SAE INTERNATIONAL

AEROSPACE RECOMMENDED PRACTICE

ARP996™ REV. A

Issued 1967-08
Revised 1986-11
Reaffirmed 2025-03

Superseding ARP996

Cooling Data for Turbine Engines in Helicopters

RATIONALE

ARP996A has been reaffirmed to comply with the SAE Five-Year Review policy.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

PREPARED BY SAE COMMITTEE S-12,
HELICOPTER POWERPLANT

1. PURPOSE: Efficient design of a turbine engine installation requires data on the ways the engine rejects heat, the temperature limits of various parts of
Ø the engine, and the changes in heat rejection from service, as well as a method of using these data. A uniform, practical method of data presentation and use is needed. Cooling margins developed by these methods would be subject to full scale testing for verification.

2. SCOPE:

2.1 Method: A tested method of data presentation and use is described herein. The method shown is a useful guide, to be used with care and to be improved with use.

2.2 Axial Nacelle Air Flow: Machine computer programs and an example problem are given for axial nacelle air flow.

2.3 Transverse Nacelle Air Flow: Calculation is given for natural and forced convection with example problems.

3. DEFINITION OF TERMS:

3.1 Nomenclature:

A = Area, ft^2
c_p = Specific heat, Btu/lb deg F
d = Equivalent diameter of annulus, ft
h = Heat transfer coefficient, Btu/hr ft^2 deg F
k = Thermal conductivity, Btu/hr ft deg F
K_e = Effective conductance of shell, Btu/hr ft^2 deg F
N_{pr} = Prandtl Number
N_{RE} = Reynolds Number
Ø q = Heat flux per unit length, Btu/hr ft
S = Radial spacing in annulus, ft = $\frac{D_2 - D_1}{2}$
T = Temperature, deg R
V = Velocity, ft/hr
W = Weight flow, lb/hr
X = Axial distance from entrance or flow disturbance, ft
ϵ_f = Equivalent emissivity for radiation between cylindrical shells. This quantity includes the view effects and the emissivities
σ = Stefan Boltzmann constant
μ = Absolute viscosity, lb/ft hr
ρ = Density, lb/ft^3

3.2 Subscripts:

a = Relating to fluid a
b = Relating to fluid b
1a = Relating to surface of shell 1 on the side toward fluid a
2a = Relating to surface of shell 2 on the side toward fluid a
2b = Relating to surface of shell 2 on the side toward fluid b
i = Zone number

3.2 (Continued):

IN = At inlet of zone
e = Effective
r = Radiation sink

3.3 Definition of Input for Listed Programs:

A = Mean air temperature of zone, deg R
A1 = Flow area of zone, ft^2
A2 = Surface area of engine zone, ft^2
A3 = Surface area of nacelle zone, ft^2
C = Specific heat, Btu/lb deg R
D = Equivalent diameter of zone, ft
E1 = Emissivity factor for zone, dimensionless
E2 = Nacelle emissivity for zone, dimensionless
H9 = Nacelle heat transfer coefficient, Btu/hr ft^2 deg R
K = Thermal conductivity, Btu/hr ft deg R
L = Length of zone, ft
M1 = Viscosity, lb/ft hr
N1 = Prandtl number, dimensionless
Q = Heat rejected, Btu/hr ft
R = Laminar heat transfer coefficient of zone, Btu/hr ft^2 def F (from equation 2a)
T = Nacelle temperature, deg R
Ø TØ = Air temperature into zone, deg R
T1 = Engine surface temperature, deg R
T3 = Ambient air temperature, deg R
W = Air weight flow, lb/hr
Z, Z1, Z2, Z3 = Convergence criteria

4. DATA TO BE SUPPLIED BY THE ENGINE MANUFACTURER:

4.1 Accessory Temperature Limits: Maximum temperature limits for accessories must be specified at designated locations on the accessories.

4.2 Engine Skin Temperature Presentation: The skin temperature of the heat producing parts of the engine must be calculated in each zone as a function
Ø of heat rejection rate. These results are to be presented in graphical form as in Fig. 1 and are to include engine zone designation, engine accessory heat rejection data, and flange leakage data. Also, the range of engine surface emissivity is supplied versus engine length, as plotted in Fig. 2.

4.3 Power Conditions: The heat rejection data presented shall be identified as measured or estimated and shall be for the following conditions:

(1) Maximum power, Sea level, 103 F day
(2) Maximum power, Sea level, 130 F day
(3) Maximum power, 6000 ft altitude, 95 F
(4) Maximum power, Sea level, standard day

4.4 Ø <u>Accessory Temperature Presentation</u>: For the case of accessories on the surface of the engine, the specified surface temperature and heat generation curves may be altered as shown by the dashed lines on the graph and may increase or decrease surface temperature.

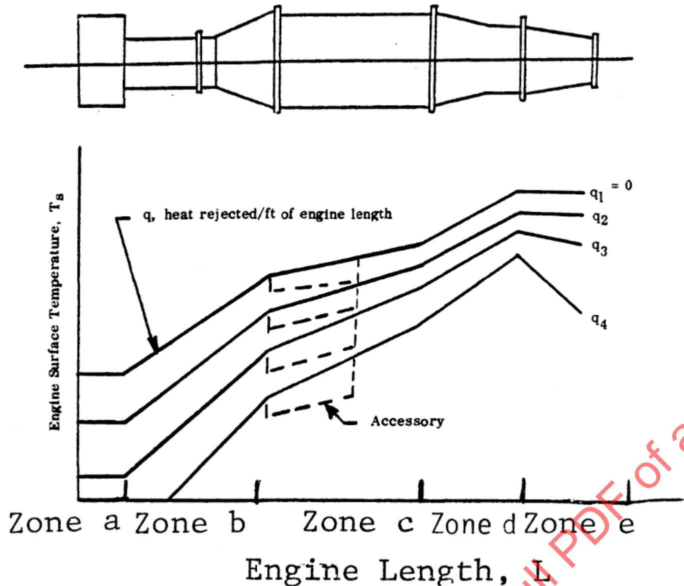

FIGURE 1

Engine Surface Temperature Versus Engine Length for Various Amounts of Heat Rejection

