0.046 \left(G_c D_H / \mu_c \right)^{-0.2}, \quad (12)$$

well over the Reynolds number range from 10^4 to 10^6 .

A value of 0.5 was used as the entrance pressure loss coefficient and 1.0 as the exit pressure loss coefficient for each coolant pass. These values belong strictly to an infinite area ratio and infinite Reynolds number, and thus are larger than those values anticipated in a heat exchanger header, yielding a slightly conservative pressure loss.

*Kays, W. M. and London, A. L., *Compact Heat Exchangers*, 2ed., McGraw-Hill Book Co., New York, New York (1955).

Two-Phase Coolant Heat Transfer Coefficients and Friction Factors

The heat transfer coefficients used for boiling heat transfer were obtained from Chen's work*. Chen postulated that the heat transfer coefficient could be *decomposed* into two parts, one due to boiling, the other due to forced convection, i.e.,

$$h = h_{\text{boil}} + h_{\text{fc}}, \quad (13)$$

where

$$h_{\text{fc}} = 0.023 F_R (\text{Re})^{0.8} \text{PR}_f^{0.4} k/D_H$$

with $\text{Re} = D_H G_c (1-x)/\mu_f$ and with F_R an empirical Reynolds number correction factor, which is a function of the Martinelli parameter $X_{tt} \left[= \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_f} \right)^{0.5} \left(\frac{\mu_f}{\mu_g} \right)^{0.1} \right]$. Here the subscripts f and g refer to the saturated liquid and vapor, respectively.

The value of h_{boil} was obtained from the Forster and Grief pool boiling correlation. This value was modified to account for the temperature gradient between the wall and saturation temperature. Since this gradient is steeper in forced convection, the mean superheat of the fluid surrounding the bubble may be considerably less than the wall superheat. This reduces bubble growth and in effect, suppresses the boiling heat transfer. This suppression factor S decreases from unity to zero as the Reynolds number goes from zero to infinity. The term h_{boil} is given by

$$h_{\text{boil}} = S \left[\frac{(q/A)_{\text{Forster-Grief}}}{T_w - T_{\text{sat}}} \right] \quad (14)$$

The empirical values S and F_R are functions of Reynolds number and the Martinelli parameter X_{tt} , respectively.

The boiling pressure drop has three components: friction, elevation, and expansion. Since the coolant goes from a region of approximately zero velocity to another low-velocity region, the net acceleration pressure drop is essentially zero.

The three components of the pressure drop were computed following the procedure of Martinelli and Nelson** who assumed

*Chen, J. C., *A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow*, ASME Paper 63-HT-34 (1963).

**Martinelli, R. C., and Nelson, D. B., *ASME Trans*, 70, p. 695 (1948).

1. The void fraction, α , and the empirical parameter $\phi_{gtt}^2 \left[= \left(\frac{dp}{dz} \right)_{TPF} / \left(\frac{dp}{dz} \right)_{GPF} \right]$ were functions of the Martinelli parameter only.
2. That data obtained from horizontal flow of air and various liquids could be used to determine this functional relationship.

Using the data of Martinelli and Lockhart*, a two-phase multiplier R was obtained and the two-phase frictional pressure drop computed from the relation

$$\Delta p = 4Rf \frac{L}{D_H} \frac{G_c^2}{2\rho_f g_c} \quad (15)$$

where, for a given fluid, R is a function of exit quality and critical pressure ratio p/p_c only.

Using the value of the void fraction, α , at the exit obtained from X_{tt} , the expansion pressure loss may be obtained from

$$\Delta p = \frac{G^2}{2g_c} \frac{\left[\frac{x^3 v_g^2}{\alpha^2} + \frac{(1-x)^3 v_f^2}{(1-\alpha)^2} \right]}{[xv_g + (1-x)v_f]} \quad (16)$$

The static pressure change is given by

$$\Delta p = R_3 \rho_f \left(\frac{g}{g_c} \right) L \sin \theta, \quad (17)$$

where θ is the angle between the flow direction and the horizontal and

$$R_3 = 1 - (1 - v_f/v_g)(1/x_e) \int_0^{x_e} \alpha \, dz. \quad (18)$$

We now have methods available to compute each of the individual parameters needed in equations 1 through 5. The next task is to utilize these parameters in producing the *best* heat exchanger design to perform a given duty.

*Lockhart, R. W. and Martinelli, R. C., *Chem. Engr. Prog.*, 45, p. 39 (1949).

DESIGN PROCEDURE

In the following design procedure, the gas-side frontal area is specified either by the designer or by the optimization routine. It is assumed throughout that this frontal area is square, and no attempt has been made to design gas coolers with rectangular gas-side frontal areas. Secondly, once the frontal area is determined, the cooler thickness is also fixed. For the tube-and-fin heat exchangers, this thickness is then increased to a thickness that contains an integral number of tube rows without changing the frontal area. This additional thickness provides more surface area than required and represents an oversized exchanger. With the plate-fin matrix, the coolant channel width is variable and may match the cooler thickness.

A principal restriction on the choice of optimum heat exchangers is the lack of cost information on available heat transfer surfaces. Thus, even with the best design procedure, one may produce a poor heat exchanger because of poor cost information. In this program the heat exchanger cost is divided into two components; one a function of heat exchanger gas-side surface area, the other a function of gas-side pumping power. Coolant pumping costs were included in the total yearly cost but were not included in the optimization procedure since they seem to contribute little to the total cost and are rarely a strong function of frontal area. Other considerations such as header perimeter, shell surface area, weight, and structure were not included in this optimization since information on the effects of these parameters on heat exchanger cost was not available. Thus all capital costs were expressed as functions of gas-side surface area.

OPTIMIZATION

In the following discussion, optimization implies minimum yearly cost, where the capital cost is amortized at some specified rate, and the coolant pumping and gas compression operating costs are specified as functions of power. In optimizing a heat exchanger, the number of passes in the sensible heat section is specified and a minimum cost exchanger with this number of passes is found. If a range of number of passes is allowed, each is optimized and that one with the lowest cost is chosen as the optimum.

The cost equation to be minimized is

$$\text{Cost} = C_A A^R + C_P P + C_P^* P^*, \quad (19)$$

with

A = gas side surface area,

P = gas compressor power, and

P^* = coolant pump power (assumed constant during optimization)

$$\frac{d\text{Cost}}{dZ} = C_A R A^{R-1} \frac{dA}{dZ} + C_P \frac{dP}{dZ} = 0, \quad (20)$$

where Z is some yet unspecified quantity. It is convenient at this point to introduce E , the *flow friction power* defined as the amount of power expended on the gas per unit of gas-side surface area. E can be readily evaluated as a function of Reynolds number, friction factor, and fluid properties from

$$E = \frac{1}{2g_c} \frac{\mu^3}{\rho^2} \left(\frac{1}{D_H}\right)^3 f \text{Re}^3. \quad (21)$$

Since h can be written as

$$h = \frac{C_P \mu}{D_H} j \text{Re} \text{Pr}^{-2/3}, \quad (22)$$

j , f , and Reynolds number may be manipulated to obtain h as a function of E for a given set of gas conditions.

Such a function is shown in figure 3 for two surfaces.

This is a particularly informative way to compare surfaces since for a given heat transfer duty, $\dot{Q}$, area, A , and log mean temperature difference ΔT_{lm} , both surfaces must have the same value of h . We see that for the same h , surface 2 has a much larger value of E , and since the power requirement is $P = E \times A$, surface 2 requires a great deal more power to accomplish the same duty as 1.

