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SIZING TUBE-FIN SPACE RADIATORS

By Jerry A. Peoples
Preliminary Design Office

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16. ABSTRACT

Temperature and size considerations of the tube-fin space radiator are characterized by charts and equations. An approach of accurately assessing rejection capability commensurate with a phase A/B level output is reviewed using the analytical techniques developed by Donald B. Mackey. A computer program, based on Mackey's equations, is also presented which sizes the rejection area for a given thermal load. The program also handles the flow and thermal considerations of the film coefficient.

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DEFINITION OF SYMBOLS

Symbol	Definition
A	Area (ft^2)
C_α	Energy absorbed from environment ($Btu/hr\text{-}ft^2$)
$C_\epsilon = \sigma(\epsilon_a + \epsilon_b)$	Radiation constant defined in Reference 2
C_p	Heat capacity (Btu/lb-°F)
D	Tube diameter (ft)
F	View factor
F_r	Area correction factor between tube-fin and rectangular duct, see Reference 2
K	Fluid conductivity (Btu/hr-ft-°F)
L	Length (ft)
LOHARP	Lockheed orbital heat rate package
$\dot{m}$	Mass flow rate (lbm/hr)
Q	Energy rate (Btu/hr)
RAD-K = C_α	Environmental flux
R, RE	Reynolds number
T	Temperature (°R)
δ	Fin thickness
ϵ	Surface emissivity or tube surface roughness (Colebrook equation of Appendix A)

DEFINITION OF SYMBOLS (Concluded)

Symbol	Definition
η	Fin efficiency
Ω	Fin effectiveness
σ	Stefan-Boltzmann constant, 0.1714×10^{-8} (Btu/hr-ft^2-°R^4)

Subscripts

eff	Effective
d	Diameter
f	At temperature T_R or fluid friction factor (Colebrook equation of Appendix A)
in	Inlet
n, F	Fin
o	Initial conditions
out	Outlet
S	Sink
R	Root
RD	Rectangular duct
TF	Tube-fin
w	Wall

SIZING TUBE-FIN SPACE RADIATORS

INTRODUCTION

The radiator area required to reject a given amount of energy can be calculated by direct application of the Stefan-Boltzmann radiation law:

$$Q = \sigma \epsilon A F T^4 \quad . \tag{1}$$

Even though this law is mathematically simple, its application to radiation sizing can become complex. To avoid complexity, equation (1) is sometimes applied by assessing the effective temperature, T_{eff}, of the radiator:

$$A = \frac{Q}{\sigma \epsilon F T_{eff}^4} \quad . \tag{2}$$

The effective temperature of the radiator is assessed on the basis of experience and empirical data. This approach is normally applied as a result of quick needs by project personnel. However, this approach does have a "built-in" capacity to produce large errors. This results from the fourth power relationship. A small error in the effective temperature is multiplied several times in the resulting area.

The sensitivity of this error can be determined qualitatively from the previously mentioned Stefan-Boltzmann relationship. The change in the required area with an accompanying change in temperature is

$$dA = -\frac{4Q}{\sigma \epsilon F T^5} dT \quad . \tag{3}$$

Normalizing these results by substituting equation (1),

$$\frac{dA}{A} = -4\frac{dT}{T} \quad . \tag{4}$$

Recognizing that dA/A is the percent change in area that results from a percent change in temperature, dT/T. A unit percent increase in temperature will result in a four unit decrease of the required area. An over estimate of the effective temperature by 4 percent will undersize the required area by 16 percent. This is a significant error, even to be tolerated in preliminary design. However, it should be noted that the percent change in temperature is based upon thermodynamic temperature. Also, equation (4) is a mathematically exact relationship where the difference between two temperatures approach zero. In practice, where finite differences are encountered, the actual multiplication error is greater than four. Consider the example where the actual effective temperature is 53.2°F (513.2°R), see Appendix A, and an assumed effective temperature of 68.5°F [(40 + 97)/2], 528.5°R. This is a percent temperature error of 2.98 percent:

$$\frac{528.5 - 513.2}{513.2} = 2.98\% \quad .$$

The percent error in area is

$$\frac{A_e - A_o}{A_o} = \left(\frac{T_o}{T_e}\right)^4 - 1 = \left(\frac{528.2}{513.2}\right)^4 - 1 = 12.21\% \quad ,$$

where A_o is the area resulting from T_o, and A_e is the area resulting from, T_e.

One of the purposes herein is to present a technique for "sizing" radiators which can be defended with rigorous engineering analysis. The techniques are presented in detail, including a computer program to accomplish the basic sizing task which is compatible with a phase A/B study effort.

SIZING VERSUS DESIGN

Radiator sizing can be characterized by fin efficiency (discussed in next section). Radiator design is characterized by the configuration to achieve a desired fin efficiency. The procedure is to, first, size the radiator based upon a desired fin efficiency; second, this fin efficiency is guaranteed by the configuration from which weight can be calculated. If this weight is unacceptable, then a tradeoff has to be made between size and weight (or the thermal load can be reduced).

The important fact to recognize is that a relationship does exist between radiator area and weight, depending upon the fin efficiency (Fig. 1). The relative scale has been selected between one and two since analytical procedures show approximately a 2 to 1 inverse relationship between area and weight. For example, if it was desirable to reduce the area of a tapered tube-fin configuration by 100 percent, then the weight must increase by 100 percent. The extra weight is manifested in the extra fin mass required to achieve a greater fin efficiency.

