RICE UNIVERSITY
Houston, Texas

HEAT EXCHANGERS FOR CONVECTIVE AND RADIATIVE ENVIRONMENTS

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Aerospace Technology Report #4

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ABSTRACT

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 invarient 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 claculations 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, ($\mathcal{E}$), 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 $\mathcal{E}$ specified by the user. A sufficiently large temperature difference (high value for $\mathcal{E}$) 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

Equations	Computer Program	Definition
$A_{a,aa}$	AA-AAA	Equation constants
A_d	AD	Duct cross sectional area for fluid flow, sq ft/duct
α_a	ALPHA A	Absorptivity of surface facing sun, nondimensional
α_b	ALPHA B	Absorptivity of surface away from sun, nondimensional
A_p	AP	Plan form area exchanging heat with the environment. (Heat may be exchanged from both sides of the extended surface), sq ft
$B_{b,bb}$	BB-BBB	Equation constants
b	BD	Constant (see Eq. 1-23)
$\mathcal{E}$	BIGE	Heat exchanger effectiveness, non-dimensional
b(L)	B(L)	Subscripted constants
C_c	CC	Equation constant (Eq. 1-24)
N_{nu}	CKH	Nusselt number, nondimensional
$K_{o,5}$	CKO,...,CK5	Heat transfer constants, (see Eq. 2-21 to 2-26)
	CM(I)	Subscripted constant in Runge-Kutta-Gill integration routine
C_p	CP	Fluid specific heat, Btu/lb R
C_1	C1	Radiative constant, $\sigma(\epsilon_a + \epsilon_b)$, Btu/hr sq ft R^4

Equations	Computer Program	Definition
$C_1(L)$	C1(L)	Radiative constant, C_1 for fin (L), Btu/hr sq ft R^4
$C_{1d}(I)$	C1D(I)	Radiative constant, C_1, for duct section (I), Btu/hr sq ft R^4
C_2	C2	Radiative constant, heat received from environment by both surfaces of a unit of fin area, Btu/hr sq ft
$C_{2d}(I)$	C2D(I)	Radiative constant C_2 for duct section (I), Btu/hr sq ft
$C_2(L)$	C2(L)	Radiative constant C_2 for fin (L), Btu/hr sq ft
C_3	C3	Environmental parameter, $C_2/C_1 T_W^4$, nondimensional
δ_h	DELTAH	Thickness of root or attachment end of extended surface, ft
D_i	DI	Duct inside diameter (or effective diameter), ft
D_o	DO	Tube outside diameter, ft
	DP	Fluid pressure change in section, lb/sq ft
	DPSUM	Accumulated sum of pressure drop, lb/sq ft
DZO	DZO	$(dZ/d\omega)_1$ for a previous attempt at convergence where heat transfer was low
DZW	DZW	$(dZ/d\omega)_1$ for a previous attempt at convergence where heat transfer was high
$(dZ/d\omega)_1$	DZ1	Initial value of $(dZ/d\omega)$ to start integration routine, nondimensional
$(dZ/d\omega)_1$	DZ1A	Initial value of $(dZ/d\omega)$ at the root of the first section, nondimensional

Equations	Computer Program	Definition
$\frac{d^2Z}{d\omega^2}$	D2Z	Function statement, nondimensional
Ω_c	EFC	Flat plate convective effectiveness, nondimensional
Ω_r	EFR	Flat plate radiative effectiveness (approximate), nondimensional
L_c	ELC	Duct width for convective heat transfer, ft
L_e	ELE	Equivalent length of a fin, ft
L_{ef}	ELEEND	Equivalent length of fin at conditions approximating the end of duct (Program 4-1), ft
$L_e(L)$	ELE(L)	Equivalent length for fin (L), ft
$L_{ef}(L)$	ELEF(L)	Equivalent length for fin (L) at conditions approximating the end of duct, or at temperature T_{wf}, ft
	ELEMID	Equivalent length for fin at wall temperature intermediate between entrance and exit (Program 4-1), ft
$L_{es}(L)$	ELES(L)	Equivalent length for fin (L) at conditions approximating the entrance to duct or at temperature, T_{ws}, ft
L_{es}	ELE1	Equivalent length for fin at conditions approximating entrance to duct (Program 4-1), ft
L_w	ELW	Duct section length, ft
L_{ws}	ELWSUM	Total duct length measured from inlet, ft
$\dot{m}(L)$	EM(L)	Subscripted constants, see Eq.(2-17)

Equations	Computer Program	Definition
N	ENT	Number of ducts in heat exchanger
ϵ_a	EPSA	Emissivity of extended surface facing sun, nondimensional
ϵ_b	EPSB	Emissivity of extended surface away from sun, nondimensional
ϵ_m	EPSM	Emissivity of body "m", nondimensional
ϵ_x	EPSX	Emissivity of external body, "x", nondimensional
η_a	ETAA	Area effectiveness, ideal/actual, nondimensional
F_a	FA	Radiative form factor between heat exchanger and body, "m", for surface facing sun, nondimensional
F_{ah}	FAH	Environmental convective parameter, nondimensional
F_{ax}	FAX	Radiative form factor between the heat exchanger surface facing the sun and a second surface near the heat exchanger, nondimensional
F_b	FB	Radiative form factor between heat exchanger and body "m" for surface away from sun, nondimensional
F_{bx}	FBX	Radiative form factor between the heat exchanger surface away from sun and a second surface near the exchanger, nondimensional
F_e	FE	Variable used in Newton's iteration procedure
f	FF	Fluid friction factor, nondimensional
	FFL	Friction factor for laminar flow, nondimensional
	FFT	Friction factor for turbulent flow, nondimensional

Equations	Computer Program	Definition
F_h	FH	Extended surface convective parameter, nondimensional
L_h	FINLH	Extended surface length of section, ft
δ_h	FINTH	Thickness of extended surface at root edge, ft
m	FM	Constant, see Eq. (1-22)
N_s	FMESH	Number of sections in which the duct length is divided, nondimensional
k	FNK	Thermal conductivity of fin material, Btu/hr ft R
	FO(I)	Subscripted integration increment
F_w	FWH, FW(K-1)	Variable used in linear convergence process
$G_{d1,3}$	GD1,...GD3	Constants for duct
$G_2(L)$	G2(L)	Subscripted constants
$G_3(L)$	G3(L)	Subscripted constants
h	H	Convective heat transfer coefficient from duct fluid to wall, Btu/hr sq ft R
h_a	HA	Convective heat transfer coefficient on the side facing the sun (if applicable) Btu/hr sq ft R
$h_a(L)$	HA(L)	Convective heat transfer coefficient to the environment for side "a" of fin (L), Btu/hr sq ft R
h_{at}	HAT	Sum of convective terms, $(h_a T_{aa} + h_b T_{ab})$, Btu/hr sq ft
h_b	HB	Convective heat transfer coefficient on the side away from the sun (if applicable), Btu/hr sq ft R

Equations	Computer Program	Definition
$h_b(L)$	HB(L)	Convective heat transfer coefficient to the environment from side"b" of fin (L), Btu/hr sq ft R
$h_d(I)$	HD(I)	Convective heat transfer coefficient to environment from duct section (I), Btu/hr sq ft R
L_h	HL	Extended surface length, ft
h_t	HT	Sum of convective heat transfer coefficients, $(h_a + h_b)$, Btu/hr sq ft R
	ISO	Switch to signal a zero condition has been encountered
	ISW	Switch to signal a wilt condition has been encountered
	ITER	The number of attempts for a starting value of $dZ/d\omega$
	ITLT	Maximum number of iterations allowed to revise initial $dZ/d\omega$
ω	OMEGA	Ratio, L/L_h where L represents distance that the heat has traveled along the fin and L_h represents the total length, nondimensional
	P	Pressure, lb/sq ft
p	PERIM	Duct internal perimeter, ft
N_{pr}	PRN	Prandtl number computed by heat transfer coefficient subroutine AICH, nondimensional
q	Q	Heat exchanged with environment, Btu/hr per tube
q_s	QS	Heat transfer to the environment from a unit of duct length, Btu/hr
	QSUM	System heat exchanged with environment, Btu/hr

Equations	Computer Program	Definition
N_{re}	RE	Reynolds number, nondimensional
	REAV	Average Reynolds number, nondimensional
ρ	RHO	Fluid density, lb/cu ft
ρ_{av}	RHOAV	Average fluid density, lb/cu ft
ρ_f	RHOFM	Density of fin material, lb/cu ft
ρ_m	RHOM	Surface reflectivity of body "m", nondimensional
ρ_d	RHOTM	Density of duct material, lb/cu ft
ρ_x	RHOX	Reflectivity of second surface, x, nondimensional
S_c	SC	Solar heat, Btu/hr sq ft
$S_c(I)$	SC(I)	Peripheral duct length for convective heat transfer from section (I), ft
$S_r(I)$	SR(I)	Effective peripheral duct length for radiative heat transfer from section (I), ft
T		Temperature at any point on an extended surface, R
T_{aa}	TAA	Ambient fluid temperature on side "a", R
$T_{aa}(L)$	TAA(L)	Ambient environmental temperature for side "a" of fin (L), R
T_{ab}	TAB	Ambient fluid temperature on side "b", R
$T_{ab}(L)$	TAB(L)	Ambient environmental temperature for side "b" of fin (L), R
$T_a(I)$	TA(I)	Ambient environmental temperature for duct section (I), R

Equations	Computer Program	Definition
T_b	TB	Bulk (mixed) fluid temperature for AICH subroutine, R
T_e	TE	Effective environmental temperature, see Eq. (1-5) to (1-7), R
T_f	TF	Temperature of fluid in duct, R
T_{fend}	TFEND	Fluid exit temperature, T
T_{f1}	TF1	Fluid inlet temperature,R
θ_m	THETAM	Angle between sun's rays and normal to body "m" surface, degrees
θ_p	THETAP	Angle between sun's rays and normal to fin surface, degrees
θ_x	THETAX	Angle between sun's rays and normal to second surface, degrees
T_m	TM	Surface temperature of body "m", R
T_s	TS	Duct wall temperature for AICH subroutine, R
T*	TSTAR	Temperature used for calculating N_{re} and N_{pr} in AICH subroutine, R
T_w	TW	Duct wall temperature, R
$T_{wf}(L)$	TWF(L)	Duct temperature at root of fin (L) for conditions approximating the exit of duct, and used to calculate, $L_{ef}(L)$, R
$T_{ws}(L)$	TWS(L)	Duct temperature at root of fin (L) for conditions approximating the entrance of duct, and used to calculate $L_{es}(L)$, R
T_x	TX	Surface temperature of body "x", R
	T(2)	Existing value of ω, nondimensional
$d\omega$	T(3)	Increment of ω used for calculations, nondimensional
Z	T(4)	Value of T/T_w, nondimensional

Equations	Computer Program	Definition
$dZ/d\omega$	T(5)	$dZ/d\omega$, nondimensional
V	V	Fluid velocity, ft/sec
V_{av}	VAV	Arithmetic mean of inlet and outlet velocity, ft/sec
μ	VISC	Viscosity of fluid, lb/hr ft
	WALTH	Duct wall thickness, ft
	WD	Weight of a tube, lb
$\dot{w}$	WDOT	Total fluid weight flow, lb/hr
$\dot{w}_d$	WDOTD	Fluid weight flow in a duct, lb/hr
	WF	Weight of extended surfaces attached to a tube, lb
	WILT	Value of $(dZ/d\omega)_1$ used for the previous iteration, nondimensional
	WL	Weight of the liquid trapped in a section of duct, lb
W_t	WT	Total weight of heat exchanger including fluid, lb
y	Y	Heat transfer rate at fin root temperature, Btu/hr sq ft
Z	Z	Temperature ratio T/T_w for integration routine, nondimensional
ζ_p	ZETAP	Profile number for rectangular plan extended surface, nondimensional

SUBSCRIPTS

a		Surface facing the sun (if applicable)
av		Arithmetic average
b		Surface in shade (if applicable)

Equations	Computer Program	Definition
(I)		Duct section "I"
(L)		Fin "L"
1		Denotes entrance to duct section (if applicable)
2		Denotes exit from duct section (if applicable)

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

$$dq = [C_1 T^4 - C_2 + h_a (T - T_{aa}) + h_b (T - T_{ab})] dA_p . \tag{1.1}$$

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,

$$F_e = C_1 T_e^{\,4} + h_t T_e - (C_2 + h_{at}) = 0 \tag{1.2}$$

where

$$h_t = h_a + h_b \tag{1.3}$$

and

$$h_{at} = h_a T_{aa} + h_b T_{ab} \quad . \tag{1.4}$$

If the convective heat transfer coefficients $h_t = h_{at} = 0$,

$$T_e = \left(\frac{C_2}{C_1} \right)^{\frac{1}{4}} . \tag{1.5}$$

If the radiative heat transfer is zero $C_1 = C_2 = 0$,

$$T_e = \frac{h_{at}}{h_t} \quad . \tag{1.6}$$

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

$$T_e = T_e' - F_e \Big/ \frac{dF_e}{dT_e} \tag{1.7}$$

where

$$\frac{dF_e}{dT_e} = 4C_1 T_e^{\,3} + h_t \quad . \tag{1.8}$$

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

$$C_1 = (\epsilon_a + \epsilon_b)\sigma = (\epsilon_a + \epsilon_b)0.1713\text{x}10^{-8} \tag{1.9}$$

[Ref. 1, Eq. 2]

and

$$C_2 = S_c(\alpha_a \cos\theta_p + F_a\alpha_a\rho_m \cos\theta_m + F_{ax}\alpha_a\rho_x \cos\theta_x + F_b\alpha_b\rho_m \cos\theta_m$$
$$+ F_{bx}\alpha_b\rho_x \cos\theta_x) + 0.1713\text{x}10^{-8}\Big[(F_a\epsilon_a + F_b\epsilon_b)\epsilon_m T_m^{\,4}$$
$$+ (F_{ax}\epsilon_x + F_b\epsilon_b)\epsilon_x T_x^{\,4}\Big] + 0.01(\epsilon_a + \epsilon_b). \tag{1.10}$$

[Ref. 1, Eq. 3]

The heat transfer from an element of fin shown in Fig. 1 can be written as

$$d_{qf} = \frac{-k\delta_h T_w}{L_h} \left(\frac{dZ}{d\omega}\right) dL = yL_h \eta_a dL = yL_e dL \tag{1.11}$$

where

$$y = C_1 T_w^{\ 4} - C_2 + T_w h_t - h_{at} \tag{1.12}$$

$$\omega = \frac{L}{L_h} \tag{1.13}$$

and

$$Z = \frac{T}{T_w} \ . \tag{1.14}$$

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

$$dq_d = D_o(C_1 T_w^{\ 4} - C_2) + L_c(h_t T_w + h_{at}) \ \ dL \tag{1.15}$$

where

$$L_c = \frac{\pi D_o}{2} - \delta_h \ . \tag{1.16}$$

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

$$L_e = \eta_a L_h \ . \tag{1.17}$$

This equation can also be expressed in terms of the dimensionless temperature gradient, $(dZ/d\omega)_1$, at the fin root, and

$$L_e = \frac{-L_h \left(\frac{dZ}{d\omega}\right)_1}{\zeta_p(1-C_3) + F_h - F_{af}} \tag{1.18}$$

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

Approximate values for the wall temperature, T_{ws}, and $(dZ/d\omega)_1$ 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 $(dZ/d\omega)_1$ 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

$$T_{fend} = T_{f1} - \mathcal{E}(T_{f1} - T_e). \tag{1.19}$$

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

$$T_{wf} = T_{fend} - 0.8(T_e - T_{fend}). \tag{1.20}$$

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

$$L_e = mT_w + b \tag{1.21}$$

where

$$m = \frac{L_{ef} - L_{es}}{T_{wf} - T_{ws}} \tag{1.22}$$

and

$$b = L_{es} - mT_{ws} \ . \tag{1.23}$$

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

$$L_e = T_w(T_w A_a + B_b) + C_c \tag{1.24}$$

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

$$L_e = T_w(T_w A_a + B_b) + C_c + mT_w + b \ . \tag{1.25}$$

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

$$N_{re} = \frac{D_i V \rho}{\mu} = \frac{D_i \dot{w}_d}{A_d \mu} \quad . \tag{1.26}$$

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

$$h = \frac{k\ N_{nu}}{D_i} \qquad \text{(1.27}$$

[Ref. 4, Eq. 7-30]

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

$$h = \frac{0.0384 \dot{w}_d C_p (N_{re})^{-1/4}}{A_d[1 + 1.5(N_{pr})^{-1/6}(N_{re})^{-1/8}(N_{pr} - 1)]} \tag{1.28}$$

