HEAT EXCHANGERS FOR
CONVECTIVE AND RADIATIVE ENVIRONMENTS

D. B. Mackay, Professor
of Aerospace Engineering
W. E. Branner, Programmer
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This document presents two IBM-7040 Fortran Programs, Numbers 4-1 and 4-2, for the analysis of steady-state liquid heat exchange systems composed of ducts and extended surfaces (fins). The system exchanges heat with its environment by radiation, convection or both, under conditions where the environment parameters are invariant with position along the tube or duct. The programs calculate heat exchanged, fluid and duct temperatures, friction pressure drop and a number of heat exchange parameters. Common input data include system dimensions, material and liquid properties, flow rate, and environmental conditions.

Program 4-1 is applicable to environments where as many as three external bodies may be present to effect the radiation heat transfer. One of these bodies is assumed to be the sun. The system configuration is limited to round tubes and symmetrical fins.

Program 4-2 is applicable to innumerable geometric configurations including non-circular ducts, irregular shaped fins and fin spacing. Each fin can have its individual material properties and environment. However, fin performance information has to be established by techniques which are not included in Program 4-2.
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INTRODUCTION

This study is a natural follow-up of the work presented in Ref. 1 in which programs using numerical integration routines are presented for calculating the combined radiative and convective heat transfer from extended surfaces (fins). Techniques have already been developed for predicting the performance of extended surface heat exchangers losing heat solely by radiation. This study, however, contains the first programs capable of analyzing convective and radiative heat exchangers, where a fluid is either heated or cooled.

In the radiative cases it was observed that the length of duct required to cool a fluid could be established using an effective (or equivalent) length for the fins and duct system. The equivalent length of a fin is the length which an imaginary strip perpendicular to the axis of the duct would have if it transferred the same amount of heat as the actual strip length but had a constant temperature equal to the duct root temperature. The total heat exchanged is the sum of that exchanged from the duct and fins. The fin equivalent length was observed to change little with a change in root temperature and would therefore remain almost constant along the duct. In Ref. 2, duct length calculations were made using an arithmetic average equivalent length. However, it was found in this study that an alternate method would give increased accuracy, as a result some of the procedures were modified.

Two programs are presented, 4-1 and 4-2. The first is limited to the case of circular tubes and symmetrical fins. The maximum complexity for the radiative environment is as illustrated in Fig. 1. Program 4-2 is very
general to make possible the solution of a large variety of problems. It will not calculate fin equivalent length. However, this information can be obtained by using the programs given in Ref. 1. Finite difference programs may also be used. In Program 4-2 the duct configuration is specified by the cross-sectional area and perimeter in order that non-circular ducts can be analyzed. By separating the fin performance from the duct length calculations, innumerable configurations can be analyzed. The fins can be of single or multisection shapes, of unsymmetrical design, of different materials, and even in different environments. These programs should find wide application in solving a host of heat transfer problems which have heretofore been considered momentarily impractical or physically impossible to solve accurately.

The solutions to purely radiative problems of Ref. 2 were simplified by using dimensionless parameters to specify the environment and the fin configuration. Numerical integration was used to obtain the fin effectiveness in terms of the parameters. Curve fitting techniques were employed to correlate the numerically calculated data so that the programs contained only systems of equations to calculate fin performance and the duct lengths. No comparable set of equations is available for calculating the effectiveness of fins in a combination radiative and convective environment. Therefore, numerical integrations are used, either directly or indirectly, to obtain the fin performance.

Considerable machine time is saved by using the integration routines as little as possible. Approximate wall temperatures are established at each end of the duct. After these temperatures are set, the heat transfer equation is integrated at each end to determine the effective fin lengths.
Following these integration steps, tests are made to determine the change in the value of the equivalent lengths, $L_e$. If less than a ten per cent change has taken place, a linear equation relating $L_e$ and $T_w$ is derived to pass through the two established points. If a greater change has taken place, a third integration is made at a temperature midway between the inlet and exit wall temperatures. A polynomial curve fit is then made through the three points. The fin equivalent length equation is used to calculate the section lengths and the temperature conditions along the duct.

The manner of specifying problem conditions in Program 4-1 may seem odd at the outset. The normally expected conditions (tube size, fluid flow rate, and fin geometry) are of course required. However, the duct length and duct section lengths are established by the program. The tangible control over these items is the heat exchanger effectiveness, $\varepsilon$, and the mesh (number of divisions to be used). The exchanger effectiveness is defined in the same way as in conventional heat exchanger or boilers, i.e. the actual temperature change divided by the maximum possible temperature change. In this case the limit of maximum heating or cooling is established by the effective environmental temperature. Since the environment is composed of both convective and radiative items, the exact effective temperature may not be readily attainable nor is it necessary in most cases. Furthermore, the machine uses approximations for obtaining the exit wall temperatures and from this recalculates the exit fluid temperature. Also, the actual effectiveness will differ from the value of $\varepsilon$ specified by the user. A sufficiently large temperature difference (high value for $\varepsilon$) can be predicted, and the duct length calculated for approximately this condition. Since the machine
can calculate the conditions in a duct of considerable extra length and for a large number of intervals in little time, the length of duct actually needed can be taken either at one of the section points calculated or by interpolating between points. The author has observed that in many cases neither the inlet nor the outlet fluid conditions are specifically known until the system performance has been established. In these cases a wider spread in temperature than needed is submitted to the machine since the required conditions can be established by plotting the output data and selecting a midsection from the curve. Two advantages are obtained by specifying the input data in this way. (1) the programming is straightforward so that a minimum of machine time is used, and (2) the likelihood of the user's specifying conditions which are impossible to solve is greatly reduced.

The "AICH" subroutine used in Ref. 2 for calculating the internal heat transfer coefficient employed several equations to cover the laminar, transition and turbulent regimes. In passing from one regime to the next, an abrupt and unrealistic difference in heat transfer coefficient was predicted. Also the heat transfer coefficient in the laminar region was higher than calculated using the popular Nusselt number. A new subroutine was written to use Nusselt number in the laminar region and equations from Ref. 4 in the turbulent region.

Duct length calculations in these programs use an iterative method in lieu of the parametric procedure used in Ref. 2. This decision resulted from the fact that discrepancies as high as 15 per cent were observed in sample problems when comparing the lengths calculated by the two methods. The error appeared to lie in the use of an average effective length for the fins.
Mathematical procedures using variable effective lengths would have been time consuming. In the interest of machine and programming time they were abandoned before being completely developed even though they have merit.

Since this work follows and is in parallel with that presented in Ref. 1, it will be assumed that the reader has access to Ref. 1. Therefore, little of the data presented in that document will be reproduced herein. It will also be assumed that the reader is acquainted with Newton's iteration method and linear techniques for obtaining solutions to complex equations. Linear and polynomial curve fittings are made to represent the data extracted from an array of equation. These techniques are assumed to be self-explanatory. A number of equations have been taken from the literature. The source of the equation and specific location are given in parentheses following the equation.

The basic assumptions made for these programs include the following:

1. Steady state conditions have been reached.

2. No heat is transferred from the outeredges of the fins (corrections can be made for this item in the second program if the user wishes to expend the extra effort to do so, but normally the accuracy of the problem does not warrant such refinement).

3. No heat is transferred in the direction parallel to the duct. (The temperature gradient in this direction is normally small enough that its effect does not materially degrade the predicted performance.)

4. The environment remains constant along a fin and throughout the length of the duct.
5. The material properties of the duct and fin are not affected by temperature or position in the system.
## NOMENCLATURE

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<td>Duct cross sectional area for fluid flow, sq ft/duct</td>
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<td>Constant (see Eq. 1-23)</td>
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<td>G3(L)</td>
<td>Subscripted constants</td>
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<td>H</td>
<td>Convective heat transfer coefficient from duct fluid to wall, Btu/hr sq ft R</td>
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<td>HA</td>
<td>Convective heat transfer coefficient on the side facing the sun (if applicable), Btu/hr sq ft R</td>
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<td>$h_a(L)$</td>
<td>HA(L)</td>
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<td>Sum of convective terms, $(h_{a}T_{aa} + h_{b}T_{ab})$, Btu/hr sq ft</td>
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<td>$h_b$</td>
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<td>Convective heat transfer coefficient on the side away from the sun (if applicable), Btu/hr sq ft R</td>
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<td>Definition</td>
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<tr>
<td>$h_b(L)$</td>
<td>HB(L)</td>
<td>Convective heat transfer coefficient to the environment from side &quot;b&quot; of fin (L), Btu/hr sq ft R</td>
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<td>$h_d(I)$</td>
<td>HD(I)</td>
<td>Convective heat transfer coefficient to environment from duct section (I), Btu/hr sq ft R</td>
</tr>
<tr>
<td>$L_h$</td>
<td>HL</td>
<td>Extended surface length, ft</td>
</tr>
<tr>
<td>$h_t$</td>
<td>HT</td>
<td>Sum of convective heat transfer coefficients, $(h_a + h_b)$, Btu/hr sq ft R</td>
</tr>
<tr>
<td>ISO</td>
<td></td>
<td>Switch to signal a zero condition has been encountered</td>
</tr>
<tr>
<td>ISW</td>
<td></td>
<td>Switch to signal a wilt condition has been encountered</td>
</tr>
<tr>
<td>ITER</td>
<td></td>
<td>The number of attempts for a starting value of $dZ/dw$</td>
</tr>
<tr>
<td>ITLT</td>
<td></td>
<td>Maximum number of iterations allowed to revise initial $dZ/dw$</td>
</tr>
<tr>
<td>$\omega$</td>
<td>OMEGA</td>
<td>Ratio, $L/L_h$ where $L$ represents distance that the heat has traveled along the fin and $L_h$ represents the total length, nondimensional</td>
</tr>
<tr>
<td>$P$</td>
<td></td>
<td>Pressure, lb/sq ft</td>
</tr>
<tr>
<td>$p$</td>
<td>PERIM</td>
<td>Duct internal perimeter, ft</td>
</tr>
<tr>
<td>$N_{pr}$</td>
<td>PRN</td>
<td>Prandtl number computed by heat transfer coefficient subroutine AICH, nondimensional</td>
</tr>
<tr>
<td>$q$</td>
<td>Q</td>
<td>Heat exchanged with environment, Btu/hr per tube</td>
</tr>
<tr>
<td>$q_s$</td>
<td>QS</td>
<td>Heat transfer to the environment from a unit of duct length, Btu/hr</td>
</tr>
<tr>
<td>QSUM</td>
<td></td>
<td>System heat exchanged with environment, Btu/hr</td>
</tr>
<tr>
<td>Equations</td>
<td>Computer Program</td>
<td>Definition</td>
</tr>
<tr>
<td>-----------</td>
<td>-----------------</td>
<td>------------</td>
</tr>
<tr>
<td>$N_{re}$</td>
<td>RE</td>
<td>Reynolds number, nondimensional</td>
</tr>
<tr>
<td></td>
<td>REAV</td>
<td>Average Reynolds number, nondimensional</td>
</tr>
<tr>
<td>$\rho$</td>
<td>RHO</td>
<td>Fluid density, lb/cu ft</td>
</tr>
<tr>
<td>$\rho_{av}$</td>
<td>RHOAV</td>
<td>Average fluid density, lb/cu ft</td>
</tr>
<tr>
<td>$\rho_f$</td>
<td>RHOFM</td>
<td>Density of fin material, lb/cu ft</td>
</tr>
<tr>
<td>$\rho_m$</td>
<td>RHOM</td>
<td>Surface reflectivity of body &quot;m&quot;, nondimensional</td>
</tr>
<tr>
<td>$\rho_d$</td>
<td>RHOST</td>
<td>Density of duct material, lb/cu ft</td>
</tr>
<tr>
<td>$\rho_x$</td>
<td>RHOX</td>
<td>Reflectivity of second surface, x, nondimensional</td>
</tr>
<tr>
<td>$S_c$</td>
<td>SC</td>
<td>Solar heat, Btu/hr sq ft</td>
</tr>
<tr>
<td>$S_c(I)$</td>
<td>SC(I)</td>
<td>Peripheral duct length for convective heat transfer from section (I), ft</td>
</tr>
<tr>
<td>$S_r(I)$</td>
<td>SR(I)</td>
<td>Effective peripheral duct length for radiative heat transfer from section (I), ft</td>
</tr>
<tr>
<td>$T$</td>
<td></td>
<td>Temperature at any point on an extended surface, R</td>
</tr>
<tr>
<td>$T_{aa}$</td>
<td>TAA</td>
<td>Ambient fluid temperature on side &quot;a&quot;, R</td>
</tr>
<tr>
<td>$T_{aa}(L)$</td>
<td>TAA(L)</td>
<td>Ambient environmental temperature for side &quot;a&quot; of fin (L), R</td>
</tr>
<tr>
<td>$T_{ab}$</td>
<td>TAB</td>
<td>Ambient fluid temperature on side &quot;b&quot;, R</td>
</tr>
<tr>
<td>$T_{ab}(L)$</td>
<td>TAB(L)</td>
<td>Ambient environmental temperature for side &quot;b&quot; of fin (L), R</td>
</tr>
<tr>
<td>$T_a(I)$</td>
<td>TA(I)</td>
<td>Ambient environmental temperature for duct section (I), R</td>
</tr>
<tr>
<td>Equations</td>
<td>Computer Program</td>
<td>Definition</td>
</tr>
<tr>
<td>-----------</td>
<td>-----------------</td>
<td>------------</td>
</tr>
<tr>
<td>$T_b$</td>
<td>TB</td>
<td>Bulk (mixed) fluid temperature for AICH subroutine, R</td>
</tr>
<tr>
<td>$T_e$</td>
<td>TE</td>
<td>Effective environmental temperature, see Eq. (1-5) to (1-7), R</td>
</tr>
<tr>
<td>$T_f$</td>
<td>TF</td>
<td>Temperature of fluid in duct, R</td>
</tr>
<tr>
<td>$T_{fend}$</td>
<td>TFEND</td>
<td>Fluid exit temperature, T</td>
</tr>
<tr>
<td>$T_{f1}$</td>
<td>TF1</td>
<td>Fluid inlet temperature, R</td>
</tr>
<tr>
<td>$\theta_m$</td>
<td>THETAM</td>
<td>Angle between sun's rays and normal to body &quot;m&quot; surface, degrees</td>
</tr>
<tr>
<td>$\theta_p$</td>
<td>THETAP</td>
<td>Angle between sun's rays and normal to fin surface, degrees</td>
</tr>
<tr>
<td>$\theta_x$</td>
<td>THETAX</td>
<td>Angle between sun's rays and normal to second surface, degrees</td>
</tr>
<tr>
<td>$T_m$</td>
<td>TM</td>
<td>Surface temperature of body &quot;m&quot;, R</td>
</tr>
<tr>
<td>$T_s$</td>
<td>TS</td>
<td>Duct wall temperature for AICH subroutine, R</td>
</tr>
<tr>
<td>$T^*$</td>
<td>TSTAR</td>
<td>Temperature used for calculating $N_{re}$ and $N_{pt}$ in AICH subroutine, R</td>
</tr>
<tr>
<td>$T_w$</td>
<td>TW</td>
<td>Duct wall temperature, R</td>
</tr>
<tr>
<td>$T_{wf}(L)$</td>
<td>TWF(L)</td>
<td>Duct temperature at root of fin (L) for conditions approximating the exit of duct, and used to calculate $L_{ef}(L)$, R</td>
</tr>
<tr>
<td>$T_{ws}(L)$</td>
<td>TWS(L)</td>
<td>Duct temperature at root of fin (L) for conditions approximating the entrance of duct, and used to calculate $L_{es}(L)$, R</td>
</tr>
<tr>
<td>$T_x$</td>
<td>TX</td>
<td>Surface temperature of body &quot;x&quot;, R</td>
</tr>
<tr>
<td>$T(2)$</td>
<td></td>
<td>Existing value of $\omega$, nondimensional</td>
</tr>
<tr>
<td>$d\omega$</td>
<td>T(3)</td>
<td>Increment of $\omega$ used for calculations, nondimensional</td>
</tr>
<tr>
<td>$Z$</td>
<td>T(4)</td>
<td>Value of $T/T_w$, nondimensional</td>
</tr>
<tr>
<td>Equations</td>
<td>Computer Program</td>
<td>Definition</td>
</tr>
<tr>
<td>-----------</td>
<td>-----------------</td>
<td>------------</td>
</tr>
<tr>
<td>$\frac{dZ}{dw}$</td>
<td>T(5)</td>
<td>$dZ/dw$, nondimensional</td>
</tr>
<tr>
<td>$V$</td>
<td>V</td>
<td>Fluid velocity, ft/sec</td>
</tr>
<tr>
<td>$V_{av}$</td>
<td>VAV</td>
<td>Arithmetic mean of inlet and outlet velocity, ft/sec</td>
</tr>
<tr>
<td>$\mu$</td>
<td>VISC</td>
<td>Viscosity of fluid, lb/hr ft</td>
</tr>
<tr>
<td>WALTH</td>
<td></td>
<td>Duct wall thickness, ft</td>
</tr>
<tr>
<td>WD</td>
<td></td>
<td>Weight of a tube, lb</td>
</tr>
<tr>
<td>$\dot{w}$</td>
<td>WDOT</td>
<td>Total fluid weight flow, lb/hr</td>
</tr>
<tr>
<td>$\dot{w}_d$</td>
<td>WDOTD</td>
<td>Fluid weight flow in a duct, lb/hr</td>
</tr>
<tr>
<td>WF</td>
<td></td>
<td>Weight of extended surfaces attached to a tube, lb</td>
</tr>
<tr>
<td>WILT</td>
<td></td>
<td>Value of $(dZ/dw)_1$ used for the previous iteration, nondimensional</td>
</tr>
<tr>
<td>WL</td>
<td></td>
<td>Weight of the liquid trapped in a section of duct, lb</td>
</tr>
<tr>
<td>$W_t$</td>
<td>WT</td>
<td>Total weight of heat exchanger including fluid, lb</td>
</tr>
<tr>
<td>$y$</td>
<td>Y</td>
<td>Heat transfer rate at fin root temperature, Btu/hr sq ft</td>
</tr>
<tr>
<td>$Z$</td>
<td>Z</td>
<td>Temperature ratio $T/T_w$ for integration routine, nondimensional</td>
</tr>
<tr>
<td>$\zeta_p$</td>
<td>ZETAP</td>
<td>Profile number for rectangular plan extended surface, nondimensional</td>
</tr>
</tbody>
</table>

**SUBSCRIPTS**

- a: Surface facing the sun (if applicable)
- av: Arithmetic average
- b: Surface in shade (if applicable)
<table>
<thead>
<tr>
<th>Equations</th>
<th>Computer Program</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>(I)</td>
<td></td>
<td>Duct section &quot;I&quot;</td>
</tr>
<tr>
<td>(L)</td>
<td></td>
<td>Fin &quot;L&quot;</td>
</tr>
<tr>
<td>1</td>
<td></td>
<td>Denotes entrance to duct section (if applicable)</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>Denotes exit from duct section (if applicable)</td>
</tr>
</tbody>
</table>
PROGRAM 4-1 DESCRIPTION

This program is adapted to systems having the geometry shown in Fig. 1, and it is described below in detail. Much of the work presented is common to the two programs and the overlapping portions will not be repeated.