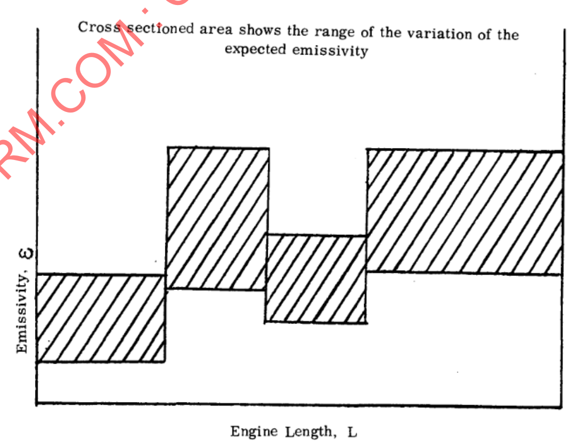

FIGURE 2

Engine Surface Emissivity Versus Length

4.5 Zonal Heat Rejection Rates: Engine heat rejection due to flange leakage and
Ø from accessories and components must also be provided. Figure 3 provides a graphical method of presenting this within each appropriate engine zone and illustrates the heat rejection rates when these zones are compartmented. Engine flange leakage is to be separately identified with the flange leakage rates at specific axial and circumferential positions of the engine.

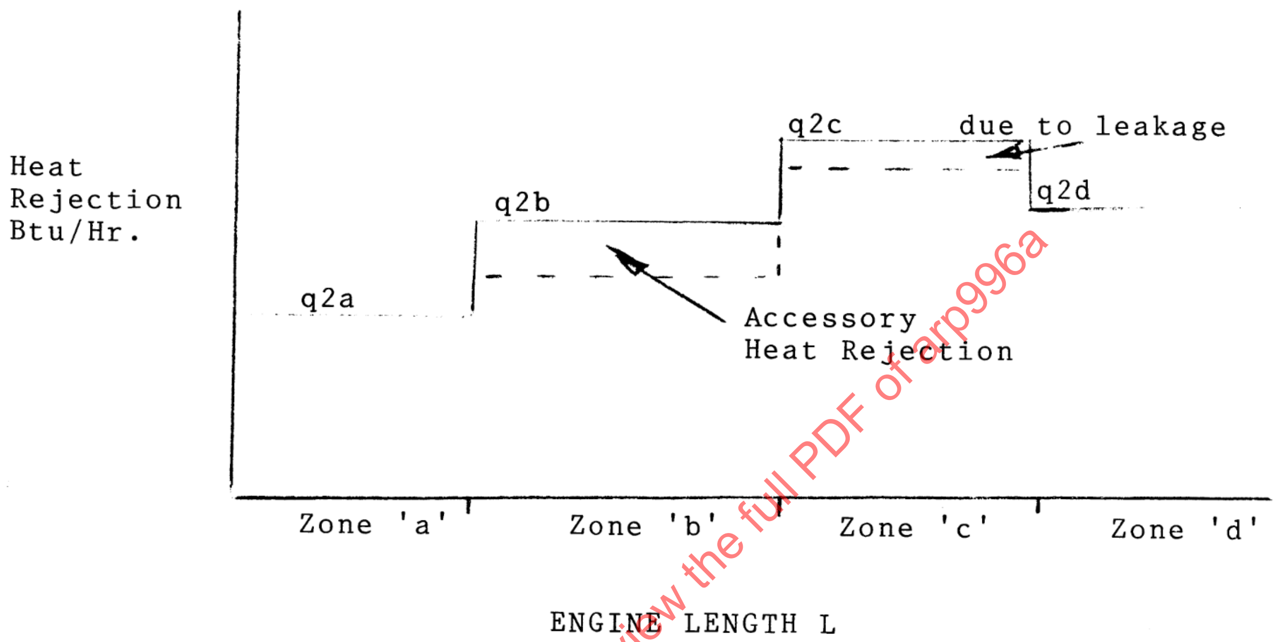

FIGURE 3

Zonal Heat Rejection Rates

5. CALCULATIONS TO BE PERFORMED BY THE AIRFRAME MANUFACTURER:

5.1 Cooling Air Required: The amount of cooling air required must be calculated
Ø to maintain the temperature of the engine accessories within acceptable limits, or maintain maximum requested engine skin temperature, as well as providing cooling for airframe components which may require it, such as hoses, structure, and cowling. Calculations must be carried out both for cooling air required to maintain engine component temperatures with the engine running throughout its operating envelope of ambient temperature and pressures and after shutdown. Generally only convective flow is available after shutdown and this condition may well become the critical case.

5.2 Samples Given: Two sample calculations are given in paragraphs 6 and 7. The first is for axial nacelle air flow and the second is for transverse air flow.

5.3 Cautions Noted: Some of the radiation equations used in the heat transfer
ø calculations may be ignored if, with experience, it can be seen that these effects are small. By the same token more extensive evaluation of the engine's external film coefficients and surface areas may have to be considered if it is shown that these effects are predominate in determining the engine heat rejection rates. The potential for error exists if struts, baffles or shields are encountered and changes in surface conditions in service use such as dirt or grease can substantially alter the emissivity of the surface.