[Ref. 4, Eq. 8.14]

where Prandtl number,

$$N_{pr} = \frac{C_p \mu}{k} \quad . \tag{1.29}$$

Both Prandtl and Reynolds numbers are evaluated at the reference temperature

$$T* = T_b - \frac{(.01\ N_{pr} + 40)(T_b - T_s)}{(N_{pr} + 72)} \quad . \tag{1.30}$$

[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

$$N_{re} = \frac{D_i V \rho}{\mu} = \frac{D_i \dot{w}_d}{A_d \mu} \quad . \tag{1.26}$$

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

$$h = \frac{k\ N_{nu}}{D_i} \tag{1.27}$$

[Ref. 4, Eq. 7-30]

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

$$h = \frac{0.0384 \dot{w}_d C_p (N_{re})^{-1/4}}{A_d [1 + 1.5 (N_{pr})^{-1/6} (N_{re})^{-1/8} (N_{pr} - 1)]} \tag{1.28}$$

[Ref. 4, Eq. 8.14]

where Prandtl number,

$$N_{pr} = \frac{C_p \mu}{k} \quad . \tag{1.29}$$

Both Prandtl and Reynolds numbers are evaluated at the reference temperature

$$T^* = T_b - \frac{(.01\ N_{pr} + 40)(T_b - T_s)}{(N_{pr} + 72)} \quad . \tag{1.30}$$

[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,

$$dq = hp(T_f - T_w)dL_w \ . \tag{1.31}$$

From the external surface to the environment,

$$dq = q_s dL_w \tag{1.32}$$

where

$$q_s = 2y[T_w(T_w A_a + B_b) + C_c + mT_w + B] + D_o(C_1 T_w^{\,4} - C_2) + L_c(h_t T_w - h_{at}). \tag{1.33}$$

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

$$F_w = q_s - hp(T_f - T_w) \tag{1.34}$$

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

$$dq = -\dot{w}_d \, C_p \, dT_f \tag{1.35}$$

Combining Eqs. (1.31) and (1.35)

$$dL_w = \frac{-\dot{w}_d \, C_p \, dT_f}{hp(T_f - T_w)} \; . \tag{1.36}$$

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

$$h_{av} = \frac{h_1 + h_2}{2} \tag{1.37}$$

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

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

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

$$\Delta T_m = \frac{(T_{f1} - T_{w1}) - (T_{f2} - T_{w2})}{\log_e \left[\dfrac{(T_{f1} - T_{w1})}{(T_{f2} - T_{w2})} \right]} \tag{1.39}$$

For a finite section length the fluid temperature change is

$$dT_f = T_{f2} - T_{f1} . \qquad (1.40)$$

with these modifications Eq. (136) becomes

$$L_w = \frac{\dot{w}_d\ C_p (T_{f1} - T_{f2})}{h_{av}\ p\ \Delta T_m} = \frac{q}{h_{av}\ p\ \Delta T_m} \qquad (1.41)$$

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

$$W_L = \Sigma \rho_{av}\ A_d\ L_w \qquad (1.42)$$

The duct weight is

$$W_d = \frac{\rho_d\ L_{ws}\ \pi}{4} (D_o^2 - D_i^2) \qquad (1.43)$$

The weight of a fin is

$$W_f = \frac{\rho_f\ L_{ws}\ L_h\ \delta_h}{2} (1 + \delta_r) \qquad (1.44)$$

and the total weight is

$$W_t = (W_L + W_d + 2W_f)N . \qquad (1.45)$$

The plan area of the system is,

$$A_p = (D_o + 2L_h)N\ L_{ws} . \qquad (1.46)$$

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

$$D_i = \frac{4A_d}{p} \quad . \tag{2.1}$$

For circular ducts

$$A_d = \frac{\pi D_i^2}{4} . \tag{2.2}$$

and

$$p = \pi D_i \quad . \tag{2.3}$$

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

$$dq(I) = \left\{S_r(I) \left[C_{1d}(I)\ T_w^4 - C_{2d}(I)\right] + S_c(I)\ h_d(I) \left[T_w - T_a(I)\right]\right\} dL. \tag{2.4}$$

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

$$dq_d = (G_{d1}\ T_w^4 + G_{d2}\ T_w - G_{d3})dL \tag{2.5}$$

where

$$G_{d1} = \sum_{I=1}^{n} S_r(I)\ C_{1d}(I) \tag{2.6}$$

$$G_{d2} = \sum_{I=1}^{n} S_c(I)\ h_d(I) \tag{2.7}$$

and

$$G_{d3} = \sum_{I=1}^{n} \left[S_r(I)\ C_{2d}(I) + S_c(I)\ h_d(I)\ T_a(I) \right] . \tag{2.8}$$

For an extended surface the heat transfer is

$$dq(L) = \left\{ C_1(L)\ T_w^{\ 4} - C_2(L) + T_w \left[h_a(L) + h_b(L) \right] - \left[h_a(L)\ T_{aa}(L) + \right.\right.$$

$$\left.\left. + h_b(L) T_{aa}(L) \right] \right\} L_e(L) dL. \tag{2.9}$$

Then, for

$$G_2(L) = h_a(L) + h_b(L) \tag{2.10}$$

and

$$G_3(L) = h_a(L)\ T_{aa}(L) + h_b(L)\ T_{ab}(L) + C_2(L), \tag{2.11}$$

$$dq(L) = \left[C_1(L)\ T_w^{\ 4} + G_2(L)\ T_w - G_3(L) \right] L_e(L)\ dL. \tag{2.12}$$

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 $T_e = T_w$

$$F_e = C_1(1)\ T_e^{\ 4} + G_2(1)\ T_e - G_3(1) = 0 \tag{2.13}$$

$$dF_e / dT_e = 4\ C_1(1) T_e^{\ 3} + G_2(1) \tag{2.14}$$

and

$$T_e = T_e' - F_e \Big/ \frac{dF_e}{dT_e} . \tag{2.15}$$

The outlet fluid temperatures T_{f2} 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

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

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

$$dq = q_s dL. \tag{2.20}$$

Where

$$q_s = K_0 + K_1 T_w + K_2 T_w^2 + K_4 T_w^4 + K_5 T_w^5. \tag{2.21}$$

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

$$K_1 = G_{d2} + \sum_{L=1}^{n'} \left[G_2(L)\ b(L) - G_3(L) \right] m(L) \tag{2.23}$$

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

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

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

and

n' = 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

$$F_w = q_s - hp(T_f - T_w)\ . \tag{2.27}$$

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

$$\Delta T = T_{W1} - T_{W2} = (T_{W1} - T_{Wend})/FMESH \ . \tag{2.28}$$

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 envolving 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 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

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2. Mackay, D. B., "Digital Programs for Establishing Steady-State Radiator Performance," ASD Technical Report 63-222, October 1963.

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4. Eckert, E. R. G., and Drake, R. M., Heat and Mass Transfer, McGraw-Hill Book Company, Inc., New York, New York, 1959.

5. Berezin, I. S., and Zhidkov, N. P. (translated by Blunn, D. M., and Booth, A. D.) Computing Methods, Vol. 11. Pergamon Press 1965 (U.S.A. edition by the Addison-Wesley Company, Inc.).

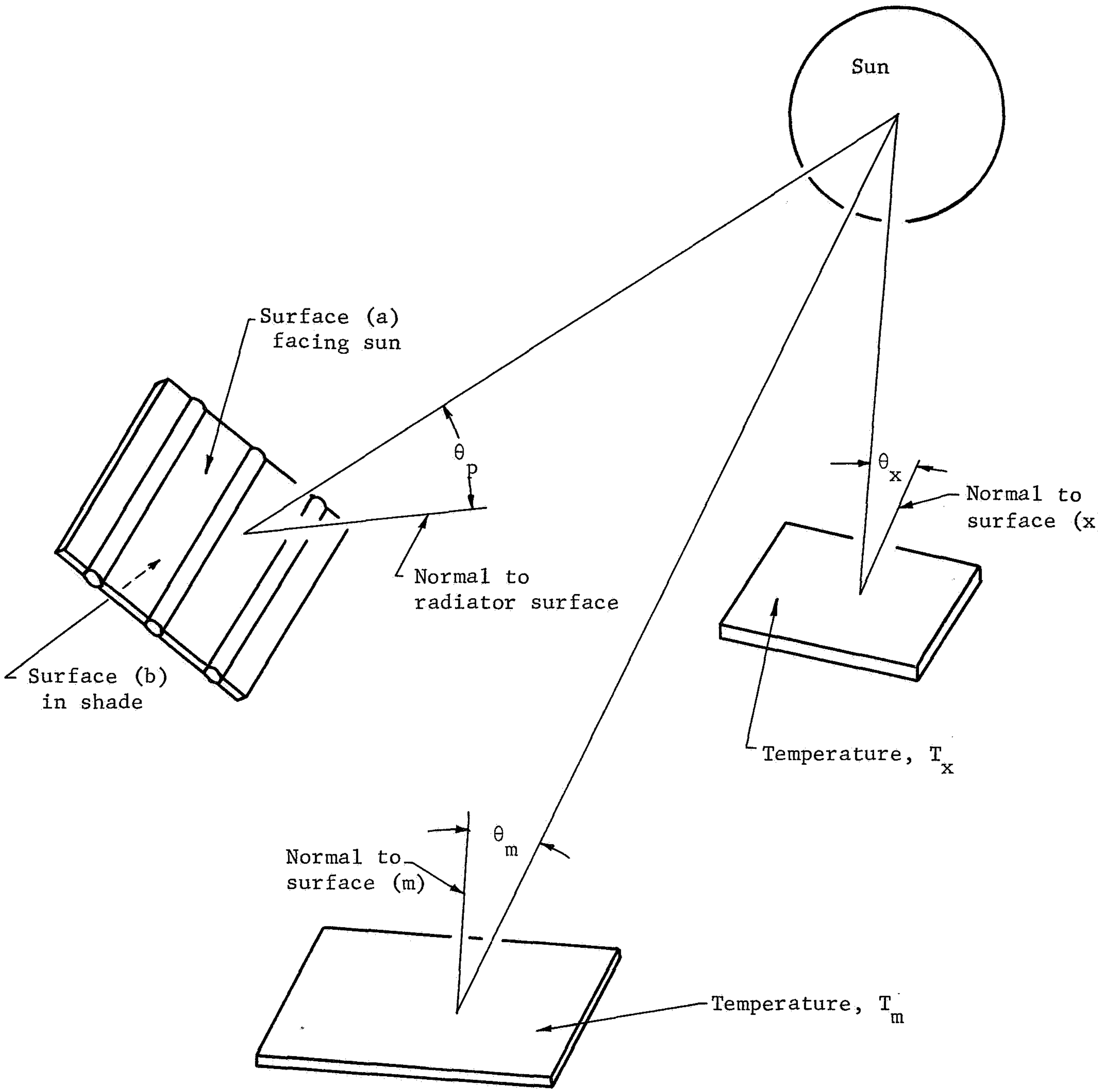
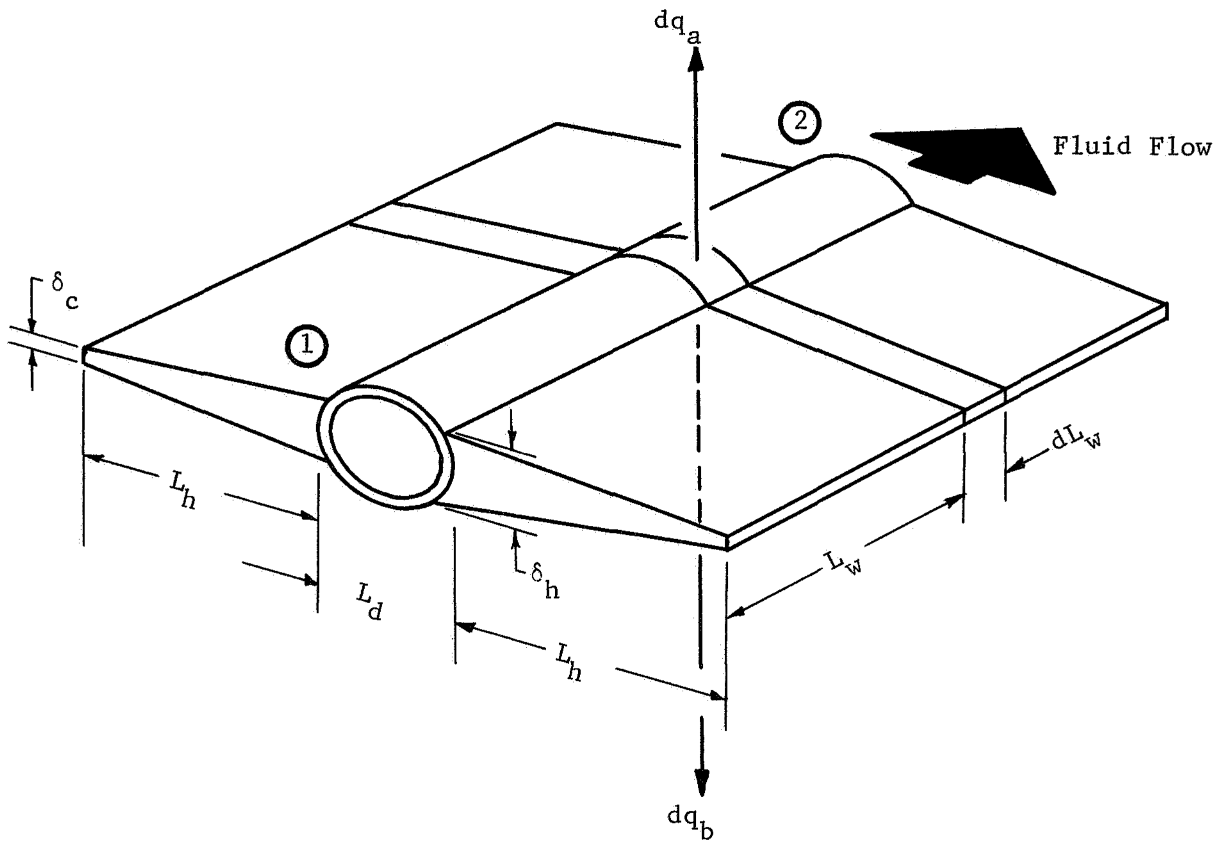