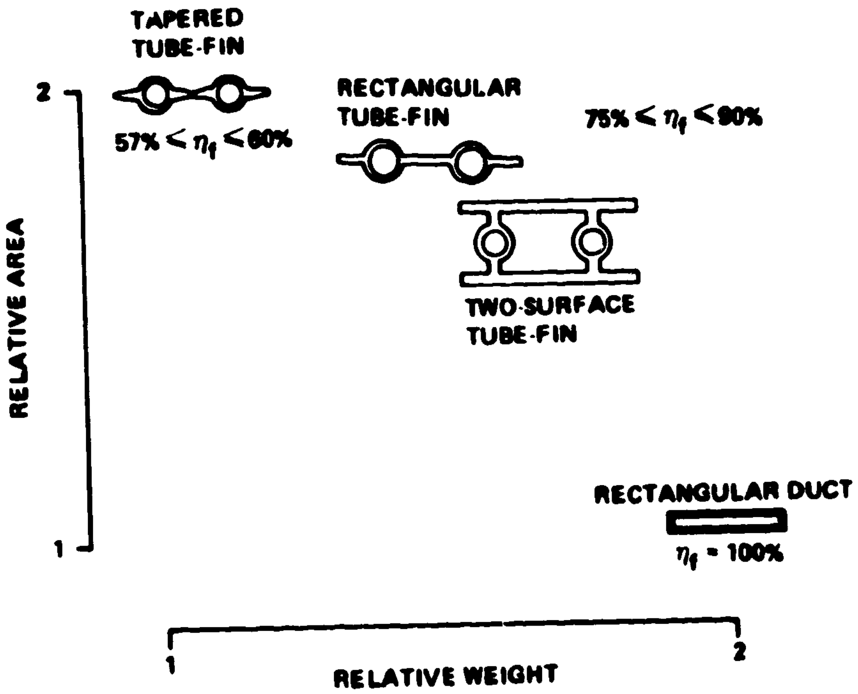

Figure 1. Radiator configuration and weight/area behavior.

The design spectrum represented in Figure 1 is bounded by two configurations. The heaviest is a rectangular duct, having (by definition) a fin efficiency of 100 percent. However, this rectangular duct will have a relatively small area for a given heat load.

The other end of the spectrum is a tapered tube-fin. It has been determined by rigorous analysis that this configuration has the optimum area to weight ratio. Neither of these two configurations, as depicted, is a practical consideration; the first is too massive and the second is structurally weak. Practical radiators are somewhere between these two extremes, with fin efficiencies between 75 and 90 percent. Generally speaking, from the tapered tube-fin configuration, this represents a 50 percent increase in weight with a 12 percent decrease in area [1]. Thus, sizing a radiator not only depends upon the heat load, but also upon the allowable weight which results from design considerations. This report is primarily concerned with sizing rather than design. Design usually occurs in phase C or D, in a primarily design effort, and sufficient data exist to select practical fin efficiencies. This allows the sizing process to proceed in support of programmatic decision. Thus, there are no long delays in specific radiator sizing and weight assessments.

FIN EFFICIENCIES AND FIN EFFECTIVENESS

A tube-fin configuration is practical because of strength and rejection capability, as previously mentioned. As a result, fin performance is important. At least two criteria exist in the literature for assessing fin performance: fin efficiency and fin effectiveness. Fin efficiency is defined as the ratio of the actual heat rejected to what would be rejected if the entire surface was at the root fin temperature, T_R. The environmental effects are accounted for by the sink temperature, T_S:

$$\eta_f = \frac{Q}{\epsilon\sigma\left[T_R^4 - T_S^4\right]A_F} \quad . \tag{5}$$

Fin effectiveness, Ω, is defined as the ratio of the actual heat rejected to what would be rejected if the entire surface was at the root fin temperature. Environmental factors are accounted for by the net heat flux absorbed by the surface from the environment:

$$\Omega = \frac{Q}{\epsilon \sigma T_R^4 A_F} \quad . \tag{6}$$

Thus, the relationship between the two efficiencies is [1]

$$\eta_f = \frac{\Omega}{\left[1 - \left(\frac{T_S}{T_R}\right)^4\right]} \quad . \tag{7}$$

The characteristic behavior of these efficiencies is usually characterized by a fin profile number which results from rigid fin analysis. The profile numbers are a dimensionless set of characteristics which are indicative of geometry, material, and local thermal conditions. The characteristics of fin efficiency are illustrated in Figure 2. Normally, practical radiators have profile numbers less than 1.0. Thus, fin efficiency is always greater than 60 percent. Typical characteristic values of radiator configurations previously discussed are illustrated in Figure 2.

It is important to recognize that the fin efficiency, as defined, is for a single root-fin temperature. In an actual radiator, the root fin temperature decreases in the direction of flow. As illustrated in Figure 2, the local value of efficiency is lowest at the inlet conditions and increase in the direction of the outlet. Thus, fin efficiency cannot be applied directly but must be integrated over the radiator area. Surprisingly, a thermal model employing fin efficiency is not readily available.

The procedure reviewed herein utilizes fin effectiveness as defined in Reference 2. The rationale for selecting this method is its ready applicability to the preliminary design function. Assumptions are employed which simplify the problem for easy equation solving computer techniques. View factors and

