

SNAP 2 RADIATIVE-CONDENSER DESIGN*

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ABSTRACT

The design of a minimum weight radiative condenser for SNAP 2 requires the simultaneous consideration of many factors. These include 2-phase mercury flow, a variety of heat transfer problems, meteoroid protection and radiator geometry effects. This paper describes the interactions between the above problem areas, and presents the method by which a minimum weight radiative condenser for SNAP-2 was designed.

I. INTRODUCTION

The heat rejection system for space power plants is a major weight item, and for high power systems may be the predominant one. For this reason, considerable effort can profitably be expended to insure the design of minimum weight systems.

The 3-kw SNAP 2 power system, currently under development by Atomics International for the Missile Projects Branch of the Atomic Energy Commission Aircraft Reactors Office, uses a mercury Rankine cycle to convert heat, generated in a nuclear reactor, to electrical energy. The Rankine cycle requires the rejection of heat by condensation of the working fluid, and because of the space environment, the heat rejection must ultimately be accomplished by thermal radiation to space.

The SNAP 2 design employs a direct-condensing radiator. In this design, the heat released in the condensation process is radiated to space from a large extended surface, consisting of

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fins which are attached to tubes containing the condensing fluid. This is the type of system considered in this report.

The large number of variables involved in the design of a radiator-condenser makes it impossible to prepare a single design curve. The different aspects of the radiator-condenser design problem are considered in Sections II, III, and IV. The data developed in these sections are then combined in Section V where a procedure leading to the determination of a minimum weight heat rejection system is developed. An example of the procedure is presented for the SNAP 2 radiator design point which requires the mercury working fluid to be condensed at 600°F.

II. TWO-PHASE FLOW

A. PRESSURE DROP

Viscous drag is the mechanism by which condensed mercury is removed from the radiative condenser. Mercury enters the condenser as saturated vapor, and is condensed at substantially constant temperature by being subjected to a constant heat flux throughout the length of the tube. After the vapor has been completely condensed the liquid is subcooled by about 200°F before leaving the radiator. Subcooling of the condensate is required to reduce the possibility of pump cavitation, and to provide a low-temperature bearing and alternator coolant.

The design of a minimum weight radiator requires an accurate prediction of the pressure drop associated with the condensing mercury as it flows through the radiator tubes. This must be done with some precision since each one psi of static pressure loss in the condenser tubes lowers the saturation temperature of the condensing mercury by about 14°F, which results in a 5% drop in radiating power at this temperature.

Many investigators have correlated pressure drop data for 2-phase flow in tubes. Of these, the correlation of Lockhart and Martinelli¹ is probably the best and most widely used. They were able to correlate pressure drop for 2-phase, 2-component flow in the nondimensional form,

$$\left(\frac{dP}{dl}\right)_{TPF} = \phi^2 \frac{2}{g} \left(\frac{dP}{dl}\right)_g, \quad (1)$$

where

$$\left(\frac{dP}{dl}\right)_{\text{TPF}} = \text{2-phase frictional pressure gradient,}$$

$$\left(\frac{dP}{dl}\right)_g = \text{pressure gradient per unit tube length of the gas (vapor) phase alone,}$$

$$\phi_g = \text{dimensionless parameter which is a function only of } \frac{\left(\frac{dP}{dl}\right)_l}{\left(\frac{dP}{dl}\right)_g} ;$$

and

$$\left(\frac{dP}{dl}\right)_l = \text{pressure gradient per unit tube length of the liquid phase alone.}$$

The experimental data of Lockhart and Martinelli were plotted in terms of

$$\phi_g \text{ and } \sqrt{\frac{\left(\frac{dP}{dl}\right)_l}{\left(\frac{dP}{dl}\right)_g}} \text{ for different combinations of flow regimes,}$$

e.g., turbulent gas-laminar liquid, turbulent gas-turbulent liquid. While these correlations were for 2-component flow, extension of these correlations to boiling or condensing (i.e., 2-phase, one-component flow) has been suggested by Martinelli and Nelson² and has been tested with some success. In support of this extrapolation, McAdams³ has found that friction arising from transfer of momentum between phases of one-component system was of little importance under his experimental conditions. Furthermore, Lockhart and Martinelli stated that their correlation was independent of flow mechanism, whether mist, annular, or stratified flow existed. Unpublished experimental work being conducted by AI and its subcontractors indicates that these correlations predict reasonable values for condensing pressure drop in tubes.