Fig. 1. Program 4-1 Heat Exchanger Radiative Environment



1 Entrance
2 Exit

Fig. 2. Program 4-1 Heat Exchanger Configuration

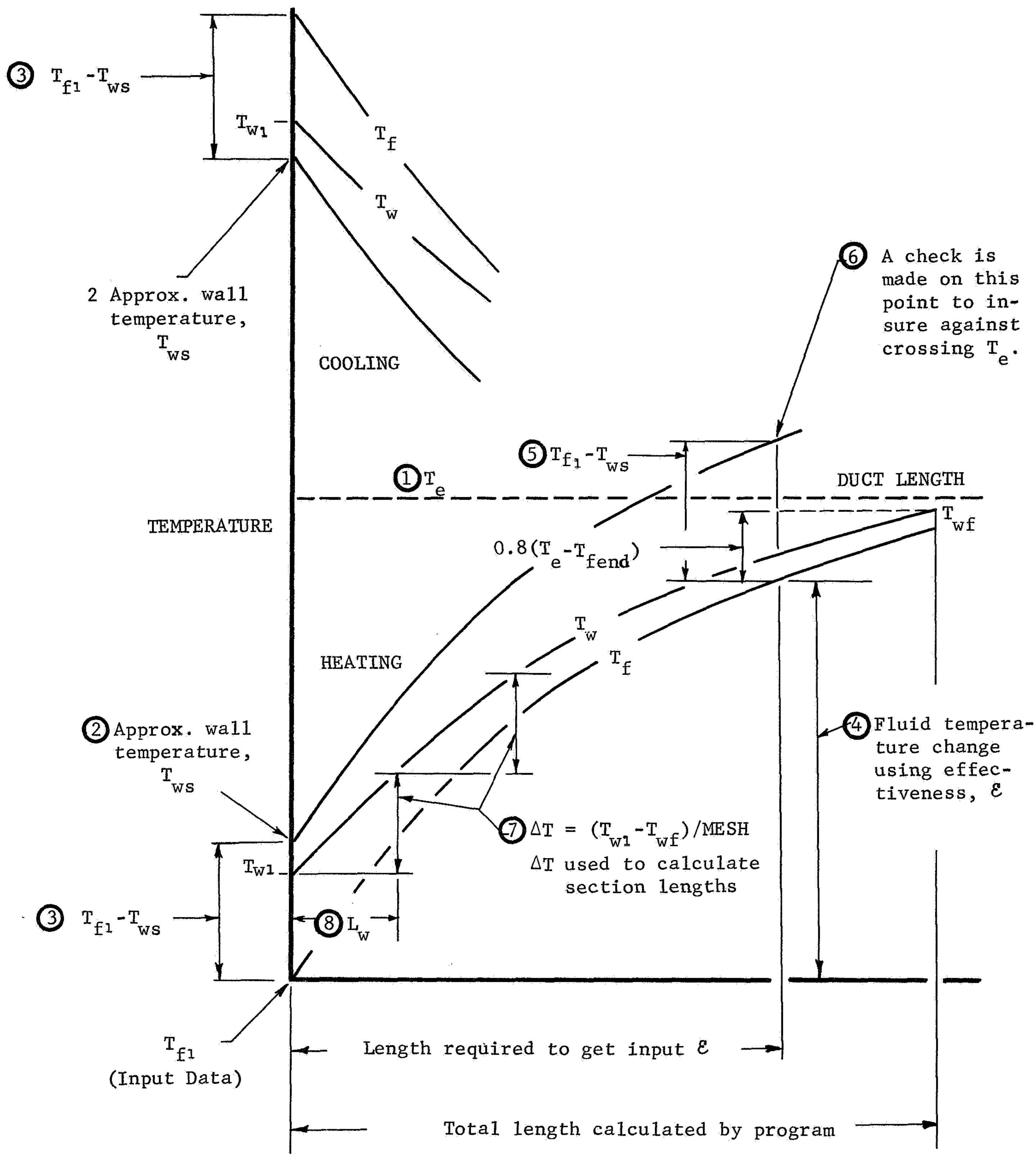


Fig. 3. Program 4-1 Calculation Sequence

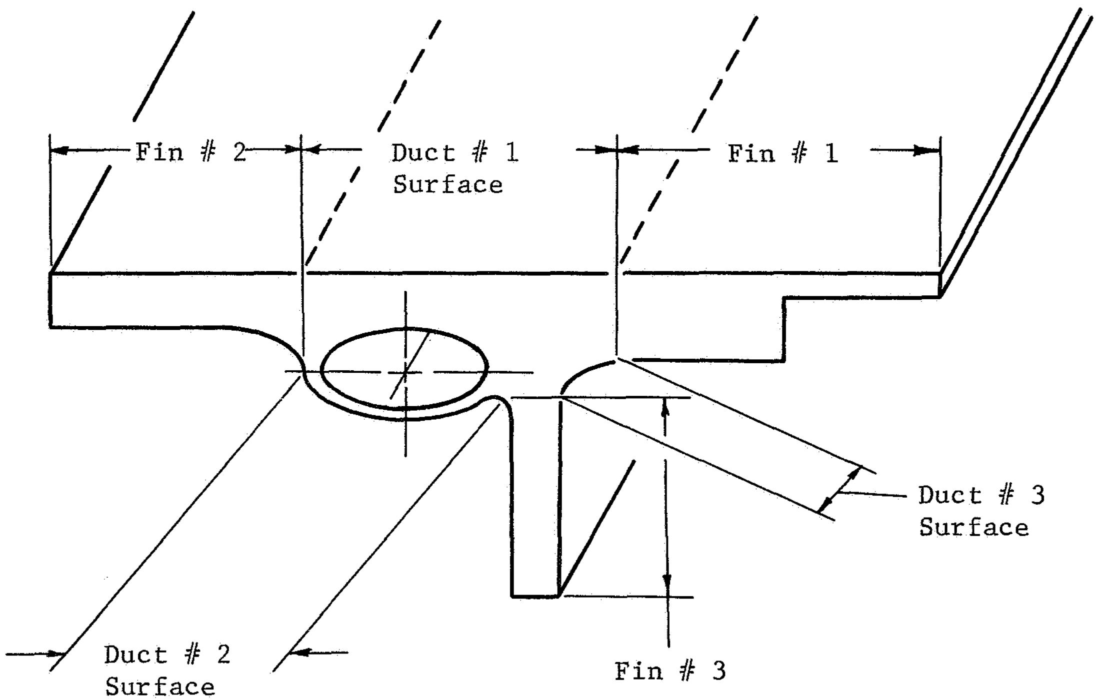


Fig. 4. Program 4-2 Representative Configuration

APPENDIX A

PROGRAM INFORMATION

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.

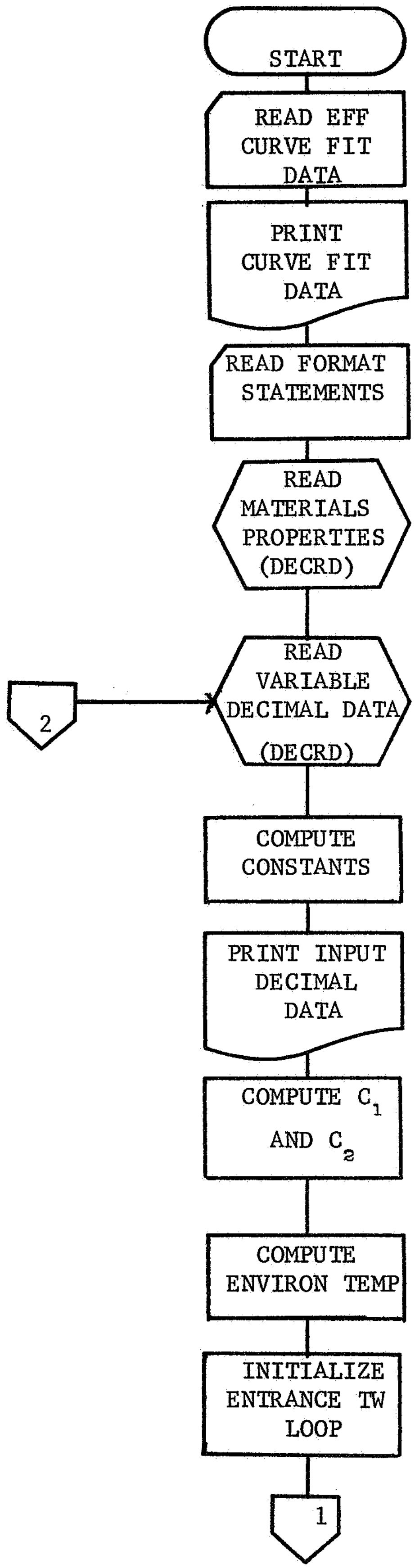


Fig. A-1 Program 4-1 Flow Diagram

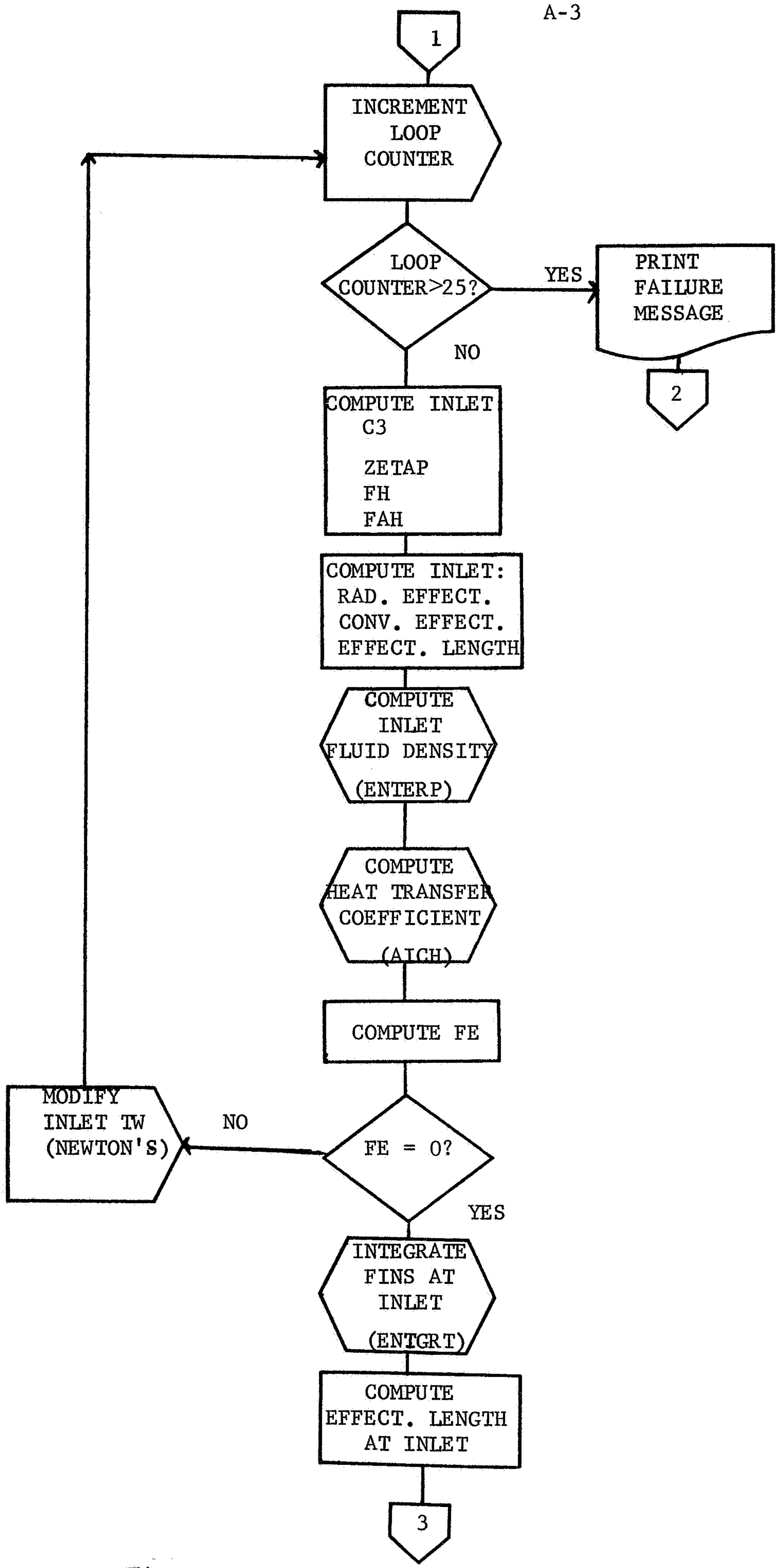


Fig. A-1 Program 4-1 Flow Diagram (cont.)

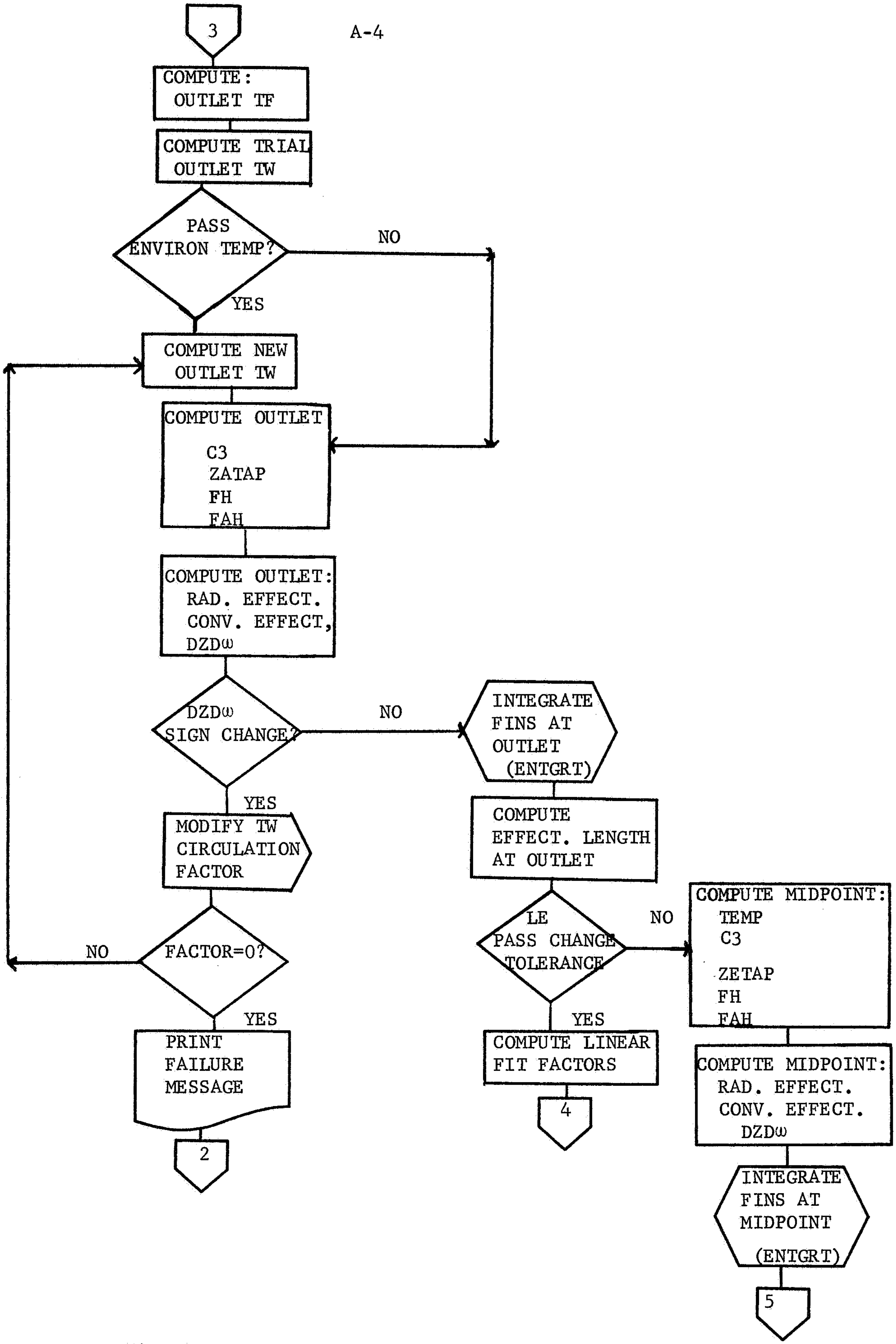
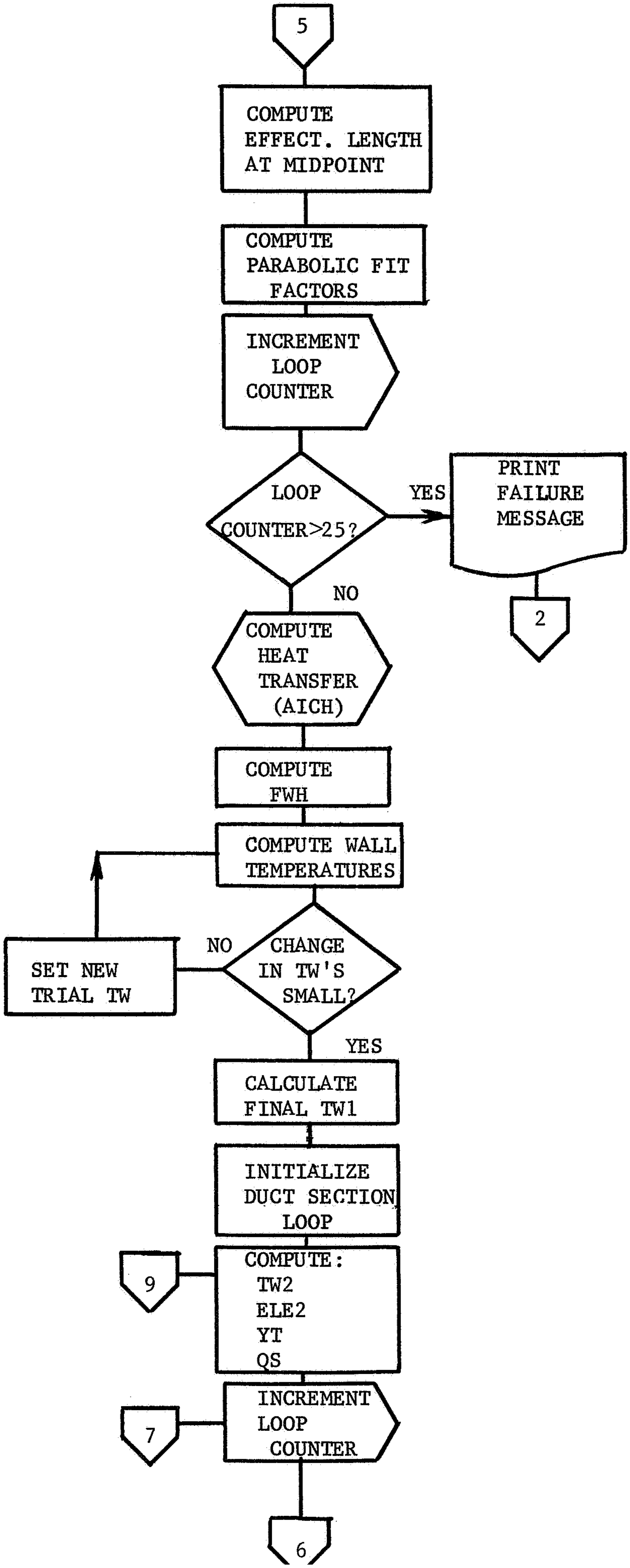
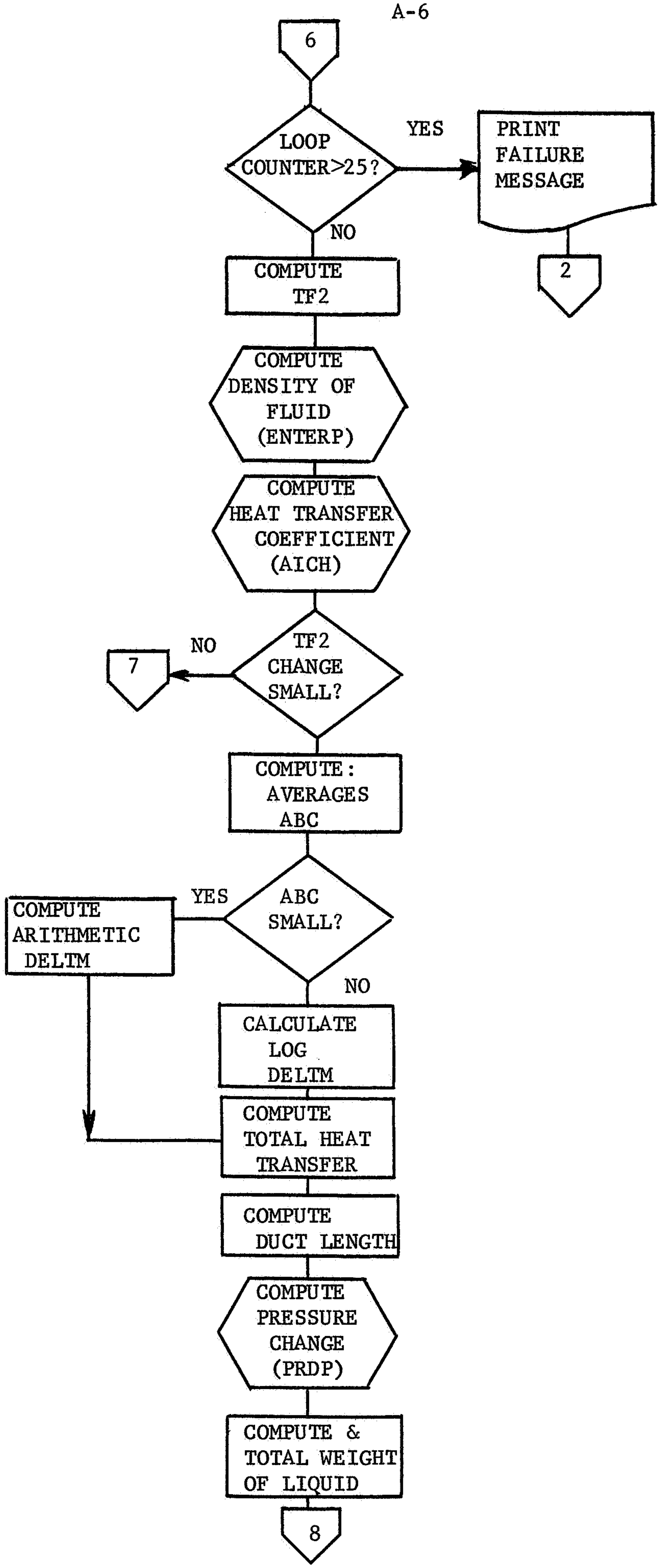