Utilizing j and f values from equation 10, equations 21 and 22 may be manipulated to obtain

$$h = \gamma E^P. \quad (23)$$

Using the flow friction power, E , as the dependent variable Z in equation 19 and substituting $P = A \cdot E$ one obtains

$$C_A R A^{R-1} \frac{dA}{dE} + C_P \frac{d}{dE} (A \cdot E) = 0. \quad (24)$$

Decomposing A into A_l , the area through which latent heat is transferred, and A_s , the sensible heat area, and noting that

$$\frac{dA_i}{dE} = \frac{Q_i}{F_i \Delta T_{lmi}} \frac{d}{dE} \left(\frac{1}{U_i}\right) = - \frac{Q_i}{F_i \Delta T_{lmi} \eta h^2} \frac{dh}{dE} = - \frac{Q_i}{F_i \Delta T_{lmi} \eta_i \gamma E^{1+P}},$$

Figure 3
HEAT TRANSFER AND FRICTIONAL POWER
REQUIREMENT CHARACTERISTICS
Comparison of Two Heat Exchanger Surfaces
on a Unit of Surface Area Basis

Equation 24 becomes

$$(C_A R A^{R-1} + C_p E) \left(\frac{Q_L}{F_L \Delta T_{LmL}} + \frac{Q_S}{F_S \Delta T_{LmS}} \right) \frac{P}{\eta \gamma E^{1+P}} = C_p A. \quad (25)$$

Equation 25 is solved iteratively to obtain the value of E which yields the lowest cost operation. Once E is specified, the values of h, j, Re, G, and flow area follow immediately. The heat transfer area, compressor power, and their associated costs, as well as the heat exchanger thickness, are now readily calculated. The heat exchanger design is now completely fixed for the number of passes specified in the sensible heat portion of the exchanger. (Because $F = 1$ in the latent heat section, there is little incentive in varying the number of passes in the boiling section.) The number of passes in the sensible heat section may now be changed and the entire optimization procedure repeated to find the optimum exchanger.

Performance Evaluation

The optimization may be by-passed and the frontal area specified if the performance of a given heat exchanger design is to be evaluated. In this way, several heat exchangers could be examined which would fit in a given space and their relative costs evaluated. There is still some optimization allowed in varying the number of passes.

PROGRAM

A complete listing of the program is presented in appendix B. The program is used as a subroutine, entitled HEX, in a user supplied program which fixes the basic design requirements of the heat exchanger. The user must not only supply the calling routine, but must furnish subroutines to calculate gas and coolant properties. The following section concerning problem setup describes these programs.

The flow diagram for subroutine HEX is shown in figure 4. Certain parameters are arbitrarily set in this subroutine. They are as follows:

1. Cost of coolant pumping power (CPP) = 6.5 mills/kwhr,
2. Coolant pump efficiency = 0.75,
3. Tube wall or plate wall thickness = 0.035 in.,
4. Metal thermal conductivity (con) = 120 Btu/hr-ft-°F, and
5. Maximum number of sensible heating passes = $\begin{cases} 5 & \text{(plate-fin)} \\ 10 & \text{(tube-fin)} \end{cases}$

Figure 4

FLOW DIAGRAM FOR SUBROUTINE HEX

PROBLEM SETUP

In order for the user to run subroutine HEX, he must prepare heat exchanger characteristic data, furnish coolant and gas properties, and set up a calling routine that will supply the pertinent data to the subroutine. In the following, the method used for the remote terminal version is described.

Heat Exchanger Surface Data

It is necessary to present the physical description of the heat exchanger matrix along with appropriate j and f versus Reynolds number data in a form usable by HEX. Subroutine /CNEW/ was written for this purpose. A listing of this subroutine is presented in figure 5. In the remote terminal version, data prepared by /CNEW/ is transferred to data file PØRE where it is permanently stored for future use in HEX. PØRE currently contains data for the matrices described by Pearson and Brock*.

Gas Properties

The user must supply a subroutine which will furnish values of specific heat at constant pressure, CPU (Btu/lb-°F); viscosity, GMU (lb/ft-sec); thermal conductivity, GKU (Btu/hr-ft-°F); gas density, RØ (lb/cu ft); Prandtl number, PR and PR3 ($= PR^{1/3}$) evaluated at a reference temperature chosen by the user.

Coolant Properties

A subroutine entitled SFP must be furnished to compute the physical properties of the coolant, it must be written in the form SUBROUTINE SFP(I,TSAT,AK,RHØF,CC,PV,PCR,UF,UG,HFG,HRØG,PRAN),

where

I = FLAG = 0 or 6 (Input);
 TSAT = saturation temperature, °F (Input);
 AK = thermal conductivity, Btu/hr-ft-°F;
 RHØF = density of the saturated liquid, lb/cu ft;
 CC = specific heat, Btu/lb-°F;
 RHØG = density of the saturated vapor, lb/cu ft;
 PV = vapor pressure corresponding to TSAT, psia;
 PCR = critical pressure (I=6), psia;
 PCR = surface tension (I=0), lb_f/ft;
 UF = viscosity of the saturated liquid, lb/ft-sec;
 UG = viscosity of the saturated vapor, lb/ft-sec;

*Report under preparation by J. T. Pearson and W. R. Brock, Union Carbide Corporation, Nuclear Division, Oak Ridge Gaseous Diffusion Plant, Oak Ridge, Tennessee.

```

      OPEN(2,OUTPUT,/CORE/)
1  WRITE(1,200)
200 FORMAT($ INPUT M AND N, WHERE M=1 FOR PLATE AND FIN SURFACES$
1/$ M=2 FOR TUBE AND PLATE FINS AND M=3 FOR TUBE AND CIRCULARS$
2$ FIN$,$/ $ N IS THE SURFACE IDENTIFIER$,$/ $ ALL DIMENSIONS$
3$ ARE IN INCHES$)
      ACCEPT(M,N)
      IF(M-2)2,3,5
5  WRITE(1,205)
205 FORMAT($INPUT LONGITUDINAL PITCH, TRANSVERSE PITCH, TUBE O.D., $
1$ FIN THICKNESS, $/$NO. OF FINS/IN., FIN O.D.$)
      ACCEPT(SL,ST,D0,T,FN,DF)
      SL=SL/12
      ST=ST/12.
      D0=D0/12
      S=D0
      DF=DF/12
      T=T/12
      W=1./12./FN
      AF=(ST-D0)*W-(DF-D0)*T
      AT=3.14159*((DF*DF-D0*D0)/2.+D0*(W-T))
      DH=4.*AF*SL/AT
      ALPHA=AT/W/ST/SL
      FAT=AT/3.14159/D0/W
      RE=DF/2
      M=2
      GO TO 4
2  WRITE(1,201)
201 FORMAT($INPUT PLATE SPACING, FIN THICKNESS, NO. OF FINS/IN., $
1$ SPLITTER THICKNESS$)
      ACCEPT(B,T,FN,S)
      B=B/12
      T=T/12.
      W=1./12./FN
      S=S/12
      AF=(B-S)*(W-T)
      AT=2.*(B-S)+4.*(W-T)
      DH=4.*AF/AT
      RE=0
      ALPHA=AT/B/W
      FAT=AT/2./W
      SL=B
      ST=W
      GO TO 4
3  WRITE(1,203)
203 FORMAT($INPUT LONGITUDINAL PITCH, TRANSVERSE PITCH, TUBE O.D., $
1$ FIN THICKNESS,$/$NO. OF FINS/IN.$)
      ACCEPT(SL,ST,D0,T,FN)
      SL=SL/12.
      ST=ST/12.