6. AXIAL NACELLE AIR FLOW:

6.1 Calculation: The following calculation to determine the cooling air weight flow and the nacelle temperature is made by assuming the engine to be divided into several axial lengths and then performing a calculation for each section. Since each section depends upon what happened in the upstream section, they are all related to each other and must be analyzed interdependently.

Fig. 4 is a sketch of an engine in an enclosure which has been divided into axial sections.

A heat balance for arbitrary section i, shell 1, is as follows:

Heat Rejected = Radiation + Convection

$$\underbrace{q_{1i}}_{\text{Shell 1}} = \underbrace{q_{12i}}_{\text{Shell 1, Shell 2}} + \underbrace{q_{1a_i}}_{\text{Shell 1, Fluid a}} \qquad (1)$$

For the case of transverse flow, see paragraph 7.

Substituting the expressions for convection and radiation heat transfer:

$$L_i \, q1_i = h_{1_{a_i}} A_{1_{1_i}} \left(T_{1_i} - T_{a_i}\right) + \epsilon_{12} \, \sigma A_{1_i} \left(T^4_{1_i} - T^4_{2_i}\right) \qquad (2)$$

6.1 (Continued):

Similarly for Shell 2.

$$\varepsilon f_{12_i} \sigma A_{1_i} \left(T^4_{1_i} - T^4_{2_i}\right)$$

$$= h_{2a_i} A_{2_i} (T_{2_i} - T_{a_i})$$

$$+ h_{2b_i} A_{2_i} (T_{2_i} - T_{2b_i}) \qquad (3)$$

$$+ \varepsilon_{2_i} \sigma A_{2_i} \left(T^4_{2_i} - T^4_{r_i}\right)$$

Heat transfer coefficient is a function of weight flow and may be substituted into the equations as h = f(W).

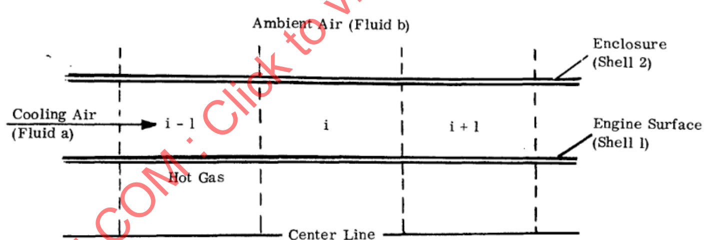

FIGURE 4

Paragraph 6.1.1 suggests possible correlation for h. From the graph of q versus L, q_{1_i} is obtained. T_{a_i} is the average air temperature of zone i.

6.1 (Continued):

Another heat balance expressing the heat gained by the cooling air may be written.

$$h_{1a_i} A_{1_i} \left(T_{1_i} - T_{a_i}\right) + h_{2a_i} A_{2_i} \left(T_{2_i} - T_{a_i}\right) = 2W_{a_i} C_p \left(T_{a_i} - T_{a_{i_{IN}}}\right) \tag{4}$$

$$T_{a_i} = \frac{T_{IN_i} + T_{OUT_i}}{2}$$

The three unknowns--T_a, T_2 and W--may be found from solving equations (2) through (4) simultaneously. Since each downstream section is affected by the section upstream from it, section 1 must be calculated first and then the others in sequence.

From equations (2) through (4), the flow W_{a1} for section 1 and the air temperature, T_{a1}, may be calculated. This weight flow is that which is needed to remove the heat rejected at section 1, q_{a1}. Applying the calculation to section 2, the weight flow W_{a2} may be calculated using $T_{ai_{IN}} = T_{IN} + 2\left(T_{a_1} - T_{IN}\right)$ in equation (4).

If $W_{a2} > W_{a1}$, W_{a2} must be used for the weight flow and W_{a1} must be discarded since it is not large enough to satisfy the cooling requirement of section 2. Using T_{a2}, section 3 may be calculated, the same process being repeated. Obviously section 1 will be overcooled, but this is necessary so that section 2 or some later section will be adequately cooled.

If $W_{a2} < W_{a1}$, W_{a1} should be used and $T_{a_{i_{IN}}} = T_{a2} + 2\left(T_{a2} - T_{a2_{IN}}\right)$ for the calculation of section 3. The largest weight flow calculated is the cooling flow necessary for the engine. Once the weight flow is determined, the enclosure (or nacelle) temperature may be determined from equations (2) and (3). The calculation need not start with zone 1 and may be shortened by starting with one of the hot downstream zones.

If many concentric shells surround the engine, the same procedure is followed except that there is an additional heat balance equation for each shell and each air flow.

6.1 (Continued):

Since the equations are non-linear and must be solved many times, a digital computer program should be quite useful here. Otherwise, many painful graphical or iterative solutions must be made.

Paragraph 6.2 gives an example problem of an engine with one enclosure divided into three axial sections.