The heat transfer from an element of fin surface depicted in Fig. 2 in a radiative and convective environment can be written as

\[ dq = \left[ C_1 T^4 - C_2 + h_a (T - T_{aa}) + h_b (T - T_{ab}) \right] dA. \]  

(1.1)

This equation is used to calculate the approximate effective environmental temperature. It is applicable to an element of fin or duct surface, but the two surfaces may have different temperatures. Since the fin is assumed to be the controlling heat exchange member, the environmental temperature for this surface is taken as the one for the system. Calculation of the true system environmental temperature did not seem worth the extra effort to obtain it. The fin environmental temperature is obtained by assuming in Eq.(1.1) that \( dq = 0 \) and \( T = T_e \). Therefore,

\[ F_e = C_1 T_e^4 + h_t T_e - (C_2 + h_{at}) = 0 \]  

(1.2)

where

\[ h_t = h_a + h_b \]  

(1.3)

and

\[ h_{at} = h_a T_{aa} + h_b T_{ab}. \]  

(1.4)

If the convective heat transfer coefficients \( h_t = h_{at} = 0 \),

\[ T_e = \left( \frac{C_2}{C_1} \right)^{\frac{1}{4}}. \]  

(1.5)
If the radiative heat transfer is zero \( C_1 = C_2 = 0 \),

\[
T_e = \frac{h_{at}}{h_t}. \tag{1.6}
\]

If both modes of heat transfer are present, Newton's method is used for obtaining \( T_e \). For these calculations

\[
T_e = T_e' - \frac{\frac{dF_e}{dT_e}}{\frac{dF_e}{dT_e}} \tag{1.7}
\]

where

\[
\frac{dF_e}{dT_e} = 4C_1T_e^3 + h_t. \tag{1.8}
\]

For the first approximation \( T_e' \) is taken equal to the average value calculated from Eqs. (1.5) and (1.6).

The constants \( C_1 \) and \( C_2 \) for the environment of Fig. 1 are

\[
C_1 = (\varepsilon_a + \varepsilon_b)\sigma = (\varepsilon_a + \varepsilon_b)0.1713\times10^{-8} \tag{1.9}
\]

[Ref. 1, Eq. 2]

and

\[
C_2 = S_e (\alpha_a \cos \theta_p + F_a \alpha_a \rho \cos \theta_m + F_{ax} \alpha_a \rho \cos \theta_x + F_{b x} \alpha_b \rho \cos \theta_b) + 0.1713\times10^{-8} \left[ (F_a \varepsilon_a + F_b \varepsilon_b) \varepsilon T_m^4 + (F_{ax} \varepsilon_a + F_{b x} \varepsilon_b) \varepsilon T_x^4 \right] + 0.01(\varepsilon_a + \varepsilon_b). \tag{1.10}
\]

[Ref. 1, Eq. 3]

The heat transfer from an element of fin shown in Fig. 1 can be written as
\[
d_{qf} = \frac{-k\frac{\delta}{h}T_w}{L_h} \left( \frac{d\ell}{d\omega} \right) dL = yL_h \eta_a dL = yL_e dL
\]

where

\[
y = C_1 T_w^{4} - C_2 + T_w h_t - h_{at}
\]

\[
w = \frac{L}{L_h}
\]

and

\[
Z = \frac{T}{T_w}
\]

Thus \( y \) is the heat exchanged per hour per unit area at the wall temperature.

The heat transfer from an element of duct can be obtained by adding the amounts resulting from both radiation and convection. However, in this program each mode of heat transfer is calculated differently. As explained in Ref. 2, the heat transferred by radiation from a fin and tube system can be readily evaluated with excellent accuracy using the projected area of the tube. In this program the heat transferred by convection is assumed to take place from the exposed area of the tube. With these areas the heat transfer is

\[
d_{qd} = D_o (C_1 T_w^{4} - C_2) + L_c (h_t T_w + h_{at}) dL
\]

where

\[
L_c = \frac{\tau D}{2} - \delta_h
\]

The equivalent length for the fin is equal to the product of the fin length, \( L_h \), and the area effectiveness, or

\[
L_e = \eta_a L_h
\]
This equation can also be expressed in terms of the dimensionless temperature gradient, \((dZ/dw)_1\), at the fin root, and

\[
L_e = -L_h \left( \frac{dZ}{dw} \right)_1 / \left[ \zeta_p (1-C_S) + F_h - F_{af} \right]
\]  

[Ref. 1, Eq. 31]  
(modified slightly)

Approximate values for the wall temperature, \(T_{ws}\), and \((dZ/dw)_1\) at the duct entrance are obtained by using the method of Ref. 2. However, the approximate equations (14) through (17) of Ref. 1 are used for calculating the fin performance. The actual value of \((dZ/dw)_1\) is then obtained at the approximate wall temperature by the numerical integration procedures described in Ref. 1, Eqs. (4) through (23).

It is also necessary to establish the numerical value for the equivalent fin length at the end of the duct and, for certain conditions, at an intermediate point, depending upon problem conditions. Fig. 3 is presented to depict the sequence of events and the procedure. Steps (1) and (2) have been explained above. Step (3) is quite obvious. The fluid temperature calculated for step (4) for both cooling and heating is

\[
T_{f\text{end}} = T_{f1} - \varepsilon(T_{f1} - T_e).
\]  

(1.19)

The exit (end) wall temperature is established by assuming \((T_{f1} - T_{ws})\) remains constant. In the event that the predicted \(T_{wf}\) curve crossed the \(T_e\) line, a more appropriate wall temperature is arbitrarily taken wherein

\[
T_{wf} = T_{f\text{end}} - 0.8(T_e - T_{f\text{end}}).
\]  

(1.20)

This wall temperature is used in recalculating the fluid temperature at the exit conditions.
The end wall temperature is used to calculate the equivalent length \( L_{ef} \) for the fins at the end. The integration procedure is almost identical to the one used at the duct entrance. The magnitude of the change in \( L_{es} \) from the entrance to \( L_{ef} \) at the end is then calculated. If less than a 10% change has occurred a linear equation is derived to pass through the points determined by the two temperatures and the two equivalent lengths.

For this case

\[
L_e = mT_w + b
\]

where

\[
m = \frac{L_{ef} - L_{es}}{T_{wf} - T_{ws}}
\]

and

\[
b = L_{es} - mT_{ws}.
\]

If more than a ten per cent change takes place, the equivalent length \( L_{em} \) at a temperature \( T_{wm} \) midway between \( T_{ws} \) and \( T_{wf} \) is calculated. A polynomial curve fit is then used wherein

\[
L_e = T_w (T_{wa} A_a + B_b) + C_c
\]

and where \( A_a, B_b, \) and \( C_c \) are computed from the three points: start, midpoint, and finish. Since the machine calculates the coefficients for only one case (\( m \) and \( b \) will be zero if the polynomial fit is used and \( A_a = B_b = C_c = 0 \) if a linear fit is used), the equation for \( L_e \) can be written as

\[
L_e = T_w (T_{wa} A_a + B_b) + C_c + mT_w + b.
\]
The convective heat transfer coefficient between the duct fluid and the duct wall is calculated by the "AICH" subroutine. Two methods are used depending upon the value of Reynolds number. This number is

\[ N_{re} = \frac{D_i V_p}{\mu} = \frac{D_i \dot{w}_d}{A_d \mu} \quad (1.26) \]

If \( N_{re} < 2000 \), the flow is considered laminar. If \( N_{re} > 2000 \), the flow is considered turbulent. In the laminar region the coefficient is calculated from Nusselt number, which is part of the input data. A table of values for circular and other shaped ducts given in Ref. 3, p 103. From the Nusselt number the fluid convective heat transfer coefficient is calculated or

\[ h = \frac{k N_{nu}}{D_i} \quad (1.27) \]

[Ref. 4, Eq. 7-30]

In the turbulent region a set of equations is used, wherein

\[ h = \frac{0.0384 \dot{w}_d C_p (N_{re})^{-1/4}}{A_d [1 + 1.5(N_{pr})^{-1/6}(N_{re})^{-1/6}(N_{pr} - 1)]} \quad (1.28) \]

[Ref. 4, Eq. 8.14]

where Prandtl number,

\[ N_{pr} = \frac{C_p \mu}{k} \quad (1.29) \]

Both Prandtl and Reynolds numbers are evaluated at the reference temperature

\[ T^* = \frac{T_b - \left(0.01 \frac{N_{pr}}{40}(T_b - T_s)\right)}{\left(N_{pr} + 72\right)} \quad (1.30) \]

[Ref. 4 Eq. (9.16)]
The convective heat transfer coefficient between the duct fluid and the duct wall is calculated by the "AICH" subroutine. Two methods are used depending upon the value of Reynolds number. This number is

$$N_{re} = \frac{D_i V \rho}{\mu} = \frac{D_i \dot{W}_d}{A_d \mu}.$$  \hspace{1cm} (1.26)

If $N_{re} < 2000$, the flow is considered laminar. If $N_{re} > 2000$, the flow is considered turbulent. In the laminar region the coefficient is calculated from Nusselt number, which is part of the input data. A table of values for circular and other shaped ducts given in Ref. 3, p 103. From the Nusselt number the fluid convective heat transfer coefficient is calculated or

$$h = \frac{k \ N_{nu}}{D_i}.$$  \hspace{1cm} (1.27) \hspace{0.5cm} \text{[Ref. 4, Eq. 7-30]}

In the turbulent region a set of equations is used, wherein

$$h = \frac{0.0384 \dot{W}_d C_p (N_{re})^{-1/4}}{A_d [1 + 1.5(N_{pr})^{-1/6}(N_{re})^{-1/6}(N_{re} - 1)]}.$$  \hspace{1cm} (1.28) \hspace{0.5cm} \text{[Ref. 4, Eq. 8.14]}

where Prandtl number,

$$N_{pr} = \frac{C_p \mu}{k}.$$  \hspace{1cm} (1.29)

Both Prandtl and Reynolds numbers are evaluated at the reference temperature

$$T^* = T_b - \frac{(.01 \ N_{pr} + 40)(T_b - T_s)}{(N_{pr} + 72)}.$$  \hspace{1cm} (1.30) \hspace{0.5cm} \text{[Ref. 4 Eq. (9.16)]}
The temperature relationship between the fluid and the wall is established by equating the internal and external heat exchange. From the fluid to the wall,

$$dq = h_p(T_f - T_w)dL_w.$$  \hfill (1.31)

From the external surface to the environment,

$$dq = q_s dL_w.$$  \hfill (1.32)

where

$$q_s = 2y[T_w(T_w A_a + B_b) + C_c + mT_w + B] + D_o (C_1T_w^4 - C_2) + L_c (h_t T_w - h_{at}).$$  \hfill (1.33)

When Eqs. (1.31) and (1.32) are combined:

$$F_w = q_s - h_p(T_f - T_w)$$  \hfill (1.34)\

where $F_w$ is a term which approaches zero in the convergence processes.

Eq. (1.34) is employed in several ways depending upon the known conditions. At the duct entrance the fluid temperature is known. The wall temperature is calculated in a linear interpolation loop followed by Aitken's method as described in Ref. 5, pp 136 to 153. The input fluid temperature and the environmental temperature, $T_e$, set the initial values for the temperature points in the interpolation loop. The wall temperature lies between these two extremes. One or the other temperature points is held during the interpolation process depending upon whether the duct fluid is being heated or cooled. Aitken's method speeds up the otherwise slow convergence.
when the true wall temperature is approached. At other stations along the duct the wall temperatures are known and the associated fluid temperatures are iteratively calculated using variable fluid properties with Eq. (1.34).

The duct length required to cause the change in the wall temperature, and the associated change in the fluid temperature is calculated from basic heat transfer equations. The heat transferred from the duct fluid in passing through an element is

$$dq = -\dot{\omega}_d c_p dT_f$$  \hspace{1cm} (1.35)

Combining Eqs. (1.31) and (1.35)

$$dL_w = \frac{-\dot{\omega}_d c_p dT_f}{h_p(T_f - T_w)}$$  \hspace{1cm} (1.36)

this equation is modified to apply to a finite length of duct. For the heat transfer coefficient an arithmetic average is used and

$$h_{av} = \frac{h_1 + h_2}{2}$$  \hspace{1cm} (1.37)

For the temperature difference between the fluid and the wall an arithmetic average is used if the change in $(T_{f1} - T_{w1})$ and $(T_{f2} - T_{w2})$ is less than five degrees in a section, or

$$\Delta T_m = (T_f - T_w)_{av} = \frac{(T_{f1} - T_{w1}) + (T_{f2} - T_{w2})}{2}$$  \hspace{1cm} (1.38)

If more than five degrees of change are calculated, a log mean temperature difference is used, or

$$\Delta T_m = \frac{(T_{f1} - T_{w1}) - (T_{f2} - T_{w2})}{\log_e \left[\frac{(T_{f1} - T_{w1})}{(T_{f2} - T_{w2})}\right]}$$  \hspace{1cm} (1.39)
For a finite section length the fluid temperature change is

\[ \text{d}T_f = T_{f2} - T_{f1} \]  \hspace{1cm} (1.40)

with these modifications Eq. (136) becomes

\[ L_w = \frac{\dot{w}_d C_p (T_{f1} - T_{f2})}{h_{av} p \Delta T_m} = \frac{q}{h_{av} p \Delta T_m} \]  \hspace{1cm} (1.41)

The weight of liquid trapped in a tube is the sum of that trapped in each section or

\[ W_L = \sum \rho_{av} A_d L_w \]  \hspace{1cm} (1.42)

The duct weight is

\[ W_d = \rho_d L_{ws} \pi (D_o^2 - D_i^2) \]  \hspace{1cm} (1.43)

The weight of a fin is

\[ W_f = \rho_f L_{ws} L_h \delta_h (1 + \delta_r) \]  \hspace{1cm} (1.44)

and the total weight is

\[ W_t = (W_L + W_d + 2W_f)N \]  \hspace{1cm} (1.45)

The plan area of the system is,

\[ A_p = (D_o + 2L_h)N L_{ws} \]  \hspace{1cm} (1.46)
Program 4-2 Description

This program is set up to handle a variety of duct and fin configurations as illustrated by Fig. 4. In order to make it flexible the number of extended surfaces attached to the duct and the divisions of the duct circumference are specified by the input data. The heat transfer to the extended surface is calculated for each surface from its equivalent lengths. The heat transfer from the duct is obtained by adding the radiative and convective heat transfer from each of the sections. The temperature is assumed constant around the periphery for all sections.

When the heat transfer to the environment from a given extended surface is independent of the other extended surfaces, the programs in Ref. 1 can be used to calculate the equivalent length for the surface. Unfortunately, when the extended surfaces are oriented in such a way that they can "see" each other, the heat transfer by radiation to the environment is restricted and no known programs are available for calculating the equivalent length. At present the user will have to approximate the reduction in heat transfer and adjust the values of $L_e$ accordingly. The approximations will be quite accurate if the interfering surfaces are insulated, or if the ambient fluid is opaque to radiation.