```

Figure 5
SUBROUTINE /CNEW/

```

*      D0=D0/12.
      T=T/12.
      W=1./FN/12.
      S=D0
      AF=(ST-D0)*(W-T)/2.
      AT=2.*(SL*ST-.7854*D0*D0)+3.14159*D0*(W-T)
      DH=8.*AF*SL/AT
      ALPHA=AT/SL/ST/W
      FAT=AT/3.14159/D0/W
      RE=SQRT(SL*ST/.7854)/2
4  CONTINUE
      WRITE(1,204)
204  FORMAT($INPUT RE(MIN),J(RE(MIN)),F(RE(MIN)),RE(MAX),J(RE(MAX)),S
1$F(RE(MAX))$)
      ACCEPT(R1,AJ1,F1,R2,AJ2,F2)
      E1=3.87E-18*F1*(R1**3)/(DH**3)
      E2=3.87E-18*F2*(R2**3)/(DH**3)
      H1=5.33E-03*AJ1*R1/DH
      H2=5.33E-03*AJ2*R2/DH
      Q=ALOG(AJ1/AJ2)/ALOG(R2/R1)
      P=ALOG(H1/H2)/ALOG(E1/E2)
      BETA=AJ1*(R1**Q)
      GAM=H1/(E1**P)
      WRITE(2,100)N,M,DH,ALPHA,FAT,SL,ST,T,P,Q,BETA,GAM,R1,R2,S,RE
100  FORMAT(2I2,14E10.4)
      DISPLAY(N,M,DH,ALPHA,FAT,SL,ST,T,P,Q,BETA,GAM,R1,R2,S,RE)
      GO TO 1
      END

```

Figure 5
SUBROUTINE /CNEW/ (Con't)

HFG = latent heat, Btu/lb; and
PRAN = Prandtl number.

Calling Routine

A main program must be written which will specify the input quantities required by HEX. It requires a common statement: COMMON. CPU,GMU,GKU,PR,PR3,RØ,RHØF,RHØG,CC,PV,PCR,UF,UG,HFG,PRAN,AK, where all of the quantities are defined by the gas and coolant property subroutines and a call statement,

```
CALL HEX(FLØ,CFLØ1,CFLØ2,X,TGI,TGØTI,TCØR,NPAS,CA,CP,N,AMØRT).
```

These variables are as follows:

FLØ = gas flow rate, lb/sec;
CFLØ1 = coolant flow rate in sensible heating section, lb/sec;
CFLØ2 = coolant flow rate in the boiling section, lb/sec;
X = exit quality;
TGI = gas inlet temperature, °F;
TGØ = gas outlet temperature, °F;
TI = coolant inlet temperature, °F;
TCØ = coolant outlet temperature, °F;
NPAS = minimum number of passes allowed in the sensible heat section;
N = surface type;
AMØRT = amortization factor;
CP = power cost, \$/kwyr;
CA = heat exchanger unit cost when the HX surface cost is given by cost = CA * A^R, and
R = exponent in the HX surface cost.

Example

A listing of a sample program along with the required subroutines is presented in figures 6 and 7. This program is used to design a heat exchanger using surface 7, a tube and plate-fin surface, to cool 20 lb/sec of exhaust gas from a turbine from 550 to 300°F while heating Freon-114 from 100 to 150°F and boiling 10% of this coolant. Figure 8 presents the dialog and program output. The user was first asked whether an optimization on the frontal area was desired. If his answer had been *no*, then the desired frontal area would have been typed at that point. He was then asked for the range of boiling passes and for the number of rows in the sensible heat section. The program then asked if he wanted an optimization on number of sensible heat passages. An affirmative answer caused the program to examine all numbers of passes from 1 (N1 = 1 in main program) to 10. Had the answer been *no*, the program would only have evaluated the case of one sensible heat pass. The results of this computation, shown in figure 8, are self-explanatory.

```

COMMON CPU,GMU,GKU,PR,PR3,R0,RH0F,RH0G,CC,PV,PCR,UF,UG,
.HFG,PRAN,AK
TGI=550
TG0=300
FL0=20
X=.1
TI=100
TC0=150
N1=1
N=7
AM0RT=.1457
CA=.75
R=1
CP=6.5*8.76
PRES=14.7
TR=(TGI+TG0)/2
CALL GAS(TR,PRES,GMU,CPU,GKU,PR,PR3,R0)
CALL SFP(6,TC0,AK,RH0F,CC,PV,PCR,UF,UG,HFG,RH0G,PRAN)
QT=FL0*CPU*(TGI-TG0)
CFL01=QT/(CC*(TC0-TCI)+X*HFG)
CFL02=CFL01
Q1=CFL01*CC*(TC0-TCI)
Q2=CFL02*X*HFG
CALL HEX(CFL0,CFL01,CFL02,X,TGI,TG0,TI,TC0,R,N1,R1,R2,
.CA,CP,N,AM0RT)
STOP
END

```

```

SUBROUTINE GAS(TR,PRES,GMU,CPU,GKU,PR,PR3,R0)
R0=PRES*144/53.35/(TR+460)
GMU=1.22E-5
CPU=.241
GKU=1.6E-2
PR=.7
PR3=PR**(1/3)
RETURN
END

```

Figure 6

EXAMPLE OF PROGRAM UTILIZING SUBROUTINE HEX
Subroutine GAS Furnishes Physical Properties of Gas Used

```

SUBROUTINE SFP(IFC,T,AK,DENSL,CP,VP,ST,VISL,VISV,HV,DENSV,PR)
  K=1
  IF(IFC-4)3,1,6
6  IFC=0
  K=4
  GO TO 3
1  IFC=0
  T=4986.8292/(12.74008-ALOG(VP))-460.
3  IF(IFC-3)5,10,5
5  TC=(T-32.)/1.8
  TK=TC+273.16
  DENSL=96.12-.05159*T-.0002674*T*T
  AK=.043842-8.1579E-05*T
  HV=55.3692+.0109428*T-.00043599*T*T
  A=(T-80.)/120.
  VISL=2.42*EXP[1706.891/(T+460.)-4.17747]
  VISV=2.42*EXP(-3.36021-603.0366/(T+460.))
  CP=.22126+.0002835*T
  IF(IFC)10,10,50
10 VP=EXP(12.74008-4986.8292/(T+460.))
  IF(IFC-3)15,50,15
15 ST=.0008907*((294.3-T)/217.3)**1.2
  CP=.22126+.0002835*T
  DENSV=8.8432*(VP/TK)
  TR=T+459.6
18 A=(.0000612*TR-.0197)*DENSV*DENSV
  B=(.002819*TR-3.789)*DENSV
  C=.06279*TR
  PHI=DENSV*(A+B+C)-VP
  IF(ABS(PHI)-.0001)40,40,20
20 DER=3.*A+2.*B+C
  DENSV=DENSV-PHI/DER
  GO TO 18
40 PR=CP*VISL/AK
  IF(K-3)50,50,51
51 ST=474.
50 CONTINUE
  RETURN
  END