Fig. A-1 Program 4-1 Flow Diagram (cont.)


5
COMPUTE EFFECT. LENGTH AT MIDPOINT
COMPUTE PARABOLIC FIT FACTORS
INCREMENT LOOP COUNTER
LOOP COUNTER>25?
YES
PRINT FAILURE MESSAGE
2
NO
COMPUTE HEAT TRANSFER (AICH)
COMPUTE FWH
COMPUTE WALL TEMPERATURES
CHANGE IN TW'S SMALL?
NO
SET NEW TRIAL TW
YES
CALCULATE FINAL TW1
INITIALIZE DUCT SECTION LOOP
9
COMPUTE: TW2 ELE2 YT QS
7
INCREMENT LOOP COUNTER
6


6
LOOP COUNTER>25?
YES
PRINT FAILURE MESSAGE
2
NO
COMPUTE TF2
COMPUTE DENSITY OF FLUID (ENTERP)
COMPUTE HEAT TRANSFER COEFFICIENT (AICH)
TF2 CHANGE SMALL?
NO
7
COMPUTE: AVERAGES ABC
ABC SMALL?
YES
COMPUTE ARITHMETIC DELTM
NO
CALCULATE LOG DELTM
COMPUTE TOTAL HEAT TRANSFER
COMPUTE DUCT LENGTH
COMPUTE PRESSURE CHANGE (PRDP)
COMPUTE & TOTAL WEIGHT OF LIQUID
8

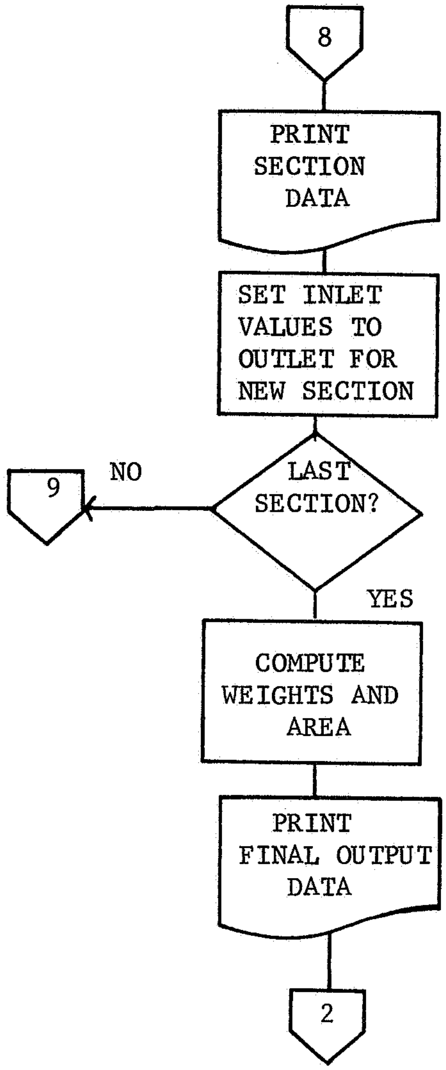
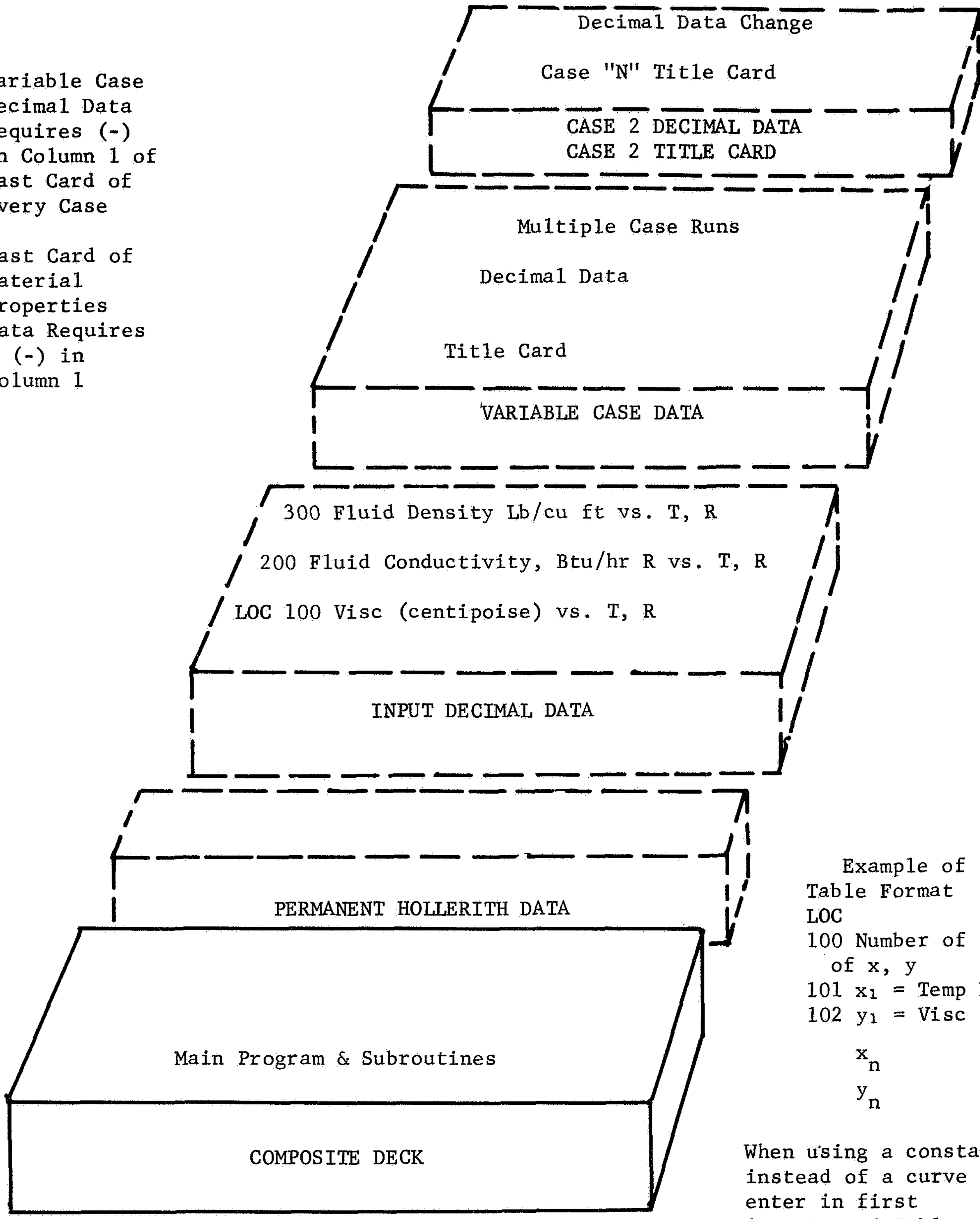


Fig. A-1 Program 4-1 Flow Diagram (cont.)

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



Example of
Table Format
LOC
100 Number of Pairs
of x, y
101 x_1 = Temp R
102 y_1 = Visc

x_n

y_n

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),DO),(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),FB),(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,WDOTD,AD,RHO,REAV,RHOAV,ELW,
     V VAV,DP,FH,FAH,ZETAP,C3,DZ1,DELTAR,RE2,TSTAR
C
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 3E12.5/1H 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)                         /3X
     V 90HLOC 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(3XI5,F13.6,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*DI**2/4.
      WDOTD=WDOT/ENT
      DELTAR=FINTC/FINTH
      HT=HA+HB
      HAT=HA*TAA+HB*TAB
      ELC=PI*DO/2.-FINTH
      TF1=DA(13)
      AA=0.
      BB=0.
      CC=0.
      FM=0.
```

Fig. A-3 Program 4-1 Listing

```
      BP=0.
      C3MID=0.
      ZETAPM=0.
      FHMID=0.
      FAHMID=0.
      ELEMID=0.
      PRINT 3002,TITLE
 3002 FORMAT (1H1,16A5///)
      PRINT F1,(J,DA(J), J=1,41)
      C1=.1713E-8*(EPSA+EPSB)
      C2= SC *(ALPHAA*COS(THETAP*.01745329)+FA*ALPHAA*RHOM*COS(THETAM*
     V.01745329)+FAX*ALPHAA*RHOX*COS(THETAX*.01745329)+FB*ALPHAB*RHOM
     V*COS(THETAM*.01745329)+FBX*ALPHAB*RHOX*COS(THETAX*.01745329))+
     VEPSM*TM**4*.1713E-8*(FA*EPSA+FB*EPSB)+EPSX*TX**4*.1713E-8*(FAX*
     VEPSA+FBX*EPSB)+.01*(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-TAA)+HB*(TE-TAB)
      IF(ABS(FE)-.001)109,109,108
  108 DFEDTE=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,ZETAP1,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*(ZETAP1*A(NA)+B(NA))+C(NA))
      EFC1=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*PERI
      TW1=TW1-FE/DFEDTW
      GO TO 115
C       COMPUTE EFFECTIVE LENGTH
  150 T(5)=DZDW1
      CALL ENTGRT
      DZDW1=DZ1
      ELE1=(-ELH*DZDW1)/(ZETAP1*(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))162,161,161
  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.) 181,180,180
  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(((DZDW1+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=FHEND
      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)
```

Fig. A-3 Program Listing (cont.)

```
      ZETAPM=C1*TWMID**3*ELH**2/(FNK*FINTH)
      FHMID=(ELH**2*HT)/(FNK*FINTH)
      FAHMID=(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-FAHMID)*EFCMID
      T(5)=DZDWM
      ZETAP=ZETAPM
      C3=C3MID
      FH=FHMID
      FAH=FAHMID
      CALL ENTGRT
      DZDWM=DZ1
      ELEMID=(-ELH*DZDWM)/(ZETAPM*(1.-C3MID)+FHMID-FAHMID)
      ZZ1=(ELE1-ELEMID)/(TW1-TWMID)
      ZZ2=(ELEMID-ELEEND)/(TWMID-TWEND)
      AA=(ZZ1-ZZ2)/(TW1-TWEND)
      BB=ZZ1-AA*(TW1+TWMID)
      CC=ELE1-AA*TW1**2-BB*TW1
C     COMPUTE ENTRANCE WALL TERMPERATURE
  325 IF ((TF1-TE).LT.0.0) 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
  323 PRINT 6000,TW1,TS
 6000 FORMAT (//3X5HTW1 =E15.8/3X6HTS   =E15.8/3X17HTW1 NOT CONVERGED )
      GO TO 80
  322 TS=TW(1)
      CALL AICH
      H=ACH
      FWH=2.*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
      FG(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(//3X7HTW(1) = E15.8/3X7HTW(2) = E15.8/3X7HTW(3) = E15.8/
     V 3X7HTW(4) = E15.8/3X5HFWH = E15.8/3X7HFG(1) = E15.8/3X7HFG(2) =
     V E15.8/3X7HFG(3) = E15.8/3X21HTWALL CONVERGE FAILED )
      STOP
   25 IF (ABS(TW(3)-TW(4))-.5) 30,30,27
   27 TW(1)=TW(4)
```

Fig. A-3 Program Listing (cont.)

```
      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)
      LC4=0
335   LC4=LC4+1
      IF(LC4-25)337,337,336
336   PRINT F8,TF2,RE2P,RE2
      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-TW2))
402   Q=WDOTD*CP*(TF1-TF2)
      ELW=Q/(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.)

```
      RHO1=RHO2
      ELE1=ELE2
  500 CONTINUE
      WD=(PI*RHOTM*ELWSUM*(DO**2-DI**2))/4.
      WF=(RHOFM*ELH*ELWSUM*FINTH*(1.+DELTAR))/2.
      WT=(WL+WD+2.*WF)*ENT
      AP=(DO+2.*ELH)*ELWSUM*ENT
      PRINT F5,C31,C3MID,C3END,ZETAP1,ZETAPM,ZETAPE,FH1,FHMID,FHEND,
     VFAH1,FAHMID,FAHEND,
     VELE1,ELEMID,ELEEND,WLSUM,WD,WF,WT,TE,ELWSUM,DPSUM,QSUM,AP,C1,C2
      GO TO 80
      END
```

Note: See following page for permanent Hollerith Listing.