The cross sectional flow area for the duct can be calculated if the duct is circular. If not, the area and duct perimeter must be included in the input data. The effective internal diameter for a non-circular duct is

$$D_i = \frac{4A_d}{p} . \quad (2.1)$$
For circular ducts

\[ A_d = \frac{\pi D_i^2}{4} \]  

(2.2)

and

\[ p = \pi D_i \]  

(2.3)

To account for the two modes of heat transfer, the calculations for a section of duct circumference are divided into two parts, convection and radiation. The division is necessary because of the use of projected areas (if applicable) for calculating radiative heat transfer. Also it is assumed that the duct itself might be of a complicated configuration requiring several sections to represent the heat transfer. For any section, the heat transfer can be written as

\[ dq(I) = \left\{ S_r(I) \left[ C_{1d}(I) T_w^4 - C_{2d}(I) \right] + S_c(I) h_d(I) \left[ T_w - T_a(I) \right] \right\} dL. \]  

(2.4)

The effective section lengths \( S_r(I) \) and \( S_c(I) \) are thus chosen independently. In some cases, for example, a duct without fins, equal values for \( S_r(I) \) and \( S_c(I) \) should be specified. The heat transfer from an element of duct circumference is

\[ dq_d = (G_{d1} T_w^4 + G_{d2} T_w - G_{d3})dL \]  

(2.5)

where

\[ G_{d1} = \sum_{I=1}^{n} S_r(I) C_{1d}(I) \]  

(2.6)

\[ G_{d2} = \sum_{I=1}^{n} S_c(I) h_d(I) \]  

(2.7)
and
\[ G_{ds} = \sum_{I=1}^{n} \left[ S_r(I) C_{a d}(I) + S_c(I) h_d(I) T_a(I) \right]. \] (2.8)

For an extended surface the heat transfer is
\[ dq(L) = \left\{ C_1(L) T_w^4 - C_2(L) + T_w \left[ h_a(L) + h_b(L) \right] - \left[ h_a(L) T_{a a}(L) + h_b(L) T_{a b}(L) \right] \right\} L_e(L) dL. \] (2.9)

Then, for
\[ G_2(L) = h_a(L) + h_b(L) \] (2.10)

and
\[ G_3(L) = h_a(L) T_{a a}(L) + h_b(L) T_{a b}(L) + C_3(L), \] (2.11)

\[ dq(L) = \left[ C_1(L) T_w^4 + C_2(L) T_w - G_3(L) \right] L_e(L) dL. \] (2.12)

Data for fin No. 1 and Eq. (2.12) are used with Newton's method for establishing the effective environmental temperature. Setting, \( dq = 0 \) and \( T_e = T_w \)

\[ F_e = C_1(1) T_e^4 + C_2(1) T_e - G_3(1) = 0 \] (2.13)

\[ \frac{dF_e}{dT_e} = 4 C_1(1) T_e^3 + C_2(1) \] (2.14)

and
\[ T_e = T_e' - F_e \left\{ \frac{dF_e}{dT_e} \right\}. \] (2.15)
The outlet fluid temperatures $T_{fa}$ is used as the first approximation for $T'_e$. The value of $T_e$ is accepted when

$$F_e \leq .001 \quad . \tag{2.16}$$

Should the system have no fins, the data for duct No. 1 and Eq. (2.4) is substituted for Eq. (2.12) above.

The value of $L_e$ at any point along the duct is evaluated from the wall temperature at that point. Approximate wall temperatures $T_{ws}$ and $T_{wf}$ at the duct entrance and exit, respectively, are used in establishing the values for $L_{es}$, and $L_{ef}$. A linear equation is derived which passes through the points established by the two lengths and the two temperatures. The slope of the line,

$$m(L) = \frac{L_{ef}(L) - L_{es}(L)}{T_{wf}(L) - T_{ws}(L)} \quad . \tag{2.17}$$

The intercept

$$b(L) = L_{es}(L) - m(L)T_{ws}(L) \quad . \tag{2.18}$$

The equation for $L_e(L)$ is

$$L_e(L) = m(L)T_w + b(L) \quad . \tag{2.19}$$

The other extended surfaces are handled as illustrated for surface (L) above.

The relationship between the fluid and wall temperatures is obtained from the system heat transfer equations. The heat from an element of duct
length can be obtained by adding the heat from the extended surfaces and the duct or

\[ dq = q_s dL. \]  (2.20)

Where

\[ q_s = K_0 + K_1 T_w + K_2 T_w^2 + K_4 T_w^4 + K_5 T_w^5. \]  (2.21)

\[ K_0 = - \left( G_{d3} + \sum_{L=1}^{n'} G_3(L) b(L) \right) \]  (2.22)

\[ K_1 = G_{d3} + \sum_{L=1}^{n'} \left[ G_2(L) b(L) - G_3(L) \right] m(L) \]  (2.23)

\[ K_2 = \sum_{L=1}^{n'} G_2(L) m(L) \]  (2.24)

\[ K_4 = G_{d1} + \sum_{L=1}^{n'} \left[ C_1(L) b(L) \right] \]  (2.25)

\[ K_5 = \sum_{L=1}^{n'} C_1(L) m(L) \]  (2.26)

and

\[ n' = \text{number of fins attached to a duct}. \]

To calculate the wall temperature from a given fluid temperature Eq. (1.35) is combined with Eq. (2.20) and

\[ F_w = q_s - h_p (T_f - T_w). \]  (2.27)

where \( F_w \to 0 \) in the iteration processes.
In this program, the fluid temperatures are specified at inlet and outlet of the duct. The corresponding wall temperatures are calculated by subroutine "TWALL" which uses a linear interpolation loop followed by Aitken's method. This procedure is similar to the method used with Eq. (1.35) at the duct inlet in Program 4.1.

The wall temperature change in a section is calculated from the overall change and the number of sections or

\[ \Delta T = T_{W1} - T_{WS} = (T_{W1} - T_{wend})/FMESH. \]  

(2.28)

Thus equal temperature drops are taken for each section.

Section lengths and heat transfer quantities are calculated as in Program 4.1. In this case the ducts may be of irregular shape and the actual fin lengths are not used by the program, therefore, the weights of the duct and fins are not calculated.
CONCLUSIONS AND RECOMMENDATIONS

These programs can be used to solve a myriad of heat exchanger problems. While they have been specifically designed for problems where the heat exchange with the environment is by the combination of convection and radiation, they can be used where the transfer is restricted to either one or the other. Problems involving only convection environments are also solvable by conventional approaches and such a program would be more economical with machine time.

The rigorous mathematical treatment of the combined effects of radiation and convection heat exchange systems has been avoided in the past because of the difficulties in solving the nonlinear differential equations. These difficulties are made acute by the large number of variables which influence the exchanger performance. Radiative exchange problems have been greatly simplified by the use of dimensionless parameters. Comparable studies with the combined effects of radiation and convection are sorely needed. Performance curves or charts would aid in the understanding of problems and in interpreting the results obtained for a given case. The programs presented herein combined with those of Ref. 1 should provide interested parties with the basic tools for making such a study.

Temperature drops in the duct walls and temperature variation effects around the duct periphery have been neglected. These effects could be taken into account in program 4-2 if desired. Wall resistance can be added to film resistance to obtain a total resistance. The overall effect can be approximated by multiplying the fluid thermal conductivity by the ratio of fluid resistance to total resistance. The peripheral temperature distribution
effects can be taken into account by adjusting the values used for $S_r(I)$ and $S_c(I)$. However, under normal conditions these effects are small enough to be neglected.

Inter-radiation effects between fins which "see" each other will introduce errors which are difficult to approximate. Very little work has been done in this field.

A number of problems such as those involving condensing or evaporating fluids or with cooling or heating gases can not be satisfactorily solved with these programs. However, the modifications to make them solvable are not extensive and therefore a continuation of work in this field is recommended.
REFERENCES


Fig. 1. Program 4-1 Heat Exchanger Radiative Environment
Fig. 2. Program 4-1 Heat Exchanger Configuration
37

2 Approx. wall temperature, $T_{ws}$

TEMPERATURE

HEATING

2 Approx. wall temperature, $T_{ws}$

COOLING

$T_{f1} - T_{ws}$

$T_{w1}$

$T_f$

$T_w$

$T_{fl}$

0.8($T_e - T_{fend}$)

4 Fluid temperature change using effectiveness, $\varepsilon$

5 $T_{f1} - T_{ws}$

6 A check is made on this point to insure against crossing $T_e$.

$\Delta T = (T_{w1} - T_{wf})/\text{MESH}$

$\Delta T$ used to calculate section lengths

3 $T_{f1} - T_{ws}$

8 $L_w$

Length required to get input $\varepsilon$

Total length calculated by program

Length required to get input $\varepsilon$

Fig. 3. Program 4-1 Calculation Sequence
Fig. 4. Program 4-2 Representative Configuration
Included in this Appendix are flow diagram, deck setup, compiled listing of the main program, and subroutines for Programs 4.1 (Figs. A-1 through A-6) and 4.2 (Figs. A-7 through A-12).

The following subroutines are common to both programs and are illustrated only once.

Subroutine ENTERP (Fig. A-13)
Subroutine DECRD (Fig. A-14)

Subroutine "DECRD" gives instructions for entering input data. However, the actual data numbers are given along with the problem input data in Appendix B.
START
READ EFF CURVE FIT DATA
PRINT CURVE FIT DATA
READ FORMAT STATEMENTS
READ MATERIALS PROPERTIES (DECRD)
READ VARIABLE DECIMAL DATA (DECRD)
COMPUTE CONSTANTS
PRINT INPUT DECIMAL DATA
COMPUTE $c_1$ AND $c_2$
COMPUTE ENVIRON TEMP
INITIALIZE ENTRANCE TW LOOP

Fig. A-1  Program 4-1 Flow Diagram
1. INCREMENT LOOP COUNTER

2. PRINT FAILURE MESSAGE

3. MODIFY INLET TW (NEWTON'S) NO

4. FE = 0? YES

5. INTEGRATE FINS AT INLET (ENTGRV) (CONT.)

6. COMPUTE EFFECT. LENGTH AT INLET

7. COMPUTE INLET C3 ZETAP FH FAH

8. COMPUTE INLET: RAD. EFFECT. CONV. EFFECT. EFFECT. LENGTH

9. COMPUTE INLET FLUID DENSITY (ENTERP)

10. COMPUTE HEAT TRANSFER COEFFICIENT (AICHP)

11. COMPUTE FE

12. NO

13. LOOP COUNTER > 25?

14. YES
Fig. A-1 Program 4-1 Flow Diagram (cont.)
5

COMPUTE EFFECT LENGTH AT MIDPOINT

COMPUTE PARABOLIC FIT FACTORS

INCREMENT LOOP COUNTER

LOOP COUNTER>25?

YES

PRINT FAILURE MESSAGE

NO

COMPUTE HEAT TRANSFER (AICH)

COMPUTE FWH

COMPUTE WALL TEMPERATURES

SET NEW TRIAL TW

CHANGE IN TW'S SMALL?

YES

CALCULATE FINAL TW1

INITIALIZE DUCT SECTION LOOP

COMPUTE: TW2 ELE2 YT QS

INCREMENT LOOP COUNTER

6
LOOP COUNTER > 25?

YES

PRINT FAILURE MESSAGE

NO

COMPUTE TF2

COMPUTE DENSITY OF FLUID (ENTERP)

COMPUTE HEAT TRANSFER COEFFICIENT (AICH)

TF2 CHANGE SMALL?

NO

COMPUTE AVERAGES ABC

YES

ABC SMALL?

YES

COMPUTE ARITHMETIC DELTM

CALCULATE LOG DELTM

COMPUTE TOTAL HEAT TRANSFER

COMPUTE DUCT LENGTH

COMPUTE PRESSURE CHANGE (PRDP)

COMPUTE & TOTAL WEIGHT OF LIQUID

NO

COMPUTE TF2

COMPUTE AVERAGES ABC

YES

ABC SMALL?

NO

CALCULATE LOG DELTM

COMPUTE TOTAL HEAT TRANSFER

COMPUTE DUCT LENGTH

COMPUTE PRESSURE CHANGE (PRDP)

COMPUTE & TOTAL WEIGHT OF LIQUID

8
SET INLET VALUES TO OUTLET FOR NEW SECTION

LAST SECTION?

COMPUTE WEIGHTS AND AREA

PRINT FINAL OUTPUT DATA

PRINT SECTION DATA
Variable Case
Decimal Data
Requires (-)
In Column 1 of
Last Card of
Every Case

Last Card of
Material
Properties
Data Requires
A (-) in
Column 1

300 Fluid Density Lb/cu ft vs. T, R
200 Fluid Conductivity, Btu/hr R vs. T, R
LOC 100 Visc (centipoise) vs. T, R

INPUT DECIMAL DATA

Example of
Table Format
LOC
100 Number of Pair of x, y
101 x₁ = Temp R
102 y₁ = Visc

When using a constant
instead of a curve
enter in first
location of Table as a
negative

100 -VISC

Fig. A-2 Composite Deck Setup Program 4-1
C PROGRAM NO. 4-1

C DIMENSION A(3), B(3), C(3), TW(6), FG(6), Y2(6), TITLE(16)
DIMENSION F1(216), F2(12), F3(24), F4(96), F5(144), F6(12), F7(12),
VF8(14), F9(12)
EQUIVALENCE (DA(1), D0), (DA(2), DI), (DA(3), ENT), (DA(4), WDOT),
V(DA(5)), ELH), (DA(6), FINTH), (DA(7), FINTC), (DA(8), RHOFM),
V(DA(9), RHOTM), (DA(11), FNK), (DA(12), CP),
V(DA(14), HA), (DA(15), HB), (DA(16), TAA), (DA(17), TAB), (DA(18), ALPHAA),
V(DA(19), ALPHAB), (DA(20), EPSA), (DA(21), EPSB), (DA(22), EPSX),
V(DA(23), FA), (DA(24), FAX), (DA(25), FBA), (DA(26), FBX), (DA(27), RHOM),
V(DA(28), RHOX), (DA(29), THETAP), (DA(30), THETAM), (DA(31), THETAX),
V(DA(32), TM), (DA(33), TX), (DA(34), EPSM), (DA(37), SC), (DA(38), BIGE),
V(DA(39), FMESH), (DA(40), P), (DA(41), CKH)
COMMON DA(400), T(5), TB, TS, ACH, RE, V, WDOT, AD, RHO, REAV, RHOAV, ELW,
V VAV, DP, FH, FAH, ZETAP, C3, DZ1, DELTAR, RE2, TSTAR

C READ EFF CURVE FIT
READ 1001, (A(I), I=1,3), (B(I), I=1,3), (C(I), I=1,3)
1001 FORMAT(3F12.0)
PRINT 1002, A, B, C
1002 FORMAT(14H1EFF CURVE FIT/1H03E12.5/1H 3E12.5)
C READ PERMANENT DATA (FORMAT STATEMENTS)
READ 1000, F1, F2, F3, F4, F5, F6, F7, F8, F9
1000 FORMAT(12A6)
C READ MATERIALS PROPERTIES
CALL DECRD(DA)
C PRINT FLUID PROPERTIES
K=100
PRINT 1235
1235 FORMAT(96H1 FLUID PROPERTIES (=) = CONSTANT, TABLE FORMAT
V= NO PTS, X1,Y1=--XN,YN, X= TEMP (R)
V 96HLOC 100 = FLUID VISC, CENTIPOISE, 200 = FLUID K BTU/HR FT R, 3
V00 = FLUID DENSITY, LB/CU FT / )
DO 1900 KKK=1,3
K8=DA(K)
K9=K+2*K8
PRINT 2500, (K1, DA(K1), K1=K, K9)
2500 FORMAT(3X15F13.6, I8, F13.6, I8, F13.6, I8, F13.6, I8, F13.6)
K=K+100
1900 CONTINUE
C READ VARIABLE DECIMAL DATA
80 READ 3001, TITLE
3001 FORMAT (16A5)
CALL DECRD(DA)
PI=3.1415926
AD=PI*D1**2/4.
WDOTU=WDOT/ENT
DELTAR=FINTC/FINTH
HT=HA+HB
HAT=HA*TAA+HB*TAB
ELC=PI*D0/2.-FINTH
TF=DA(13)
AA=0.
BB=0.
CC=0.
FM=0.