If the nacelle wall provides an appreciable thermal resistance, it may be accounted for by expressing the resistance as a conductance, K_e, and rewriting the equations (2) through (4) as follows. This results in four simultaneous non-linear equations. The nacelle resistance has been neglected in the example problem. Also the radiation sink temperature, T_{r_i}, was assumed equal to the air temperature, T_{b_i}

$$L_i \, q1_i = h_{1a_i} A_{1_i} (T_{1a_i} - T_{a_i}) + \varepsilon_{f_{12}} \sigma A_{1_i} \left(T^4_{1a_i} - T^4_{2_i}\right) \tag{5}$$

$$\varepsilon_{f_{12}} \sigma A_{1_i} \left(T^4_{1a_i} - T^4_{2a_i}\right) = h_{2a_i} A_{2_i} \left(T_{2a_i} - T_{a_i}\right) + K_{2_i} A_{2_i} \left(T_{2a_i} - T_{2b_i}\right) \tag{6}$$

$$K_{2_i} A_{2_i} \left(T_{2a_i} - T_{2b_i}\right) = h_{2b_i} A_{2_i} \left(T_{2b_i} - T_{b_i}\right) + \varepsilon_{2_i} \sigma A_{2_i} \left(T^4_{2b_i} - T^4_{r_i}\right) \tag{7}$$

$$h_{1a_i} A_{1_i} \left(T_{1a_i} - T_{a_i}\right) + h_{2a_i} A_{2_i} \left(T_{2a_i} - T_{a_i}\right) = 2W_{a_i} C_p \left(T_{a_i} - T_{a_{i_{IN}}}\right) \tag{8}$$

6.1.1 Correlations for Convection Heat Transfer Coefficients: The forced convection heat transfer coefficient to be used in the annulus between the engine and its enclosure may be calculated from the following equations:

If $N_{RE} > 2000$,

$$h = 0.023\ N_{RE}^{0.8}\ N_{PR}^{0.4}\ \left(\frac{k}{d}\right) \tag{1a}$$

If $N_{RE} < 2000$, the average h for a section, i, starting at axial location X_1 and ending at X_2 is:

$$h = \frac{3.65k}{d} - \frac{0.13k\ N_{RE}\ N_{PR}}{X_2 - X_1} \times \left[Ln\left(\frac{(X_1)^{0.8} + 0.016(d\ N_{RE}\ N_{PR})^{0.8}}{(X_2)^{0.8} + 0.016(d\ N_{RE}\ N_{PR})^{0.8}}\right)\right] \tag{2a}$$

where:

$$N_{RE} = \frac{Wd}{\mu A}$$

$$d = \frac{4A}{P} = \frac{4\ (\text{Flow Area})}{\text{Wetted Perimeter}} \tag{3a}$$

The axial distance, X, is measured from the most recent upstream flow disturbance.

6.2 Example Problem: This is worked as an example to illustrate the use of the
Ø data and the calculation method. Data is specified for an engine consisting of three zones. The data shown below for the example problem is hypothetical but ordinarily would be obtained from a graph similar to Fig. 5.

From the graphs, the heat rejection, emissivity of the engine and the allowable surface temperature corresponding to the heat rejection may be determined. These values are tabulated in Table I.

6.2 (Continued):

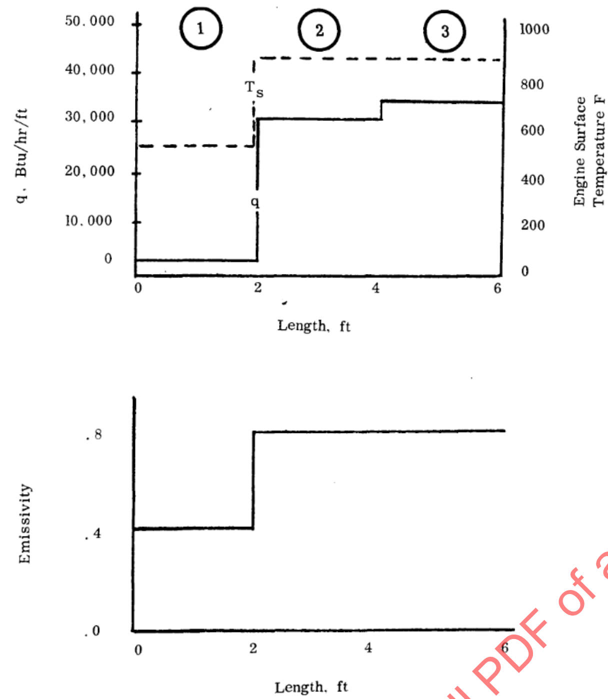

FIGURE 5

Table I

Zone	q, Btu/hr ft	T_S, deg F	ε
1	2050	500	.4
2	30390	840	.8
3	34054	840	.8

The airframe engineer must provide the temperature of the cooling air supply, the ambient temperature and the outside heat transfer coefficient on the nacelle. These values are listed below for the example problem:

$$T_{b_1} = T_{b_2} = T_{b_3} = 100 \text{ F}$$

Cooling air in = 100 F

$$h_{2b_1} = h_{2b_2} = h_{2b_3} = 2 \text{ Btu/hr ft}^2 \text{ deg F}$$

$$\varepsilon_2 = 0.8$$

6.2 (Continued):

The flow area between the engine and nacelle will be assumed constant with engine length.

Nacelle diameter = 14 in. = D_2

Engine diameter = 12 in. = D_1

$$\text{Spacing, } S = \frac{D_2 - D_1}{2} = 1 \text{ in.}$$

The emissivity factor for zone 1 is:

$$\varepsilon_{f_{12_1}} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{D_1}{D_2}\left(\frac{1}{\varepsilon_2} - 1\right)} \tag{1b}$$

$$\varepsilon_{f_{12_1}} = \frac{1}{\frac{1}{0.4} + \frac{12}{14}\left(\frac{1}{0.8} - 1\right)} = .37$$

Assume $N_{RE} > 2000$ to start the calculation. From paragraph 6.1.1:

$$h_{1a_1} = 0.023 \left[\frac{\frac{2}{12} W_{a_1}}{(0.046)(0.27)}\right]^{0.8} (0.7)^{0.4} \left(\frac{0.0158}{2/12}\right) \tag{2b}$$

$$h_{1a_1} = 0.0156\, W_{a_1}^{0.8} = h_{2a_1}$$

Writing the heat balance equations (2) through (4) for zone 1:

Equation 2

$$2050\, L_1 = 0.0156\, W_{a_1}^{0.8} \left(\frac{12}{12}\right) \pi L_1 (960 - T_{a_1}) \tag{3b}$$

$$+ (\pi) \left(\frac{12}{12}\right) L_1 (0.36)(0.1713 \times 10^{-8}) \left[960^4 - T^4_{2_1}\right]$$

Equation 3

$$(\pi) \left(\frac{12}{12}\right) (0.36)\, L_1 (0.1713 \times 10^{-8}) \left[960^4 - T^4_{2_1}\right] \tag{4b}$$

$$= 0.0156\, W_a^{0.8}\, \pi \left(\frac{14}{12}\right) (L_1)(T_{2_1} - T_{a_1}) + (2)(\pi)\left(\frac{14}{12}\right)$$

$$+ (2)(\pi)\left(\frac{14}{12}\right) L_1 (T_{2_1} - 560)$$

6.2 (Continued):

$$+ (0.8)(0.1713 \times 10^{-8})(\pi)\left(\frac{14}{12}\right)(L_1)\left[T^4_{2_1} - 560^4\right]$$

Equation 4

$$0.0156\, W_{a_1}^{0.8} (\pi)\left(\frac{12}{12}\right)(2)(960 - T_{a_1}) \quad (5b)$$

$$+ (2)\left(\frac{14}{12}\right)(\pi)(2)(T_{2_1} - T_{a_1})$$

$$= W_{a_1}(0.24)(T_{a_1} - 560)(2)$$

Solving (3b), (4b) and (5b):

$$T_{a_1} = 592.4\ R$$

$$T_{2_1} = 655.5\ R$$

$$W_{a_1} = 117.9\ lb/hr$$

Checking Reynolds number:

$$N_{RE} = \frac{\left(dW_{a_1}\right)}{(\mu)(A)} = \frac{(117.8)}{(0.049)} \frac{\left(\frac{2}{12}\right)}{(.27)} = 1360$$

Since $N_{RE} < 2000$, equation (2a) for the laminar convection heat transfer coefficient will be tried.

$$h_{1a_1} = \frac{3.65k}{d} - \frac{0.13k\, N_{RE}\, N_{PR}}{X_2 - X_1} \times \text{Ln}\ \frac{(X_1)^{0.8} + 0.016\,(d\, N_{RE}\, N_{PR})^{0.8}}{(X_2)^{0.8} + 0.016\,(d\, N_{RE}\, N_{PR})^{0.8}}$$

$$= \frac{(3.65)(0.0158)}{\frac{2}{12}} - \frac{(0.13)(0.0158)(1360)(0.7)}{2}$$

6.2 (Continued):

$$\times \left[\text{Ln} \left(\frac{0.016 \left[\left(\frac{2}{12}\right) (1360)(0.7) \right]^{0.8}}{(2)^{0.8} + (0.016) \left[(1360) \left(\frac{2}{12}\right) (0.7) \right]^{0.8}} \right) \right]$$

$$= 1.5 \text{ Btu/hr ft}^2 \text{ deg F} \qquad (2a)$$

Replacing $0.0158\ W_{a_1}^{0.8}$ by 1.5 in equations (3b) through (5b) and solving again:

$$T_{a_1} = 779.6 \text{ R}$$

$$T_{2_1} = 695.5 \text{ R}$$

$$W_{a_1} = 24.3 \text{ lb/hr}$$

This weight flow is much lower than previously, therefore, another h_{1a_1} must be calculated using equation (2a). This results in a new h_{1a_1} = 0.81 Btu/hr ft^2 deg F. Again the equations (3b) through (5b) must be solved and so on around the loop until successive calculations produce no change in the weight flow. The converged calculation produces the following values:

$$T_{a_1} = 670.95 \text{ R}$$

$$T_{2_1} = 644.44 \text{ R}$$

$$W_{a_1} = 29.09 \text{ lb/hr}$$

The final N_{RE} = 344.4 showing that the flow is laminar. The air temperature entering zone 2 is then $T_{IN_1} + 2\ (T_{a_1} - T_{IN_1})$ = 560 + 2 (670.95 - 560) = 781.9 R. Then using this inlet temperature to zone 2 and equations (3) through (5) and (2a) the values of T_{a_2}, W_{a_2} and T_{2_2} may be calculated iteratively. This produces a N_{RE} > 2000. Therefore, the laminar calculation must be abandoned and a turbulent calculation made.

6.2 (Continued):

The turbulent heat transfer coefficients for zone 2 are calculated and T_{a_2}, T_{2_2} and W_2 solved for using equations (1a) and (3) through (5).

$$T_{a_2} = 801.7 \text{ R}$$

$$T_{2_2} = 872.2 \text{ R}$$

$$W_{a_2} = 5406.7 \text{ lb/hr}$$

Since $W_{a_2} > W_{a_1}$, the temperatures for zones 1 and 2 must be recalculated using the higher weight flow and equations (3) and (4). The revised numbers are:

$$T_{a_1} = 571.8 \text{ R}$$

$$T_{a_2} = 612.4 \text{ R}$$

$$T_{2_1} = 586.3 \text{ R}$$

$$T_{2_2} = 745.8 \text{ R}$$

$$W_{a_2} = 5406.7 \text{ lb/hr}$$

The temperature entering zone 2 has decreased from its previous value of 781 R to 560 + 2 (571.8 - 560) = 583.6 R. This is a large enough decrease to warrant recalculation of the weight flow needed for zone 2. This produces the following corrected numbers for zones 1 and 2.