```

Figure 7

TYPICAL SUBROUTINE TO FURNISH PHYSICAL PROPERTY DATA FOR HEX

```

*****
DO YOU WANT TO OPTIMIZE? YES
NUMBER OF BOILING PASSES WILL VARY FROM 1
TO 1
NUMBER OF PASSES PER ROW IN THE SENSIBLE SECTION = 1
DO YOU WANT TO OPTIMIZE THE NUMBER OF PASSES? YES
*****
TUBE-FIN HEAT EXCHANGER TYPE 7
WIDTH = 6.58
LENGTH = 6.76
THERE ARE 1 PASSES WITH 2. ROWS PER PASS IN THE SENSIBLE
HEAT SECTION AND 1 PASSES WITH 1 ROWS PER PASS IN THE
BOILING SECTION. THERE ARE 79. TUBES PER ROW
HX AREA = 1729.55
FRONTAL AREA = 44.48
COOLER THICKNESS, INCHES = 2.5992
TH1 = 1.7328 INCHES
TH2 = 0.8664 INCHES
*****
SINGLE PHASE FLOW RATE (LB/SEC) = 27.21
SINGLE PHASE DELP = 0.675
SINGLE PHASE VELOCITY = 3.4343
SINGLE PHASE REYNOLDS NO. = .46E+05
TWO PHASE FLOW RATE (LB/SEC) = 27.21
TWO PHASE DELP = 8.870
TWO PHASE VELOCITY = 6.9555
TWO PHASE REYNOLDS NUMBER = .94E+05
PUMP POWER = 0.824 HORSEPOWER
SINGLE PHASE COEFFICIENT = 241.3637
TWO PHASE COEFFICIENT = 1249.9789
*****
FIN EFFICIENCY = 0.9132
MASS VELOCITY, G = 0.8393
GAS REYNOLDS NO. = .82E+03
PRESSURE DROP, PSI = 0.0064
COMPRESSOR POWER REQUIRED = 0.75 HP
GAS SIDE HEAT TRANSFER COEFFICIENT = 12.4306
TGP = 523.3556
*****
HEAT TRANSFERED = 1704.53 HORSEPOWER
U1 = 7.1818
U2 = 10.2281
*****
AMORTIZATION FACTOR = 0.1457
GAS SIDE PUMPING COST = 31.95
COOLANT PUMPING COST = 32.31
CAPITAL COST = 1297.16 DOLLARS
YEARLY HEAT EXCHANGER COST = 189.00
YEARLY COST = 253.26

```

Figure 8

REMOTE CONSOLE DIALOG AND PROGRAM OUTPUT FOR SAMPLE PROGRAM
 Sample Program Is Illustrated in Figure 6

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APPENDIX A
FIN EFFICIENCY

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FIN EFFICIENCY

As a result of conduction along a fin, the fin temperature approaches the fluid temperature as the distance from the fin's base is increased. The temperature drop between the root and the tip of the fin reduces the fin efficiency, since such a drop reduces the mean effective temperature difference between the heat transfer surface and the fluid. The fin efficiency, η_f , has been defined as the ratio of the actual heat transferred by the fin to that transferred by an isothermal fin with the same root temperature.

A more convenient parameter is the efficiency of the heat transfer surface, η , which is defined by

$$\eta = 1 - (1 - \eta_f) A_f/A, \quad (\text{A-1})$$

where A_f/A is a fraction of the heat transfer area contributed by the fins.

Expressions for the efficiency of circular fins and plate fins are derived in this appendix subject to the following assumptions:

1. The fin thickness, δ , is much less than the fin length, L ;
2. There is no heat transfer from the fin tip;
3. The root of the fin is at constant temperature, T_0 ;
4. The fluid surrounding the fin is at constant temperature, T_∞ ; and
5. The heat transfer coefficient, h , is constant.

A heat balance on the fin section shown in figure A-1 may be written as

$$q(x)A(x) = q(x+dx) A(x+dx) + 2 P(x)dx h(T - T_\infty),$$

which leads to

$$k \frac{d}{dx} \left[A(x) \frac{dT}{dx} \right] = h P(T - T_\infty) \quad (\text{A-2})$$

Circular Fins

Applying equation A-2 to the circular fin shown in figure A-2 leads to the following differential equation:

Figure A-1
FIN SECTION

Figure A-2
CIRCULAR FIN

$$r^2 \frac{d^2\theta}{dr^2} + \frac{d\theta}{dr} = m^2 r^2 \theta = 0, \quad (A-3)$$

$$\text{with } \theta = \frac{T - T_\infty}{T_0 - T_\infty} \text{ and } m = \sqrt{\frac{2h}{k\delta}}$$

The solution to equation A-3 subject to boundary conditions

$$1. \quad \theta(r_0) = \theta_0 = 1$$

$$2. \quad \frac{d\theta}{dr}(r_e) = 0$$

is

$$\theta = \frac{I_0(mr) K_1(mr_e) - I_1(mr_e) K_0(mr)}{I_0(mr_0) K_1(mr_e) - I_1(mr_e) K_0(mr_0)}$$

where I and K are the modified Bessel functions of the first and second kind.

The fin efficiency is given by

$$\eta_f = \frac{-k 2\pi r_0 \frac{dT}{dr}(r = r_0)}{2h(\pi r_e^2 - \pi r_0^2)(T_0 - T_\infty)},$$

and becomes

$$\eta_f = \frac{2r_0}{m(r_e^2 - r_0^2)} \frac{I_1(mr_e) K_1(mr_0) - I_1(mr_0) K_1(mr_e)}{I_0(mr_0) K_1(mr_e) - I_1(mr_e) K_0(mr_0)}. \quad (A-4)$$

For the case of the tube and plate fin shown in figure A-3, the efficiency can be approximated by equation A-4 if the effective radius is given by

$$r_e = \sqrt{\frac{S_L S_T}{\pi}}$$

Plain Plate-Fin Surfaces

The assumptions previously discussed are valid for the geometry shown in figure A-4.

Figure A-4

PLAIN PLATE-FIN SURFACE GEOMETRY

Equation A-2 is valid for both the fin and the splitter plate with $A(x)$ equal to δ or s , respectively. Equation A-2 becomes

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \quad \text{in the fin}$$

and

$$\frac{d^2\psi}{dy^2} - p^2\psi = 0 \quad \text{in the splitter}$$

where

$$\theta = \frac{T_{\text{fin}} - T_{\infty}}{T_0 - T_{\infty}},$$

$$\psi = \frac{T_{\text{splitter}} - T_{\infty}}{T_0 - T_{\infty}},$$

$$m = \sqrt{\frac{2h}{k_{\text{fin}} \delta}}, \text{ and}$$

$$p = \sqrt{\frac{2h}{k_s \Delta}}.$$

These equations are solved to the following boundary conditions.

1. $\theta(0) = 1$,
2. $\theta(L/2) = \psi(0)$
3. $-k_f \delta \frac{d\theta}{dx}(L/2) = -k_s s \frac{d\psi}{dy}(0)$, and
4. $\frac{d\psi}{dy}(\frac{\ell}{2}) = 0$.

The fin efficiency becomes

$$\eta_f = \frac{k_f \delta m}{h(L + \ell)} \left[\frac{G + \tanh(\frac{mL}{2})}{1 + G \tanh(\frac{mL}{2})} \right] \quad (\text{A-5})$$

where

$$G = \frac{k_S}{k_f} \frac{s}{\delta} \frac{p}{m} \tanh\left(\frac{pl}{2}\right)$$

Equations A-4 and A-5 were combined with equation A-1 to obtain the efficiency of the heat transfer surfaces used in the code.