Fig. A-3 Program 4-1 Listing (cont.)

```
 .41199999  .00533150   .00002362
-.87831999 -.09625650  -.00406435
 1.00000000 .6246049    .23347166
(11H INPUT DATA//I9,F15.5,22H OUTSIDE DIAMETER (FT)/I9,F15.5,21H INSIDE
DIAMETER (FT)/I9,F15.5,13H NO. OF TUBES/I9,F15.5,21H WEIGHT FLOW (LBS/HR
)/I9,F15.5,16H FIN LENGTH (FT)/I9,F15.5,27H FIN THICKNESS AT ROOT (FT)/
I9,F15.5,31H FIN THICKNESS AT FAR EDGE (FT)/I9,F15.5,36H DENSITY OF FIN
MATERIAL (LBS/CU FT)/I9,F15.5,37H DENSITY OF TUBE MATERIAL (LBS/CU FT)/
I9,F15.5,9H NOT USED /I9,F15.5,32H THERM COND OF FIN (BTU/FT HR R)/I9,F1
5.5,32H SPECIFIC HEAT OF FLD (BTU/LB R)/I9,F15.5,  27H FLUID TEMP AT ENT

RANCE (R)/I9,F15.5,52H HEAT TRANSFER COEFFICIENT SIDE A (HA, BTU/HR SQ F
T)/I9,F15.5,52H HEAT TRANSFER COEFFICIENT SIDE B (HB, BTU/HR SQ FT)/I9,
F15.5,24H AMBIENT TEMP SIDE A (R)/I9,F15.5,24H AMBIENT TEMP SIDE B (R)/
I9,F15.5,7H ALPHAA/I9,F15.5,7H ALPHAB/I9,F15.5,5H EPSA/I9,F15.5,5H EPSB/
I9,F15.5,5H EPSX/I9,F15.5,3H FA/I9,F15.5,4H FAX/I9,F15.5,3H FB/I9,F15.5,
4H FBX/I9,F15.5,5H RHOM/I9,F15.5,5H RHOX/I9,F15.5,13H THETAP (DEG)/I9,
F15.5,13H THETAM (DEG)/I9,F15.5,13H THETAX (DEG)/I9,F15.5,7H TM (R)/
I9,F15.5,7H TX (R)/I9,F15.5,5H EPSM/I9,F15.5,16H ITERATION LIMIT/I9,
F15.5,25H NO. OF INTEGRATION STEPS/I9,F15.5,30H SOLAR CONSTANT (BTU/HR S
Q FT)/I9,F15.5,29H HEAT EXCHANGER EFFECTIVENESS/I9,F15.5,19H NO. OF SUBS
ECTIONS/I9,F15.5,21H PRESSURE (LBS/SQ FT)/I9,F15.5,12H NUSSELT NO.)
(/34H0ENVIRON TEMP CONVERG. FAILED. TE=E12.5)
(41H0INITIAL INLET WALL TEMP CONVERG. FAILED./5H TW1=E12.5,7H ZETAP=
E12.5,4H C3=E12.5,4H FH=E12.5,5H FAH=E12.5)
(12H1SECTION NO.I3//6H0INLET,12X,7H OUTLET/E12.5,6X,E12.5,15H FLUID TEMP
 (R)/E12.5,6X,E12.5,14H WALL TEMP (R)/E12.5,6X,E12.5,23H FIN EFFECT LENG
TH (FT) /E12.5,6X,E12.5,43H HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R) /
/                                     /18X,E12.5,47H HEAT TRANSFER COEFFI
CIENT AVG (BTU/HR FT SQ R)/18X,E12.5,17H REYNOLDS NO. AVG/18X,E12.5,22H
VELOCITY AVG (FT/SEC)/18X,E12.5,20H SECTION LENGTH (FT)/18X,E12.5,28H PR
ESSURE CHANGE (LBS/SQ FT)/18X,E12.5,28H HEAT TRANSFER (BTU/HR TUBE)/18X,
E12.5,18H FLUID DENSITY AVG/18X,E12.5,19H WT OF LIQUID (LBS))
(18H1FINAL OUTPUT DATA//6H0INLET 12X,9H MIDPOINT, 9X,7H OUTLET/E12.5,6X,
E12.5,6X,E12.5,19H ENVIRON PARAM (C3)/E12.5,6X,E12.5,6X,E12.5,12H PROFIL
E NO./E12.5,6X,E12.5,6X,E12.5,22H CONVECTIVE PARAM (FH)/E12.5,6X,E12.5,
6X,E12.5,23H CONVECTIVE PARAM (FAH)/E12.5,6X,E12.5,6X,E12.5,23H FIN EFFE
CT LENGTH (FT)//
18X,E12.5,23H TOT WT OF LIQUID (LBS)/18X,E12.5,21H WEIGHT OF DUCT (LBS)/
18X,E12.5,21H WEIGHT OF FINS (LBS)/18X,E12.5,19H TOTAL WEIGHT (LBS)/
18X,E12.5,17H ENVIRON TEMP (R)/18X,E12.5,18H TOTAL LENGTH (FT)/18X,E12.5
,32H TOTAL PRESSURE DROP (LBS/SQ FT)/18X,E12.5,29H TOTAL HEAT TRANSFER (
BTU/HR)/18X,E12.5,18H PLAN AREA (SQ FT)/
18X,E12.5,24H RADIATION CONSTANT (C1)/18X,E12.5,24H RADIATION CONSTANT (
C2))
(44H0FINAL INLET WALL TEMP CONVERG. FAILED. TW1=E12.5)
(/38H0INTEGRATION CONVERGENCE FAILED. DZ1A=E12.5,5H DZ1=E12.5)
(/36H FLUID TEMP CONVERGENCE FAILED. TF2=E12.5,5X,5HRE2P=E12.5,5X,5HRE2=
  E12.5/)
(/42H0HEAT EXCHANGER EFFECTIVENESS IS TOO LARGE)
```

Note: This Hollerith Listing follows Subroutine Listings. See Fig. A-2.

Fig. A-3 Program 4-1 Listing (cont.)

```
      SUBROUTINE AICH
      EQUIVALENCE (DA(1),DO),(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,FH,FAH,ZETAP,C3,DZ1,DELTAR,RE2,TSTAR
      V=WDOTD/(RHO*AD*3600.)
      VISC=ENTERP(TB,DA(100))*2.4190297
      REN=WDOTD*DI/AD
      RE=REN/VISC
      IF (RE-2000.) 410,410,500
  410 FLK=ENTERP(TB,DA(200))
      ACH=CKH*FLK/DI
      RETURN
  500 VISC=ENTERP(TS,DA(100))*2.4190297
      FLK=ENTERP(TS,DA(200))
      PRN=CP*VISC/FLK
      TSTAR=TB-((.1*PRN+40.)*(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**(-.16667)*
     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
C
      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-5 Subroutine PRDP

```
      SUBROUTINE ENTGRT
      DIMENSION CM(5),FO(5)
      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
      D2Z(T2,T4,T5)=(T5*(1.-DELTAR)+ZETAP*(T4**4-C3)+FH*T4-FAH)/
     V(1.-T2*(1.-DELTAR))
C
      ITER = 0
      ITLT=DA(35)
      IS0=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(38H0INTEGRATION CONVERGENCE FAILED. DZ1A=E12.5,5H DZ1=
     V  E12.5)
      RETURN
 1130 DO 1390   J=1,MESH
      FO(2)=0.
      FO(3)=T(3)/2.
      FO(4)=T(3)/2.
      FO(5)=T(3)
      CM(1)=0.
      DO 2  I=2,5
      CM(I)=D2Z(T(2)+FO(I),T(4)+FO(I)*T(5),T(5)+FO(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 DZ0=DZ1
      T(5)=1.10*DZ1
      IF(ISW)1210,1239,1210
 1210 IF((ABS(DZW)-ABS(DZ0))/ABS(T(5))-.0025)1400,1400,1239
 1239 IS0=1
      IF(ITLT-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(DZ0))/ABS(T(5))-.0025)1400,1400,1270
 1270 ISW=1

      IF(ITLT-ITER) 1125,1125,1050
 1380 WILT = T(5)
 1390 CONTINUE
      GO TO 1260
 1400 RETURN
      END
```

Fig. A-6 Subroutine ENTGRT (cont.)

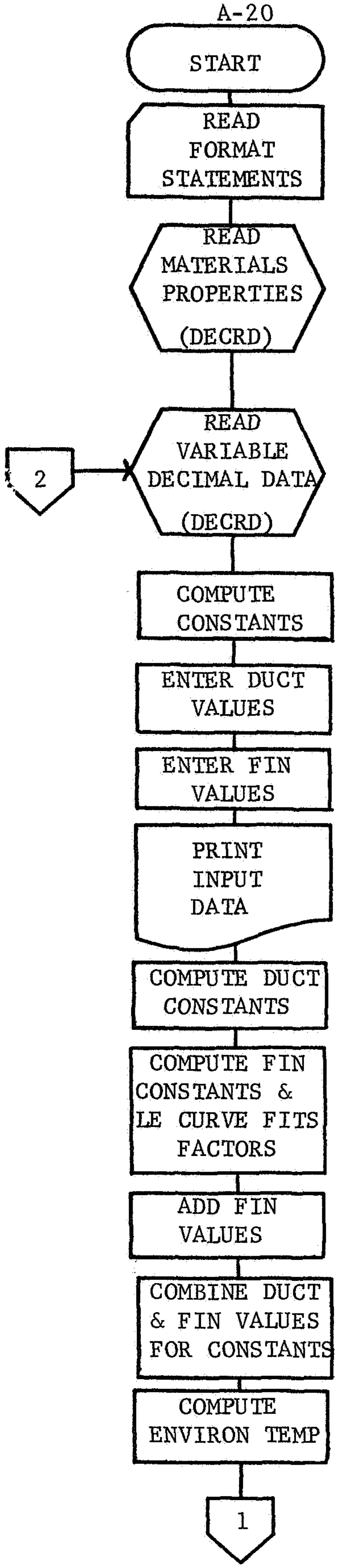


Fig. A-7 Program 4-2 Flow Diagram

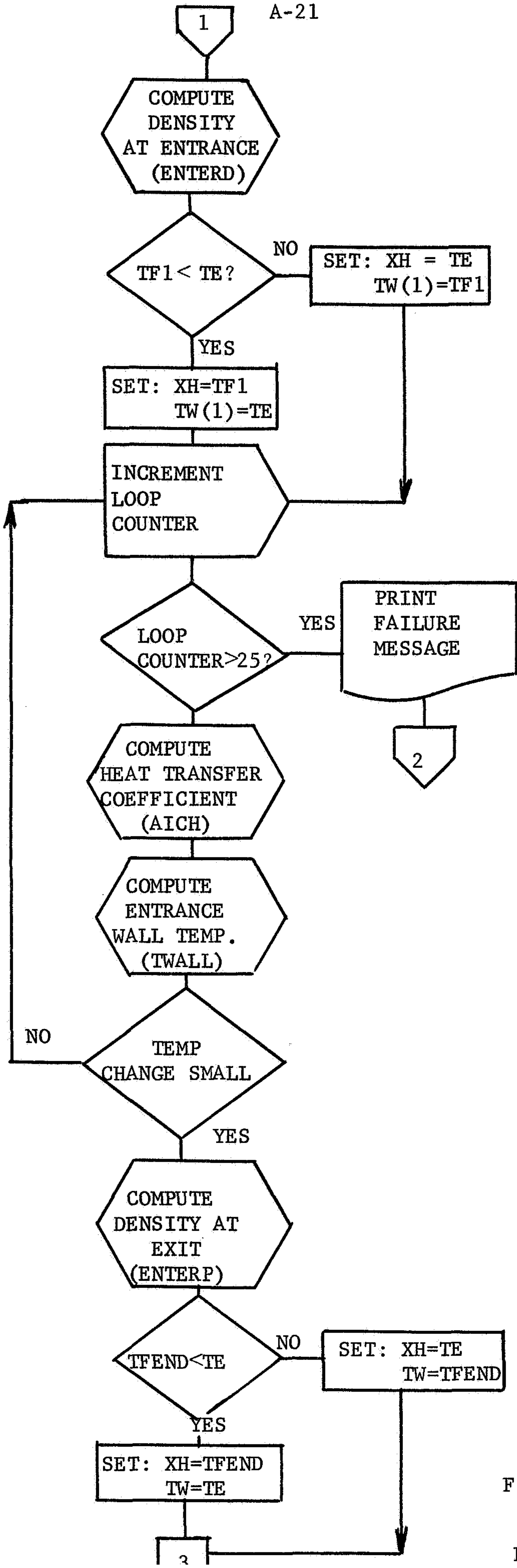


Fig. A-7 Program 4-2

Flow Diagram (cont.)

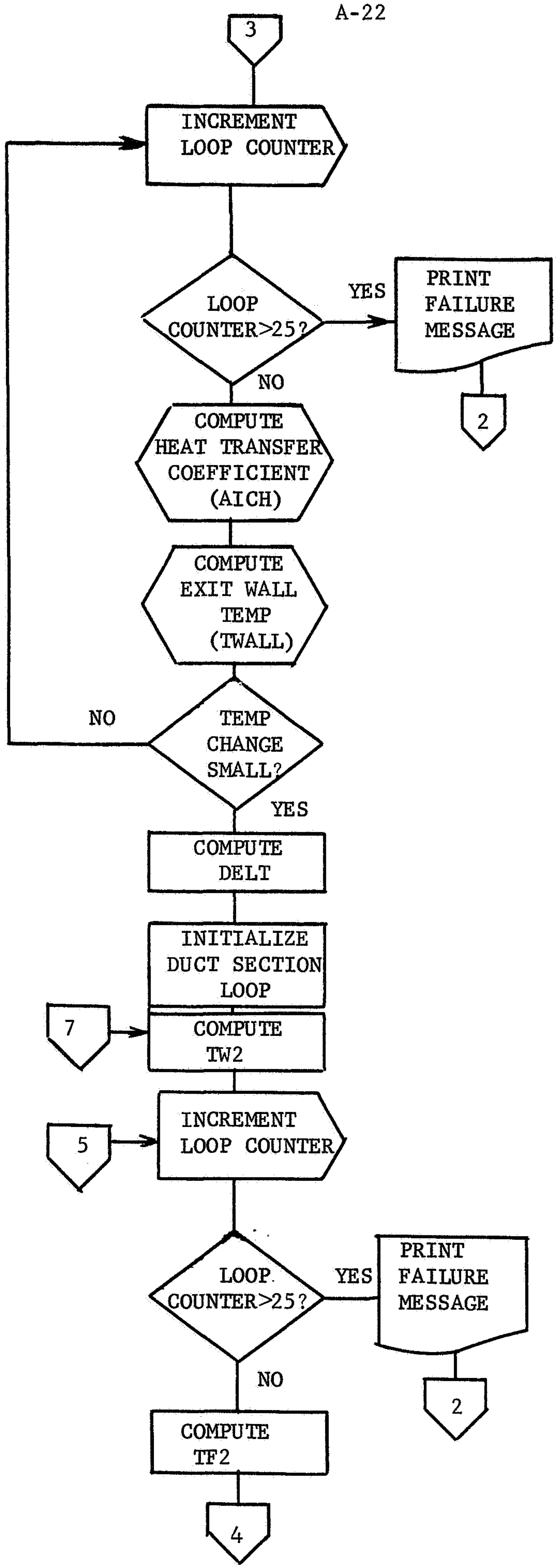


Fig. A-7 Program 4-2 Flow Diagram (cont.)

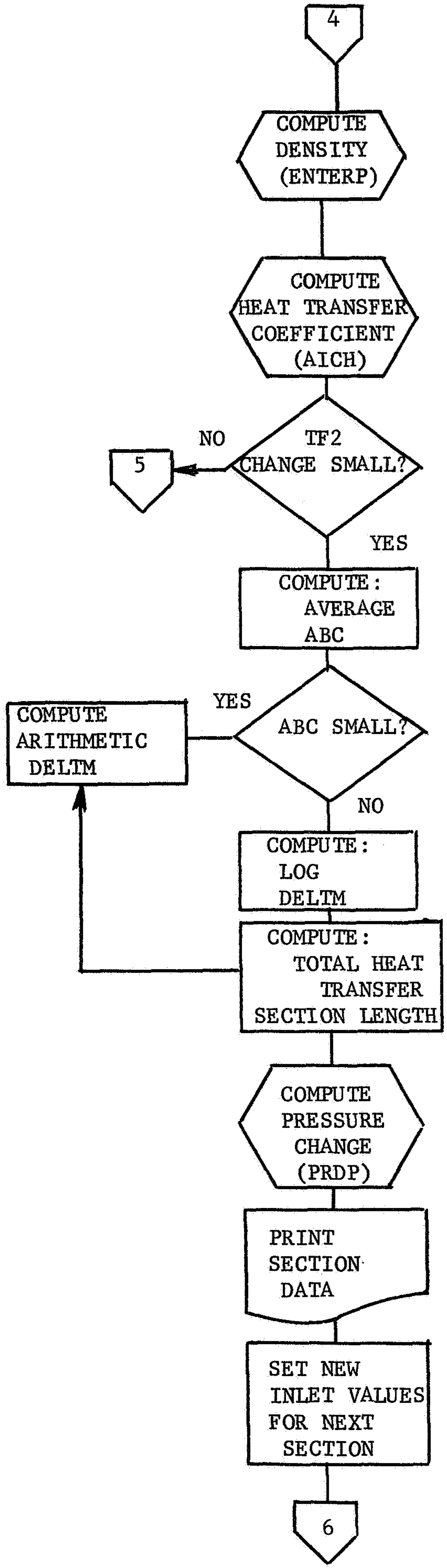


Fig. A-7 Program 4-2 Flow Daigram (cont.)

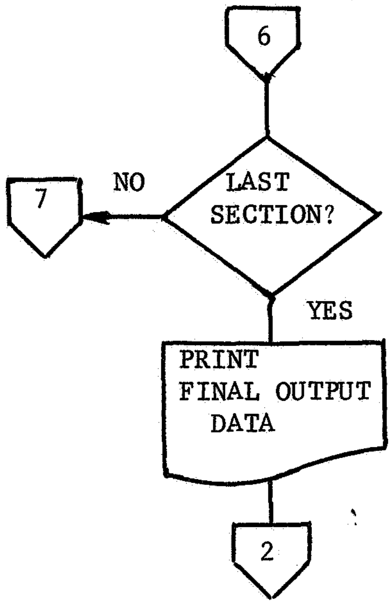
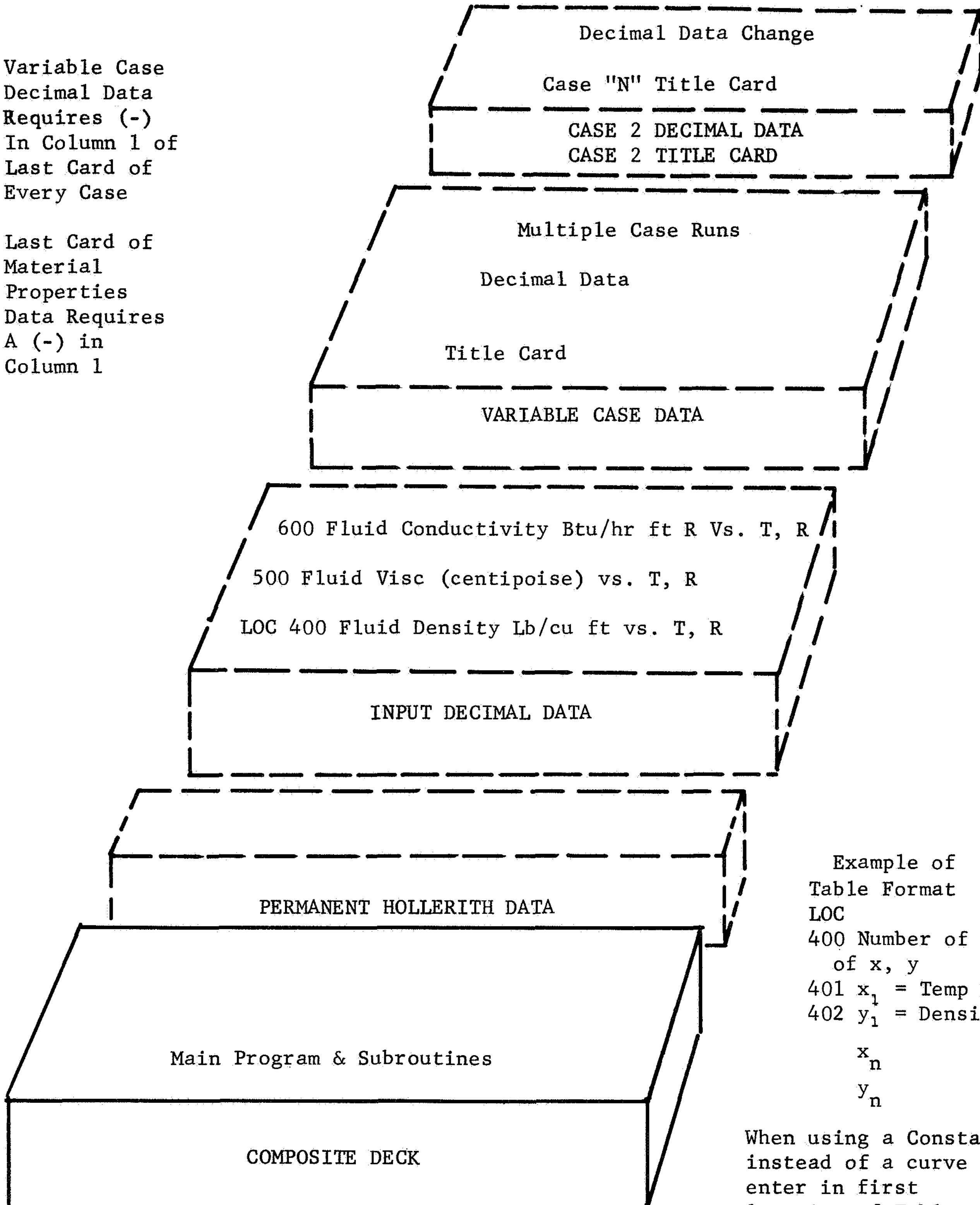


Fig. A-7 Program 4-2 Flow Diagram (cont.)

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



Example of
Table Format
LOC
400 Number of Pairs
of x, y
401 x_1 = Temp R
402 y_1 = Density

x_n

y_n

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