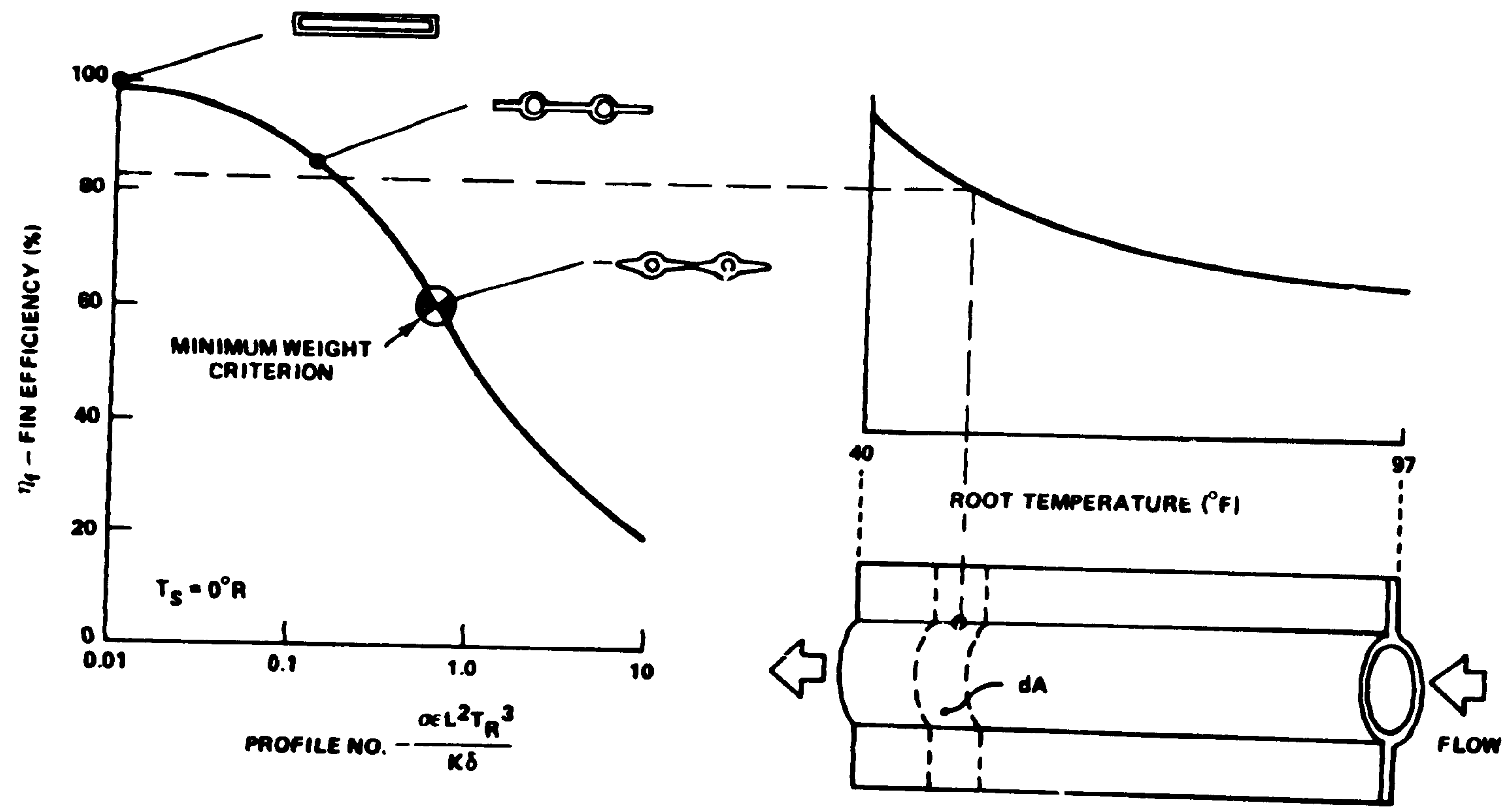

Figure 2. Fin efficiency and its variation in the flow direction.

Rad-K's are inputs which can be evaluated by other programs such as LOHARP. If accuracy is not extremely important, view factors and Rad-K's can be evaluated by charts in combination with experience.

The basis for the method employed is the equivalent width relationship, employed by Mackey [2] between a rectangular duct and a tube-fin configuration. The procedure is to calculate the required area of a rectangular duct for a given heat rejection load. This area is then modified by the equivalent length relationship to establish the area required for the equivalent tube fin configuration. To demonstrate this technique, consider the rectangular duct in Figure 3 with rejection area A_{RD}. The equivalent tube-fin configuration has an area A_{TF} proportional to $(L_d + 2L_n)$. The relationship between these two areas for the same heat rejection capability is

$$\frac{A_{TF}}{A_{RD}} = \frac{2(L_d + 2L_n)}{2L_e} , \qquad (8)$$

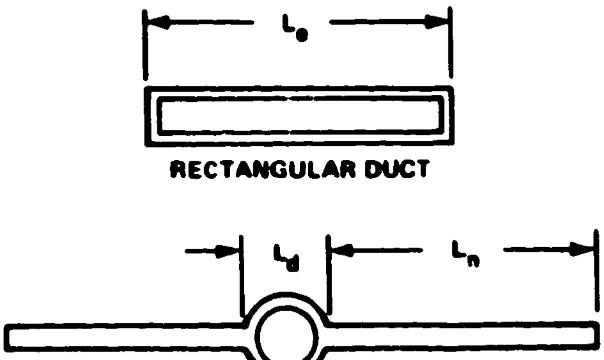

Figure 3. Equivalent rectangular duct for tube-fin.

where L_e is the equivalent rectangular duct width as defined by Mackey [2]:

$$L_e = \left\{ L_d + \frac{2\Omega L_n}{\left[1 - \frac{Q_{ABS}}{2\sigma\epsilon T_w^4} \right]} \right\} F_r \quad . \tag{9}$$

Mackey notes that the greatest value of F_r is 1.07 which is for the optimum area to weight configuration. The smallest value is 1.00. Mackey provides a chart for F_r as a function of profile number. Thus, if the equivalent area of a rectangular duct is known, then the adjustment can be made to find the area of a tube-fin configuration. Before the assessment is made, an evaluation of the configuration and environmental conditions must be made.

This procedure is considered to be valid for rectangular and tapered fins. If a more complex configuration is involved, the procedure is still valid; however, within the computer program, provisions are made to account for the temperature gradient between the heat transfer fluid and the radiating surface.