APPENDIX B
COMPLETE LISTING OF PROGRAM
USED AS SUBROUTINE HEX

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SUBROUTINE HEX(FL0,CFL01,CFL02,X,TGI,TG0,TI,TC0,R,NPAS,
R1,R2,CA,CP,N,AM0RT)
COMMON CPU,GMU,GKU,PR,PR3,R0,RH0F,RH0G,CC,PV,PCR,UF,UG,
HFG,PRAN,AK
CST(PA,A,R)=PA*(A**R)
DCST(PA,A,R)=PA*R*(A**R)/A
NUMB=1
IFLAG=0
ETA1P=1
ETA2P=1.
N2=1
NPP=1
CPP=4.5
NC1=1
N0=1HN
QT=FL0*CPU*(TGI-TG0)
Q1=CFL01*CC*(TC0-TI)
Q2=CFL02*X*HFG
NST=NPAS
NSP=10
TGP=TGI-Q2/FL0/CPU
CG=FL0*CPU/CFL01
F2=1
OPEN(2,INPUT,/P0RE/)
DO 1 I=1,N
1 READ(2,200)J,M,D,ALPHA,FAT,SL,ST,T,P,Q,BETA,GAM,RMIN,RMAX,S,RE
200 FORMAT(2I2,14E10.4)
CLOSE(2)
IF (M.EQ.2) FAT=FAT*S/(S-.005833333)
IF (M.EQ.2) S=S-.00583333
SIGMA=ALPHA*D/4.
GAM=GAM*GKU*((PR/.956)**(1./3.))*((R0/.137)**(2.*P))*((1.511E-05
./GMU)**(3.*P))/5.41E-03
EPS=(TGP-TG0)/(TGP-TI)
DT1=1
WRITE(1,243)
F1=1
WRITE(1,205)
READ(0,240)JP
IF (N0.NE.JP) GO TO 2
WRITE(1,241)
READ(0,201)AFP
201 FORMAT(F8.3)
2 SC0ST=1.E50
IF (M.EQ.2) GO TO 3
WRITE(1,207)
READ(0,202)HTT

```

```

      HT=HTT/12
      ALPHA=ALPHA*SL/(SL*HT+.0058333)
      SIGMA=ALPHA*D/4
202  FORMAT(F10.4)
      WRITE(1,270)
      READ(0,240)JKL
      IF(N0.EQ.JKL)GO TO 3
      WRITE(1,271)
      READ(0,272)FPI
      FPI=12.*FPI
      WRITE(1,273)
      READ(0,272)FT
      FT=FT/12.
      IFLAG=1
      FAT=FAT/(1+FPI*(HT-FT))
3  CONTINUE
      WRITE(1,290)
      READ(0,291)NBPI
      WRITE(1,292)
      READ(0,293)NBPF
      WRITE(1,294)
      READ(0,295)NPP
290  FORMAT($      NUMBER OF BOILING PASSES WILL VARY FROM $,Z)
291  FORMAT(I2)
292  FORMAT($      TO $,Z)
293  FORMAT(I2)
294  FORMAT(5X$NUMBER OF PASSES PER ROW IN THE SENSIBLE SECTION = $,Z)
295  FORMAT(I2)
      WRITE(1,203)
203  FORMAT($      DO YOU WANT TO OPTIMIZE THE NUMBER OF PASSES?$,
      .Z)
      READ(0,240)JT
      IF (N0.EQ.JT) GO TO 23
      IF(M.NE.2)NSP=5
4  DO 17 N1=NST,NSP
      DO 17 N2=NBPI,NBPF
      R1=.01
      R2=.01
      IF (Q1.EQ.0) GO TO 6
5  CALL AREA(CC,CG,N1,TU1,EPS,F1,2)
      DT1=(TG0-TI -TGP+TC0)/ALOG((TG0-TI)/(TGP-TC0))
6  DT2=1
      IF (Q2.EQ.0) GO TO 7
      DT2=(TGI-TGP)/ALOG((TGI-TC0)/(TGP-TC0))
7  AP1=Q1/F1/DT1*3600.
      AP2=Q2/DT2*3600.
      CON=120.
204  FORMAT(//)
      PP=CP
      PA=AMORT*CA

```

```

      IF (M-2) 9,8,9
8  RZ=S/2
   ASTK=1./FAT
   G0 T0 10
9  RE=SL
   ASTK=D/2./((SL-S)
   RZ=ST
10 CONTINUE
205 FORMAT($      D0 YOU WANT TO OPTIMIZE? $,Z)
   IF (N0.E0.JP) G0 T0 22
   E=P*PA/PP/(1.-P)
11 H=GAM*(E**P)
   FEF=ETA(C0N,H,T,RE,RZ,M,S)
   FEF=FEF*(1.-ASTK)+ASTK
   IF(IFLAG.LE.0)G0 T0 35
   H1P=1/R1
   H2P=1/R2
   Z1=HT/2*SQRT(2.*H1P/C0N/FT)
   ETA1P=TANH(Z1)/Z1
   Z2=HT/2*SQRT(2.*H2P/C0N/FT)
   ETA2P=TANH(Z2)/Z2
35  U1=1./(R1*FAT/ETA1P+1./H/FEF)
   U2=1./(R2*FAT/ETA2P+1./H/FEF)
   A1=AP1/U1
   A2=AP2/U2
   A12=A1+A2
   A12N=A12**R
   AP12=AP1+AP2
   E1=AP12*P*DCST(PA,A12,R)/(FEF*H*PP*(A12-AP12*P/FEF/H))
   IF (ABS(E1-E)/E1.LT.1.E-04) G0 T0 12
   E=E1
   G0 T0 11
12 CONTINUE
   AX=A1+A2
   POW=AX*E
   DP=5.12*R0*POW/FL0
   G=(H*D/(GKU*PR3 *BETA))**(1./(1-Q))*GMU/D
   REN=G*D/GMU
206 FORMAT($ RE OUTSIDE ACCEPTABLE RANGE, RE = $,E10.4)
   AFL0=FL0/G
   V0L=AX/ALPHA
   AFR=AFL0/SIGMA
   THICK=V0L/AFR*12.
   TH1=THICK*A1/AX
   TH2=THICK*A2/AX
207 FORMAT( $      C00LANT CHANNEL THICKNESS (INCHES) = $,Z)
   CALL DLP(TH1,TH2,N1,CFL02,CFL01,AFR,RH0F,RH0G,UF,UG,V,SL,ST,X,PV,
1PCR,PVCC,S,M,AK,PRAN,R1,L,N,V2,D2,HL,HT,NU,C1,C2,SPD,
2TPD,REN1,REN2,IFLAG,FPI,FT,N2,NPP)
   THICK=TH1+TH2

```

```

VØL=AFR*THICK/12.
A12=VØL*ALPHA
PØW=A12*E
DP=5.12*RØ*PØW/FLØ
IF(M.NE.1)GØ TØ 30
DEP=CLØS(REN,SIGMA)*G*G/RØ/64.4/144
PØW=PØW*(DEP+DP)/DP
DP=DP+DEP
30 HEF=FEF*H*FAT
IF (CFLØ2.EØ.Ø.) GØ TØ 14
H2=Ø
Z=Ø
DX=X/5.
DØ 13 I=1,5
Z=Z+DX
NUMB=1
CALL HTR(TCØ,Z,D2,V2,NUMB,DT2,HEF,H2P)
H2=H2+H2P
13 CØNTINUE
H2P=H2/5
RR=R2
R2=1./H2P
IF (ABS(RR-R2)/RR.GT..Ø5Ø) L=2
14 IF (L-2) 16,15,16
15 CØNTINUE
IF (NØ.EØ.JP) GØ TØ 22
GØ TØ 11
16 PC1=CPP*8.76*PVCC/.75
PC2=PP*PØW
C3=C3T(PA,A12,R)
CAPIT=CA*A12**R
IF(CFLØ1.EØ.Ø.Ø) PC1=Ø.Ø
CØST=PC1+PC2+C3
IF(M.NE.2)GØ TØ 25
IF(NC1.GT.2)GØ TØ 24
IF(C1.LT.1.9)NC1=NC1+1
25 IF (CØST.GT.SCØST) GØ TØ 17
SA=A1+A2
SG=G
SVØL=SA/ALPHA
STT=SVØL/AFR*12
STT1=STT*A1/SA
STT2=STT*A2/SA
STH1=TH1
STH2=TH2
STH=THICK
SA12=A12
SC3=C3
SPC2=PC2
SPC1=PC1

```