```
C        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(12),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),TWS(10),G2(10),G3(10),EM(10),B(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,TB,TS,V,DP,PI,DI
      COMMON TWAL,TW    ,CK0,CK1,CK2,CK4,CK5,H,PERIM,XH,RE2,TSTAR
C       READ FORMAT STATEMENTS
      READ 1,F1,F2,F3,F4,F5,F6,F7,F8,F9,F10,F11,F12,F13,F14,F15,F16,
     VF17,F18,F19,F20,F21,F22,F23,F24
    1 FORMAT (12A6)
C       READ MATERIALS PROPERTIES
      CALL DECRD (DA)
C     PRINT FLUID PROPERTIES
      K=400
      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 K,BTU/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,I8,F13.6,I8,F13.6,I8,F13.6)
      K=K+100
 2900 CONTINUE
C
    5 READ 2400,TITLE
 2400 FORMAT (16A5)
      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 WDOTD=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,ND)
      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)+HB(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+G2(L)*EM(L)
      SUM4=SUM4+C1(L)*B(L)
  500 SUM5=SUM5+C1(L)*EM(L)
      CK0=-(GD3+SUM0)
      CK1=GD2+SUM1
      CK2=SUM2
      CK4=GD1+SUM4
      CK5=SUM5
C     COMPUTE ENVIRONMENT TEMP
      LC=0
  600 TE=TF1-1.2*(TF1-TFEND)
  640 LC=LC+1
      IF (LC-25) 1001,1001,651
  651 PRINT 652,TE
  652 FORMAT(/32H INITILIZATION OF TE FAILED. TE=E12.8)
      GO TO 5
 1001 IF(ELE(1).EQ.0.0) GO TO 1020
      FE=C1(1)*TE**4+G2(1)*TE-G3(1)
      IF(ABS(FE)-.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 -C2D(1))+SC(1)*HD(1)*(TE-TA(1))
      IF(ABS(FE)-.001)1003,1003,1021
 1021 DFEDTE=4.0*SR(1)*C1D(1)*TE **3+SC(1)*HD(1)
 1030 TE=TE-FE/DFEDTE
 1041 GO TO 640
C     COMPUTE ENTRANCE WALL TEMP
 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(//3X5HTW1 =E15.8/3X6HTW1P =E15.8/3X17HTW1 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 TWENDP=TWAL
      CALL TWALL
      LC3=LC3+1
      IF(LC3.LT.25) GO TO 7000
      PRINT 8000,TWEND,TWENDP
 8000 FORMAT(//3X7HTWEND =E15.8/3X7HTWENDP=E15.8//3X19HTWEND NOT CONVERG
     VED)
      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=ACH1
 1070 LC4=LC4+1
      IF(LC4-25)1075,1075,1074
 1074 PRINT F22,TF2P,TF2,RE2P,RE2
      GO TO 5
 1075 TF2P=TF2
      RE2P=RE2
      TF2=TW2+(1./(H*PERIM))*(CK0+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)1080,1080,1070
 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 PRDP
      DPSUM=DPSUM+DP
      WL=RHOAV*AD*ELW
      WLSUM=WLSUM+WL
      PRINT F23,I,TF1,TF2,TW1,TW2,ACH1,ACH2,RE1,RE2,VAV,ELW,Q,HAV,REAV,
     V RHOAV,WL,DP
      TF1=TF2
      TW1=TW2
      V1=V2
      RE1=RE2
      RHO1=RHO2
      ACH1=ACH2
 2000 CONTINUE
      PRINT F24,DPSUM,QSUM,WLSUM,TE,ELWSUM
      GO TO 5
      END
```

Note: See following page for permanent Hollerith Listing.

Fig. A-9 Program 4-2 Listing (cont)

```
(28H ENVIRON TEMP CONVERG FAILED/4H TE=E12.8,5H CK0=E12.8,5H CK1=E12.8,
5H CK2=E12.8,5H CK4=E12.8,5H CK5=E12.8)
(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)
(30H EXIT WALL TEMP CONVERG FAILED/7H TWEND=E12.8,5H CK0=E12.8,5H CK1=
E12.8,5H CK2=E12.8,5H CK4=E12.8,5H CK5=E12.8)
(11H INPUT DATA/I9,F15.8,21H NO. OF DUCT SECTIONS/I9,F15.8,12H NO. OF FI
NS/I9,F15.8,28H DUCT PERIMETER (FT) *OPTION/I9,F15.8,26H DUCT AREA (SQ F
T) *OPTION/I9,F15.8,32H EFFECTIVE DIAMETER (FT) *OPTION/I9,F15.8,9H NOT
USED/I9,F15.8,22H OUTSIDE DIAMETER (FT)/I9,F15.8,26H TOTAL WEIGHT FLOW (
LB/HR)/I9,F15.8,20H WALL THICKNESS (FT)/I9,F15.8,13H NO. OF DUCTS/I9,
F15.8,34H SPECIFIC HEAT OF FLUID (BTU/LB R)/I9,F15.8,27H FLUID TEMP AT E
NTRANCE (R)/I9,F15.8,23H FLUID TEMP AT EXIT (R)/I9,F15.8,19H NO. OF SUBS
ECTIONS/I9,F15.8,13H NUSSELTS NO.)
(//18H INPUT DUCT VALUES/    4H SEC5X,43H EFFECT PERIFERAL LENGTH FOR RA
DIATION (FT)/(I3,7X,F15.8))
(     4H SEC5X,44H EFFECT PERIFERAL LENGTH FOR CONVECTION (FT)/(I3,7X,
F15.8))
( 4H SEC5X,42H RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)/(I3,7X,E15.8))
(     4H SEC5X,37H RADIATION CONSTANT C2 (BTU/HR SQ FT)/(I3,7X,F15.8))
(     4H SEC5X,48H CONVECTIVE HEAT TRANSFER COEFF (BTU/HR SQ FT R)/(I3,7X
,F15.8))
(     4H SEC5X,17H AMBIENT TEMP (R)/(I3,7X,F15.8))
(17H1INPUT FIN VALUES/    4H FIN5X,42H RADIATION CONSTANT C1 (BTU/HR SQ
FT R**4)/(I3,7X,E15.8))
(     4H FIN5X,37H RADIATION CONSTANT C2 (BTU/HR SQ FT)/(I3,7X,F15.8))
(     4H FIN5X,55H CONVECTIVE HEAT TRANSFER COEFF SIDE A (BTU/HR SQ FT R)
/(I3,7X,F15.8))
(     4H FIN5X,55H CONVECTIVE HEAT TRANSFER COEFF SIDE B (BTU/HR SQ FT R)
/(I3,7X,F15.8))
(     4H FIN5X,24H AMBIENT TEMP SIDE A (R)/(I3,7X,F15.8))
(     4H FIN5X,24H AMBIENT TEMP SIDE B (R)/(I3,7X,F15.8))
(4H FIN5X,10H NOT USED  /(I3))
(     4H FIN5X,27H EFFECT LENGTH AT EXIT (FT)/(I3,7X,F15.8))
(     4H FIN5X,31H EFFECT LENGTH AT ENTRANCE (FT)/(I3,7X,F15.8))
(     4H FIN5X,27H DUCT WALL TEMP AT EXIT (R)/(I3,7X,F15.8))
(     4H FIN5X,31H DUCT WALL TEMP AT ENTRANCE (R)/(I3,7X,F15.8))
(40H0OUTLET FLUID TEMP CONVERG FAILED. TF2P=E12.8,5H TF2=E12.8,6H RE2P=,
E12.8,5H RE2= E12.8)
(8H1SECTION I3 //6H INLET14X,7H OUTLET/F15.8,5X,F15.8,15H FLUID TEMP (R)
/F15.8,5X,F15.8,14H WALL TEMP (R)/F15.8,5X,F15.8,36H HEAT TRANSFER COEFF
. (BTU/HR SQ FT)  /F15.8,5X,F15.8,13H REYNOLDS NO. //20X,F15.8,22H VELOC
ITY AVG (FT/SEC)  /
20X,F15.8,20H SECTION LENGTH (FT)/20X,F15.8,23H HEAT TRANSFER (BTU/HR)/
20X,F15.8,39H HEAT TRANSFER COEFF AVG (BTU/HR SQ FT)/20X,F15.8,
17H REYNOLDS NO. AVG/20X,F15.8,18H FLUID DENSITY AVG/20X,F15.8,
22H WEIGHT OF LIQUID (LB)/20X,F15.8,27H PRESSURE CHANGE (LB/SQ FT))
(18H1FINAL OUTPUT DATA/20X,F15.8,31H PRESSURE CHANGE SUM (LB/SQ FT)/
20X,F15.8,29H TOTAL HEAT TRANSFER (BTU/HR)/20X,F15.8,28H TOTAL WEIGHT OF
 LIQUID (LB)/20X,F15.8,17H ENVIRON TEMP (R)/20X,F15.8,18H TOTAL LENGTH (
FT))
```

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),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,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),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,TB,TS,V,DP,PI,DI
      COMMON TWAL,TW    ,CK0,CK1,CK2,CK4,CK5,H,PERIM,XH,RE2,TSTAR
      V=WDOTD/(RHO*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
```

Fig. A-11 Subroutine AICH for Program 4-2

```
      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),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),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,TB,TS,V,DP,PI,DI
      COMMON TWAL,TW     ,CK0,CK1,CK2,CK4,CK5,H,PERIM,XH
      FWH=CK0+CK1*XH+CK2*XH**2+CK4*XH**4+CK5*XH**5-H*PERIM*(TB-XH)
      L=0
   20 DO 10  K=2,4
      L=L+1
      FW(K-1)=CK0+CK1*TW(K-1)+CK2*TW(K-1)**2+CK4*TW(K-1)**4+CK5*
     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/
     V 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
    1 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

FORTRAN SOURCE LIST