Fig. A-3 Program 4-1 Listing
BP=0.
C3MID=0.
ZETAPM=0.
FHMI=0.
FAHMD=0.
ELEMD=0.
PRINT 3002
3002 FORMAT (1H1,16A5//)
PRINT F1,(J,DA(J),J=1,41)
C1=.1713E-8*(EPSA+EPSP)
C2= SC *(ALPHAA*COS(THETAP*.01745329)+FA*ALPHAA*RHOM*COS(THETAM*.01745329)+FB*ALPHAB*RHOM
V*COS(THETAM*.01745329)+FBX*ALPHAB*RHOX*COS(THETAX*.01745329)+
VEPSM*TM**4*.1713E-8*(FA*EPSCA+FB*EPSB)+EPSX*TX**4*.1713E-8*(FA*
VEPSA+FBX*EPSB)+.81*(EPSA+EPSB)

C COMPUTE ENVIRONMENTAL TEMPERATURE
IF(C1)101,103,101
101 IF(HA)104,102,104
102 TE=SQRT(SQRT(C2/C1))
GO TO 109
103 TE=HAT/HT
GO TO 109
104 LC1=0
TE=.5*(SQRT(SQRT(C2/C1)))+HAT/HT)
105 LC1=LC1+1
IF(LC1-25)107,107,106
106 PRINT F2,TE
GO TO 80
107 FE=C1*TE**4-C2+HA*(TE-TA)+HB*(TE-TB)
IF(ABS(FE)<.001)109,109,108
DFEDT=4.*C1*TE**3+HT
TE=TE-FE/DFEDTE
GO TO 105
C COMPUTE APPROXIMATE INLET WALL TEMPERATURE
109 TW1=TF1-2*(TF1-TE)
PERIM=PI*DI
LC2=0
115 LC2=LC2+1
IF(LC2-25)121,121,120
120 PRINT F3,TW1,2ZETAP1,C31,FH1,FAH1
GO TO 80
121 C31=C2/(C1*TW1**4)
ZETAP1=C1*TW1**3*ELH**2/(FNK*FINTH)
FH1=(ELH**2*HT)/(FNK*FINTH)
FAH1=(ELH**2*HAT)/(FNK*FINTH*TW1)
IF(ZETAP1=100.)126,125,125
125 NA=3
GO TO 127
126 IF(ZETAP1)128,127,128
128 NA=2.+ALOG10(ZETAP1)
IF(NA,LT,1) NA=1
127 EFR1=(1.-C31)*(ZETAP1*(ZETAP1A(NA)+B(NA))+C(NA))
EC1=TANH(FH1**.5)/FH1**.5
DZDW1=-ZETAP1*EFR1-(FH1-FAH1)*EFC1
FH=FH1
FAH=FAH1
ZETAP=ZETAP1
C3=C31
ELE1=(-ELH*DZDW1)/(ZETAP1*(1.-C31)+FH=FAH)

Fig. A-3 Program 4-1 Listing (cont.)
TB=TF1
TS=Tw1
RHO=ENTERP(TF1,DA(300))
RHO1=RHO
CALL AICH
ACH1=ACH
RE1=RE
V1=V
Y=C1*TW1**4+C2+TW1*HT=HAT
FE=2.**Y*ELE1+DO*(C1*TW1**4+C2)+ELC*(HT*TW1-HAT)-PERIM*ACH1*
V(TF1-TW1)
IF(ABS(FE)-.001)150,150,140
140 DFEDTw=(4.*C1*TW1**3+HT)*2.*ELE1+4.*C1*DO*TW1**3+ELC*HT+ACH1*PERIM*ACH1*
TW1=TW1-FE/DFEDTw
GO TO 115
C COMPUTE EFFECTIVE LENGTH
150 T(5)=DZDW1
CALL ENTGRT
DZDW1=DZ1
ELE1=(-ELH*DZDW1)/(ZETAPI*(1.-C31)+FH=FAH)
C COMPUTE OUTLET END CONDITIONS
TFEND=TF1-BIGE*(TF1-TE)
TWEND=TFEND-(TF1-TW1)
FW=.8
IF(ABS(TF1-TWEND)-ABS(TF1-TE))161,161,161
161 TWEND=TFEND+FW*(TE-TFEND)
162 C3END=C2/(C1*TWEND**4)
ZETAPE=C1*TWEND**3*ELH**2/(FNK*FINTH)
FHEND=(ELH**2*HT)/(FNK*FINTH)
FAHEND=(ELH**2*HAT)/(FNK*FINTH*TWEND)
IF(ZETAPE-100,180,180,180
180 NA=3
GO TO 183
181 IF(ZETAPE)184,183,184
184 NA=2.+ALOG10(ZETAPE)
IF(NA.LT.1) NA=1
183 EFREND=(1.-C3END)*(ZETAPE*(ZETAPE*A(NA)+B(NA)+C(NA))
EFCEND=TANH(FHEND**.5)/FHEND**.5
DZDWE=-ZETAPE*EFREND-(FHEND-FAHEND)*EFCEND
IF((DZDWE1+DZDWE)/DZDW1)-1.)190,190,192
190 FW=FW-.1
IF(FW.EQ.0.) GO TO 191
GO TO 161
191 PRINT F9
GO TO 80
192 T(5)= DZDWE.
ZETAP=ZETAPE
C3=C3END
FH=HEND
FAH=FAHEND
CALL ENTGRT
DZDWE=DZ1
ELEEND=(-ELH*DZDWE)/(ZETAPE*(1.-C3END)+FHEND-FAHEND)
IF (ABS(ELEEND-ELE1)/ELE1)-.10) 200,200,201
200 FM=(ELEEND-ELE1)/(TWEND-TW1)
BP=ELE1-FM*TW1
GO TO 325
201 TWMID=.5*(TWEND+TW1)
C3MID=C2/(C1*TWMID**4)
ZETAPM=C1*TWMID**3*ELH**2/(FNK*FINTH)
FHMID=(ELH**2*HT)/(FNK*FINTH)
FAHMIK=(ELH**2*HAT)/(FNK*FINTH*TWMID)
IF((ZETAPM-100.)301,300,300
300 NA=3
GO TO 303
301 IF(ZETAPM)304,303,304
304 NA=2.+ALOG10(ZETAPM)
IF(NA.LT.1) NA=1
303 EFRMID=(1.-C3MID)*(ZETAPM*(ZETAPM*A(NA)+B(NA))+C(NA))
EFCMID=TANH(FHMID**5)/FHMID**5
DZDWM=-ZETAPM*EFRMID-(FHMID-FAHMIK)*EFCMID
T(5)=DZDWM
ZETAP=ZETAPM
C3=C3MID
FH=FHMID
FAH=FAHMIK
CALL ENTGRT
DZDWM=DZ1
ELEMID=(ELH*DZDWM)/(ZETAPM*(1.-C3MID)+FHMID-FAHMIK)
ZZ1=(ELE1-ELEMID)/(TW1-TWMID)
ZZ2=(ELEMID-ELEEND)/(TWMID-TWEND)
AA=(ZZ1-2ZZ2)/(TW1-TWEND)
BB=ZZ1-AA*(TW1+TWMID)
CC=ELE1-AA*(TW1**2-BB*TW1
C COMPUTE ENTRANCE WALL TEMPERATURE
325 IF (((TF1-TE)<=0T.0) GO TO 310
XH=TE
TW(1)=TF1
GO TO 320
310 XH=TF1
TW(1)=TE
320 LC3=0
Y1=C1*XH**4+C2+XH*HT-HAT
321 LC3=LC3+1
IF (LC3=25) 322,322,323
322 TS=Tld(l)
CALL AICH
WACH
FWH=Z.*Y1*(XH*(XH*AA+BB)+CC+FM*XH+BP)+DO*(C1*XH**4-C2)+ELC*(HT*
V XH-HAT)-(H*PERIM)*(TF1-XH)
L=0
20 DO 10 K=2,4
L=L+1
Y2(K-1)=C1*TW(K-1)**4=C2+TW(K-1)*HT-HAT
F6(K-1)=2.*Y2(K-1)*(TW(K-1)*(TW(K-1)*AA+BB)+CC+FM*TW(K-1)+BP)+DO*
V (C1*TW(K-1)**4=C2)+ELC*(HT*TW(K-1)-HAT)-(H*PERIM)*(TF1-TW(K-1))
10 TW(K)=(XH*FG(K-1)-TW(K-1)*FWH)/(FG(K-1)-FWH)
IF (L.LT.20) GO TO 25
PRINT 50,TW(1),TW(2),TW(3),TW(4),FWH,FG(1),FG(2),FG(3)
50 FORMAT(/3XHTW(1) = E15.8/3XHTW(2) = E15.8/3XHTW(3) = E15.8/
V 3XHTW(4) = E15.8/3XHTW(5) = E15.8/3XHTW(6) = E15.8/3XHTW(7) = E15.8/
V E15.8/3XHTW(8) = E15.8/3XHTW(9) = E15.8/3XHTW(10) = E15.8/3XHTW(11) = E15.8/
V 3XHTW(12) = E15.8/3XHTW(13) = E15.8/3XHTW(14) = E15.8/3XHTW(15) = E15.8/
V E15.8/3XHTW(16) = E15.8/3XHTW(17) = E15.8/3XHTW(18) = E15.8/
V E15.8/3XHTW(19) = E15.8/3XHTW(20) = E15.8/
STOP
25 IF (ABS(TW(3)-TW(4))<=.5) 30,30,27
27 TW(1)=TW(4)
GO TO 20
30 TW(1)=TW(2)-(TW(3)-TW(2))**2/(TW(4)+TW(2)-2.*TW(3))
    IF (ABS(TS-TW(1))-.2) 330,330,321
330 ACH1=ACH
    TW1=TW(1)
    DELT=(TW1-TWEND)/FMESH
    ELE1=TW1*(TW1*AA+BB+FM) + CC + BP
    NCOUNT=FMESH
    QSUM=0.
    DPSUM=0.
    ELWSUM=0.
    WLSUM=0.
    DO 500 J=1,NCOUNT
        ACH2=ACH1
        TW2=TW1-DELT
        ELE2=TW2*(TW2*AA+BB+FM) + CC + BP
        YT=C1+TW2**4-C2+TW2*HT-HAT
        QS=(2.*YT)*ELE2 + DO*(C1*TW2**4-C2) + ELC*(HT*TW2-HAT)
    500 CONTINUE
    LC4=LC4+1
    IF (LC4-.25) 337,337,336
336 PRINT F8*TF*PRE*P*RE
    GO TO 80
337 TF2P=TF2
    RE2P=RE2
    TF2=TW2+QS/(ACH2*PERIM)
    TB=TF2
    TS=TW2
    RHO=ENTERP(TF2,DA(300))
    RHO2=RHO
    CALL AICH
    ACH2=ACH
    RE2=RE
    V2=V
    IF (ABS(TF2P-TF2)-.25) 340,340,335
340 HAV=.5*(ACH1+ACH2)
    REAV=.5*(RE1+RE2)
    VAV=.5*(V1+V2)
    RHOAV=.5*(RHO1+RHO2)
    ABC=(TF1-TW1)-(TF2-TW2)
    IF (ABS(ABC)-5.0) 400,400,401
400 DELTM=(TF1-TW1+TF2-TW2)/2.0
    GO TO 402
401 DELTM=ABC/ALOG((TF1-TW1)/(TF2-TF2))
402 Q=WDOTD*CP*(TF1-TF2)
    ELW=G/(HAV*PERIM*DELTM)
    ELWSUM=ELWSUM+ELW
    QSUM=QSUM+Q*ENT
    CALL PRDP
    DPSUM=DPSUM+DP
    WL=RHOAV*PI*DI**2/4.*ELW
    WLSUM=WLSUM+WL
    PRINT F4*J,TF1,TF2,TW1,TW2,ELE1,ELE2,ACH1,ACH2,HAV,REAV,VAV,ELW,
    V DP,Q,RHOAV,WL
    TF1=TF2
    TW1=TW2
    ACH1=ACH2
    RE1=RE2
    V1=V2

Fig. A-3 Program Listing (cont.)
RH01=RH02
ELE1=ELE2
500 CONTINUE
WD=(PI*RHOTM*ELWSUM*(DO**2-DI**2))/4.
WF=(RHOFM*ELH*ELWSUM*FINTH*(1.+DELTAR))/2.
WF=(WL+WD+2.*WF)*ENT
AP=(DO+2.*ELH)*ELWSUM*ENT
PRINT F5,C1,C3MID,C3END,ZETAP1,ZETAPM,ZETAPE,FH1,FHMID,FHEND,
VFAH1,FAHMID,FAHEND,
VELE1,ELEMID,ELEEND,WSUM,WD,WF,WT,TE,ELWSUM,DPSUM,QSUM,AP,C1,C2
GO TO 80
END

Note: See following page for permanent Hollerith Listing.
### Input Data

<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>I/F</td>
<td>Outside Diameter (FT)</td>
<td>/F15.5,22H</td>
</tr>
<tr>
<td></td>
<td>Inside Diameter (FT)</td>
<td>/F15.5,13H</td>
</tr>
<tr>
<td></td>
<td>No. of Tubes</td>
<td>/F15.5,12H</td>
</tr>
<tr>
<td></td>
<td>Weight Flow (LBS/Hr)</td>
<td>/F15.5,32H</td>
</tr>
<tr>
<td></td>
<td>Fin Length (FT)</td>
<td>/F15.5,16H</td>
</tr>
<tr>
<td></td>
<td>Fin Thickness at Root (FT)</td>
<td>/F15.5,2H</td>
</tr>
<tr>
<td></td>
<td>Fin Thickness at Far Edge (FT)</td>
<td>/F15.5,31H</td>
</tr>
<tr>
<td></td>
<td>Density of Tube Material (LBS/CF)</td>
<td>/F15.5,36H</td>
</tr>
<tr>
<td></td>
<td>Not Used</td>
<td>/F15.5,9H</td>
</tr>
<tr>
<td></td>
<td>Specific Heat of Fin (BTU/FT HR)</td>
<td>/F15.5,27H</td>
</tr>
</tbody>
</table>

### Outside Diameter and Inside Diameter

- **Outside Diameter (FT):** 15.522H
- **Inside Diameter (FT):** 15.513H

### Number of Tubes

- **No. of Tubes:** 15.512H

### Weight Flow

- **Weight Flow (LBS/Hr):** 15.532H

### Specific Heat

- **Specific Heat of Fin (BTU/FT HR):** 15.527H

### Fluid Temperature

- **Fluid Temperature at Entrance (R):** 15.552H

### Heat Transfer Coefficient

- **Heat Transfer Coefficient Side A (BTU/LB HR):** 15.54H
- **Heat Transfer Coefficient Side B (BTU/LB HR):** 15.54H

### Ambient Temperature

- **Ambient Temperature Side A (R):** 15.54H
- **Ambient Temperature Side B (R):** 15.54H

### Alpha

- **Alpha A (FAH):** 15.513H
- **Alpha B (FAH):** 15.513H

### Epsilon

- **Epsilon A (FAH):** 15.531H
- **Epsilon B (FAH):** 15.531H

### Theta

- **Theta A (DEG):** 15.513H
- **Theta B (DEG):** 15.513H

### Rho

- **Rho A (LBS/CF):** 15.531H
- **Rho B (LBS/CF):** 15.531H

### Heat Exchanger Effectiveness

- **Heat Exchanger Effectiveness:** 15.518H

### Environment Temperature

- **Environment Temperature (R):** 15.518H

### Section No.

- **Section No. 1:** 15.518H
- **Section No. 2:** 15.518H
- **Section No. 3:** 15.518H

### Heat Transfer Coefficient

- **Heat Transfer Coefficient (BTU/HR FT SQ R):** 15.54H

### Total Weight of Liquid

- **Total Weight of Liquid (LBS):** 15.521H

### Heat Transfer

- **Heat Transfer (BTU/HR Tube):** 15.521H

### Plan Area

- **Plan Area (FT):** 15.524H

### Radiation Constant

- **Radiation Constant (C1):** 15.524H

### Heat Exchanger Effectiveness

- **Heat Exchanger Effectiveness:** 15.524H

---

**Note:** This Hollerith Listing follows Subroutine Listings. See Fig. A-2.
SUBROUTINE AICH
EQUIVALENCE (DA(1),D0),(DA(2),DI),(DA(12),CP),(DA(41),CKH)
COMMON DA(400),T(5),TB,TS,ACH,RE,V,WDOTD,AD,RHO,REAV,RHOAV,ELW,
V,VAV,DP,FP,FAH,ZETAP,C3,DZ1,DELTAR,RE2,TSTAR
V=WDOTD/(RHO*AD*3600.)
VISC=ENTERP(TB,DA(100))*2.4190297
RE=WDOTD*DI/AD
RE=REN/VISC
IF (RE<2000.) 410,410,500
410 FLK=ENTERP(TB,DA(200))
ACH=CHK*FLK/DI
RETURN
500 VISC=ENTERP(TS,DA(100))*2.4190297
FLK=ENTERP(TS,DA(200))
PRN=CP*VISC/FLK
TSTAR=TB-((.1*PRN+.4)*(TB-TS)/(PRN+.72.))
VISC=ENTERP(TSTAR,DA(100))*2.4190297
RE=REN/VISC
ACH=(CP*WDOTD*(.0384*RE**(-.25)))/(AD*(1.+1.5*PRN**(.016667)*
VRE**(-.125)*(PRN-.1.)))
RETURN
END

Fig. A-4  Subroutine AICH for Program 4-2
SUBROUTINE PRDP
EQUIVALENCE (DA(2),DI)
COMMON DA(400),T(5),TB,TS,ACH,RE,V,WDOTD,AD,RHO,REAV,RHOAV,ELW,
V,VAV,DP,FH,FAH,ZETAP,C3,DZ1,DELTAR

REAL RE,REAV
REAL FFT=0.0055*(1.+(1./DI+1.E6/RE)**.3333)
IF(REAV<2000.)2500,2500,2600
2500 FF=FFL
       GO TO 2900
2600 IF(REAV<3500.)2700,2700,2800
2700 FF=.5*(FFL+FFT)
       GO TO 2900
2800 FF=FFT
2900 DP=RHOAV*VAV**2*FF*ELW/(DI*64.34)
       RETURN
END

Fig. A-5 Subroutine PRDP
SUBROUTINE ENTGRT
DIMENSION CM(5),FO(5)
COMMON DA(400),T(5),TB*TS,ACH*RE,V,WUOTD*AD,RHO,REAV,RHOAV,ELW,
V*VAV*DP*FH*FAH,ZETAP*C3,DZ1,DELTAR

DZ2(T2,T4,T5)=(T5*(1.*DELTAR)+ZETAP*(T4**4-C3)+FH*T4-FAH)/
V(1.*T2*(1.*DELTAR))

C
ITER = 0
ITLT=DA(35)
ISO=0
ISW=0
DZ0=0.
DZW=0.
DZ1A=T(5)

1050 WILT = 500.*
T(4) = 1.*
IF(DA(36))1061,1060,1061
1060 DA(36)=25.
1061 T(3)=1./DA(36)
T(2) = 0.*
MESH=DA(36)
DZ1=T(5)
ITER=ITER+1
IF (ITLT - ITER) 1125,1130,1130
1125 PRINT 1500,DZ1A,DZ1
1500 FORMAT(38H0)INTEGRATION CONVERGENCE FAILED. DZ1A=E12.*5,H DZ1=
V E12.*5
RETURN

1130 DO 1390 J=1,MESH
F0(2)=0.*
F0(3)=T(3)/2.*
F0(4)=T(3)/2.*
F0(5)=T(3)
CM(1)=0.*
DO 2 I=2,5
CM(I)=DZ2(T(2)+F0(I),T(4)+F0(I)*T(5),T(5)+F0(I)*CM(I-1))
2 CONTINUE

T(2)=T(2)+T(3)
DT4 =T(3)/6.*((CM(2)+CM(3)+CM(4))*T(3)+T(3)*T(5)
T(4)=T(4)+DT4
DT5 =T(3)/6.*((CM(2)+2.*CM(3)+2.*CM(4)+CM(5))
T(5)=T(5)+DT5
1200 IF(ABS(DZ1)+ABS(T(5))=ABS(DZ1+T(5)))1250,1250,1205
1205 IF(ISW)1225,1235,1225
1225 DZ0=DZ1
T(5)=.5*(DZ0+DZW)
GO TO 1210

Fig. A-6 Subroutine ENTGRT
1235 DZO=DZ1
   T(5)=1.10*DZ1
   IF(ISW)1210,1239,1210
1210 IF((ABS(DZW)-ABS(DZO))/ABS(T(5))-.0025)1400,1400,1239
1239 IS0=1
   IF(IVAL-ITER) 1125,1125,1050
C*** TEST FOR WILT
1250 IF (ABS (WILT)-ABS (T(5))) 1260, 1380,1380
1260 IF(IS0)1278,1277,1278
1277 T(5)=.90*DZ1
   DZW=DZ1
   GO TO 1265
1278 DZW=DZ1
   T(5)=.5*(DZ0+DZW)
1265 IF((ABS(DZW)-ABS(DZO))/ABS(T(5))-.0025)1400,1400,1270
1270 ISW=1
   IF(IVAL-ITER) 1125,1125,1050
1380 WILT = T(5)
1390 CONTINUE
   GO TO 1260
1400 RETURN
END

Fig. A-6 Subroutine ENTGRT (cont.)
Fig. A-7  Program 4-2  Flow Diagram
COMPUTE DENSITY AT ENTRANCE (ENTERD)

IF TF1 < TE?