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The Lockhart and Martinelli correlation corresponding to the SNAP 2 condenser conditions can be represented by:⁴

$$\phi = 1.76 X^{0.0825}, \quad (2)$$

where

$$X = \sqrt{\frac{\left(\frac{dP}{dl}\right)_l}{\left(\frac{dP}{dl}\right)_g}} \quad (3)$$

By combining Equations 1,2, and 3 with the appropriate values of $(dP/dl)_g$ and $(dP/dl)_l$ and integrating the resulting expression to account for the changing flow conditions throughout the length of the tube, an expression for the frictional pressure drop can be derived.⁴ The result is given by:

$$\left(\frac{\Delta P}{L}\right)_{TPF} = \frac{0.42 W_T^{1.684} \mu_g^{0.316}}{N D \rho_g g_c} \left(\frac{\rho_g \mu_l}{\rho_l \mu_g}\right)^{0.0825} \quad (4)$$

where

W_T = weight flow

μ_g = viscosity of vapor

μ_l = viscosity of liquid

N = number of tubes

D = inside tube diameter

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ρ_g = density of vapor

ρ_l = density of liquid

g_c = gravitational constant.

Any consistent system of units may be used.

In addition to the frictional pressure drop given by Equation 4 there is a pressure rise due to the momentum loss of the high velocity mercury vapor as it traverses the condenser tube. Since the fluid velocity is essentially zero at the exit of the condenser this is given by:

$$(\Delta P)_M = - \left(\frac{1.62W_T^2}{\rho_g g_c D^4 N^2} \right), \quad (5)$$

where the negative sign indicates a pressure rise.

Equations 4 and 5 were used to calculate the pressure drop for the condensers which are considered in this report. It must be emphasized that this analysis is valid only if the static pressure drop is a small fraction of the inlet pressure.

III. HEAT TRANSFER

The heat transfer problem associated with the design of a radiative-condenser is complicated in that it includes condensing heat transfer under a complex and changing flow regime, heat conduction through the tube walls to the fins, and thermal radiation from the surface of the fins to the space environment. In addition, for the design under consideration in this report, there is a requirement that subcooling of the condensate be achieved. The need to design the lightest-weight system to do the job requires that all the above phenomena be thoroughly analyzed and experimentally verified.

A. CONDENSING

Very high heat-transfer coefficients have been measured condensing mercury. Values of over 50,000 Btu/hr-ft²-°F have been reported by Misra and Bonilla.⁵ Condensing heat flux for

a typical SNAP 2 configuration will be about 5,000 Btu/hr-ft². If h were only of the order of 5,000, the temperature drop across the condensing mercury would be only 1°F. An experimental program has been initiated, as part of the SNAP 2 development, to investigate the effects of the zero-gravity environment on boiling and condensing. Tests made in parabolic-flight aircraft at WADD show that condensation in "zero-g" is a highly stable process, and there has been no evidence to suggest that there will be any materially different heat transfer coefficients than those obtained under the influence of gravity. Since the thermal resistance of condensation is small, it has been neglected in the analysis.

B. TUBE TO FIN HEAT TRANSFER

The temperature drop between the tube and the fin will depend on the nature of the bond between them. Work has been undertaken at Atomics International to determine suitable ways of bonding steel tubes to aluminum. Preliminary analysis has indicated that with any reasonable bond, the thermal resistance is quite small, and hence, was not included in this analysis.