```
ISN       SOURCE STATEMENT

  0 $IBFTC 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(I12, 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(SIGN(1.,DRBU(I))) 7,11,6
 23     11 CALL EXIT
 24        END
```

Fig. A-14   Subroutine DECRD

APPENDIX B

SAMPLE PROBLEMS

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

$$\mathcal{E} = \frac{T_{f1} - T_{f2}}{T_{f1} - T_e} = \frac{660 - 549.94}{660 - 534.36} = 0.87$$

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:

$$\epsilon_a = 0.6$$

$$\epsilon_x = 1.0$$

$$T_x = 1000 \text{ R}$$

$$F_x = 1.0$$

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

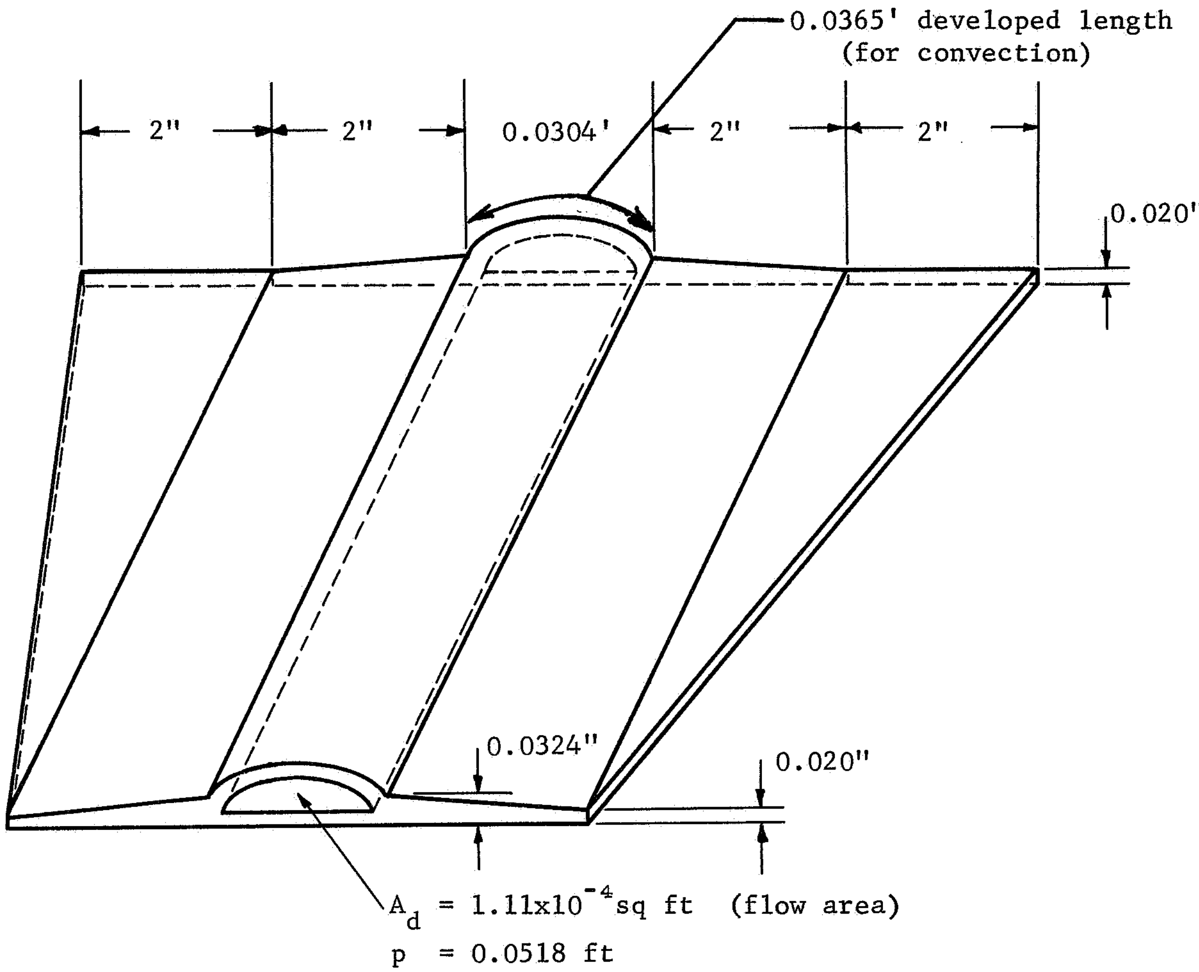
$$C_1 = \epsilon_a \sigma = (0.8)(0.1613)(10^{-8}) = 0.137(10^{-8})$$

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

One hundred degrees fluid temperature rise in passing through the duct was estimated for a first trail. 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



Fluid - 250 lb/hr - "Coolanol"

Material - Aluminum - k = 118 Btu/hr ft R

T_{f1} = 760 R

T_{f2} = 710 R

T_{w1} = 730 R (estimated)

T_{w2} = 680 R (estimated)

L_{e1} = 0.1559' (calculated by Program 2-1)

L_{e2} = 0.2697' (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.

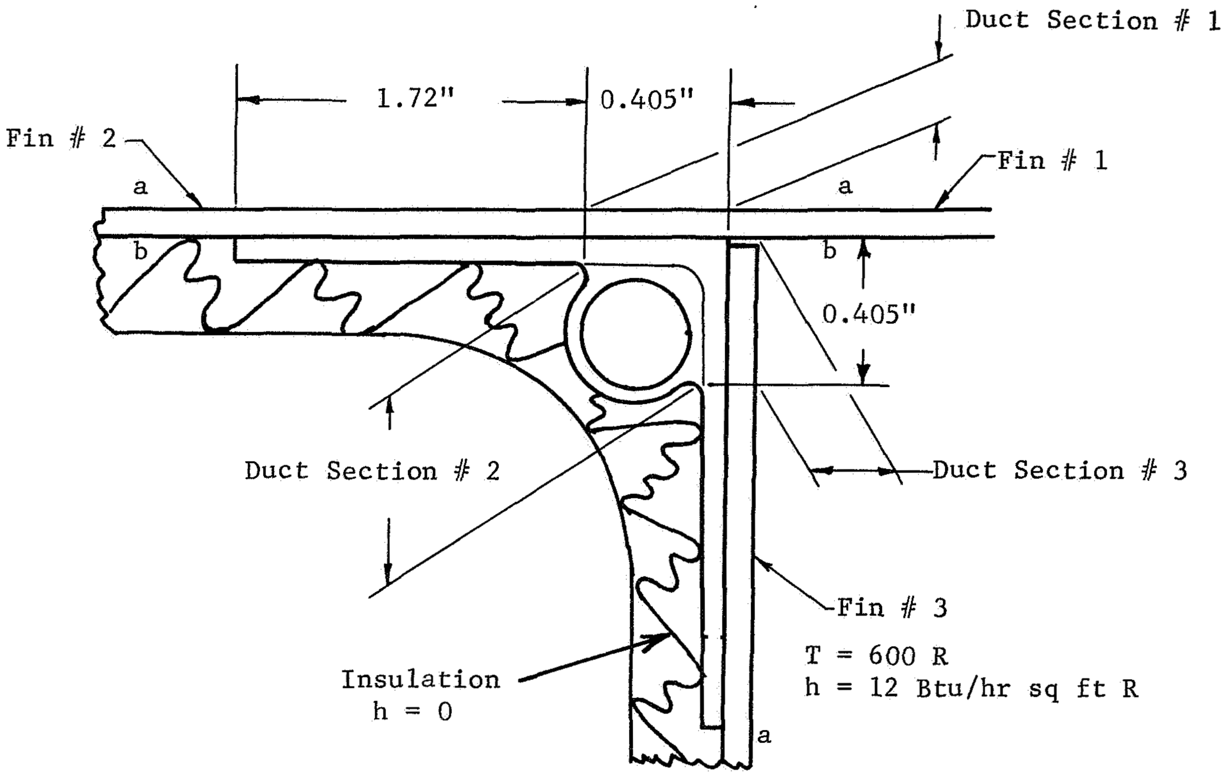
The calculated wall temperatures on the entrance and exit are not close to the estimated values. Some improvement in accuracy would result from rerunning the program using problem 3 data. Still better accuracy could be attained if the duct length were divided into a number of sections and the duct and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



<u>System</u>

T_{f1} = 800 R

T_{f2} = 650 R

$\dot{w}_d$ = 1000 lb/hr

<u>Fin # 1 and Fin # 2</u>

$T_m = T_a$ = 530 R

h_a = 7 Btu/hr sq ft R

$C_1 = 0.1456 \times 10^{-8}$

C_2 = 146.1 Btu/hr sq ft

<u>Sketch 2</u>

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

100	4.000000	101	492.000000	102	1.786000	103	660.000000	104	0.305000
105	860.000000	106	0.132300	107	1060.000000	108	0.088900		
200	4.000000	201	492.000000	202	0.337000	203	660.000000	204	0.393000
205	860.000000	206	0.382000	207	1060.000000	208	0.293000		
300	4.000000	301	492.000000	302	62.540000	303	660.000000	304	60.200000
305	860.000000	306	53.620000	307	1060.000000	308	42.370000		

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

400	2.000000	401	0.000000	402	69.000000	403	2000.000000	404	20.200000
500	13.000000	501	0.000000	502	1000.000000	503	410.000000	504	148.799999
505	435.000000	506	44.640000	507	450.000000	508	26.700000	509	475.000000
510	15.250000	511	500.000000	512	9.750000	513	510.000000	514	8.180000
515	525.000000	516	6.200000	517	550.000000	518	4.000000	519	560.000000
520	3.270000	521	600.000000	522	1.800000	523	660.000000	524	1.488000
525	2000.000000	526	0.005000						
600	2.000000	601	0.000000	602	0.103500	603	2000.000000	604	0.003500

"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

1	0.04167	OUTSIDE DIAMETER (FT)
2	0.03083	INSIDE DIAMETER (FT)
3	12.00000	NO. OF TUBES
4	250.00000	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 FLD (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	ALPHAA
19	0.00000	ALPHAB
20	0.85000	EPSA
21	0.85000	EPSB
22	0.00000	EPSX
23	1.00000	FA
24	0.00000	FAX
25	1.00000	FB
26	0.00000	FBX
27	0.00000	RHOM
28	0.00000	RHOX
29	0.00000	THETAP (DEG)
30	0.00000	THETAM (DEG)
31	0.00000	THETAX (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	NUSSELT NO.

EFF CURVE FIT

```
 0.41200E 00 0.53315E-02 0.23620E-04
-0.87832E 00-0.96256E-01-0.40643E-02
 0.10000E 01 0.62460E 00 0.23347E 00
```

Fig. B-2 Input Data - Problem 1

```
SECTION NO.  1

INLET               OUTLET
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               OUTLET
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

INLET	OUTLET	
0.58741E 03	0.54994E 03	FLUID TEMP (R)
0.56479E 03	0.54320E 03	WALL TEMP (R)
0.10838E 00	0.10865E 00	FIN EFFECT LENGTH (FT)
0.52199E 02	0.50432E 02	HEAT TRANSFER COEFFICIENT (BTU/HR FT SQ R)

0.51315E 02	HEAT TRANSFER COEFFICIENT AVG (BTU/HR FT SQ R)
0.32763E 03	REYNOLDS NO. AVG
0.12609E 00	VELOCITY AVG (FT/SEC)
0.11969E 02	SECTION LENGTH (FT)
0.11518E 01	PRESSURE CHANGE (LBS/SQ FT)
0.78048E 03	HEAT TRANSFER (BTU/HR TUBE)
0.61472E 02	FLUID DENSITY AVG
0.54937E 00	WT OF LIQUID (LBS)

FINAL OUTPUT DATA

INLET	MIDPOINT	OUTLET	
0.58505E 00	0.	0.90636E 00	ENVIRON PARAM (C3)
0.50953E-01	0.	0.36694E-01	PROFILE NO.
0.94339E 00	0.	0.94339E 00	CONVECTIVE PARAM (FH)
0.83283E 00	0.	0.92915E 00	CONVECTIVE PARAM (FAH)
0.10865E 00	0.	0.10865E 00	FIN EFFECT LENGTH (FT)

0.92176E 00	TOT WT OF LIQUID (LBS)
0.60993E 01	WEIGHT OF DUCT (LBS)
0.14992E 02	WEIGHT OF FINS (LBS)
0.43959E 03	TOTAL WEIGHT (LBS)
0.53436E 03	ENVIRON TEMP (R)
0.20178E 02	TOTAL LENGTH (FT)
0.16065E 01	TOTAL PRESSURE DROP (LBS/SQ FT)
0.27514E 05	TOTAL HEAT TRANSFER (BTU/HR)
0.80694E 02	PLAN AREA (SQ FT)
0.29121E-08	RADIATION CONSTANT (C1)
0.22980E 03	RADIATION CONSTANT (C2)

Fig. B-3 Output Data - Problem 1 (cont)

FXAMPLE 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 DATA

INPUT DUCT VALUES

SFC EFFFCT PERIFERAL LENGTH FOR RADIATION (FT) 20
1 0.09817000
SEC 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 1028.00000000
SEC CONVECTIVE HEAT TRANSFER COEFF (BTU/HR SQ FT R) 24
1 1.30000000
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

INLET               OUTLET
  560.00000000        611.32028961 FLUID TEMP (R)
  572.25449371        619.60701752 WALL TEMP (R)
  177.45729256        239.59266663 HEAT TRANSFER COEFF. (BTU/HR SQ FTR)
 2193.12750244       3964.59719849 REYNOLDS NO.

                        4.36191505 VELOCITY AVG (FT/SEC)
                       69.34133530 SECTION LENGTH (FT)
                    -8981.05065918 HEAT TRANSFER (BTU/HR)
                      208.52497864 HEAT TRANSFER COEFF AVG (BTU/HR SQ FTR)
                     3078.86233521 REYNOLDS NO. AVG
                       54.70989227 FLUID DENSITY AVG
                        1.10410248 WEIGHT OF LIQUID (LB)
                     1873.22721863 PRESSURE CHANGE (LB/SQ FT)

SECTION   2

INLET               OUTLET
  611.32028961        659.99997711 FLUID TEMP (R)
  619.60701752        666.95954132 WALL TEMP (R)
  239.59266663        254.78985596 HEAT TRANSFER COEFF. (BTU/HR SQ FTR)
 3964.59719849       4603.47137451 REYNOLDS NO.

                        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 FTR)
                     4284.03424072 REYNOLDS NO. AVG
                       53.48989296 FLUID DENSITY AVG
                        1.16375335 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           = 0.21430382E 00
     ZETAP        = 0.49383479E-01
     FINAL Z      = 0.97934200E 00
     FINAL DZ1    =-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

```
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) = 0.68000000E 03


     SECTION NUMBER =     1


     12 THICKNESS ROOT EDGE     =  0.32400000E-01  (INCHES)
     13 THICKNESS FAR EDGE      =  0.20000000E-01  (INCHES)
     14 FIN LENGTH              =  0.20000000E 01  (INCHES)
     15 THERMAL CONDUCTIVITY    =  0.11800000E 03  (BTU/HR-FT-DEG(R))

     29 C1 (ENTERED AS DATA)    =  0.14560000E-08
     30 C2 (ENTERED AS DATA)    =  0.88609999E 02


INPUT DATA FOR CONVECTION
     31    0.0000000 HEAT TRANSFER COEFFICIENT SIDE A
     32    0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
     33    0.0000000 AMBIENT TEMP A (DEG(R))
     34    0.0000000 AMBIENT TEMP B (DEG(R))

     SECTION NUMBER =     2


     42 THICKNESS ROOT EDGE     =  0.20000000E-01  (INCHES)
     43 THICKNESS FAR EDGE      =  0.20000000E-01  (INCHES)
     44 FIN LENGTH              =  0.20000000E 01  (INCHES)
     45 THERMAL CONDUCTIVITY    =  0.11800000E 03  (BTU/HR-FT-DEG(R))

     59 C1 (ENTERED AS DATA)    =  0.14560000E-08
     60 C2 (ENTERED AS DATA)    =  0.88609999E 02


INPUT DATA FOR CONVECTION
     61    0.0000000 HEAT TRANSFER COEFFICIENT SIDE A
     62    0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
     63    0.0000000 AMBIENT TEMP A (DEG(R))
     64    0.0000000 AMBIENT TEMP B (DEG(R))


      ***CONVERGENCE ACCOMPLISHED

           EFFECTIVE LENGTH =    3.235892 (INCHES)
```

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

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

```
EXAMPLE PROBLEM


INPUT DATA
      1         1.00000000 NO. OF DUCT SECTIONS
      2         2.00000000 NO. OF FINS
      3         0.05180000 DUCT PERIMETER (FT) *OPTION
      4         0.00011100 DUCT AREA (SQ FT) *OPTION
      5         0.00000000 EFFECTIVE DIAMETER (FT) *OPTION
      6         0.00000000 NOT USED
      7         0.00000000 OUTSIDE DIAMETER (FT)
      8        20.00000000 TOTAL WEIGHT FLOW (LB/HR)
      9         0.00000000 WALL THICKNESS (FT)
     10         1.00000000 NO. OF DUCTS
     11         0.70000000 SPECIFIC HEAT OF FLUID (BTU/LB R)
     12       760.00000000 FLUID TEMP AT ENTRANCE (R)
     13       710.00000000 FLUID TEMP AT EXIT (R)
     14         2.00000000 NO. OF SUBSECTIONS
     15         3.50000000 NUSSELTS NO.


INPUT DUCT VALUES                                                  INPUT DATA
SEC        EFFECT PERIFERAL LENGTH FOR RADIATION (FT)
 1              0.03040000                                                 20
SEC        EFFECT PERIFERAL LENGTH FOR CONVECTION (FT)
 1              0.03650000                                                 21
SEC        RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)
 1           0.14560000E-08                                                22
SEC        RADIATION CONSTANT C2 (BTU/HR SQ FT)
 1              88.60999966                                                23
SEC        CONVECTIVE HEAT TRANSFER COEFF (BTU/HR SQ FT R)
 1              0.00000000                                                 24
SEC        AMBIENT TEMP (R)
 1              0.00000000                                                 25
```