YES

SET: XH = TF1
Tw(1) = TE

INCREMENT LOOP COUNTER

LOOP COUNTER > 25?

YES

PRINT FAILURE MESSAGE

COMPUTE HEAT TRANSFER COEFFICIENT (AICH)

COMPUTE ENTRANCE WALL TEMP. (TWall)

NO

TEMP CHANGE SMALL

YES

COMPUTE DENSITY AT EXIT (ENTERP)

TFEND < TE?

NO

SET: XH = TE
Tw = TFEND

YES

SET: XH = TFEND
Tw = TE

Fig. A-7  Program 4-2

Flow Diagram (cont.)
Fig. A-7  Program 4-2 Flow Diagram (cont.)
A-23

4

COMPUTE DENSITY (ENTERP)

COMPUTE HEAT TRANSFER COEFFICIENT (AICH)

TF2 CHANGE SMALL?

NO

5

YES

COMPUTE: AVERAGE ABC

YES

ABC SMALL?

COMPUTE: ARITHMETIC DELTM

NO

COMPUTE: LOG DELTM

COMPUTE: TOTAL HEAT TRANSFER SECTION LENGTH

COMPUTE: PRESSURE CHANGE (PRDP)

PRINT SECTION DATA

SET NEW INLET VALUES FOR NEXT SECTION

6

Fig. A-7  Program 4-2  Flow Diagram (cont.)
Fig. A-7  Program 4-2  Flow Diagram (cont.)
Variable Case
Decimal Data
Requires (-)
In Column 1 of
Last Card of
Every Case

Last Card of
Material
Properties
Data Requires
A (-) in
Column 1

Decimal Data Change
Case "N" Title Card
CASE 2 DECIMAL DATA
CASE 2 TITLE CARD

Multiple Case Runs
Decimal Data
Title Card

VARIABLE CASE DATA

600 Fluid Conductivity Btu/hr ft R Vs. T, R
500 Fluid Visc (centipoise) vs. T, R
LOC 400 Fluid Density Lb/cu ft vs. T, R

INPUT DECIMAL DATA

PERMANENT HOLLERITH DATA

Main Program & Subroutines

COMPOSITE DECK

Example of
Table Format
LOC
400 Number of Pairs
of x, y
401 \( x_1 = \text{Temp R} \)
402 \( y_1 = \text{Density} \)
\( \begin{align*}
  x_n \\
  y_n
\end{align*} \)

When using a Constant
instead of a curve
enter in first
location of Table as a
negative
400 -RHO

Fig. A-8 Composite Deck Setup Program 4-2
PROGRAM NO. 4-2

C DIMENSION F1(24), F2(24), F3(24), F4(96), F5(24), F6(24), F7(12), F8(12),
     VF9(24), F10(12), F11(24), F12(12), F13(24), F14(24), F15(12), F16(12),
     VF17(12), F18(12), F19(12), F20(12), F21(24), F22(24), F23(96), F24(48)
     DIMENSION TW(5), FW(5), TITLE(16)
     DIMENSION DA(700), SR(6), SC(6), C1D(6), C2D(6), HD(6), TA(6), C1(10),
     VC2(10), HA(10), HB(10), TAA(10), TAB(10), ELE(10), ELEF(10), ELES(10),
     VTWF(10), TW(S10), G2(10), G3(10), EM(10), B(10), ELEAV(10)
     EQUIVALENCE (DA(7), DO), (DA(8), WDOT), (DA(9), WALT), (DA(10), ENT),
     V(DA(11), CP), (DA(12), TF1), (DA(13), TFEND), (DA(14), FMESH),
     V(DA(15), CKH)
     COMMON DA, REAV, RH0AV, VAV, ELW, WDOTD, RH0, AD, RE, ACH, TB, TS, V, DP, PI, DI
     COMMON TWAL, TW, CK0, CK1, CK2, CK4, CK5, H, PERIM, XH, RE2, TSTAR

C READ FORMAT STATEMENTS
C READ 1, F1, F2, F3, F4, F5, F6, F7, F8, F9, F10, F11, F12, F13, F14, F15, F16,
     VF17, VF18, VF19, VF20, VF21, VF22, VF23, VF24
1 FORMAT (12A6)
C READ MATERIALS PROPERTIES
C CALL DECRD (DA)
C PRINT FLUID PROPERTIES
C K=400
C PRINT 2300
2300 FORMAT (96H1, FLUID PROPERTIES (-) = CONSTANT, TABLE FORMAT
     V= NO PTS, X1, Y1, --XN,YN, X = TEMP (R) /3X
     V 90HLOC 400 = FLUID DENSITY, LB/CU FT, 500 = FLUID VISC., CENTIPOISE
     V 600 = FLUID KBTU/HR FT R /
     DO 2900 KKK=1,3
     K8=DA(K)
     K9=K+2*K8
     PRINT 2500, (K1, DA(K1), K1=K, K9)
2500 FORMAT (3XI5, F13.6, I8, F13.6, I0, F13.6, I8, F13.6, I8, F13.6)
     K=K+100
2900 CONTINUE
C 5 READ 2400, TITLE
2400 FORMAT (16A5)
C CALL DECRD (DA)
C
PI=3.1415926
IF(DA(3))10,15,10
10 PERIM=DA(3)
     AD=DA(4)
     GO TO 20
15 DI=DA(5)
     AD=PI*DI**2/4.
     PERIM=PI*DI
     GO TO 30
20 DI=4.*AD/PERIM
30 WDOTU=WDOT/ENT
C ENTER DUCT VALUES
ND=DA(1)
NP=20
DO 100 I=1, ND
SR(I)=DA(NP)
SC(I)=DA(NP+1)
C1D(I)=DA(NP+2)

Fig. A-9 Program 4-2 Listing
C2D(I)=DA(NP+3)
HD(I)=DA(NP+4)
TA(I)=DA(NP+5)

100 NP=NP+10

C
ENTER FIN VALUES
NF=DA(2)
N=80
DO 200 L=1,NF
C1(L)=DA(N)
C2(L)=DA(N+1)
HA(L)=DA(N+2)
HB(L)=DA(N+3)
TAA(L)=DA(N+4)
TAB(L)=DA(N+5)
ELEF(L)=DA(N+7)
ELES(L)=DA(N+8)
TWF(L)=DA(N+9)
TWS(L)=DA(N+10)

200 N=N+20
PRINT 2600*TITLE

2600 FORMAT (1H1,16A5//)
PRINT F4*(J,DA(J),J=1,15)
PRINT F5*(K*SR(K),K=1*ND)
PRINT F6*(K*SC(K),K=1*ND)
PRINT F7*(K*C1D(K),K=1*ND)
PRINT F8*(K*C2D(K),K=1*ND)
PRINT F9*(K*HD(K),K=1*ND)
PRINT F10,(K*TA(K),K=1*NF)
PRINT F11,(K*C1(K),K=1*NF)
PRINT F12,(K*C2(K),K=1*NF)
PRINT F13,(K*HA(K),K=1*NF)
PRINT F14,(K*HB(K),K=1*NF)
PRINT F15,(K*TAA(K),K=1*NF)
PRINT F16,(K*TAB(K),K=1*NF)
PRINT F17,(K*K=1*NF)
PRINT F18,(K*ELEF(K),K=1*NF)
PRINT F19,(K*ELES(K),K=1*NF)
PRINT F20,(K*TWF(K),K=1*NF)
PRINT F21,(K*TWS(K),K=1*NF)
GD1=0.
GD2=0.
GD3=0.
DO 300 I=1,ND
GD1=GD1+SR(I)*C1D(I)
GD2=GD2+SC(I)*HD(I)

300 GD3=GD3+(SR(I)*C2D(I)+SC(I)*HD(I)*TA(I))
DO 400 L=1,NF
G2(L)=HA(L)+HB(L)
G3(L)=HA(L)+TAA(L)+TAB(L)*C2(L)
EM(L)=(ELEF(L)-ELES(L))/(TWF(L)-TWS(L))

400 B(L)=ELES(L)-EM(L)*TWS(L)
SUM0=0.
SUM1=0.
SUM2=0.
SUM4=0.
SUM5=0.
DO 500 L=1,NF
SUM0=SUM0+G3(L)*B(L)
SUM1=SUM1+(G2(L)*B(L)-G3(L)*EM(L))

Fig. A-9 Program 4-2 Listing (cont.)
SUM2=SUM2+GZ(L)*E#(L)
SUM4=SUM4+C1(L)*B(L)
SUM5=SUM5+C1(L)*EM(L)
CKO=-(GD3+SUMO)
CK1=GD2+SUM1
CK2=SUM2
CK4=GD1+SUM4
CKS=SUMS
Lc=0
LC=LC+1
IF (LC<25) 1001,1001,651
500 PRINT 652,TE
652 FORMAT(/32H INITIALIZATION OF TE FAILED. TE=E12.8)
GO TO 5
1001 IF(ELE(1),EQ.0.0) GO TO 1020
IF(ABS(TE)<.001)1003,1003,1002
1002 DFEDTE=4.0*C1(1)*TE**3+G2(1)
GO TO 1030
1020 FE=SR(1)*(C1D(1)*TE**4 +G2D(1)) + SC(1)*HD(1)*(TE-TA(1))
IF(ABS(TE)<.001)1003,1003,1002
1021 DFEDTE=4.0*SR(1)*C1D(1)*TE **3+SC(1)*HD(1)
1030 TE=TE-TE/DFEDTE
1041 GO TO 640
C COMPUTE ENVIRONMENT TEMP
600 TE=TF1-1.2*(TF1-TFEND)
640 LC=LC+1
IF (LC=25) 1001,1001,651
1003 RHO=ENTERP(TF1,DA(400))
RHO1=RHO
TB=TF1
IF((TF1-TE).LT.0.0) GO TO 1004
XH=TE
TW(1)=TF1
GO TO 1006
1004 XH=TF1
TW(1)=TE
1006 LC2=0
1007 LC2=LC2+1
IF(LC2.LT.25) GO TO 1107
PRINT 6000+TW1,TW1P
6000 FORMAT(/32X5HTW1 =E15.8/3X6HTW1P =E15.8/3X17WTW1 NOT CONVERGED)
GO TO 5
1107 TS=TW(1)
TW1P=TW(1)
CALL AICH
H=ACH
CALL TWALL
TW(1)=TWAL
IF(ABS(TW1P-TWAL)<.2)1008,1008,1007
1008 RE1=RE
ACH1=ACH
V1=V
TW1=TWAL
C COMPUTE EXIT WALL TEMP
RHO=ENTERP(TFEND,DA(400))
TB=TFEND
IF((TFEND-TE).LT.0.0) GO TO 1009
XH=TE
TW(1)=TFEND
Fig A-9 Program 4-2 Listing (cont.)
GO TO 1010
1009 XH=TFEND
     TW(1)=TE
     LC3=0
1010 TWEND=TWAL
     CALL TWALL
     LC3=LC3+1
     IF(LC3.LT.25) GO TO 7000
     PRINT 8000,TWEND,TWENDP
8000 FORMAT(//'3XHTWEND =E15.8/3XHTWENDP=E15.8//'3X19HTWEND NOT CONVERGED)
     GO TO 5
7000 TWEND=TWAL
     TW(1)=TWAL
     TS=TWEND
     CALL AICH
     H=ACH
     IF(ABS(TWENDP-TWEND)>.2)1013,1013,1010
1013 DELT=(TW1-TWEND)/FMESH
     MESH=FMESH
     ELWSUM=0.
     DPSUM=0.
     QSUM=0.
     WLSUM=0.
     DO 2000 I=1,MESH
     TW2=TW1 - DELT
     LC4=0
     H=ACH
     1070 LC4=LC4+1
     IF(LC4=25)1075,1075,1074
     PRINT F22,TF2P,TF2,RE2P,RE2
     GO TO 5
1075 TF2P=TF2
     RE2P=RE2
     TF2=TW2+(1./H*PERIM)*CK1*TW2+CK2*TW2**2+CK4*TW2**4+
     VCK5*TW2**5)
     TB=TF2
     TS=TW2
     RHO=ENTERP(TF2,DA(400))
     RHO2=RHO
     CALL AICH
     H=ACH
     ACH2=ACH
     RE2=RE
     V2=V
     IF(ABS(TF2P-TF2)-.25)1080,1080,1080
1080 HAV=.5*(ACH1+ACH2)
     ABC=(TF1-TW1)-(TF2-TW2)
     IF (ABS(ABC)-5.0) 1400,1400,1500
1400 DELTM=(TF1-TW1+TF2-TW2)/2.0
     GO TO 1700
1500 DELTM=ABC/ALOG((TF1-TW1)/(TF2-TW2))
1700 Q=WDOTD*CP*(TF1-TF2)
     ELW=Q/(HAV*PERIM*DELTM)
     ELWSUM=ELWSUM+ELW
     QSUM=QSUM+Q*ENT
     REAV=.5*(RE1+RE2)
     RHOAV=.5*(RHO1+RHO2)
     VAV=.5*(V1+V2)

Fig. A-9  Program 4-2 Listing (cont)
CALL PDP
DPSUM=DPSUM+DP
WL=RHOAV*AD*ELW
WLSUM=WLSUM+WL
PRINT F23,I,TF1,TF2,TW1,TW2,ACH1,ACH2,RE1,RE2,VAV,ELW,G,HAV,REAV,
V RHOAV,WL,DP
TF1=TF2
TW1=TW2
V1=V2
RE1=RE2
RHO1=RHO2
ACH1=ACH2
2000 CONTINUE
PRINT F24,DPSUM,QSUM,WSUM,TE,ELWSUM
GO TO 5
END

Note: See following page for permanent Hollerith Listing.
(28H ENVIRON TEMP CONVERG FAILED/4H TE=12.8*5H CK0=E12.8*5H CK1=E12.8*5H CK2=E12.8*5H CK4=E12.8*5H CK5=E12.8*5H)
(34H ENTRANCE WALL TEMP CONVERG FAILED/5H TW1=E12.8*5H CK0=E12.8*5H CK1=E12.8*5H CK2=E12.8*5H CK4=E12.8*5H CK5=E12.8*5H)
(30H EXIT WALL TEMP CONVERG FAILED/7H TEWEND=E12.8*5H CK0=E12.8*5H CK1=E12.8*5H CK2=E12.8*5H CK4=E12.8*5H CK5=E12.8*5H)

(11H INPUT DATA/19T F15.8*21H NO. OF DUCT SECTIONS/19T F15.8*12H NO. OF FI NS/19T F15.8*28H DUCT PERIMETER (FT) *OPTION/19T F15.8*26H DUCT AREA (SQ FT) *OPTION/19T F15.8*32H EFFECTIVE DIAMETER (FT) *OPTION/19T F15.8*9H NOT USED/19T F15.8*22H OUTSIDE DIAMETER (FT)/19T F15.8*26H TOTAL WEIGHT FLOW (LB/HR)/19T F15.8*20H WALL THICKNESS (FT)/56H EFFECT PERIFERAL LENGTH FOR RA DIATION (FT)/(13T7X15.8)

(17H INPUT FIN VALUES/ 4H FINSX*42H RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)/(13T7X15.8)
( 4H FINSX*37H RADIATION CONSTANT C2 (BTU/HR SQ FT)/(13T7X15.8)
( 4H FINSX*55H CONVECTIVE HEAT TRANSFER COEFF SIDE A (BTU/HR SQ FT R)/(13T7X15.8)
( 4H FINSX*55H CONVECTIVE HEAT TRANSFER COEFF SIDE B (BTU/HR SQ FT R)

Note: This Hollerith Listing follows Subroutine Listings. See Fig. A-8.