C. RADIANT HEAT TRANSFER

1. Temperature

Radiant heat transfer is described by the following equation:

$$Q = \epsilon \sigma A \left(T^4 - T_{\text{sink}}^4 \right), \quad (6)$$

where

Q = radiative power (Btu/hr),

ϵ = surface emissivity,

A = surface area (ft²),

σ = Stephen Boltzman constant = 0.171×10^{-8}
Btu/hr-ft²-°R⁴,

T = radiating surface temperature (°R), and

T_{sink} = radiative sink temperature (°R).

The exact determination of T_{sink} is a complicated calculation requiring a detailed knowledge of the specific mission and conditions involved. A detailed study^o for a satellite orbiting the earth at a 300-mile altitude, reveals that the effective radiative sink temperature is about 0°F . This value is sufficiently low in comparison with the range of T under consideration in this report (600°F) as to permit the neglect of this term. With this assumption, Equation 6 is simplified to

$$Q = \epsilon \sigma AT^4 \quad . \quad (7)$$

Equation 7 is plotted in Figure 1 for values of ϵ from 0.2 to 1.0.

2. Fin Effectiveness

A quantity termed the fin effectiveness is introduced to assist in the evaluation of the performance of the tube-and-fin radiator. It is defined as the ratio of the heat rejected by the fin to that which would be rejected if the entire fin were maintained at the base temperature. The model used to calculate the fin effectiveness is shown in Figure 2. Expressed mathematically:

$$\eta \equiv \frac{\int_0^{B/2} T_x^4 dx}{T_o^4 B/2} \quad (8)$$

where

η = fin effectiveness

B = tube spacing (ft)

T_o = temperature at the base of the fin ($^{\circ}\text{R}$)

T_x = temperature at a point on the fin ($^{\circ}\text{R}$)

x = distance along fin (ft)

This equation is derived in Reference 4 and the results are given in Figure 3 as a function of the dimensionless parameter M_R , defined as:

$$M_R = \frac{B^2 \epsilon \sigma T_o^3}{kt} \quad (9)$$

where

k = conductivity of fin material

t = fin thickness

By using the curve in Figure 3, the fin effectiveness may be evaluated for a given material, geometry, and base temperature (T_o).

3. Radiator Area Requirements

a. Condensing

For the condensing portion of the radiator, the tube temperature remains constant until the fluid is completely condensed. This will be true only if the static pressure drop is kept small since the condensate and condensing vapor are always in thermal equilibrium. If this condition is met, the area requirements for the condensing portion of the radiator can be obtained by combining Equation 7 with the definition of fin effectiveness.

$$Q = \epsilon \sigma \eta AT_o^4 \quad (10)$$

Figure 1 can be used directly to obtain the required area if the ϵ as shown is considered to be equal to the product, $\epsilon \times \eta$.

b. Subcooler

In the subcooler section the radiant heat rejection is accompanied by a sensible heat loss of the fluid. The temperature decrease of the fluid results in temperature gradients both perpendicular and parallel to the direction of fluid flow. By combining the model for the condensing (constant temperature) fin with that of a radiator which experiences a coolant

temperature drop, (Figure 2B) an expression can be derived to give the area requirements for the tube-fin subcooler configuration.⁴ The result is given by:

$$Q = 3 \epsilon \sigma \eta A \Delta T \left[\frac{1}{(T_{in} - \Delta T)^3} - \frac{1}{T_{in}^3} \right]^{-1} \quad (11)$$

where

Q = radiative power (Btu/hr) ,

A = area required (ft²),

T_{in} = fluid temperature into subcooler (°R),

T_{out} = fluid temperature out of subcooler (°R),

ΔT = temperature drop of fluid passing through the subcooler (°R) = ($T_{in} - T_{out}$), and

η = fin effectiveness (evaluated at some intermediate temperature).

Equation 11 is plotted in Figure 4 for various values of T_{in} and ΔT . This figure can be used to determine the area requirements for the subcooler section.

IV. METEOROID PROTECTION

Meteoroids can cause failure of the system by puncture of radiator fluid lines, causing loss of working fluid. At the present time there is considerable doubt associated with the methods used to calculate the amount of armor required to provide adequate meteoroid protection. The best effort to date to correlate the small amount of experimental data with a reasonable analytical model has been by Bjork.^{7,8} While admittedly his results are probably conservative, his approach appears justifiable in view of the meager experimental data available. It is essentially his procedure which has been followed to determine the armor requirements for meteoroid protection of the radiator.