The computer program for sizing purposes given in Appendix A calculates the area required by a rectangular duct. It is then necessary to manipulate this value by the equivalent length concept to establish the required area for a tube-fin configuration. Combining equations (8) and (9) with Q_{ABS} equal to zero:

$$A_{TF} = A_{RD} \frac{1 + 2\frac{L_n}{L_d}}{\left(1 + 2\Omega \frac{L_n}{L_d}\right) F_r} \quad . \tag{10}$$

A typical fin effectiveness for low temperature radiators is 70 percent. Usually the ratio of L_n / L_d will be approximately 2. If, F_r is 1.04,

$$A_{TF} = A_{RD}[1.265] \quad . \tag{11}$$

Thus, the manipulation required is very simple to arrive at the desired tube-fin configuration.

SPECIAL RADIATOR RELATIONSHIPS

There is a special case of radiator design of particular interest that arises when the absorbed flux can be assumed to be zero. Under this assumption the radiator equation simplifies, and several expressions result which can serve as a guide in developing a philosophy for particular radiator problems.

The first of these is the relationship between heat rejection area and thermal load:

$$A_{Rej} = \frac{Q}{3\sigma\epsilon} \frac{\left[\frac{1}{T_o^3} - \frac{1}{T_{in}^3}\right]}{\left[T_{in} - T_o\right]} \quad . \tag{12}$$

The rejection area is that of a rectangular duct. The equivalent length modification can be applied if a tube-fin configuration is desired. The importance of equation (12) is the sensitivity of the radiator inlet and exit temperatures. It is not apparent, but for a given inlet temperature, the required rejection area decreases as the exit temperature increases. However, as the exit temperature increases, the required flow rate through the system also increases. These facts are illustrated in Figure 4. The ordinate scale has been normalized.

Mass flow rate is normalized to 13 687 lb/hr which occurs at an exit temperature of 80°F. Area is normalized to 537 ft^2 (Ω = 0.70) which occurs at 0°F exit temperature. These data were actually obtained from the computer program of Appendix A. The effects of the heat transfer film coefficient, as a result of flow rate, is accounted. However, this is an insensitive consideration for Reynolds numbers above 300. The radiator area is sized by the radiator thermal resistance. In practical applications, the pump size or pump power may not be allowable. Within an allowable mass flow range, however, there is some flexibility in reducing radiator area.

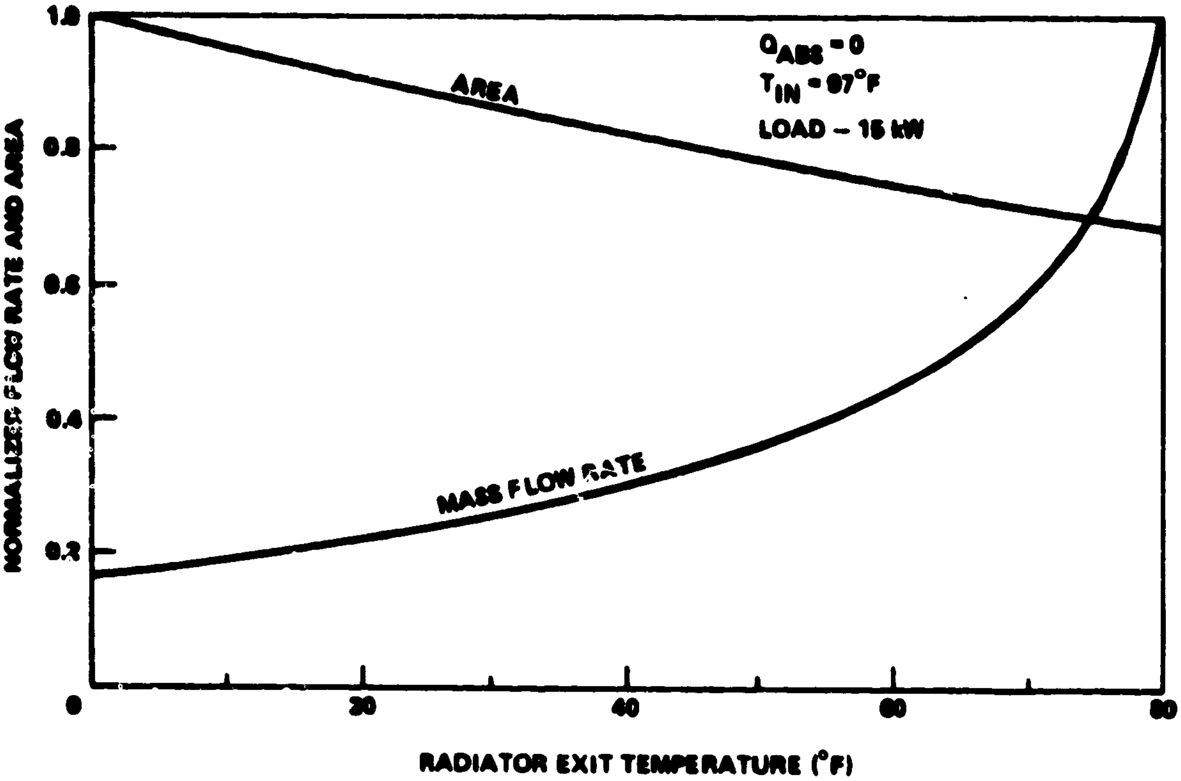

Figure 4. Influence of exit radiator temperature on area and flow rate for a tube-fin configuration.