$$T_{a_1} = 595.5 \text{ R}$$

$$T_{a_2} = 616.3 \text{ R}$$

$$T_{2_1} = 608.1 \text{ R}$$

6.2 (Continued):

$$T_{2_2} = 782.6\ R$$

$$W_{a_2} = 3488\ lb/hr$$

Since T_{a_1} is within 25 F of its previous value, additional correction of the weight flow appears to be unnecessary. Using equations (2) through (4) for zone 3, the following quantities are calculated:

$$T_{a_3} = 695.8\ R$$

$$T_{2_3} = 807.6\ R$$

$$W_{a_3} = 5010\ lb/hr$$

Since $W_{a_3} > W_{a_{2_1}}$, a cool flow of 5010 lb/hr will be used. The final temperatures corresponding to this flow are tabulated in Table II.

Table II

Zone	W, lb/hr	T_a, deg R	T_2, deg R
1	5010	595	605
2	5010	634	766
3	5010	688	803

The temperatures and weight flow may be calculated more accurately by solving equations (2) through (4) again for zone 3 and using the resulting weight flow to calculate the temperatures of all the zones. This procedure may be repeated until any desired degree of convergence is attained.

6.2.1 Machine Computer Programs: It should be remembered that the numbers in
ø this problem are meant to be illustrative, but not necessarily realistic. With the use of a digital computer, the laborious solution of the simultaneous nonlinear equations will not be difficult and many more zones may be considered. It should be noted however that since the relationship to engineering units is masked by the computer, they may be easily misapplied.

6.2.1 (Continued):

Listings of three programs used to solve the nonlinear simultaneous equations of the sample problem follow. The programs are written for the GE625 digital computer in the language called BASIC. These programs solve only one zone at a time and hence many runs are necessary to solve the complete problem. The programs do not automatically sequence.

Program 1 is for solving the three equations when a turbulent heat transfer coefficient is used for h_{1_a} and h_{2_a}.

Program 2 solves the equations when laminar flow exists.

Program 3 solves equations (3) and (4) when W is known.

These programs are general and are not restricted to the sample problem. A definition of their input is given in paragraph 3 - Definition of Terms.

6.2.1.1 Program 1: Solves the three equations when a turbulent heat transfer coefficient is used for h_{1_a} and h_{2_a} and is a function of $W^{0.8}$.

PROGRAM 1

```
NLSMEQ    15:Ø9    SCH    THU    4/28/6

Ø5Ø    REM SOLUTION OF SIMULTANEOUS NON-LINEAR EQUATIONS
Ø51    REM NEWTON'S METHOD B. BRADDY X353Ø
Ø52    REM PROGRAM NUMBER 1
1ØØ    READ K,D,M1,A1,N1,T1,A2,E1,Q,H9,A3,T3,E2,TØ,C,L,A,T,W,
       Z1,Z2,Z3
11Ø    LET B =0.Ø23*(K/D)*((D/(M1*A1))↑.8)*N1↑.4
12Ø    LET C1 = A2*T1*B
13Ø    LET C2 = A2*B
14Ø    LET S =0.1713E-8
15Ø    LET C4 = E1*S*A2
16Ø    LET D1 = C4*T1↑4-Q*L
17Ø    LET D2 = C4*T1↑4+H9*A3*T3+E2*S*T3↑4*A3
18Ø    LET D3 = C4+E2*S*A3
19Ø    LET C5 = B*A3
2ØØ    LET C6 = H9*A3
21Ø    LET D5 = B*(A2+A3)
22Ø    LET D6 = 2*C
23Ø    LET D7 = D6*TØ
24Ø    LET F = C1*W↑0.8-C2*A*W↑0.8-C4*T↑4+D1
25Ø    LET G = D2-D3*T↑4-C5*T*W↑0.8+C5*A*W↑0.8-C6*T
26Ø    LET H = C1*W↑0.8-D5*A*W↑0.8+C5*T*W↑0.8-D6*A*W+D7*W
27Ø    LET F1 = -C2*W↑0.8
28Ø    LET F2 = -4*C4*T↑3
29Ø    LET F3 = 0.8*(C1-C2*A)/W↑0.2
```

6.2.1.1 (Continued):

```
3ØØ     LET G1 = C5*W ↑0.8
31Ø     LET G2 = -4*D3*T ↑ 3-C5*W ↑ 0.8-C6
32Ø     LET G3 =0.8*C5*(A-T)/W ↑0.2
33Ø     LET H1 = -D5*W ↑ 0.8-D6*W
34Ø     LET H2 = C5*W ↑0.8
35Ø     LET H3 =(0.8*(C1-D5*A+C5*T)/W ↑0.2)-D6*A+D7
36Ø     LET Y = F1*G2*H3-F1*H2*G3-G1*F2*H3+G1*H2*F3+H1*F2*G3-H1*G2*F3
37Ø     IF Y = Ø THEN 49Ø
38Ø     LET X1 = -F*G2*H3-G*H2*F3-H*F2*G3+H*G2*F3+G*F2*H3+F*H2*G3
39Ø     LET X2 = -F1*G*H3+F1*H*G3+G1*F*H3-H1*F*G3+H1*G*F3-G1*H*F3
4ØØ     LET X3 = -F1*G2*H+F1*H2*G+G1*F2*H-G1*H2*F-H1*F2*G+H1*G2*F
41Ø     LET A = A+X1/Y
42Ø     LET T = T+X2/Y
43Ø     LET W = W+X3/Y
45Ø     IF ABS (X1/Y) > Z1 THEN 24Ø
46Ø     IF ABS (X2/Y) > Z2 THEN 24Ø
47Ø     IF ABS (X3/Y) > Z3 THEN 24Ø
472     PRINT "AIR TEMPERATURE-DEG. R =" A
474     PRINT "NACELLE TEMPERATURE-DEG. R =" T
476     PRINT "AIR WEIGHT FLOW-LBS./SEC. =" W
48Ø     GO TO 999
49Ø     PRINT "NO UNIQUE SOLUTION"
5ØØ     DATA0.Ø168,0.16667,0.Ø52,0.27,0.7,96Ø,6.28,0.36, 2Ø5Ø
5Ø1     DATA 2,7.33,56Ø,0.8,56Ø,0.24,2,62Ø,664,122,0.1,0.1,0.1
999     END
```