```

SP0W=P0W
SHTR=1./R1
SH2P=H2P
STPD=TPD
SSPD=SPD
SHL=HL
UN=NU
SCAPIT=CAPIT
SC1=C1
SC2=C2
SREN=REN
SREN1=REN1
SREN2=REN2
SPVC=PVCC*1.341
SAFR=AFR
SU1=U1
SU2=U2
SDP=DP
SH=H
SV=V
SV2=V2
SC0ST=C0ST
N11=N1
N22=N2
17 CONTINUE
24 NC1=1
TH1=STH1
TH2=STH2
THICK=STH
A12=SA12
C3=SC3
PC2=SPC2
PC1=SPC1
P0W=SP0W*1.341
TPD=STPD
SPD=SSPD
HL=SHL
C1=SC1
C2=SC2
AFR=SAFR
U1=SU1
U2=SU2
DP=SDP
H=SH
V=SV
V2=SV2
C0ST=SC0ST
WRITE(1,243)
IF (M-2) 18,19,18
IF (M-2) 18,19,18

```



```

18 WRITE(1,208)N
208 FORMAT(5X,$PLATE-FIN HEAT EXCHANGER TYPE $,I2)
WRITE(1,209)HL
209 FORMAT(5X,$LENGTH =$,F6.2)
WW=HL
WRITE(1,210)WW
210 FORMAT(5X,$WIDTH =$,F6.2)
HTT=HT*12
WRITE(1,211)N11
211 FORMAT(5X,$THERE ARE $,I4,$PASSES IN THE SENSIBLE HEAT SECTIONS)
WRITE(1,296)N22
296 FORMAT($ THERE ARE $,I4,$PASSES IN THE BOILING SECTIONS)
WRITE(1,212)NU,HTT
212 FORMAT(5X,$THERE ARE $,I4,$ COOLANT CHANNELS $,F6.3,$IN. THICK$)
G0 TO 20
19 WRITE(1,213)N
213 FORMAT(5X,$TUBE-FIN HEAT EXCHANGER TYPE $,I2)
WRITE(1,210)HL
WID=AFR/HL
WRITE(1,209)WID
WRITE(1,214)N11,C1,N22,C2,UN
214 FORMAT(5X,$THERE ARE $,I3,$ PASSES WITH $,F5.0,$ ROWS PER PASS$
1,$ IN THE SENSIBLE $,/5X,$HEAT SECTION AND$,I3,$ PASSES WITH$
2,I3,$ ROWS PER PASS IN THE$,/5X,$BOILING SECTION. THERE ARE$,
3F5.0,$ TUBES PER ROW$)
20 N1=N11
QT=FL0*CPU *(TGI-TG0)*778/550
WRITE(1,224)SA12
WRITE(1,221)AFR
WRITE(1,220)THICK
WRITE(1,228)TH1
WRITE(1,229)TH2
WRITE(1,243)
WRITE(1,249)CFL01
WRITE(1,215)SPD
WRITE(1,234)SV
WRITE(1,244)SREN1
WRITE(1,250)CFL02
WRITE(1,216)TPD
WRITE(1,235)SV2
WRITE(1,245)SREN2
WRITE(1,217)SPVC
WRITE(1,238)SHTR
WRITE(1,239)SH2P
WRITE(1,243)
WRITE(1,2222) FEF
2222 FORMAT(5X,$FIN EFFICIENCY =$,F6.4)
WRITE(1,222)SG
WRITE(1,246)SREN
IF (SREN.LT.RMIN.0R.SREN.GT.RMAX) WRITE(1,206)SREN

```

```

WRITE(1,227)DP
WRITE(1,223)P0W
WRITE(1,237)H
WRITE(1,226)TGP
WRITE(1,243)
WRITE(1,225)QT
WRITE(1,230)U1
WRITE(1,236)SU2
WRITE(1,243)
WRITE(1,247)AM0RT
WRITE(1,231)PC2
WRITE(1,232)PC1
WRITE(1,248)SCAPIT
WRITE(1,233)C3
WRITE(1,219)C0ST
WRITE(1,242)
215 FORMAT(5X,$SINGLE PHASE DELP =$,F7.3)
216 FORMAT(5X,$TWO PHASE DELP =$,F7.3)
217 FORMAT(5X,$PUMP POWER =$,F7.3,$ HORSEPOWER$)
218 FORMAT(///)
219 FORMAT(5X,$YEARLY COST =$,F11.2)
220 FORMAT(5X,$COOLER THICKNESS,INCHES =$,F9.4)
221 FORMAT(5X,$FRONTAL AREA =$,F11.2)
222 FORMAT(5X,$MASS VELOCITY,G=$,F9.4)
223 FORMAT(5X,$COMPRESSOR POWER REQUIRED =$,F11.2,$ HP$)
224 FORMAT(5X,$HX AREA =$,F11.2)
225 FORMAT(5X,$HEAT TRANSFERED =$,F11.2,$ HORSEPOWER $)
226 FORMAT(5X,$TGP=$,F9.4)
227 FORMAT(5X,$PRESSURE DROP,PSI =$,F9.4)
228 FORMAT(5X,$TH1 =$,F9.4,$ INCHES$)
229 FORMAT(5X,$TH2 =$,F9.4,$ INCHES$)
230 FORMAT(5X,$ U1 =$,F9.4)
231 FORMAT(5X,$GAS SIDE PUMPING COST =$,F11.2)
232 FORMAT(5X,$COOLANT PUMPING COST =$,F11.2)
233 FORMAT(5X,$YEARLY HEAT EXCHANGER COST =$,F11.2)
234 FORMAT($ SINGLE PHASE VELOCITY =$,F9.4)
235 FORMAT($ TWO PHASE VELOCITY =$,F9.4)
236 FORMAT($ U2=$,F9.4)
237 FORMAT($ GAS SIDE HEAT TRANSFER COEFFICIENT =$,F9.4)
238 FORMAT($ SINGLE PHASE COEFFICIENT =$,F9.4)
239 FORMAT($ TWO PHASE COEFFICIENT =$,F9.4)
240 FORMAT(A1)
241 FORMAT($ FRONTAL AREA =$,Z)
243 FORMAT($ *****$)
244 FORMAT(5X,$SINGLE PHASE REYNOLDS NO. =$,E9.2)
245 FORMAT(5X,$TWO PHASE REYNOLDS NUMBER =$,E9.2)
246 FORMAT(5X,$GAS REYNOLDS NO. =$,E9.2)
247 FORMAT($ AMORTIZATION FACTOR= $,F7.4)
248 FORMAT($ CAPITAL COST =$,F11.2,$ DOLLAR$)
249 FORMAT(5X,$SINGLE PHASE FLOW RATE (LB/SEC) =$,F8.2)

```