The effective temperature of the radiator is of interest to the engineer even though it has little practical value. The effective temperature is defined by equation (2). The primary purpose for presenting a rigorous expression for this temperature is to demonstrate, to some level, how errors can occur by assessing it by experience or average values:

$$T_{eff} = \sqrt[4]{\frac{3[T_{in} - T_o]}{\left[\frac{1}{T_o^3} - \frac{1}{T_{in}^3}\right]}} \quad . \qquad (13)$$

The effective temperature as computed by the radiator program is presented in Figure 5.

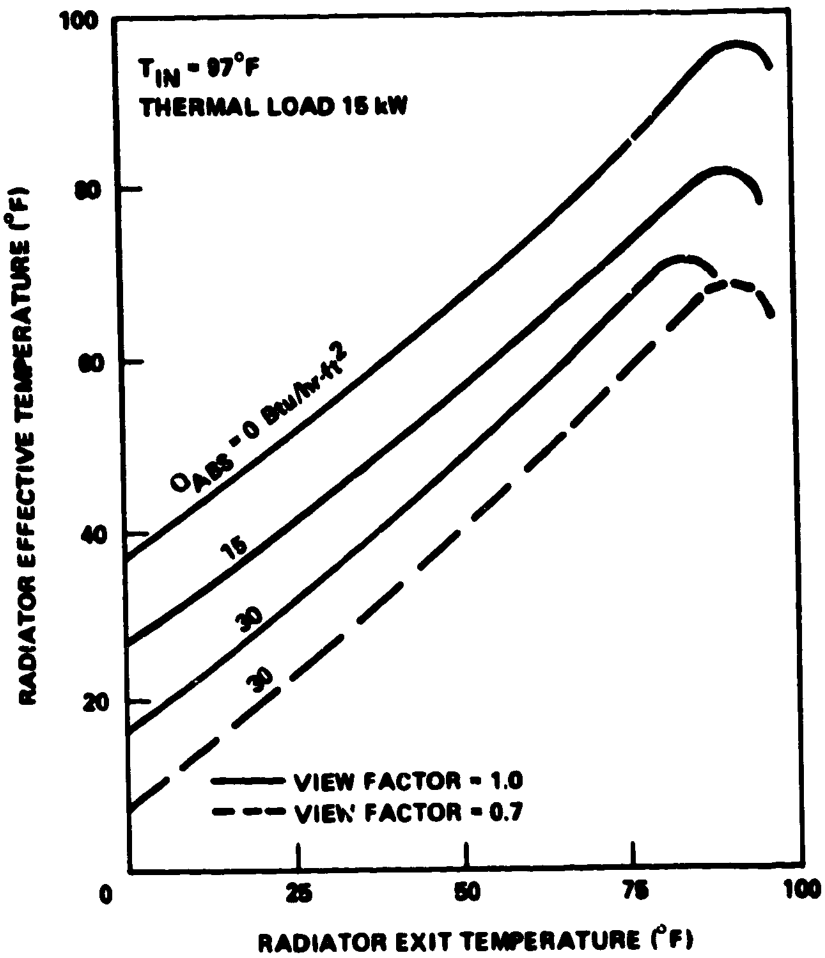

Figure 5. Sensitivity of effective temperature to exit temperature and absorbed flux.

Equation (13) is specially for zero absorbed flux. However, Figure 5 has additional data to illustrate how view factor and absorber flux can affect the effective temperature. To illustrate further the sensitivity of effective temperature, the dashed line is for an absorbed flux of 30 Btu/ft^2, but the view factor has been decreased to 0.70. On the basis of these values, much wisdom and knowledge would be required to properly assess the effective radiator temperature. Note for high Q_{ABS} and a view factor of 0.70, the effective temperature can be outside the temperature range of the inlet and outlet temperature.

This fact can completely discourage the use of applying effective temperature based on the average of inlet and exit temperatures. The effect of view factor and absorbed flux upon the required area to reject 15 kW is illustrated in Figure 6. There is an aggravation effect of absorbed flux for view factors less than one. However, area is much more sensitive to view factor.

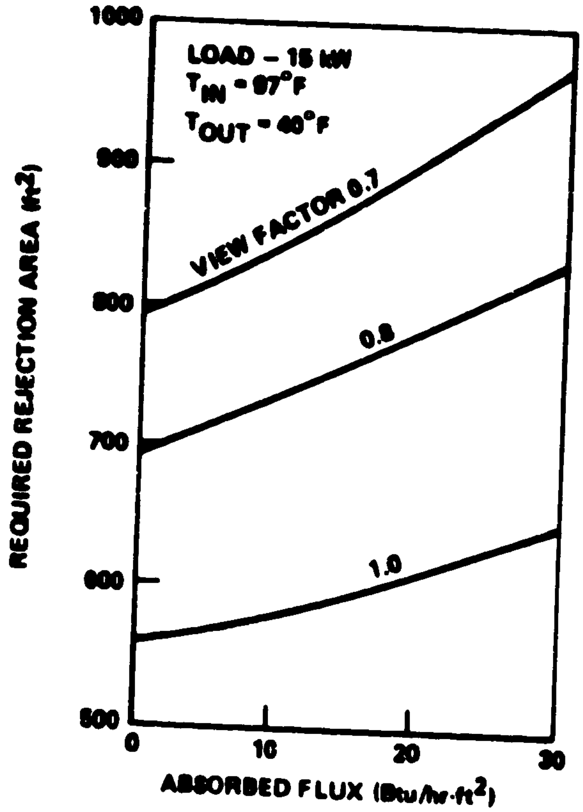

Figure 6. Sensitivity of radiator area to view factor and absorbed flux.