6.2.1.2 Program 2: Solves the equations when laminar flow exists and the heat transfer coefficient does not depend upon W.

PROGRAM 2

```
NLSMEL  15:24   SCH   THU   4/28/6

Ø11     REM NEWTON'S METHOD B. BRADDY X353Ø
Ø5Ø     REM SOLN OF NLIN SIMEQ, NU = LAMINAR
Ø52     REM PROGRAM NUMBER 2
1ØØ     READ K,D,T1,A2,E1,Q,H9,A3,T3,E2,TØ,C,L,A,T,W,Z1,Z2,Z3,R
11Ø     LET B = R
12Ø     LET C1 = A2*T1*B
13Ø     LET C2 = A2*B
14Ø     LET S = 0.1713E-8
15Ø     C4 = E1*S*A2
16Ø     LET D1 = C4*T1 ↑ 4-Q*L
17Ø     LET D2 = C4*T1 ↑ 4+H9*A3*T3+E2*S*T3 ↑ 4*A3
18Ø     LET D3 = C4+E2*S*A3
19Ø     LET C5 = B*A3
2ØØ     LET C6 = H9*A3
21Ø     LET D5 = B*(A2+A3)
```

6.2.1.2 (Continued):

```
220     LET D6 = 2*C
230     LET D7 = D6*T0
232     LET J1 = C1+D1
234     LET J2 = C5+C6
236     LET J3 = D5+D6
240     LET F = J1-C2*A-C4*T↑4
250     LET G = D2-J2*T-D3*T↑4+C5*A
260     LET H = C1-D5*A+C5*T-D6*A*W+D7*W
270     LET F1 = -C2
280     LET F2 = -4*C4*T↑3
290     LET F3 = 0
300     LET G1 = C5
310     LET G2 = -J2-4*D3*T↑3
320     LET G3 = 0
330     LET H1 = -J3
340     LET H2 = C5
350     LET H3 = -D6*A+D7
360     LET Y = F1*G2*H3-F1*H2*G3-G1*F2*H3+G1*H2*F3+H1*F2*G3-H1*G2*F3
370     IF Y = 0 THEN 490
380     LET X1 = -F*G2*H3-G*H2*F3-H*F2*G3+H*G2*F3+G*F2*H3+F*H2*G3
390     LET X2 = -F1*G*H3+F1*H*G3+G1*F*H3-H1*F*G3+H1*G*F3-G1*H*F3
400     LET X3 = -F1*G2*H+F1*H2*G+G1*F2*H-G1*H2*F-H1*F2*G+H1*G2*F
410     LET A = A+X1/Y
420     LET T = T+X2/Y
430     LET W = W+X3/Y
450     IF ABS (X1/Y) > Z1 THEN 240
460     IF ABS (X2/Y) > Z2 THEN 240
470     IF ABS (X3/Y) > Z3 THEN 240
472     PRINT "AIR TEMPERATURE-DEG. R =" A
474     PRINT "NACELLE TEMPERATURE-DEG. R =" T
476     PRINT "AIR WEIGHT FLOW-LBS./SEC. =" W
480     GO TO 999
490     PRINT "NO UNIQUE SOLUTION"
500     DATA 0.0168,0.16667,960,6.28,0.36
501     DATA 2050,2,7.33,560,0.8,560
502     DATA 0.24,2,620,664,122
503     DATA 0.1,0.1,0.1,2.03
999     END
```

6.2.1.3 Program 3: Solves equation (3) and equation (4) when W is known.

PROGRAM 3

```
NONJON    15:41    SCH    THU    4/28/6

050     REM SOLN OF SIMEQ (NONLINEAR)
051     REM B. BRADDY X3530
052     REM PROGRAM NUMBER 3
```