```

250 F0RMAT(5X,$TW0 PHASE FL0W RATE (LB/SEC) =$,F8.2)
270 F0RMAT($      D0 Y0U WANT FINS IN THE C00LANT PASSAGES?$,Z)
271 F0RMAT($      NUMBER 0F FINS/INCH    $,Z)
272 F0RMAT(F10.5)
273 F0RMAT($      FIN THICKNESS(INCHES)...$,Z)
21  RETURN
242 F0RMAT(////)
22  AFL0=AFR*SIGMA
    G=FL0/AFL0
    REN=G*D/GMU
    H=GKU*PR3*BETA/D*((G*D/GMU)**(1.-Q)
    FEF=ETA(C0N,H,T,RE,RZ,M,S)
    FEF=FEF*(1.-ASTK)+ASTK
    IF(IFLAG.LE.0)G0 T0 36
    H1P=1/R1
    H2P=1/R2
    Z1=HT/2*SQRT(2.*H1P/C0N/FT)
    ETA1P=TANH(Z1)/Z1
    Z2=HT/2*SQRT(2.*H2P/C0N/FT)
    ETA2P=TANH(Z2)/Z2
36  U1=1./(R1*FAT/ETA1P+1./H/FEF)
    U2=1./(R2*FAT/ETA2P+1./H/FEF)
    A1=AP1/U1
    A2=AP2/U2
    A12=A1+A2
    A12N=A12**R
    AP12=AP1+AP2
    E=(H/GAM)**(1./P)
    CONTINUE
    G0 T0 10
23  NST=NPAS
    NSP=NPAS
    G0 T0 4
    END

```

```

FUNCTION CL0S(R,S)
D=1.086-3.125E-06*R
C=.5566+.05556/(1.136-S)
E=1.-2.*D*S+S*S
CK=(1.-2.*C+C*C*(2.*D-1))/C/C
CL0S=CK+E
RETURN
END

```

```

FUNCTION ETA(TK,H,T,RE,R0,M,S)
  IF (M-1) 2,2,1
1 CONTINUE
  HM =SQRT(2.*H/TK/T)
  AE=HM*RE
  A0=HM*R0
  CALL BESSEL(AE,1,AE1,IF,3)
  CALL BESSEL(AE,1,AE1K,IF,4)
  CALL BESSEL(A0,0,A00I,IF,3)
  CALL BESSEL(A0,0,A00K,IF,4)
  CALL BESSEL(A0,1,A01I,IF,3)
  CALL BESSEL(A0,1,A01K,IF,4)
  C=AE1/AE1K
  C0N=(A01I-C*A01K)/(A00I+C*A00K)
  ETA=-2.*R0*C0N/HM/(RE*RE-R0*R0)
  RETURN
2 B=RE
  W=R0
  AM=2.*H/TK /T
  AM=SQRT(AM)
  AP=2.*H/TK /S
  AP=SQRT(AP)
  C0N=S/T*AP/AM*TANH(AP*W/2.)
  Z=TANH(AM*B/2.)
  C0N=(C0N+Z)/(1.+C0N*Z)
  ETA=TK *T*AM*C0N/H/(B+W)
  RETURN
END

```

```

SUBROUTINE AREA(CC,CG,N,ANTU,EPS,F,J)
  IF (J-1) 1,2,3
1  STOP
2  CALL EFF(ANTU,CG,CC,N,EPS,F)
  RETURN
3  TU1=0
  E1=0.
  TU2=10.
  CALL EFF(TU2,CG,CC,N,E2,F)
4  TU=(E1-EPS)*(TU1-TU2)/(E2-E1)+TU1
  IF (TU.GT.20.) STOP
  CALL EFF(TU,CG,CC,N,E3,F)
  IF (ABS(EPS-E3)-1.E-03) 8,8,5
5  IF (E3-EPS) 6,6,7
6  E1=E3
  TU1=TU
  GO TO 4
7  E2=E3
  TU2=TU
  GO TO 4
8  ANTU=TU
  RETURN
  END

```

```

SUBROUTINE EFF(AN,CG,CC,N,EPS,F)
  IF (CG) 1,1,2
1  EPS=1.-EXP(-AN)
  F=1.
  RETURN
2  ZB=CG/CC
  TN=N
  ZA=1./ZB
  IF (CC-CG) 3,4,4
3  PN=AN/TN
  CR=1
  Z=ZA
  GO TO 5
4  PN=AN/TN
  CR=ZA
  Z=ZB
5  ZK=1.-EXP(-PN)
  IF (N-2) 6,7,8

```

```

6 EPS=1.-EXP(-ZB*ZK)
  EP =EPS*CR
  EPS=EPS*ZA
  G0 T0 9
7 P=ZK/2.+(1.-ZK/2.)*EXP(2.*ZK*ZB)
  EPS=1.-1./P
  EP =EPS*CR
  EPS=EPS*ZA
  G0 T0 9
8 IF (N.GT.3) G0 T0 11
  C1=1.-ZK/2.
  C2=EXP(3.*ZK*ZB)
  C3=ZK*(1.-ZK/4.)
  C4=ZK*ZK*ZB*C1
  C5=C1*C1*C2+(C3-C4)*EXP(ZK*ZB)
  EPS=1.-1./C5
  EP =EPS*CR
  EPS=EPS*ZA
9 IF (Z.EQ.1) G0 T0 10
  F=ALOG((1.-Z*EP)/(1.-EP))
  F=F/AN/(1.-Z)*CR*Z
  RETURN
10 F=EP/(1.-EP)/AN
  RETURN
11 CONTINUE
  IF (ZA.EQ.1.) G0 T0 12
  TEA=EXP((ZA-1.)*AN*ZB)
  EPS=(1.-TEA)/(1.-ZA*TEA)
  EPS=ZA*EPS
  F=1
  RETURN
12 EPS=AN/(1.+AN)
  F=1.
  RETURN
  END

```

```

SUBROUTINE DLP (TH1,TH2,N1,CFL02,CFL01,AFR,RH0F,RH0G,UF,UG,V,SL,
1ST,X,P,PRC,P0W,S,M,AK,PRAN,RI,L,N,V2,D2Q,HL,HT,NU,C1,C2,SPD,
2TPD,REN, R0N,IFLAG,FPI,FT,N2,NPP)
  L=3
  D=TH1/N1/12
  T22=TH2/12/N2
  IF (M-2) 1,17,1
1 HL=SQRT(AFR)
  NU=HL/(SL+HT+.0058333)
  W=NU*HT

```