Fig. A-9  Program 4-2 Listing (cont)
SUBROUTINE PRDP
DIMENSION F1(24), F2(24), F3(24), F4(96), F5(24), F6(24), F7(12), F8(12),
VF9(12), F10(12), F11(24), F12(12), F13(12), F14(12), F15(12), F16(12),
VF17(12), F18(12), F19(12), F20(12), F21(12), F22(12), F23(72), F24(48)
DIMENSION TW(5), FW(5)
DIMENSION DA(700), SR(6), SC(6), C1D(6), C2D(6), HD(6), TA(6), C1(10),
VC2(10), HA(10), HB(10), TAA(10), TAB(10), ELE(10), ELEF(10), ELES(10),
VTWF(10), TWS(10), G2(10), G3(10), EM(10), B(10), ELEAV(10)
EQUIVALENCE (DA(7), DO), (DA(8), WDOT), (DA(9), WALT), (DA(10), ENT),
V(DA(11), CP), (DA(12), TF1), (DA(13), TFEND), (DA(14), FMESH),
V(DA(15), CH)
COMMON DA, REAV, RHOAv, VAV, ELW, WDOTd, RHO, AD, RE, ACH, TB, TS, V, DP, PI, DI
COMMON TWAL, TW, CK0, CK1, CK2, CK4, CK5, H, PERIM, XH

RE=REAV
FFL=.64/RE
FFT=.0055*(1+ (.1/DI+1.E6/RE)**.3333)
IF(REAV-2000.), 2500, 2500, 2600
2500 FF=FFL
GO TO 2900
2600 IF(REAV-3500.), 2700, 2700, 2800
2700 FF=.5*(FFL+FFT)
GO TO 2900
2800 FF=FFT
2900 DP=RHOAv*VAV**2*FF*ELW/(DI*64.34)
RETURN
END

Fig. A-10 Subroutine PRDP
SUBROUTINE AICH
DIMENSION TW(5), FW(5)
DIMENSION DA(700), SR(6), SC(6), C1D(6), C2D(6), HD(6), TA(6), C1(10),
   VC2(10), HA(10), HB(10), TAA(10), TAB(10), ELE(10), ELEF(10), ELES(10),
   VTWF(10), TWS(10), G2(10), G3(10), EM(10), B(10), ELEAV(10)
EQUIVALENCE (DA(7), DO), (DA(8), WDOTD), (DA(9), WALT), (DA(10), ENT),
   V(DA(11), CP), (DA(12), TF1), (DA(13), TFEND), (DA(14), FMESH),
   V(DA(15), CKH)
COMMON DA, REAV, RH0AV, VAV, ELW, WDOTD, RH0, AD, RE, ACH, TB, TS, V, DP, PI, DI
COMMON TWAL, TW, CK0, CK1, CK2, CK4, CK5, PERIM, XH, RE2, TSTAR
V=WDOTD/(RH0*AD*3600.)
VISC=ENTERP(TB, DA(500)) * 2.419027
REN=WDOTD*DI/AD
RE=REN/VISC
IF(RE=2000.) 410 410, 500
410 FLK=ENTERP(TB, DA(600))
   ACH=CKH*FLK/DI
RETURN
500 VISC=ENTERP(TS, DA(500)) * 2.419027
   FLK=ENTERP(TS, DA(600))
   PRN=CP*VISC/FLK
   TSTAR=TB-((.1*PRN+40.) *(TB-TS)/(PRN+72.))
   VISC=ENTERP(TSTAR, DA(500)) * 2.4190297
   RE=REN/VISC
600 ACH=(CP*WDOTD*(.0384*RE**(-.25)))/(AD*(1.1.5*PRN**(-.16667)*
   VRE**(-.125)*(PRN-1.))
RETURN
END
SUBROUTINE TWALL
DIMENSION F1(24),F2(24),F3(24),F4(96),F5(24),F6(24),F7(12),F8(12),
       VF9(12),F10(12),F11(24),F12(12),F13(12),F14(12),F15(12),F16(12),
       VF17(12),F18(12),F19(12),F20(12),F21(12),F22(12),F23(72),F24(48)
DIMENSION TW(5),FW(5)
DIMENSION DA(700),SR(6),SC(6),CD(6),C2D(6),HD(6),TA(6),C1(10),
       VC2(10),HA(10),HB(10),TAA(10),TAB(10),ELE(10),ELEF(10),ELES(10),
       VTWF(10),TWS(10),G2(10),G3(10),EM1(10),ELEAV(10)
EQUIVALENCE (DA(7),DO),(DA(8),WDOT),(DA(9),WALTH),(DA(10),ENT),
       V(DA(11),CP),(DA(12),TF1),(DA(13),TFEND),(DA(14),FMESH),
       V(DA(15),CKH)
COMMON DA,REAV,RHOAV,VAV,ELW,WDOTD,RHO,AD,RE,ACH,TH,TS,VD,PI,DID
COMMON TWAL,TW,C0,C1,C2,C4,C5,H,PERIM,XH
FWH=C0+C1*XH+C2*XH**2+C4*XH**4+C5*XH**5-H*PERIM*(TB-XH)
L=0
20 DO 10 K=2,4
   L=L+1
   FW(K-1)=C0+C1*TW(K-1)+C2*TW(K-1)**2+C4*TW(K-1)**4+C5*
   VTW(K-1)**5-H*PERIM*(TB-TW(K-1))
10 TW(K)=(XH*FW(K-1)-TW(K-1)*FWH)/(FW(K-1)-FWH)
   IF(L.LT.20) GO TO 25
   PRINT 50, TW(2),TW(3),FWH,FW(1),FW(2)
50 FORMAT(/3X7HTW(2) = E15.8/3X7HTW(3) = E15.8/3X7HFWH = E15.8/
     3X7HFW(1) = E15.8/3X7HFW(2) = E15.8/3X21HTWALL CONVERGE FAILED )
   STOP
25 IF(ABS(TW(3)-TW(4))-.5)30,30,27
27 TW(1)=TW(4)
   GO TO 20
30 TWAL=TW(2)-(TW(3)-TW(2))**2/(TW(4)+TW(2)-2.*TW(3))
   RETURN
END

Fig. A-12 Subroutine TWALL
FUNCTION ENTERP(X,TAB)
DIMENSION TAB(101)

IF(TAB(1))9,9,8
9 ENTERP=-TAB(1)
   RETURN
8 N=TAB(1)
   DO 5 I=1,N
7 IF(TAB(2*I)-X)5*4,3
3 IF(I-1)6*6,7
7 ENTERP=TAB(2*I-1)+(X-TAB(2*I-2))*(TAB(2*I+1)-TAB(2*I-1))/
     V(TAB(2*I)-TAB(2*I-2))
   RETURN
4 ENTERP=TAB(2*I+1)
   RETURN
5 CONTINUE
M=2*N+1
K=M
105 PRINT 10,X,TAB(K),(TAB(J),J=1,M)
10 FORMAT(//39H LIMITS OF TABLE EXCEEDED BY ARGUMENT = F12.4/
     VF12.4,24H = VALUE USED FROM TABLE/(5F12.4))
   ENTERP=TAB(K)
   RETURN
6 M=2*N+1
K=2
GO TO 105
END

Fig. A-13 Subroutine ENTERP
A-36

FORTRAN SOURCE LIST

ISN
SOURCE STATEMENT

0 $IRFTC DECRD  DECK
1 SUBROUTINE DECRD(DATA)
   C READS A VARIABLE NUMBER OF ITEMS OF FLOATING-POINT DATA INTO
   C SPECIFIED ELEMENTS OF AN ARRAY IN BLOCKS OF 5 CONSECUTIVE ITEMS.
   C ONE OR MORE BLANK FIELDS ON A DATA CARD CAUSE THE VALUES IN CORE
   C TO REMAIN UNCHANGED
   C THE FORTRAN INTEGER INDEX IN THE FIRST FIELD OF EACH CARD DEFINES
   C THE POSITION OF THE ARRAY OF THE FIRST ITEM OF EACH BLOCK OF FIVE.
   C THE BLOCKS NEED NOT BE SEQUENTIAL NOR CONTINUOUS.
   C THE INDEX IS PLACED AT THE END OF ITS FIELD. IT MAY NOT BE ZERO
   C OR BLANK. IT SHALL NOT CONTAIN A DECIMAL POINT.
   C A DECIMAL POINT MUST ALWAYS BE PLACED IN EACH DATA ITEM. THEREFORE,
   C THE VALUE MAY BE PLACED ANYWHERE IN EACH OF THE FIELDS PER CARD.
   C THE DECIMAL SCALE, IF ANY, FOR A DATA ITEM MUST BE PLACED AT THE
   C END OF THE FIELD.
   C A 0. MUST BE ENTERED TO READ IN A ZERO. A -0. IS THE SAME AS A
   C BLANK FIELD. THE READING OF DATA IS TERMINATED BY ENTERING A
   C NEGATIVE INDEX ON THE LAST CARD OF EACH SERIES.
   C TO USE THE ROUTINE ---- CALL DECRD(DATA)
2 DIMENSION DRBU (5), DATA (6)
3 1 READ 2, IND, (DRBU(I), I=1,5)
11 2 FORMAT(12, 5E12.0)
12 3 J = IABS(IND)
13 4 DO 7 I=1,5
14 5 IF(DRBU(I)) 6,10,6
15 6 DATA(J) = DRBU(I)
16 7 J = J+1
20 8 IF(IND) 9,11,1
21 9 RETURN
22 10 IF(SIGN1,DRBU(I))) 7,11,6
23 11 CALL EXIT
24 END

Fig. A-14  Subroutine DECRD
Four sample problems are presented to demonstrate the capabilities and limitations of the programs. These problems are also useful for check out should the program deck be reproduced or modified for use on another computer.

Two fluids, either water or "Coolanol", were used in the problems. The properties of these fluids and the method of entering the data is shown in Fig. B-1. Data properties at various temperatures are given. The program uses a linear interpolation between data points.

**Problem 1: Hot Water Panel**

A panel consisting of 12 steel tubes, 4-inches on centers are joined together with 1/8" steel plates. Two hundred-fifty pounds of water per hour at 200 F. enters the system. A detail listing of the items affecting the performance and the data locations are tabulated in the input data shown in Fig. B-2.

Three computation sections are chosen for this illustration and a heat exchanger effectiveness of 0.5 is assumed. As shown in the output data, Fig. B-3, the overall tube lengths are 20.178 feet. The water flow in the tubes is laminar with a convective heat transfer coefficient in the entrance of 55.624 Btu/hr sq ft R. For these conditions, the tube wall temperature is considerably below that of the water. It can also be observed that over half of the total length is in the third section where the water temperature approaches ambient. For the case shown, the actual exchanger effectiveness is
This is considerably higher than the value of 0.5 estimated in the input data. This difference has no effect in the accuracy for the calculated problem solution since the output data is based on the 0.87 value. It does point out the fact that for most cases the calculated data will be at a higher effectiveness than estimated.

The output data contains many items affecting the performance and the heat transfer parameters so that design changes can be made if desired.

Problem 2: Tubular Heat Exchanger

A 3/8 inch O.D. tube transports "Coolanol" through a low pressure gas enclosure having a low convective heat transfer coefficient but high gas and wall temperatures. It is required to find the temperature rise in the "Coolanol" while passing through 15 feet of tube length.

Program 4-2 is directly applicable to this problem. However, Program 4-1 could be used, but two modifications would be required: (1) the program input data for radiative heat transfer will have to be altered, and (2) fins will have to be added to the tube. Program 4-1 uses the projected tube diameter for calculating radiative heat transfer, however, this would not yield the correct solution to the problem. This item can be accounted for by multiplying the radiative constants $C_1$ and $C_2$ by the factor $\pi/2$. Extremely small dummy fins with external surface properties equal to that of the duct but of essentially no length or thickness could be entered as data. If sufficiently small, these fins would have negligible effect on the overall
heat transfer from the duct. With the above changes Program 4-1 could be used.

In using Program 4-2 the values of the radiative and environmental parameters $C_1$ and $C_2$ are required. The problem conditions used to calculate these parameters are:

\[
\begin{align*}
\epsilon_a &= 0.6 \\
\epsilon_x &= 1.0 \\
T_x &= 1000 \text{ R} \\
F_x &= 1.0
\end{align*}
\]

Using these values in Eqs. (1.9) and (1.10)

\[
C_1 = \epsilon_a \sigma = (0.8)(0.1613)(10^{-8}) = 0.137(10^{-8})
\]

\[
C_2 = 0.1713(10^{-8})(1000^4)(1.0)(0.6) = 1028
\]

One hundred degrees fluid temperature rise in passing through the duct was estimated for a first trial. The remaining input items for this problem are shown in Fig. B-4. The program output data is shown on Fig. B-5. Only two section lengths were calculated for this illustration. As shown 144.0959 feet of tubing are required to heat the "Coolanol" 100 F. This tube is longer than the problem value and therefore the fluid will be heated less than the 100 degrees estimated for the input data. Other temperature changes could be selected and the program rerun. In most instances several section lengths would have been specified so that a curve of output length and temperature could be plotted. The performance could be visually analyzed, and the proper design selections made.
Problem 3: Noncircular Duct and Tapered Fin Lengths

Problem 3, illustrated in Sketch 1 uses Program 4.2. The program assumed the equivalent length of the extended surfaces to vary linearly with

\[ 0.0365' \text{ developed length (for convection)} \]

\[ \frac{0.0365'}{2''} = 0.0304' \]

\[ p = 0.0518 \text{ ft} \]

\[ A_d = 1.11 \times 10^{-4} \text{ sq ft (flow area)} \]

Fluid - 250 lb/hr - "Coolanol"
Material - Aluminum - \( k = 118 \text{ Btu/hr ft R} \)

\[ T_{f1} = 760 \text{ R} \]
\[ T_{f2} = 710 \text{ R} \]
\[ T_{w1} = 730 \text{ R (estimated)} \]
\[ T_{wa} = 680 \text{ R (estimated)} \]
\[ L_{e1} = 0.1559' \text{ (calculated by Program 2-1)} \]
\[ L_{e2} = 0.2697' \text{ (calculated by Program 2-4)} \]

Sketch 1
tube temperature while in this problem the actual fin length varies linearly with the tube length. The equivalent length, however, is a nonlinear function and very little is known regarding its true behavior. However, it seems appropriate to test the program's ability to achieve convergence in some of the loops and to solve such a problem. The calculated answers should be regarded as being an approximation.

A listing of the input data to obtain the fin performance is shown in Figs. B-6 and B-7, while the input data used by program 4-2 is shown in Fig. B-8 and the output data in Fig. B-9.

The calculated wall temperatures on the entrance and exit are not close to the estimated values. Some improvement in accuracy would result from rerunning the program using problem 3 data. Still better accuracy could be attained if the duct length were divided into a number of sections and the duct and fin equivalent lengths calculated for each section.

Problem 4: Heating Coil in Paraffin Tank

A coil carrying "Coolanol" is brazed into a tank structure as shown with much of the data in the Sketch 2 below. The coil itself has a 1/2" outside diameter, and a 0.025" wall.

The fin input and output data at approximately inlet conditions is shown in Figs. B-10, B-11 and B-12. Program data at outlet conditions was also obtained but the details were omitted. The final input and output problem data using program 4-2 is shown in Figs. B-13 and B-14.

As shown by the output data considerable heat is transferred by this system 105,000 Btu/hr. A high temperature drop between the fluid and the tube wall is evident despite the high Reynolds number (approximately 10,000) and the
high heat transfer coefficient (approximately 240 Btu/hr ft R). This temperature drop is due to the high rate of heat transfer from the fins and the fact that the actual tube area through which the heat is flowing is small. The small diameter tube creates a large pressure drop, 1535.92 lb/sq ft (10.66 psi). It therefore seems advisable to examine other tube diameters in the event that other sizes could produce better results. The problem does demonstrate the wide capabilities of the program and the manner in which the data might be examined and the configurations chosen to attain a suitable performance compromise.
Problem 5: Flow at Critical Reynolds Number

Problem 1 is recalculated except that the flow rate is increased to 800 lbs/hr. This problem is introduced to illustrate the difficulties that might be encountered and to recognize them from the output data. The input and output data is shown in Figs. B-15 and B-16. The calculated average Reynolds number in the first section is 2,434.5 which, according to the program test, it is in the turbulent region. Convergence failed in the second section. In the calculated Reynolds numbers for the last two passes through the loop are 1,803.7 and 1,612.7. Both of these numbers appear to be considerably below the critical value of 2000. Convergence might have been accomplished at the exit of this section had these values been lower. The peculiar situation encountered in this problem arises from the fact that the fluid heat transfer coefficient is calculated alternately with the equations provided in the program for turbulent and laminar regions. If laminar, a low coefficient and high liquid temperature is predicted. The next attempt predicts a Reynolds number based on bulk temperature higher than 2000 and the calculated heat transfer coefficient is made with the turbulent equations. In these equations the Reynolds number is recalculated at an intermediate temperature, T*. This accounts for the fact that both of the Reynolds numbers are considerably below the critical value. Additional program data (not shown) was printed out before the difficulty was isolated.