A. METEOROID FLUX

The flux of meteoroids given in Reference 8 is:

$$\phi = 10^{-12} M^{-10/9} \quad , \quad (12)$$

where

ϕ = number of particles of mass greater than M
per square meter per second

M = mass of particle in grams.

B. METEOROID PENETRATION

The theoretical model postulated by Bjork⁷ predicts the depth of penetration of projectiles impinging upon thick targets at hypersonic velocities. The calculations based on this model agree in both size and shape (hemispherical) with experimental craters produced in aluminum at 6.3 km/sec and in iron at 6.8 km/sec. The results of the theoretical analysis are summarized by the equation:

$$p = K' (Mv)^{1/3} \quad (13)$$

where

p = penetration depth (cm),

M = mass of projectile (grams),

v = projectile velocity in (km/sec), and

K' = constant depending upon materials; for aluminum
projectiles impinging on aluminum targets -
K' = 1.09
for iron projectiles impinging on iron targets -
K' = 0.606.

The modification of Equation 13 to account for projectiles which have densities different than the target material is uncertain,⁹ as is the value of the meteoroid density itself. Some evidence exists which indicates that the meteoroid density may be that of stone, ~2.3 g/cc. Hence Equation 13 will

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be a good approximation for stone meteoroids impinging upon aluminum, but will predict too high a value for the depth of penetration in materials which have a density significantly greater than that of aluminum.

While Equation 13 was derived for thick targets, enough information was obtained from Bjork's analysis¹⁰ to deduce that if a projectile penetrates a depth, p , in a thick target it will just penetrate a sheet of the same target material which is $1.5 p$ thick, thus:

$$t_1 = 1.5 p \quad (14)$$

where

t_1 = required thickness of material (cm).

Equations 12 and 13 may be combined to obtain a penetrating flux which is given as:

$$\psi = 10^{-12} t_1^{-10/3} K^{10/3} v^{10/9} \quad (15)$$

where

ψ = the number of penetrations of target thickness t , per square meter per second

$K = 1.5K'$.

C. ARMOR REQUIREMENTS

The meteoroid punctures may be expected to follow a Poisson distribution which is given by:

$$P(n) = \frac{(\psi A' \tau')^n \exp -(\psi A' \tau')}{n!} \quad (16)$$

where

$P(n)$ = is the probability that n punctures will occur in time τ' of a sensitive area A' .

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For the case of the fluid type radiator under consideration, the allowable number of punctures of the tube walls is zero, hence Equation 16 reduces to:

$$P(o) = \exp^{-\psi A' \tau'} \quad (17)$$

where

$P(o)$ = the probability that no punctures will occur in time τ' of a sensitive area A' .

Combining Equations 15 and 17 yields:

$$t_1 = \frac{2.5 \times 10^{-4} K_v^{-1/3} (A' \tau')^{3/10}}{\ln \left[\frac{1}{P(o)} \right]^{3/10}} \quad (18)$$

For values of $P(o) >$ than 0.90 there is little error incurred by making the approximation:

$$t_1 = 2.5 \times 10^{-4} K_v^{-1/3} \left[\frac{A' \tau'}{1 - P(o)} \right]^{3/10} \quad (19)$$

Assuming an average meteoroid velocity of 30 km/sec and converting to engineering units, Equation 19 reduces to:

for aluminum

$$t' = 43.6 \left[\frac{A\tau}{1 - P(o)} \right]^{0.3} \quad (20)$$

for steel

$$t' = 24.2 \left[\frac{A\tau}{1 - P(o)} \right]^{0.3} \quad (21)$$

where

t' = required thickness in mils,

A = area in ft^2

τ = time of exposure in years, and

$P(o)$ = probability that no punctures will occur in time τ of area A .