Fig. B-8 Input Data - Problem 3

```
INPUT FIN VALUES                                                  INPUT DATA
FIN      RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)
 1         0.14560000E-08                                                 80
 2         0.14560000E-08                                                100
FIN      RADIATION CONSTANT C2 (BTU/HR SQ FT)
 1            88.60999966                                                 81
 2            88.60999966                                                101
FIN      CONVECTIVE HEAT TRANSFER COEFF SIDE A (BTU/HR SQ FT R)
 1              0.00000000                                                82
 2              0.00000000                                               102
FIN      CONVECTIVE HEAT TRANSFER COEFF SIDE B (BTU/HR SQ FT R)
 1              0.00000000                                                83
 2              0.00000000                                               103
FIN      AMBIENT TEMP SIDE A (R)
 1              0.00000000                                                84
 2              0.00000000                                               104
FIN      AMBIENT TEMP SIDE B (R)
 1              0.00000000                                                85
 2              0.00000000                                               105
FIN      NOT USED
 1                                                                        86
 2                                                                       106
FIN      EFFFCT LENGTH AT EXIT (FT)
 1              0.26970000                                                87
 2              0.26970000                                               107
FIN      EFFFCT LENGTH AT ENTRANCE (FT)
 1              0.15590000                                                88
 2              0.15590000                                               108
FIN      DUCT WALL TEMP AT EXIT (R)
 1            680.00000000                                                89
 2            680.00000000                                               109
FIN      DUCT WALL TEMP AT ENTRANCE (R)
 1            730.00000000                                                90
 2            730.00000000                                               110
```

Fig. B-8 Input Data - Problem 3 (cont)

```
SECTION   1

INLET                 OUTLET
  760.00000000          736.14452362 FLUID TEMP (R)
  669.13888550          650.29351044 WALL TEMP (R)
   26.74583364           27.23288274 HEAT TRANSFER COEFF. (BTU/HR SQ FT R)
  463.53445435          454.81632614 REYNOLDS NO.

                          0.98629791 VELOCITY AVG (FT/SEC)
                          2.70441812 SECTION LENGTH (FT)
                        333.97666931 HEAT TRANSFER (BTU/HR)
                         26.98935819 HEAT TRANSFER COEFF AVG (BTU/HR SQ FT R)
                        459.17538834 REYNOLDS NO. AVG
                         50.74703693 FLUID DENSITY AVG
                          0.01523377 WEIGHT OF LIQUID (LB)
                         33.74176884 PRESSURE CHANGE (LB/SQ FT)


SECTION   2

INLET                 OUTLET
  736.14452362          710.00511932 FLUID TEMP (R)
  650.29351044          631.44813538 WALL TEMP (R)
   27.23288274           27.76656246 HEAT TRANSFER COEFF. (BTU/HR SQ FT R)
  454.81632614          445.63246536 REYNOLDS NO.

                          0.97458974 VELOCITY AVG (FT/SEC)
                          3.12721592 SECTION LENGTH (FT)
                        365.95166016 HEAT TRANSFER (BTU/HR)
                         27.49972248 HEAT TRANSFER COEFF AVG (BTU/HR SQ FT R)
                        450.22439575 REYNOLDS NO. AVG
                         51.35697412 FLUID DENSITY AVG
                          0.01782708 WEIGHT OF LIQUID (LB)
                         39.32037449 PRESSURE CHANGE (LB/SQ FT)


 FINAL OUTPUT DATA
                         73.06214333 PRESSURE CHANGE SUM (LB/SQ FT)
                        699.92832947 TOTAL HEAT TRANSFER (BTU/HR)
                          0.03306086 TOTAL WEIGHT OF LIQUID (LB)
                        496.68486404 ENVIRON TEMP (R)
                          5.83163404 TOTAL LENGTH (FT)
```

Fig. B-9 Output Data - Problem 3

```
EFFECTIVENESS FOR TRAPEZOIDAL PLATE FINS

     1      0.000000 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    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    530.000000 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
```

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

```
INPUT DATA FOR CONVECTION STUDY
    31      7.000000 HEAT TRANSFER COEFFICIENT SIDE A
    32     12.000000 HEAT TRANSFER COEFFICIENT SIDE B
    33    530.000000 AMBIENT TEMP A
    34    600.000000 AMBIENT TEMP B

              C1            = 0.14560500E-08
              C2            = 0.14605317E 03
              C3            = 0.30066325E 00
              ZETAP         = 0.14016908E 01
              FINAL Z       = 0.75187499E 00
              FINAL DZ1     =-0.16360223E 01
              FINAL OMEGA   = 0.67999998E 00
              FH            = 0.41666667E 02
              FAH           = 0.31480841E 02
              C2A           = 0.14605317E 03

          TEMP RATIO (TE/TH)  =   0.75474E 00
          EFF ENVIRON TEMP  =   0.57361E 03 (DEG(R))

          FINAL EFFECTIVENESS =   0.141188

          AREA EFFECTIVENESS=     0.146517

          EFFECTIVE LENGTH =     2.19776 (INCHES)
```

Fig. B-10   Fin No. 1 (at inlet) - Problem 4

SECTION NUMBER = 1

```
12 THICKNESS ROOT EDGE    =   0.190000  (INCHES)
13 THICKNESS FAR EDGE     =   0.190000  (INCHES)
14 FIN LENGTH             =   1.720000  (INCHES)
15 THERMAL CONDUCTIVITY   =  90.000000  (BTU/HR-FT-DEG(R))
16 FAX                    =   0.500000
17 ALPHA A                =   0.200000
18 EPS A                  =   0.850000
19 FBX                    =   0.000000
20 ALPHA B                =   0.000000
21 EPS B                  =   0.000000
22 TX                     = 530.000000  (DEGREES RANKINE)
23 THETA X                =  90.000000  (DEGREES)
24 RHO X                  =   0.100000
25 THETA P                =   0.000000  (DEGREES)
26 EPS X                  =   1.000000
```

INPUT DATA FOR CONVECTION

```
31    7.0000000 HEAT TRANSFER COEFFICIENT SIDE A
32    0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
33  530.0000000 AMBIENT TEMP A (DEG(R))
34    0.0000000 AMBIENT TEMP B (DEG(R))
```

COMPUTED VALUES OF C1,C2,C3

```
C1 = 0.14560500E-08
C2 = 0.14605317E 03
C3 = 0.30066325E 00
```

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

SECTION NUMBER = 2

```
42 THICKNESS ROOT EDGE    =   0.095000  (INCHES)
43 THICKNESS FAR EDGE     =   0.095000  (INCHES)
44 FIN LENGTH             =  15.000000  (INCHES)
45 THERMAL CONDUCTIVITY   =  90.000000  (BTU/HR-FT-DEG(R))
46 FAX                    =   0.500000
47 ALPHA A                =   0.200000
48 EPS A                  =   0.850000
49 FBX                    =   0.000000
50 ALPHA B                =   0.000000
51 EPS B                  =   0.000000
52 TX                     = 530.000000  (DEGREES RANKINE)
53 THETA X                =  90.000000  (DEGREES)
54 RHO X                  =   0.100000
55 THETA P                =   0.000000  (DEGREES)
56 EPS X                  =   1.000000
```

INPUT DATA FOR CONVECTION

```
61    7.0000000 HEAT TRANSFER COEFFICIENT SIDE A
62    0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
63  530.0000000 AMBIENT TEMP A (DEG(R))
64    0.0000000 AMBIENT TEMP B (DEG(R))
```

EFFECTIVE LENGTH = 4.033377 (INCHES)

Fig. B-11 Fin No. 2 (at inlet) - Problem 4

```
SECTION NUMBER =     1

12 THICKNESS ROOT EDGE     =    0.190000  (INCHES)
13 THICKNESS FAR EDGE      =    0.190000  (INCHES)
14 FIN LENGTH              =    1.720000  (INCHES)
15 THERMAL CONDUCTIVITY    =   90.000000  (BTU/HR-FT-DEG(R))
16 FAX                     =    0.000000
17 ALPHA A                 =    0.000000
18 EPS A                   =    0.000000
19 FBX                     =    0.000000
20 ALPHA B                 =    0.000000
21 EPS B                   =    0.000000
22 TX                      =    0.000000  (DEGREES RANKINE)
23 THETA X                 =    0.000000  (DEGREES)
24 RHO X                   =    0.000000
25 THETA P                 =    0.000000  (DEGREES)
26 EPS X                   =    0.000000


INPUT DATA FOR CONVECTION
      31  12.0000000 HEAT TRANSFER COEFFICIENT SIDE A
      32   0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
      33 600.0000000 AMBIENT TEMP A (DEG(R))
      34   0.0000000 AMBIENT TEMP B (DEG(R))

SECTION NUMBER =     2

42 THICKNESS ROOT EDGE     =    0.095000  (INCHES)
43 THICKNESS FAR EDGE      =    0.095000  (INCHES)
44 FIN LENGTH              =   15.000000  (INCHES)
45 THERMAL CONDUCTIVITY    =   90.000000  (BTU/HR-FT-DEG(R))
46 FAX                     =    0.000000
47 ALPHA A                 =    0.000000
48 EPS A                   =    0.000000
49 FBX                     =    0.000000
50 ALPHA B                 =    0.000000
51 EPS B                   =    0.000000
52 TX                      =    0.000000  (DEGREES RANKINE)
53 THETA X                 =    0.000000  (DEGREES)
54 RHO X                   =    0.000000
55 THETA P                 =    0.000000  (DEGREES)
56 EPS X                   =    0.000000


INPUT DATA FOR CONVECTION
      61  12.0000000 HEAT TRANSFER COEFFICIENT SIDE A
      62   0.0000000 HEAT TRANSFER COEFFICIENT SIDE B
      63 600.0000000 AMBIENT TEMP A (DEG(R))
      64   0.0000000 AMBIENT TEMP B (DEG(R))

          EFFECTIVE LENGTH =    3.542568 (INCHES)
```

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

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

EXAMPLE PROBLEM .

INPUT DATA

1	3.00000000	NO. OF DUCT SECTIONS
2	3.00000000	NO. OF FINS
3	0.00000000	DUCT PERIMETER (FT) *OPTION
4	0.00000000	DUCT AREA (SQ FT) *OPTION
5	0.03750000	EFFECTIVE DIAMETER (FT) *OPTION
6	0.00000000	NOT USED
7	0.04200000	OUTSIDE DIAMETER (FT)
8	1000.00000000	TOTAL WEIGHT FLOW (LB/HR)
9	0.00000000	WALL THICKNESS (FT)
10	1.00000000	NO. OF DUCTS
11	0.70000000	SPECIFIC HEAT OF FLUID (BTU/LB R)
12	800.00000000	FLUID TEMP AT ENTRANCE (R)
13	650.00000000	FLUID TEMP AT EXIT (R)
14	3.00000000	NO. OF SUBSECTIONS
15	4.36400002	NUSSELTS NO.

INPUT DUCT VALUES

SEC		INPUT DATA
SEC	EFFECT PERIFERAL LENGTH FOR RADIATION (FT)	
1	0.03375000	20
2	0.00000000	30
3	0.03375000	40
SEC	EFFECT PERIFERAL LENGTH FOR CONVECTION (FT)	
1	0.03375000	21
2	0.00000000	31
3	0.03375000	41
SEC	RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)	
1	0.14560000E-08	22
2	0.	32
3	0.	42
SEC	RADIATION CONSTANT C2 (BTU/HR SQ FT)	
1	146.10000038	23
2	0.00000000	33
3	0.00000000	43
SEC	CONVECTIVE HEAT TRANSFER COEFF (BTU/HR SQ FT R)	
1	7.00000000	24
2	0.00000000	34
3	12.00000000	44
SEC	AMBIENT TEMP (R)	
1	530.00000000	25
2	0.00000000	35
3	600.00000000	45

Fig. B-13 Input Data - Problem 4

```
INPUT FIN VALUES                                                          INPUT DATA
FIN      RADIATION CONSTANT C1 (BTU/HR SQ FT R**4)
 1        0.14560000E-08                                                          80
 2        0.14560000E-08                                                         100
 3        0.                                                                     120
FIN      RADIATION CONSTANT C2 (BTU/HR SQ FT)
 1          146.10000038                                                          81
 2          146.10000038                                                         101
 3            0.00000000                                                         121
FIN      CONVECTIVE HEAT TRANSFER COEFF SIDE A (BTU/HR SQ FT R)
 1            7.00000000                                                          82
 2            7.00000000                                                         102
 3           12.00000000                                                         122
FIN      CONVECTIVE HEAT TRANSFER COEFF SIDE B (BTU/HR SQ FT R)
 1           12.00000000                                                          83
 2            0.00000000                                                         103
 3            0.00000000                                                         123
FIN      AMBIENT TEMP SIDE A (R)
 1          530.00000000                                                          84
 2          530.00000000                                                         104
 3          600.00000000                                                         124
FIN      AMBIENT TEMP SIDE B (R)
 1          600.00000000                                                          85
 2            0.00000000                                                         105
 3            0.00000000                                                         125
FIN      NOT USED
 1                                                                                86
 2                                                                               106
 3                                                                               126
FIN      EFFECT LENGTH AT EXIT (FT)
 1            0.18600000                                                          87
 2            0.34900000                                                         107
 3            0.29520000                                                         127
FIN      EFFECT LENGTH AT ENTRANCE (FT)
 1            0.18300000                                                          88
 2            0.33600000                                                         108
 3            0.29520000                                                         128
FIN      DUCT WALL TEMP AT EXIT (R)
 1          620.00000000                                                          89
 2          620.00000000                                                         109
 3          620.00000000                                                         129
FIN      DUCT WALL TEMP AT ENTRANCE (R)
 1          760.00000000                                                          90
 2          760.00000000                                                         130
 3          760.00000000                                                         150
```

Fig. B-13 Input Data - Problem 4 (cont)

```
SECTION  1

INLET                 OUTLET
  800.00000000          749.37258148 FLUID TEMP (R)
  736.59178162          700.28737640 WALL TEMP (R)
  240.97404099          241.90506363 HEAT TRANSFER COEFF. (BTU/HR SQ FTR)
10309.24023438         9946.94372559 REYNOLDS NO.

                          5.02104044 VELOCITY AVG (FT/SEC)
                         22.27206993 SECTION LENGTH (FT)
                      35439.19287109 HEAT TRANSFER (BTU/HR)
                        241.43955231 HEAT TRANSFER COEFF AVG (BTU/HR SQ FTR)
                      10128.09191895 REYNOLDS NO. AVG
                         50.09765434 FLUID DENSITY AVG
                          1.23233952 WEIGHT OF LIQUID (LB)
                        363.09560776 PRESSURE CHANGE (LB/SQ FT)
SECTION  2

INLET                 OUTLET
  749.37258148          698.96055603 FLUID TEMP (R)
  700.28737640          663.98297119 WALL TEMP (R)
  241.90506363          242.80842400 HEAT TRANSFER COEFF. (BTU/HR SQ FTR)
 9946.94372559         9610.03039551 REYNOLDS NO.

                          4.90042037 VELOCITY AVG (FT/SEC)
                         29.68588352 SECTION LENGTH (FT)
                      35288.41748047 HEAT TRANSFER (BTU/HR)
                        242.35674286 HEAT TRANSFER COEFF AVG (BTU/HR SQ FTR)
                       9778.48706055 REYNOLDS NO. AVG
                         51.33033562 FLUID DENSITY AVG
                          1.68297043 WEIGHT OF LIQUID (LB)
                        476.79207230 PRESSURE CHANGE (LB/SQ FT)

SECTION  3

INLET                 OUTLET
  698.96055603          650.00070953 FLUID TEMP (R)
  663.98297119          627.67856598 WALL TEMP (R)
  242.80842400          229.92062378 HEAT TRANSFER COEFF. (BTU/HR SQ FTR)
 9610.03039551         8850.16125488 REYNOLDS NO.

                          4.78728259 VELOCITY AVG (FT/SEC)
                         43.67850208 SECTION LENGTH (FT)
                      34271.89208984 HEAT TRANSFER (BTU/HR)
                        236.36452293 HEAT TRANSFER COEFF AVG (BTU/HR SQ FTR)
                       9230.09582520 REYNOLDS NO. AVG
                         52.54267263 FLUID DENSITY AVG
                          2.53473344 WEIGHT OF LIQUID (LB)
                        696.03276825 PRESSURE CHANGE (LB/SQ FT)
FINAL OUTPUT DATA
                       1535.92044067 PRESSURE CHANGE SUM (LB/SQ FT)
                     104999.50195313 TOTAL HEAT TRANSFER (BTU/HR)
                          5.45004338 TOTAL WEIGHT OF LIQUID (LB)
                        533.96514130 ENVIRON TEMP (R)
                         95.63645554 TOTAL LENGTH (FT)

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 FLD (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	ALPHAA
19	0.00000	ALPHAB
20	0.85000	EPSA
21	0.85000	EPSB
22	0.00000	EPSX
23	1.00000	FA
24	0.00000	FAX
25	1.00000	FB
26	0.00000	FBX
27	0.00000	RHOM
28	0.00000	RHOX
29	0.00000	THETAP (DEG)
30	0.00000	THETAM (DEG)
31	0.00000	THETAX (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	NUSSELT 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