With an actual system the flow in the vicinity of the critical Reynolds number could be either laminar or turbulent. Testing would be required to establish the operating performance, which may change from one test to another. For this reason, systems designed for operation in the transition region are usually avoided.
**FLUID PROPERTIES**

(-) = CONSTANT

**TABLE FORMAT = NO PTS, X1,Y1,--XN,YN.  X = TEMP (R)**

**LOC 100 = FLUID VISC, CENTIPOISE, 200 = FLUID K, BTU/HR FT R, 300 = FLUID DENSITY, LB/CU FT**

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Viscosity</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>4.00000000</td>
<td>1.00000000</td>
</tr>
<tr>
<td>105</td>
<td>0.51323000</td>
<td>1.00000000</td>
</tr>
<tr>
<td>200</td>
<td>0.33700000</td>
<td>1.00000000</td>
</tr>
<tr>
<td>205</td>
<td>0.38200000</td>
<td>1.00000000</td>
</tr>
<tr>
<td>300</td>
<td>62.5400000</td>
<td>60.2000000</td>
</tr>
<tr>
<td>305</td>
<td>53.6200000</td>
<td>60.2000000</td>
</tr>
</tbody>
</table>

**Water**

**FLUID PROPERTIES**

(-) = CONSTANT

**TABLE FORMAT = NO PTS, X1,Y1,--XN,YN.  X = TEMP (R)**

**LOC 400 = FLUID DENSITY, 500 = FLUID VISC, CENTIPOISE, 600 = FLUID K**

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Viscosity</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>2.00000000</td>
<td>502.000000</td>
</tr>
<tr>
<td>500</td>
<td>13.0000000</td>
<td>502.000000</td>
</tr>
<tr>
<td>505</td>
<td>435.000000</td>
<td>502.000000</td>
</tr>
<tr>
<td>510</td>
<td>15.2500000</td>
<td>502.000000</td>
</tr>
<tr>
<td>515</td>
<td>525.000000</td>
<td>502.000000</td>
</tr>
<tr>
<td>520</td>
<td>3.27000000</td>
<td>502.000000</td>
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<tr>
<td>525</td>
<td>2000.00000</td>
<td>502.000000</td>
</tr>
<tr>
<td>600</td>
<td>2.00000000</td>
<td>502.000000</td>
</tr>
</tbody>
</table>

"Coolanol"

*Note: For example, if a constant density of 62.4 were to be used for water, the Data Card would read 300 -62.4*

**Fig. B-1  Fluid Properties**
### Example Problem

**Input Data**

<table>
<thead>
<tr>
<th>Line</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Outside Diameter (FT)</td>
<td>0.04167</td>
</tr>
<tr>
<td>2</td>
<td>Inside Diameter (FT)</td>
<td>0.03083</td>
</tr>
<tr>
<td>3</td>
<td>No. of Tubes</td>
<td>1200000</td>
</tr>
<tr>
<td>4</td>
<td>Weight Flow (LBS/HR)</td>
<td>2500000</td>
</tr>
<tr>
<td>5</td>
<td>Fin Length (FT)</td>
<td>0.14580</td>
</tr>
<tr>
<td>6</td>
<td>Fin Thickness at Root (FT)</td>
<td>0.01040</td>
</tr>
<tr>
<td>7</td>
<td>Fin Thickness at Far Edge (FT)</td>
<td>0.01040</td>
</tr>
<tr>
<td>8</td>
<td>Density of Fin Material (LBS/CU FT)</td>
<td>4900000</td>
</tr>
<tr>
<td>9</td>
<td>Density of Tube Material (LBS/CU FT)</td>
<td>4900000</td>
</tr>
<tr>
<td>10</td>
<td>NOT USED</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Thrm Cond of Fin (BTU/FT HR R)</td>
<td>2500000</td>
</tr>
<tr>
<td>12</td>
<td>Specific Heat of Fld (BTU/LB R)</td>
<td>1000000</td>
</tr>
<tr>
<td>13</td>
<td>Fluid Temp at Entrance (R)</td>
<td>6400000</td>
</tr>
<tr>
<td>14</td>
<td>Heat Transfer Coefficient Side A (HA, BTU/HR SQ FT)</td>
<td>6000000</td>
</tr>
<tr>
<td>15</td>
<td>Heat Transfer Coefficient Side B (HB, BTU/HR SQ FT)</td>
<td>6000000</td>
</tr>
<tr>
<td>16</td>
<td>Ambient Temp Side A (R)</td>
<td>5350000</td>
</tr>
<tr>
<td>17</td>
<td>Ambient Temp Side B (R)</td>
<td>5350000</td>
</tr>
<tr>
<td>18</td>
<td>Alphaa</td>
<td>0.00000</td>
</tr>
<tr>
<td>19</td>
<td>Alphab</td>
<td>0.00000</td>
</tr>
<tr>
<td>20</td>
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</tr>
<tr>
<td>21</td>
<td>Epsb</td>
<td>0.85000</td>
</tr>
<tr>
<td>22</td>
<td>Epsx</td>
<td>0.00000</td>
</tr>
<tr>
<td>23</td>
<td>Falpha</td>
<td>1.00000</td>
</tr>
<tr>
<td>24</td>
<td>Fax</td>
<td>0.00000</td>
</tr>
<tr>
<td>25</td>
<td>Fb</td>
<td>1.00000</td>
</tr>
<tr>
<td>26</td>
<td>Frx</td>
<td>0.00000</td>
</tr>
<tr>
<td>27</td>
<td>Rhdb</td>
<td>0.00000</td>
</tr>
<tr>
<td>28</td>
<td>Rhdx</td>
<td>0.00000</td>
</tr>
<tr>
<td>29</td>
<td>Thetap (DEG)</td>
<td>0.00000</td>
</tr>
<tr>
<td>30</td>
<td>Thetam (DEG)</td>
<td>0.00000</td>
</tr>
<tr>
<td>31</td>
<td>Thetax (DEG)</td>
<td>0.00000</td>
</tr>
<tr>
<td>32</td>
<td>TM (R)</td>
<td>5300000</td>
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<td>Solar Constant (BTU/HR SQ FT)</td>
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<td>Heat Exchanger Effectiveness</td>
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**Eff Curve Fit**

\[
0.41200F 0.53315E-02 0.23620F-04 \\
-0.87832E-02 0.96256F-01 -0.40643E-02 \\
0.10000F 0.627460E 0.23347E 00
\]

Fig. B-2 Input Data - Problem 1
SECTION NO. 1

INLET
0.66000E+03  0.62405E+03 FLUID TEMP (R)
0.60797E+03  0.58638E+03 WALL TEMP (R)
0.10783E+00  0.10810E+00 FIN EFFECT LENGTH (FT)
0.55624E+02  0.53928E+02 HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R)

0.54776E+02 HEAT TRANSFER COEFFICIENT AVG (BTU/HR FT SQ R)
0.86895E+03 REYNOLDS NO. AVG
0.12822E+00 VELOCITY AVG (FT/SEC)
0.31746E+01 SECTION LENGTH (FT)
0.11713E+00 PRESSURE CHANGE (LBS/SQ FT)
0.74892E+03 HEAT TRANSFER (BTU/HR TUBE)
0.60450E+02 FLUID DENSITY AVG
0.14329E+00 WT OF LIQUID (LBS)

SECTION NO. 2

INLET
0.62405E+03  0.58741E+03 FLUID TEMP (R)
0.58638E+03  0.56479E+03 WALL TEMP (R)
0.10810E+00  0.10838E+00 FIN EFFECT LENGTH (FT)
0.53928E+02  0.52199E+02 HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R)

0.53063E+02 HEAT TRANSFER COEFFICIENT AVG (BTU/HR FT SQ R)
0.47411E+03 REYNOLDS NO. AVG
0.12715E+00 VELOCITY AVG (FT/SEC)
0.50339E+01 SECTION LENGTH (FT)
0.33758E+00 PRESSURE CHANGE (LBS/SQ FT)
0.76343E+03 HEAT TRANSFER (BTU/HR TUBE)
0.60956E+02 FLUID DENSITY AVG
0.22911E+00 WT OF LIQUID (LBS)

Fig. B-3 Output Data - Problem 1
SECTION NO. 3

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<th>INLET</th>
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<th>OUTLET</th>
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<td>0.58741F 03</td>
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<td>0.56479F 03</td>
<td>0.54320F 03 WALL TEMP (R)</td>
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<td>0.10838F 00</td>
<td>0.10865E 00 FIN EFFECT LENGTH (FT)</td>
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<td>0.52199F 02</td>
<td>0.50432E 02 HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R)</td>
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0.51315E 02 HEAT TRANSFER COEFFICIENT AVG (BTU/HR FT SQ R)
0.32763E 03 REYNOLDS NO. AVG
0.12609E 00 VFLOCITY AVG (FT/SEC)
0.11969E 02 SECTION LENGTH (FT)
0.11518F 01 PRESSURE CHANGE (LBS/SQ FT)
0.78049E 03 HEAT TRANSFER (BTU/HR TUBE)
0.61472E 02 FLUID DENSITY AVG
0.54937E 00 WT OF LIQUID (LBS)

FIG. B-3 OUTPUT DATA

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<th>INLET</th>
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<th>OUTLET</th>
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<td>0.58505F 00</td>
<td>0.90636E 00 ENVIRON PARAM (C3)</td>
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<tr>
<td>0.50953F-01</td>
<td>0.36694E-01 PROFILE NO.</td>
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<td>0.94339F 00</td>
<td>0.94339E 00 CONVECTIVE PARAM (FHI)</td>
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<td>0.83283F 00</td>
<td>0.92915E 00 CONVECTIVE PARAM (FAH)</td>
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<td>0.10865F 00</td>
<td>0.10865E 00 FIN EFFECT LENGTH (FT)</td>
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<tr>
<td>0.92176E 00</td>
<td>TOT WT OF LIQUID (LBS)</td>
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<tr>
<td>0.60993E 01</td>
<td>WEIGHT OF DUCT (LBS)</td>
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</tr>
<tr>
<td>0.14992E 02</td>
<td>WEIGHT OF FINS (LBS)</td>
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<tr>
<td>0.43959F 03</td>
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<td>0.53436E 03</td>
<td>ENVIRON TEMP (R)</td>
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<td>0.20178F 02</td>
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<td>0.16065F 01</td>
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<td>0.27514F 05</td>
<td>TOTAL HEAT TRANSFER (BTU/HR)</td>
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<td>0.80694E 02</td>
<td>PLAN AREA (SQ FT)</td>
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<td>0.9121E-08</td>
<td>RADIATION CONSTANT (C1)</td>
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<td>0.22980F 03</td>
<td>RADIATION CONSTANT (C2)</td>
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Fig. B-3 Output Data - Problem 1 (cont)
EXAMPLE PROBLEM

INPUT DATA
1  1.00000000 NO. OF DUCT SECTIONS
2  0.00000000 NO. OF FINS
3  0.00000000 DUCT PERIMETER (FT) *OPTION
4  0.00000000 DUCT AREA (SQ FT) *OPTION
5  0.01925000 EFFECTIVE DIAMETER (FT) *OPTION
6  0.00000000 NOT USED
7  0.03125000 OUTSIDE DIAMETER (FT)
8  250.00000000 TOTAL WEIGHT FLOW (LB/HR)
9  0.00600000 WALL THICKNESS (FT)
10 1.00000000 NO. OF DUCTS
11 0.70000000 SPECIFIC HEAT OF FLUID (BTU/LB R)
12 560.00000000 FLUID TEMP AT ENTRANCE (R)
13 660.00000000 FLUID TEMP AT EXIT (R)
14 2.00000000 NO. OF SUBSECTIONS
15 3.50000000 NUSSELTS NO.

INPUT DUCT VALUES
SFC EFFECT PERIFERAL LENGTH FOR RADIATION (FT) 20
  1 0.09817000
SFC EFFECT PERIFERAL LENGTH FOR CONVECTION (FT) 21
  1 0.09817000
SEC RADIATION CONSTANT C1 (BTU/HR SQ FT R**4) 22
  1 0.13700000E-08
SEC RADIATION CONSTANT C2 (BTU/HR SQ FT) 23
  1 1029.00000000
SEC Convective Heat Transfer Coeff (BTU/HR SQ FT R) 24
  1 1.00000000
SEC Ambient Temp (R) 25
  1 925.00000000

INPUT FIN VALUES
FIN Radiation Constant C1 (BTU/HR SQ FT R**4) 80
  1 0.
FIN Radiation Constant C2 (BTU/HR SQ FT) 81
  1 0.00000000
FIN Convective Heat Transfer Coeff Side A (BTU/HR SQ FT R) 82
  1 0.00000000
FIN Convective Heat Transfer Coeff Side B (BTU/HR SQ FT R) 83
  1 0.00000000
FIN Ambient Temp Side A (R) 84
  1 0.00000000
FIN Ambient Temp Side B (R) 85
  1 0.00000000
FIN NOT USED 86
  1
FIN Effect Length at Exit (FT) 87
  1 0.00000000
FIN Effect Length at Entrance (FT) 88
  1 0.00000000
FIN Duct Wall Temp at Exit (R) 89
  1 0.00000000
FIN Duct Wall Temp at Entrance (R) 90
  1 0.00000000

Fig. B-4 Input Data - Problem 2
### Section 1

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<td>572.25449371</td>
<td>619.60701752 WALL TEMP (R)</td>
</tr>
<tr>
<td>177.45729256</td>
<td>239.59266663 HEAT TRANSFER COEFF. (BTU/HR SQ FT)</td>
</tr>
<tr>
<td>2193.12750244</td>
<td>3964.59719849 REYNOLDS NO.</td>
</tr>
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</table>

- 36191505 VELOCITY AVG (FT/SEC)
- 69.34133530 SECTION LENGTH (FT)
- 8581.0565918 HEAT TRANSFER (BTU/HR)
- 208.52497864 HEAT TRANSFER COEFF AVG (BTU/HR SQ FT)
- 3078.86233521 REYNOLDS NO. AVG
- 54.70989227 FLUID DENSITY AVG
- 1.10410248 WEIGHT OF LIQUID (LB)

### Section 2

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<td>619.60701752</td>
<td>666.95754132 WALL TEMP (R)</td>
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<tr>
<td>239.59266663</td>
<td>254.78985596 HEAT TRANSFER COEFF. (BTU/HR SQ FT)</td>
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<td>3964.59719849</td>
<td>4603.47137451 REYNOLDS NO.</td>
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- 4.46136743 VELOCITY AVG (FT/SEC)
- 74.75459385 SECTION LENGTH (FT)
- 8518.94531250 HEAT TRANSFER (BTU/HR)
- 247.19126129 HEAT TRANSFER COEFF AVG (BTU/HR SQ FT)
- 4274.03424072 REYNOLDS NO. AVG
- 53.48989296 FLUID DENSITY AVG
- 2.16375335 WEIGHT OF LIQUID (LB)
- 2545.14883423 PRESSURE CHANGE (LB/SQ FT)

### Final Output Data

- 4418.37603760 PRESSURE CHANGE SUM (LB/SQ FT)
- 17499.99584961 TOTAL HEAT TRANSFER (BTU/HR)
- 2.26785582 TOTAL WEIGHT OF LIQUID (LB)
- 929.41597748 ENVIRON TEMP (R)
- 144.09592819 TOTAL LENGTH (FT)

---

Fig. B-5 Output Data - Problem 2
EFFECTIVENESS FOR TRAPEZOIDAL PLATE FINS AND OPTIONAL TEMP PROF

1  -1.000000 CODE FOR INITIAL VALUES
2   1.000000 CODE FOR TEMP PROFILE
3   15.000000 ITERATION LIMITS
4   0.617000 THICKNESS RATIO
5   2.000000 FIN LENGTH (INCHES)
6   730.000000 ROOT EDGE TEMP (DEG RANKINE)

**C1 AND C2 ENTERED AS DATA
7   0.14560000E-08 C1
8   0.88609999E 02 C2
9   0.11800000E 03 FIN THER COND (BTU/HR-FT-DEG(R))

25  0.32400000E-01 ROOT EDGE THICKNESS (INCHES)

C1   = 0.14560000E-08
C2   = 0.88609999E 02
C3   = 0.21430392E 00
ZETA = 0.49283479E-01
FINAL Z = 0.97934200E 00
FINAL DZ = 0.36285591E-01
FINAL OMEGA = 0.99999998E 00
FH   = 0.
FAH  = 0.

TEMP RATIO (TE/TH) = 0.68039E 00
EFF ENVIRON TEMP = 0.49668E 03 (DEG(R))

FINAL EFFECTIVENESS = 0.734772
AREA EFFECTIVENESS = 0.935186
EFFECTIVE LENGTH = 1.87037 (INCHES)

Q    = 0.50635E 02 BTU/HR (FOOT OF LENGTH)

Note: Data Calculated with Program 2-1, Ref. 1

Fig. B-6   Fin Data at Entrance - Problem 3
B-15

Effective Width for Multisection Fins and Optional Temp Prof

1  1.00 Code for Temp Prof
2  2.00 Number of Sections
3  15.00 Iteration Limit

4  0.00000000 Initial DZ/DW (Optional)

Root Edge Temp(R) = 0.68000000F 03

Section Number = 1

12 Thickness Root Edge = 0.3240000000F-01 (INCHES)
13 Thickness Far Edge = 0.2000000000F-01 (INCHES)
14 Fin Length = 0.2000000000F 01 (INCHES)
15 Thermal Conductivity = 0.1180000000E 03 (BTU/HR-FT-DEG(R))

29 C1 (Entered as Data) = 0.1456000000F-08
30 C2 (Entered as Data) = 0.88609999E 02

Input Data for Convection

31 0.00000000 Heat Transfer Coefficient Side A
32 0.00000000 Heat Transfer Coefficient Side B
33 0.00000000 Ambient Temp A (DEG(R))
34 0.00000000 Ambient Temp B (DEG(R))

Section Number = 2

42 Thickness Root Edge = 0.2000000000F-01 (INCHES)
43 Thickness Far Edge = 0.2000000000F-01 (INCHES)
44 Fin Length = 0.2000000000F 01 (INCHES)
45 Thermal Conductivity = 0.1180000000E 03 (BTU/HR-FT-DEG(R))

59 C1 (Entered as Data) = 0.1456000000F-08
60 C2 (Entered as Data) = 0.88609999E 02

Input Data for Convection

61 0.00000000 Heat Transfer Coefficient Side A
62 0.00000000 Heat Transfer Coefficient Side B
63 0.00000000 Ambient Temp A (DEG(R))
64 0.00000000 Ambient Temp B (DEG(R))

***Convergence Accomplished

Effective Length = 3.735892 (INCHES)

Note: Data Calculated with Program 2-4, Ref. 1

Fig. B-7 Fin Data at Exit - Problem 3
EXAMPLE PROBLEM

INPUT DATA

1.00000000 NO. OF DUCT SECTIONS
2.00000000 NO. OF FINS
0.05180000 DUCT PERIMETER (FT) *OPTION
0.00110000 DUCT AREA (SQ FT) *OPTION
0.00000000 EFFECTIVE DIAMETER (FT) *OPTION
0.00000000 NOT USED
0.00000000 OUTSIDE DIAMETER (FT)
20.00000000 TOTAL WEIGHT FLOW (LB/HR)
0.00000000 WALL THICKNESS (FT)
1.00000000 NO. OF DUCTS
0.70000000 SPECIFIC HEAT OF FLUID (BTU/LB R)
760.00000000 FLUID TEMP AT ENTRANCE (R)
710.00000000 FLUID TEMP AT EXIT (R)
2.00000000 NO. OF SUBSECTIONS
3.50000000 NUSSELT'S NO.