Sometimes, it is necessary to know the temperature distribution along the direction of flow of the radiator. The development of such a relationship is given in Appendix B. The results are

$$\frac{L}{L_o} = \frac{\left[\left(\frac{T_{in}}{T}\right)^3 - 1\right]}{\left[\left(\frac{T_{in}}{T_o}\right)^3 - 1\right]} , \qquad (14)$$

which gives the temperature, T, which occurs at position L feet from the radiator inlet. This normalized form is appropriate since it allows a convenient plot as shown in Figure 7. The temperature distribution is almost linear.

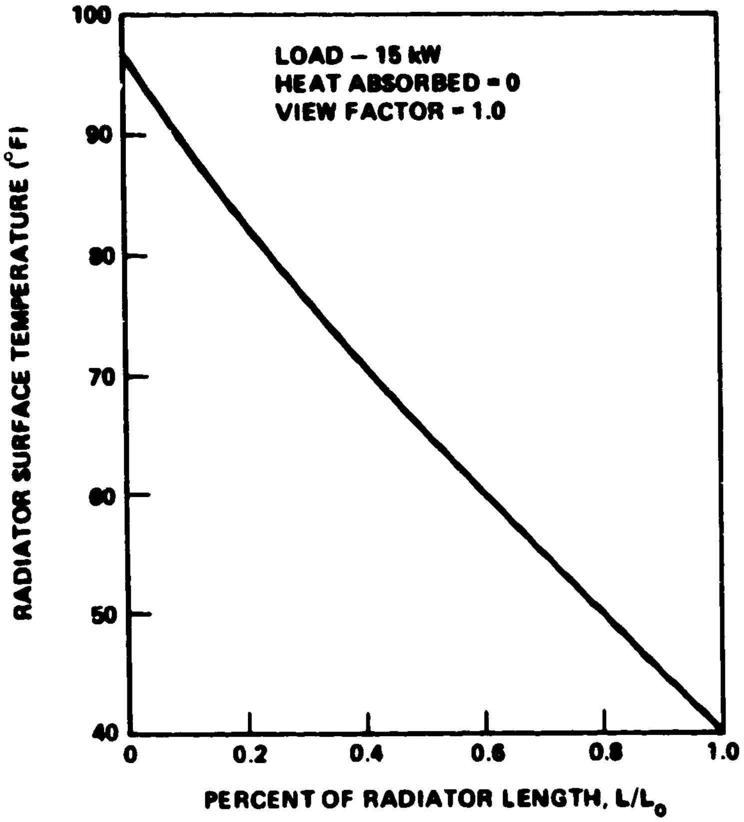

Figure 7. Radiator temperature distribution in direction of flow.

REFERENCES

1. Anderson, A. F.: Radiator Design for Space Vehicles, Airesearch Manufacturing Company, 1963.

2. Mackey, Donald B.: Design of Space Power Plants. Prentice-Hall, Inc., 1963.

APPENDIX A

RADIATOR SIZING PROGRAM

This computer program is an equation solving procedure for one of two equations. The first is statement 49, which applies whenever the absorbed flux cannot be assumed zero. The second is statement 60, which applies when the absorbed flux is zero. Both equations are reported in Reference 2.

Statements 3 through 27 determine the mass flow rate and resulting flow characteristics.

The classical relationship between friction factor and Reynolds number is well known as the Moody diagram. It can readily be found in reference books on fluid flow. For laminar flow the Hagan-Poiseuille equation is transformed into the more manageable form shown on line 12 whereas those values needed in the transition and turbulent regions (RE > 2100) require an iterative process. This is readily apparent from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -0.86 \ln \left(\frac{\epsilon / D}{3.7} + \frac{2.51}{R\sqrt{f}} \right) \quad .$$

For a first approximation the right hand term containing the friction factor is ignored and the resulting friction factor is used for the next approximation. From this point a convergent routine is employed. These are illustrated in the program in lines 14 through 20. The pipe diameter is an input value and the roughness height is built into the program. Reynolds number is calculated in the classical manner from input values. The Darcy-Weisbach equation is used to find the pressure difference which is used in the power equation.

The following are definitions of the input statement 2:

TF1 — Fluid Inlet Temperature, °F

TF2 — Fluid Exit Temperature, °F

TW1 — Wall Temperature at Inlet, °F

TW2 — Wall Temperature at Exit, °F

Q — Thermal Load, Btu/sec

CP — Heat Capacity of Fluid, Btu/lb-°F

RHO — Mass Density of Fluid, lbm/ft^3

XMU — Viscosity of Fluid, lbm/ft-hr

XK — Conductivity of Fluid, Btu/hr-ft-°F

X — Fluid Thickness (Duct Thickness), in.

E — Surface Emissivity

LD — Radiator Length Perpendicular to Flow, ft

VF — View Factor

CA — Absorbed Flux, $Btu/hr\text{-}ft^2\text{-}°F$

C — Case Identification Number

The wall temperatures, TW1 and TW2, are evaluated on the basis of previous data or other calculations. The program computes the radiator length, XW, statement 53 based on area.

Basically, the program equations account for the temperature gradient in the direction of flow. The rectangular duct necessitates a zero gradient perpendicular to the flow direction. Statement 41 calculates the fin effectiveness based on a tube-fin configuration based on minima area to weight ratio. In computing the equivalent area for a tube-fin configuration, the fin effectiveness used should be no less than this optimum value.