6.2.1.3 (Continued):

```
10Ø    READ K,D,A1,M1,N1,T1,A2,E1,H9,A3,T3,E2,TØ,C,L,T,W,Z
11Ø    LET B = 0.Ø23*(K/D)*((D/(M1*A1))↑0.8)*N1↑0.4
12Ø    LET C1 = A2*T1*B
14Ø    LET S = 0.1713E-8
15Ø    LET C4 = E1*S*A2
17Ø    LET D2 = C4*T1↑4+H9*A3*T3+E2*S*T3↑4*A3
18Ø    LET D3 - C4+E2*S*A3
19Ø    LET C5 = B*A3
2ØØ    LET C6 = H9*A3
21Ø    LET D5 = B*(A2+A3)
22Ø    LET D6 = 2*C
23Ø    LET D7 = D6*TØ
24Ø    LET U = C1*W↑0.8+D7*W
25Ø    LET V = D5*W↑0.8+D6*W
26Ø    LET A = (D3*T↑4+C5*W↑0.8*T+C6*T-D2)/(C5*W↑0.8)
27Ø    LET A6 = (4*D3*T↑3+C5*W↑0.8+C6)/(C5*W↑0.8)
28Ø    LET H = U-V*A+C5*W↑0.8*T
29Ø    LET H1 = -V*A6+C5*W↑0.8
3ØØ    LET T = T-H/H1
31Ø    LET A = (D3*T↑4+C5*W↑0.8*T+C6*T-D2)/(C5*W↑0.8)
33Ø    IF ABS (H/H1) > Z THEN 26Ø
332    PRINT "AIR TEMPERATURE-DEG. R =" A
334    PRINT "NACELLE TEMPERATURE-DEG. R =" T
34Ø    GO TO 999
35Ø    DATA 0.Ø168,0.16667,0.27,0.Ø52,0.7
351    DATA 96Ø,6.28,0.36,2,7.33
352    DATA 56Ø,0.8,56Ø,0.24,2
353    DATA 664,199,0.Ø1
999    END
```

7. TRANSVERSE NACELLE AIR FLOW:

7.1 Calculation: Transverse cooling may be used in certain regions of the engine outer shell. With this method cooling air will enter the annulus between inner and outer shells and travel peripherally to the exit port. Schematically, this cooling method is shown in Fig. 6.

The path lengths shown are semi-circular. Shorter path lengths may be used, and the sections joined together as in the case of axial flow shown previously. This would allow circumferential variations of engine heat transfer co-efficients and temperatures. Also, the nacelle itself is a heat rejecting shell with circumferential variations in heat transfer coefficients and temperature. These situations are not known to require inclusion at this time. Accordingly, the simple model shown will be treated. It should be adequate for most applications.

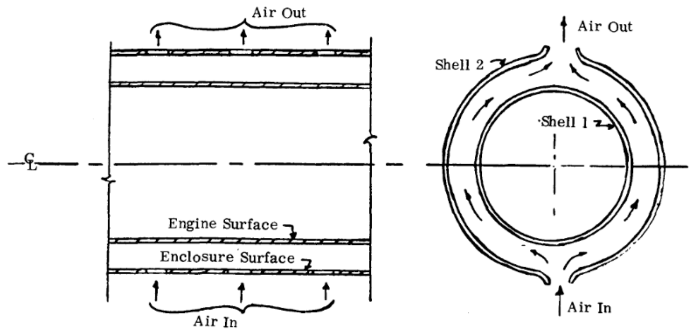

FIGURE 6

The same basic axial flow equations, equations (2), (3), and (4), in paragraph 6.1, can be applied to the transverse cooling case. The heat transfer coefficient equations presented in paragraph 6 do not apply for the flow between shells 1 and 2.

Heat transfer to the coolant circulating in the peripheral direction may be by natural convection: i.e., relying upon the density gradients existing in the annulus, or by forced convection, which requires a prime mover to circulate the cooling air. For this cooling method, the following relationships may be used to evaluate heat transfer coefficients for either natural or forced convection type flows.

7.1.1 Natural Convection:

$$h\left(\frac{\text{Btu}}{\text{hr - ft}^2\text{ - deg F}}\right) = 0.27\left(\frac{\Delta T}{D}\right)^{1/4} \quad (1c)$$

where:

ΔT = temperature difference between shell surface and mean film
ΔT = 1/2 $(T_{SHELL} - T_{AIR})$
D = diameter of shell, ft

7.1.2 Forced Convection: For flow which is not fully developed, use the following relationships for cases of laminar and for turbulent flow:

(a) Laminar Flow ($\bar{N}_{Re_L} < 3.2 \times 10^5$)

$$\frac{hL}{K} = 0.664\,(\bar{N}_{Re_L})^{1/2}\,(N_{Pr})^{1/3} \quad (2c)$$

7.1.2 (Continued):

where:

$$N_{Re_L} = \frac{\rho\ VL}{\mu} = \frac{WL}{A\mu}$$

L = path length for coolant in annulus, $\frac{\pi D}{2}$ for model presented

(b) Turbulent Flow ($\bar{N}_{Re_L} > 3.2 \times 10^5$)

$$\frac{hL}{K} = 0.037 \left[\left(\bar{N}_{Re_L} \right)^{0.8} - 15{,}500 \right] (N_{Pr})^{1/3} \qquad (3c)$$

where:

$$N_{Re_L} = \frac{WL}{A\mu}$$

L = path length of coolant in annulus, $\frac{\pi D}{2}$ for model presented

It is highly unlikely that fully developed flow will exist in the annular space between the engine and the nacelle because of the presence of flow interruptions such as oil lines, accessories, etc., along the flow passage. Also, the annulus passage wall separation would have to be less than 3/4 in. in order for fully developed flow to start. The conditions necessary for fully developed flow would not normally be present in an actual engine cooling passage. The equations presented in the test should be adequate for most applications. For annulus spacing below 3/4 in., the possibility of fully developed flow should be considered and appropriate heat transfer equations used.

7.2 Example Problem: Two sample calculations for transverse cooling are presented herewith, one for natural convection cooling and one for forced convection cooling.

7.2.1 Natural Convection: Consider the following sample design data:

q = 4100 Btu/hr ft (Supplied by Engine Mfr.)
$T_2 = T_1$ = 500 F = 960 R (Supplied by Engine Mfr.)
ε_1 = 0.4 (Supplied by Engine Mfr.)
T_b = 100 F (Assumed - Airframe Mfr. to Determine)
T_{AIR-IN} = 100 F (Assumed - Airframe Mfr. to Determine)