INPUT DUCT VALUES

SFC EFFECT PERIFERAL LENGTH FOR RADIATION (FT) 1 0.03040000
SFC EFFECT PERIFERAL LENGTH FOR CONVECTION (FT) 1 0.03650000
SFC RADIATION CONSTANT C1 (BTU/HR SQ FT R**4) 1 0.14560000*0-08
SFC RADIATION CONSTANT C2 (BTU/HR SQ FT) 1 88.60999966
SFC CONVETIVE HEAT TRANSFER COEFF (BTU/HR SQ FT R) 1 0.00000000
SFC AMBIENT TEMP (R) 1 0.00000000

INPUT DATA

20
21
22
23
24
25

Fig. B-8 Input Data - Problem 3
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<th>FIN</th>
<th>INPUT FIN VALUES</th>
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Fig. B-8 Input Data - Problem 3 (cont)
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| SECTION 2 |

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<td>0.01782708 WEIGHT OF LIQUID (LB)</td>
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<td>39.32037449 PRESSURE CHANGE (LB/SQ FT)</td>
<td></td>
</tr>
</tbody>
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<table>
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<th>FINAL OUTPUT DATA</th>
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<tr>
<td>73.06214333 PRESSURE CHANGE SUM (LB/SQ FT)</td>
</tr>
<tr>
<td>699.92832947 TOTAL HEAT TRANSFER (BTU/HR)</td>
</tr>
<tr>
<td>0.03305686 TOTAL WEIGHT OF LIQUID (LB)</td>
</tr>
<tr>
<td>496.68486404 ENVIRON TEMP (R)</td>
</tr>
<tr>
<td>5.83163404 TOTAL LENGTH (FT)</td>
</tr>
</tbody>
</table>

Fig. B-9 Output Data - Problem 3
EFFECTIVENESS FOR TRAPEZOIDAL PLATE FINS

1. 0.600000 CODE FOR INITIAL VALUES
2. 1.000000 CODE FOR TEMP PROFILE
3. 15.000000 ITERATION LIMITS
4. 1.000000 THICKNESS RATIO
5. 15.000000 FIN LENGTH (INCHES)
6. 1.760.000000 ROOT EDGE TEMP (DEG RANKINE)

**THE FOLLOWING QUANTITIES ARE ENTERED AS DATA
9. 90.000000 FIN THER COND (BTU/HR-FT-DEG(R))
10. 0.500000 FA
11. 0.200000 ALPHA A
12. 0.850000 EPS A
13. 0.000000 FAX
14. 0.000000 FB
15. 0.000000 ALPHA B
16. 0.000000 EPS B
17. 0.000000 FBX
18. 5.300000 TM (DEG RANKINE)
19. 90.000000 THETA M (DEG)
20. 0.100000 RHC M
21. 0.000000 TX (DEG RANKINE)
22. 0.000000 THETA X (DEG)
23. 0.000000 RHC X
24. 0.000000 THETA P (DEG)
25. 0.095000 ROOT EDGE THICKNESS (INCHES)
39. 0.000000 EPSX

INPUT DATA FOR CONVECTION STUDY
31. 7.000000 HEAT TRANSFER COEFFICIENT SIDE A
32. 12.000000 HEAT TRANSFER COEFFICIENT SIDE B
33. 536.000000 AMBIENT TEMP A
34. 600.000000 AMBIENT TEMP B

\[
\begin{align*}
C1 &= 0.14560500E-08 \\
C2 &= 0.146005317E-03 \\
C3 &= 0.3066325E-00 \\
\text{ZETAP} &= 0.14016068E-01 \\
\text{FINAL Z} &= 0.7517499E-00 \\
\text{FINAL D71} &= -0.16260223E-01 \\
\text{FINAL OMEGA} &= 0.6759959E-00 \\
FH &= 0.41666667E-02 \\
FAH &= 0.31480894E-02 \\
C2A &= 0.14605317E-03 \\
\text{TEMP RATIO (TE/TH)} &= 0.75474E+00 \\
\text{EFF ENVIRON TEMP} &= 0.57361E-03 \text{ (DEG(R))} \\
\text{FINAL EFFECTIVENESS} &= 0.141189 \\
\text{AREA EFFECTIVENESS} &= 0.146517 \\
\text{EFFECTIVE LENGTH} &= 2.15776 \text{ (INCHES)}
\end{align*}
\]

Fig. B-10   Fin No. 1 (at inlet) - Problem 4
**SECTION NUMBER = 1**

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<tbody>
<tr>
<td>12 THICKNESS ROOT EDGE</td>
<td>0.190000 (INCHES)</td>
</tr>
<tr>
<td>13 THICKNESS FAR EDGE</td>
<td>0.190000 (INCHES)</td>
</tr>
<tr>
<td>14 FIN LENGTH</td>
<td>1.720000 (INCHES)</td>
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<tr>
<td>15 THERMAL CONDUCTIVITY</td>
<td>90.000000 (BTU/HR-FT-DEG(R))</td>
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<tr>
<td>16 FAX</td>
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</tr>
<tr>
<td>17 ALPHA A</td>
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<tr>
<td>18 EPS A</td>
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<tr>
<td>19 FBX</td>
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<tr>
<td>20 ALPHA B</td>
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<tr>
<td>21 EPS B</td>
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</tr>
<tr>
<td>22 TX</td>
<td>530.000000 (DEGREES RANKINE)</td>
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<tr>
<td>23 THETA X</td>
<td>90.000000 (DEGREES)</td>
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<tr>
<td>24 RHO X</td>
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<tr>
<td>25 THETA P</td>
<td>0.000000 (DEGREES)</td>
</tr>
<tr>
<td>26 EPS X</td>
<td>1.000000</td>
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</tbody>
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**INPUT DATA FOR CONVECTION**

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<tbody>
<tr>
<td>31</td>
<td>7.0000000 HEAT TRANSFER COEFFICIENT SIDE A</td>
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<tr>
<td>32</td>
<td>0.0000000 HEAT TRANSFER COEFFICIENT SIDE B</td>
</tr>
<tr>
<td>33</td>
<td>530.000000 AMBIENT TEMP A (DEG(R))</td>
</tr>
<tr>
<td>34</td>
<td>0.000000 AMBIENT TEMP B (DEG(R))</td>
</tr>
</tbody>
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**COMPUTED VALUES OF C1, C2, C3**

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<td>0.14560500E-08</td>
</tr>
<tr>
<td>C2</td>
<td>0.14605317E 03</td>
</tr>
<tr>
<td>C3</td>
<td>0.30066325E 00</td>
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**SECTION NUMBER = 2**

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</thead>
<tbody>
<tr>
<td>42 THICKNESS ROOT EDGE</td>
<td>0.195000 (INCHES)</td>
</tr>
<tr>
<td>43 THICKNESS FAR EDGE</td>
<td>0.195000 (INCHES)</td>
</tr>
<tr>
<td>44 FIN LENGTH</td>
<td>15.000000 (INCHES)</td>
</tr>
<tr>
<td>45 THERMAL CONDUCTIVITY</td>
<td>90.000000 (BTU/HR-FT-DEG(R))</td>
</tr>
<tr>
<td>46 FAX</td>
<td>0.500000</td>
</tr>
<tr>
<td>47 ALPHA A</td>
<td>0.200000</td>
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<tr>
<td>48 EPS A</td>
<td>0.850000</td>
</tr>
<tr>
<td>49 FBX</td>
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<tr>
<td>50 ALPHA B</td>
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<tr>
<td>51 EPS B</td>
<td>0.000000</td>
</tr>
<tr>
<td>52 TX</td>
<td>530.000000 (DEGREES RANKINE)</td>
</tr>
<tr>
<td>53 THETA X</td>
<td>90.000000 (DEGREES)</td>
</tr>
<tr>
<td>54 RHO X</td>
<td>0.100000</td>
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<tr>
<td>55 THETA P</td>
<td>0.000000 (DEGREES)</td>
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<tr>
<td>56 EPS X</td>
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**INPUT DATA FOR CONVECTION**

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<tr>
<td>61</td>
<td>7.0000000 HEAT TRANSFER COEFFICIENT SIDE A</td>
</tr>
<tr>
<td>62</td>
<td>0.0000000 HEAT TRANSFER COEFFICIENT SIDE B</td>
</tr>
<tr>
<td>63</td>
<td>530.000000 AMBIENT TEMP A (DEG(R))</td>
</tr>
<tr>
<td>64</td>
<td>0.000000 AMBIENT TEMP B (DEG(R))</td>
</tr>
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</table>

**EFFECTIVE LENGTH = 4.033377 (INCHES)**

---

Note: Data Calculated with Program 2-4, Ref. 1

---

**Fig. B-11**  Fin No. 2 (at inlet) - Problem 4
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</tr>
</thead>
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<td>0.000000</td>
<td>0.000000</td>
<td></td>
</tr>
</tbody>
</table>

**INPUT DATA FOR CONVECTION**

- Heat transfer coefficient Side A: 12.000000
- Heat transfer coefficient Side B: 0.000000
- Ambient Temp A: 600.000000 (DEG(R))
- Ambient Temp B: 0.000000 (DEG(R))

<table>
<thead>
<tr>
<th>Section Number</th>
<th>Effective Length [INCHES]</th>
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<tbody>
<tr>
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Note: Data Calculated with Program 2-4, Ref. 1.

Fig. B-12   Fin No. 3 - Problem 4
EXAMPLE PROBLEM

INPUT DATA

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<th>Value</th>
<th>Description</th>
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<tbody>
<tr>
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<td>3.000000000</td>
<td>NO. OF DUCT SECTIONS</td>
</tr>
<tr>
<td>2</td>
<td>3.000000000</td>
<td>NO. OF FINS</td>
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<tr>
<td>3</td>
<td>0.000000000</td>
<td>DUCT PERIMETER (FT) *OPTION</td>
</tr>
<tr>
<td>4</td>
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<td>DUCT AREA (SQ FT) *OPTION</td>
</tr>
<tr>
<td>5</td>
<td>0.037500000</td>
<td>EFFECTIVE DIAMETER (FT) *OPTION</td>
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<tr>
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<tr>
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<tr>
<td>8</td>
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<td>TOTAL WEIGHT FLOW (LB/HR)</td>
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<tr>
<td>9</td>
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</thead>
<tbody>
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<table>
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<tr>
<th>SEC</th>
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<table>
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<tr>
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<th>Value</th>
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</thead>
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<table>
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Fig. B-13 Input Data - Problem 4
<table>
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<tr>
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<td>DUCT WALL TEMP AT ENTRANCE (R)</td>
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Fig. B-13 Input Data - Problem 4 (cont)
## SECTION 1

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<th>WALL TEMP (R)</th>
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<tr>
<td>10309.24023438</td>
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</tbody>
</table>

- **INLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

- **OUTLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

**SECTION DATA:**

- **VELOCITY AVG (FT/SEC):** 5.02104044
- **SECTION LENGTH (FT):** 22.27206993
- **HEAT TRANSFER (BTU/HR):** 35439.19287109
- **HEAT TRANSFER COEFF AVG (BTU/HR SQ FT):** 241.90506363
- **REYNOLDS NO. AVG:** 10128.09191895
- **WALL TEMP (R):** 749.037258148
- **BLK NFAT TRANSFER COEFFO:** 9946.94372559
- **REYNOLDS NO.:** 50.09765434
- **FLOW DENSITY AVG:** 1.23233952
- **WEIGHT OF LIQUID (LB):** 363.09560776
- **PRESSURE CHANGE (LB/SQ FT):**

## SECTION 2

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</tr>
</thead>
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- **INLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

- **OUTLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

**SECTION DATA:**

- **VELOCITY AVG (FT/SEC):** 4.90042037
- **SECTION LENGTH (FT):** 29.68588352
- **HEAT TRANSFER (BTU/HR):** 35288.41748047
- **HEAT TRANSFER COEFF AVG (BTU/HR SQ FT):** 242.80842400
- **REYNOLDS NO. AVG:** 9778.48706055
- **WALL TEMP (R):** 698.96055603
- **BLK NFAT TRANSFER COEFFO:** 9610.03039551
- **REYNOLDS NO.:** 51.33033562
- **WALL DENSITY AVG:** 1.68297043
- **WEIGHT OF LIQUID (LB):** 476.79207230
- **PRESSURE CHANGE (LB/SQ FT):**

## SECTION 3

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<th>HEAT TRANSFER COEFF. (BTU/HR SQ FT)</th>
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</table>

- **INLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

- **OUTLET**
  - FLUID TEMP (R)
  - WALL TEMP (R)
  - HEAT TRANSFER COEFF. (BTU/HR SQ FT)

**SECTION DATA:**

- **VELOCITY AVG (FT/SEC):** 4.78728259
- **SECTION LENGTH (FT):** 43.67850208
- **HEAT TRANSFER (BTU/HR):** 34271.89208984
- **HEAT TRANSFER COEFF AVG (BTU/HR SQ FT):** 236.36452293
- **REYNOLDS NO. AVG:** 9230.09582520
- **WALL DENSITY AVG:** 2.54267263
- **WEIGHT OF LIQUID (LB):** 696.03276825
- **PRESSURE CHANGE (LB/SQ FT):**

**FINAL OUTPUT DATA:**

- **PRESSURE CHANGE SUM (LB/SQ FT):** 1535.92044067
- **TOTAL HEAT TRANSFER (BTU/HR):** 104999.50195313
- **TOTAL WEIGHT OF LIQUID (LB):** 5.45004338
- **ENVIRON TEMP (R):** 533.96514130
- **TOTAL LENGTH (FT):** 95.63645554

**END-OF-DATA ENCOUNTERED ON SYSTEM INPUT FILE.**

---

Fig. B-14 Output Data - Problem 4
EXAMPLE PROBLEM

INPUT DATA

1 0.04167 OUTSIDE DIAMETER (FT)
2 0.03083 INSIDE DIAMETER (FT)
3 12.00000 NO. OF TUBES
4 800.00000 WEIGHT FLOW (LBS/HR)
5 0.14580 FIN LENGTH (FT)
6 0.01040 FIN THICKNESS AT ROOT (FT)
7 0.01040 FIN THICKNESS AT FAR EDGE (FT)
8 490.00000 DENSITY OF FIN MATERIAL (LBS/CU FT)
9 490.00000 DENSITY OF TUBE MATERIAL (LBS/CU FT)
10 0.00000 NOT USED
11 26.00000 THERM COND OF FIN (BTU/FT HR R)
12 1.00000 SPECIFIC HEAT OF FLUID (BTU/LB R)
13 660.00000 FLUID TEMP AT ENTRANCE (R)
14 6.00000 HEAT TRANSFER COEFFICIENT SIDE A (HA. BTU/HR SQ FT R)
15 6.00000 HEAT TRANSFER COEFFICIENT SIDE B (HB. BTU/HR SQ FT R)
16 535.00000 AMBIENT TEMP SIDE A (R)
17 535.00000 AMBIENT TEMP SIDE B (R)
18 0.00000 ALPHA A
19 0.00000 ALPHAB
20 0.85000 EPS A
21 0.85000 EPS B
22 0.00000 EPS X
23 1.00000 FA
24 0.00000 FAX
25 1.00000 FB
26 0.00000 FBX
27 0.00000 RHOM
28 0.00000 RHONX
29 0.00000 TETRAP (DEG)
30 0.00000 TETRAM (DEG)
31 0.00000 TETTAX (DEG)
32 530.00000 TM (R)
33 0.00000 TX (R)
34 1.00000 EPSM
35 15.00000 ITERATION LIMIT
36 15.00000 NO. OF INTEGRATION STEPS
37 0.00000 SOLAR CONSTANT (BTU/HR SQ FT)
38 0.50000 HEAT EXCHANGER EFFECTIVENESS
39 3.00000 NO. OF SUBSECTIONS
40 3000.00000 PRESSURE (LBS/SQ FT)
41 4.36400 NUSSLT NO.

Fig. B-15 Input Data - Problem 5
SECTION NO. 1

INLET           OUTLET
0.66000E 03     0.63764E 03 FLUID TEMP (R)
0.64286E 03     0.62148E 03 WALL TEMP (R)
0.10681E 00     0.10728E 00 FIN EFFECT LENGTH (FT)
0.25062E 03     0.21226E 03 HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R)

0.23144E 03 HEAT TRANSFER COEFFICIENT AVG (BTU/HR FT SQ R)
0.24345E 04 REYNOLDS NO. AVG
0.41093E 00 VELOCITY AVG (FT/SEC)
0.39945E 01 SECTION LENGTH (FT)
0.74671E 00 PRESSURE CHANGE (LBS/SQ FT)
0.14908E 04 HEAT TRANSFER (BTU/HR TUBE)
0.60356E 02 FLUID DENSITY AVG
0.18001E 00 WT OF LIQUID (LBS)

FLUID TEMP CONVERGENCE FAILED. TF2 = 0.61455E 03 RE2P = 0.18037E 04 RE2 = 0.16127E 04

Fig. B-16  Output Data - Problem 5