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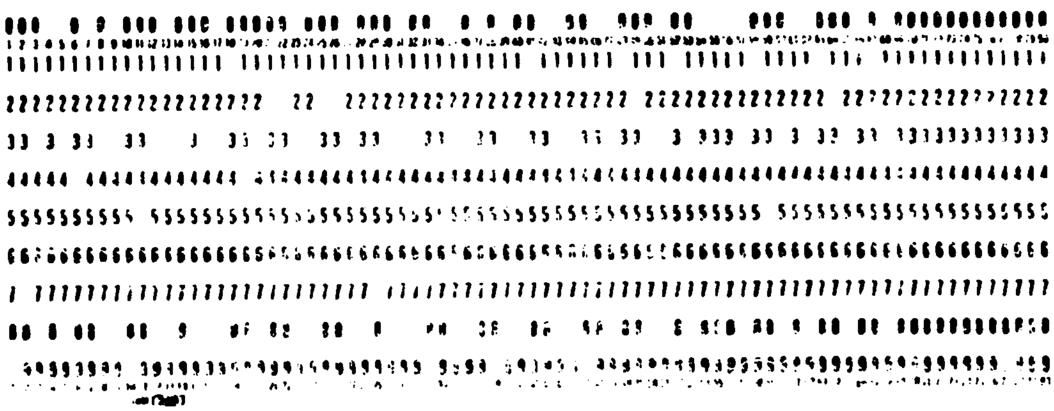
```
      REAL LD
  111 READ (5,1,END=599)TF1,TF2,TW1,TW2,Q,CP,RHO,XMU,XK,X,E,LD,VF,CA,C
      DTG=TF1-TF2
      XMDOT = Q*3600./(CP*DTG)
  112 FORMAT (/////)
      DF=2.*X/12.
      EP=.00015
      RUF=EF/DF
      VEL=2.7858*12.*XMDOT/(RHO*(X *LD)*(10.**4))
      RE=DF*VEL*RHO*3600./XMU
      IF(2100.-RE)942,942,941
  941 FF=64./RE
      GO TO 945
  942 FN=-1.1628/(ALOG(RUF/3.7))
  943 F=-1.1628/(ALOG((RUF/3.7)+(2.51/(RE*FN))))
      EF=F**2.
      DIFF=(EF-FN**2.)/FN**2.
      IF(ABS(DIFF)-0.1)945,945,944
  944 FN=F
      GO TO 943
  945 F=EF
      H=F*VEL**2/(DF*64.4)
   95 DELP=H*RHO/144.
      P=.0F424*XMDOT*DELP/RHO
      PN = XMU*CP/XK
      G=12.*XMDOT/(LD*X)
      HF=.0144*CP*G**.8/(DF**.2)
      Z=(TW2+460.)/(TW1+460.)
C
C   PROGRAM SET UP FOR TWO SIDE RADIATOR.THE EMISSIVITY
C   OF BOTH SIDES ARE EQUAL.IF NOT CHANGE FOLLOWING
C   EQUATION TO: CE=.1714E-08*VF*ET.WHERE .ET. IS
C   THE SUM OF THE TWO EMISSIVITIES.
C
      CE=.1714E-08*VF*(2.*E)
C
C   FOR ONE SIDE RADIATOR . READ REJECTION AREA .AR. AS
C   RADIATOR SIZE .I.E. LENGTH X WIDTH REQD.=AR
C
      CC=CA/(CE*(TW1+460.)**4)
      EF=.57-.55*CC
      IF (CA) 69,69,60
   60 A1=(CC**.25+Z)*(CC**.25-1.)
      A2=(CC**.25-Z)*(CC**.25+1.)
      AA=ALOG(A1/A2)
      BB=ATAN(Z/(CC**.25))-ATAN(1./(CC**.25))
      YX=ALOG((1.-CC)/(Z**4-CC))
      YY=2.*((CE*CA**3)**.25)
      A=((Q*3600.)/DTG)*(YX/(2.*HF)+((.5*AA+BB)/YY))
      GO TO 61
   69 A=(Q*3600./DTG)*((1./(2.*HF))*ALOG(1./(Z**4))
     *+(1./(3.*CE*(TW1+460.)**3)*(1./(Z**3)-1.)))
   61 XW=A/LD
      TP=P*XW
      TE=((Q*3600./(A*CE))**.25)-460.
      SPW=2.*A/(60.*.01757*3)
      AR=2.*A
      DP=DELP*XW
```

```
      PRINT 100
      PRINT 97
   97 FORMAT (6X,36H INPUTS                            )
      PRINT 98
   98 FORMAT (/)
      PRINT 29,C
   29 FORMAT (10X,36H CASE I.D. NO.............        ,F10.3)
      PRINT 94
      PRINT 30,TF1
   30 FORMAT (10X,36H FLUID INLET TEMP.  F              ,F10.3)
      PRINT 31,TF2
   31 FORMAT (10X,36H FLUID EXIT TEMP.  F               ,F10.3)
      PRINT 32,TW1
   32 FORMAT (10X,36H RAD. WALL TEMP. @ INLET  F        ,F10.3)
      PRINT 33,TW2
   33 FORMAT (10X,36H RAD. WALL TEMP. @ OUTLET  F       ,F10.3)
      PRINT 34,Q
   34 FORMAT (10X,36H RAD. THERMAL LOAD  BTU/SEC        ,F10.3)
      PRINT 35,CP
   35 FORMAT (10X,36H FLUID HEAT CAPCITY  BTU/LB F      ,F10.3)
      PRINT 36,RHO
   36 FORMAT (10X,36H FLUID DENSITY  LB/FT3             ,F10.3)
      PRINT 37,XMU
   37 FORMAT (10X,36H FLUID VISCOCITY  LB/FT HR         ,F10.3)
      PRINT 38,XK
   38 FORMAT (10X,36H FLUID CONDUCTIVITY  BTU/HR FT F   ,F10.3)
      PRINT 39,Y
   39 FORMAT (10X,36H FLUID FOU. THICKNESS  IN          ,F10.3)
      PRINT 40,E
   40 FORMAT (10X,36H RAD. SURFACE EMISSIVITY           ,F10.3)
      PRINT 41,LD
   41 FORMAT (10X,36H RAD. LENGTH PER. TO FLOW   FT     ,F10.3)
      PRINT 42,VF
   42 FORMAT (10X,36H VIEW FACTOR                       ,F10.3)
      PRINT 43,CA
   43 FORMAT (10X,36H TOTAL ENVIR. ABSORBED BTU/HR FT2  ,F10.3)
      PRINT 113
  113 FORMAT (//)
      PRINT 131
  131 FORMAT (6X,36H OUTPUT                            )
      PRINT 98
      PRINT 9,AR
    9 FORMAT (10X,36H REQD. PROJECTION AREA  FT 2       ,F11.2)
      PRINT 10,A
   10 FORMAT (10X,36H RADIATOR SIZE  FT 2               ,F11.2)
      PRINT 11,XW
   11 FORMAT (10X,36H RADIATOR LENGTH  FT               ,F11.2)
      PRINT 12,RE
   12 FORMAT (10X,36H REYNOLDS NUMBER                   ,F10.3)
      PRINT 13,HF
   13 FORMAT (10X,36H FILM COEFF.  BTU/HR FT2 F         ,F10.3)
      PRINT 14,P
   14 FORMAT (10X,36H UNIT FLOW  POWER  WATTS/FT        ,F10.3)
      PRINT 15,TP
   15 FORMAT (10X,36H TOTAL FLOW POWER  WATTS           ,F10.3)
      PRINT 16,DTQ
   16 FORMAT (10X,36H FLUID TEMP. DIFF.   F             ,F10.3)
      PRINT 17,PN
```

```
   17 FORMAT (10X,35H PRANDL  NUMBER                           ,F10.3)
      PRINT 18,SPW
   18 FORMAT (10X,35H SPECIFIC AREA    FT2/KW                  ,F10.3)
      PRINT 19,TE
   19 FORMAT (10X,35H EQUIVALENT FAC. TEMP.  F                 ,F10.3)
      PRINT 20,DP
   20 FORMAT (10X,35H PRESSURE DROP  PSI                       ,F10.3)
      PRINT 21,XMDOT
   21 FORMAT (10X,35H MASS FLOW RATE  LB/HR                    ,F12.3)
      PRINT 22,G
   22 FORMAT (10X,35H MASS VELOCITY  LB/HR FT2                 ,F10.3)
      PRINT 23,EF
   23 FORMAT (10X,35H SURFACE FIN EFF.,MIN. WEIGHT             ,F10.3)
  100 FORMAT (1H1)
  1[illegible] FORMAT ( )
      GO TO 111
  999 STOP
      END
```