```

      IF (D) 2,2,3
2  V=0
   V2=CFL02/T22/RH0F/W
   G0 T0 4
3  V=CFL01/D/RH0F/W*NPP
4  DEQ=2.*HT*D/(HT+D)
   IF(IFLAG.LE.0) G0 T0 20
   DEQ=2.*(1./FPI-FT)*HT/(1./FPI-FT+HT)
20 IF (DEQ.EQ.0) DEQ=1
   D=T22
   IF (D) 5,5,6
5  D20=1
   V2=0
   G0 T0 7
6  D20=2.*D*HT/(D+HT)
   IF(IFLAG.LE.0) G0 T0 21
   D20=2.*(1./FPI-FT)*HT/(1./FPI-FT+HT)
21 V2=CFL02/RH0F/D/W
7  REN=RH0F*DEQ*V/UF*3600
   IF (REN.EQ.0) G0 T0 8
   R1=.023*AK/DEQ*(REN**.8)*PRAN**.33333
   R1=1./R1
   IF (ABS(R1-RI).GT..025*RI) G0 T0 18
8  CONTINUE
9  IF (REN) 11,11,10
10 F=.178/(REN**.2)
11 R2N=D20*V2*RH0F/UF*3600.
   IF (R2N) 13,13,12
12 F2=.178/(R2N**.2)
13 H1=RH0F*V*V/64.4/144.
   H2=RH0F*V2*V2/64.4/144.
   DELP=F*HL*H1/DEQ
   SPDP=N1*DELP*NPP
   IF (X) 14,14,15
14 R1=1
   R3=1./2./RH0F
   G0 T0 16
15 CALL TPP(RH0F,RH0G,UG,UF ,P,PRC,R1,R2,R3,R4,X)
16 TPDP=F2*HL/D20 *H2*R1*N2
   DPIN=.5*H1
   DPEL=2.*H1*(N1-1)
   DP0UT=H1
   D2PIN=.5*H2
   D2P0=R3*RH0F*H2*2.
   SPD =DPIN+SPDP+DPEL+DP0UT
   TPD=D2PIN+TPDP+D2P0
   P0W1=SPD*.195*CFL01/RH0F
   P0W2=.195*TPD*CFL02/RH0F
   P0W=P0W2+P0W1
   DELP=SPD+TPD

```

```

      RETURN
17 CALL HEAT(D,T22,CFL01,CFL02,ST,SL,RH0F,S,V,AFR,TN1,TN2,UN,C1,C2,
    .RLT,V2,HL)
    TH1=12.*N1*D
    TH2=12*T22*N2
    NU=UN
    V=NPP*V
    DEQ=S
    D2Q=S
    GO TO 7
    RETURN
18 RI=R1
    L=2
    RETURN
    END

```

```

      SUBROUTINE HEAT (TH1,TH2,CFL01,CFL02,ST,SL,RH0F,D,V1,FR,TN1,TN2,
    .RN,C1,C2,RLT,V2,H)
    C1=0
    C2=0
    W=INT(SQRT(FR)/ST+.9)*ST
    B=D
    PA=3.1416*B**2/4.
    IF (TH1.EQ.0.0) GO TO 1
    C1=INT (TH1/SL+.99)
    TH1=C1*SL
    TA=W/ST*PA*C1
    V1=CFL01/(RH0F*TA)
    RN=INT (TA/(PA*C1))
    TN1=C1*RN
    IF (TH2) 4,4,1
1  C2=INT (TH2/SL+.99)
    TA=W/ST*PA*C2
    RN=INT (TA/(PA*C2))
    TH2=C2*SL
    V2=CFL02/PA/RN/C2/RH0F
    IF (TH1.NE.0) GO TO 2
    V1=0
2  TN2=C2*RN+1.E-50
3  H=RN*ST
    RLT=FR/(RN*ST)
    RETURN
4  V2=0
    TN2=0
    GO TO 3
    END

```



```

SUBROUTINE HTR(T,X,D,V,NUM,DT,HEF,H)
DT1=.2*DT
CALL CHEN(T,X,D,V,DT1,NUM,H)
F1=H*DT1-HEF*(DT-DT1)
DT2=.3*DT
1 CALL CHEN(T,X,D,V,DT2,NUM,H)
F2=H*DT2-HEF*(DT-DT2)
DEL=F2*(DT2-DT1)/(F1-F2)
F1=F2
DT1=DT2
DT2=DT2+DEL
IF (ABS(DEL/DT2).GT..01) GO TO 1
RETURN
END

```

```

SUBROUTINE CHEN(T,X,D,V,DT,NUM,H)
IF (NUM.GT.1) GO TO 1
CALL HEN(T,X,V,NUM,C0N,P,D,HC)
NUM=2
1 TT=T+DT
CALL SFP(3,TT,A,A,A,VP,A,A,A,A,A,A)
DP=(VP-P)*144.
H=C0N*(DT**.24)*(DP**.75)
H=H+HC
RETURN
END

```

```

SUBROUTINE HEN(T,X,V,NUM,C0N,VP,D,HC)
CALL SFP(0,T,AK,RH0L,CP,VP,ST,VISL,VISV,HV,RH0V,PR)
XTT=((((1.-X)/X)**.9)*((VISL/VISV)**.1)*SQRT(RH0V/RH0L))
F=RNFXTT)
F=RNFXTT)
RTP=RH0L*V*D*(1.-X)*3600/VISL*(F**1.25)
HC=.023*(RTP**.8)*(PR**.4)*AK/D
S=SUP(RTP)
C0N=.172*S*(AK**.79)*(CP**.45)*(RH0L**.49)/((ST**.5)*(VISL**.29)
.*(HV**.24)*(RH0V**.24))
RETURN
END

```

```

FUNCTION RNF(XTT)
Z=1./XTT
F=2.9*Z**.71
IF (F.LT.1.) F=1.
RNF=F
RETURN
END

```

```

FUNCTION SUP(R)
IF (R.LT.8100) G0 T0 1
IF (R.GT.3.5E05) G0 T0 2
SUP=3.1236-.236*ALOG(R)
RETURN
1 SUP=1
RETURN
2 SUP=.1
RETURN
END

```

```

SUBROUTINE TPP(RH0F,RH0G,UG,UF,P,PCR,R1,R2,R3,R4,XE)
PP=P
PR=PCR
C0N=((RH0G/RH0F)**.5)*((UF/UG)**.1)
N=XE/.01
DX=XE/N
X=0.
ALP1=0.
FCT1=1.
SUM1=0.
SUM2=0.
1 X=X+DX
SUM1=SUM1+(FCT+FCT1)*DX/2.
SUM2=SUM2+(ALP+ALP1)*DX/2.
FCT1=FCT
ALP1=ALP
CALL FUNC(X,FCT,ALP,C0N,PCR,P)
IF (ABS(X-XE)-.0001) 2,2,1
2 V=SUM1
R4=1.-(1.-RH0G/RH0F)*SUM2/XE
R2=XE*XE/RH0G/ALP+((1.-XE)**2)/RH0F/(1.-ALP)-1./RH0F
R3=XE*XE*XE/ALP/ALP/RH0G/RH0G
R3=R3+((1.-XE)**3)/(1.-ALP)/(1.-ALP)/RH0F/RH0F
R3=R3/2./(XE/RH0G+(1.-XE)/RH0F)
R1=V/XE
RETURN
END

```

```

SUBROUTINE FUNC(X,FCT,ALP,C0N,PCR,P)
Z=C0N*(((1.-X)/X)**.9)
PP=PHI(Z,P,PCR)
FCT=(((1.-X)**1.80)*PP*PP
ALP=1.-1./PP
RETURN
END

```

```

FUNCTION PHI(X,P,PCR)
A=F(X)
B=G(X)
PHI=A+(B-A)*P/PCR
RETURN
END

```

```

FUNCTION G(X)
T=X**1.1111
G=(1.+1./T)**.9
RETURN
END

```

```

FUNCTION F(X)
IF (X.GT.150) GO TO 1
IF (X.L1..001) GO TO 2
T=ALOG(X)
T2=T*T
T3=T2*T
T4=T2*T2
T5=T4*T
T6=T5*T
T7=T6*T
YY=1.4445569-.5073967*T+.058361463*T2-.0004458546*T3
YY=YY-.00050072494*T4+.0000025389716*T5+.000001807095*T6
F=EXP(YY)
RETURN
1 F=1.
RETURN
2 F=1./X
RETURN
END

```

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