INPUTS

CASE I.D. NO.***********	12.CIC
FLUID INLET TEMP. F	97.CCC
FLUID EXIT TEMP. F	40.000
RAD. WALL TEMP. @ INLET F	95.CCC
RAD. WALL TEMP. @ OUTLET F	38.000
RAD. THERMAL LOAD BTU/SEC	14.220
FLUID HEAT CAPCITY BTU/LB F	.220
FLUID DENSITY LB/FT3	87.CCC
FLUID VISOCITY LB/FT HR	.90J
FLUID CONDUCTIVITY BTU/HR FT F	.C9C
FLUID EQU. THICKNESS IN	.100
RAD. SURFACE EMISSIVITY	.9CC
RAD. LENGTH PER. TO FLOW FT	12.000
VIEW FACTOR	1.CCC
TOTAL ENVIR. ABSORBED BTU/HR FT2	15.000

OUTPUT

REQD. REJECTION AREA FT 2	478.27
RADIATOR SIZE FT 2	239.14
RADIATOR LENGTH FT	19.93
REYNOLDS NUMBER	758.436
FILM COEFF. BTU/HR FT2 F	35.086
UNIT FLOW POWER WATTS/FT	.C02
TOTAL FLOW POWER WATTS	.041
FLUID TEMP. DIFF. F	57.CCC
PRANDL NUMBER	2.200
SPECIFIC AREA FT2/KW	31.905
EQUIVALENT RAD. TEMP. F	53.237
PRESSURE DROP PSI	.C16
MASS FLOW RATE LB/HR	4082.297
MASS VELOCITY LB/HR FT2	4C822.966
SURFACE FIN EFF.,MIN. WEIGHT	.542

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APPENDIX B

RADIATOR TEMPERATURE DISTRIBUTION

In a differential radiator length, dL, the radiator temperature will change dT in accordance with the following energy balance:

$$\left[C_\epsilon T^4 - C_\alpha\right] L_d \, dL = -\dot{m} Cp \, dT \quad . \tag{B-1}$$

This equation is presented in Reference 2 by Mackey. In this form it assumes the fluid temperature is the same as the radiator wall temperature. If the environmental factor, C_ϵ, can be assumed zero, the equation can be readily integrated:

$$L = \frac{\dot{m} Cp}{3 L_d C_\epsilon} \left[\frac{1}{T^3} - \frac{1}{T_{in}^{\;3}}\right] \quad . \tag{B-2}$$

In this form, at a point on the radiator having temperature, T, the radiator length must be L.

Equation (B-2) can be combined with equation (12) to yield

$$\frac{L}{L_o} = \frac{\left[\frac{T_{in}}{T}\right]^3 - 1}{\left[\frac{T_{in}}{T_o}\right]^3 - 1} \quad . \tag{B-3}$$

In this form, L/L_o is the decimal value of the total length, L_o. The radiator temperature, T, corresponds to the decimal length, L/L_o.

APPROVAL

SIZING TUBE-FIN SPACE RADIATORS

By Jerry A. Peoples

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

Charles R. Darwin
Director, Preliminary Design Office

James T. Murphy
Director, Program Development

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