These equations are plotted in Figures 5 and 6 for aluminum and steel respectively for various values of $P(o)$ from 0.90 to 0.99 and for τ equal to one year. These curves were then used to determine the tube armor thickness required for radiator designs considered in another section of this report.

Another interesting approach to the meteoroid protection problem is that of the meteoroid "bumper." The usual concept has a piece of material placed around, and at some distance from the point to be protected. The meteoroid, by passing through this light piece of material, shatters and its mass is dispersed over a larger area. Preliminary calculations indicate that this is indeed the case but insufficient analytical techniques or experimental data exist at the present time to base radiator designs on this approach.

V. MINIMUM WEIGHT RADIATOR-CONDENSER DESIGN

To illustrate how the various design variables are combined to determine a minimum weight heat rejection system, a specific example is presented. The condition selected corresponds to the design point of the SNAP-2 power system and are listed below:

- a) Power - 3-kw electrical,
- b) Working fluid - mercury,
- c) Cycle overall efficiency - 6.0%,
- d) Condensing temperature - 600°F,
- e) Vehicle skin geometry - 3.8 ft equivalent-diameter truncated cone,
- f) Required subcooling $\Delta T = 180^\circ\text{F}$,

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- g) Surface emissivity - 0.90,
- h) Condenser static pressure drop - 1.75 psi,
- i) Meteoritic protection - 90% probability of no puncture for one-year life, and
- j) Materials - fins: aluminum
tubes: steel, eccentric bore (20-mil minimum wall thickness)

The radiator geometry selected is the conical nose-cone configuration shown in Figure 7. The shape was selected so as to have the radiator lie entirely within the shadow of the reactor shield, thus minimizing the shield weights. The small diameter of the cone was set by the reactor and its shield dimensions at 2-1/2 ft; the large diameter was set by the missile diameter which was taken as 5 ft. Such a geometry may be adequately described in terms of an equivalent cylinder with a diameter equal to $\frac{D_1 + D_2}{2}$.¹¹

The tube and fin detail are shown in Figure 8. The meteoroid armor is made up of the fin material and the steel tubes containing the condensate. The steel tubes are made eccentric so as to provide the necessary meteoroid armor in one direction only. Meteoroids which enter from the opposite side of the radiator were assumed to be sufficiently dispersed¹⁰ so as to require no armor and the minimum thickness of tube wall was set by corrosion and strength considerations at 20 mils.

A system of equations was then developed, expressing analytically the heat transfer, fluid flow, and meteoroid protection requirements of the radiator. Weights were calculated by a digital computer for wide ranges of pressure drop, number of tubes, and fin effectiveness. A typical result is shown in Figure 9, for a constant value of fin effectiveness equal to 0.83. From Figure 9 it is seen that the weight curve passes through a broad minimum at about 114 pounds and 36 tubes. Figure 10 illustrates the effect of varying fin effectiveness and shows the radiator weight going through a minimum for a fin effectiveness of 0.79. However, system requirements dictate a maximum area of 110 ft², corresponding to $\eta = 0.83$. It is seen that the weight penalty imposed by the arbitrary area restriction is small.

The minimum-weight design specifications for the SNAP-2 system are listed in Table I.

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TABLE I

SNAP-2 RADIATOR SPECIFICATIONS

Area	110 ft ²
Geometry	Truncated Cone, 5 ft diameter tapered to 2-1/2 ft (9.4 ft long)
Shell Thickness (Fins)	28 mils (aluminum)
Number of Tubes	36 (steel)
Tube Dimensions	0.366 in OD 0.273 in ID
Fin Effectiveness	0.83
Weight	114 lb

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NOMENCLATURE

- A = area (ft^2)
 A' = area (m^2)
 B = tube spacing (ft)
 c_p = heat capacity (Btu/lb- $^{\circ}\text{F}$)
 D = inside tube diameter (ft)
 D_1 = diameter of radiator cone (apex)
 D_2 = diameter of radiator cone (base)
 g_c = gravitational constant ($32.2 \frac{\text{lb-mass-ft}}{\text{lb-force-sec}^2}$)
 $K = 1.5K'$
 K' = constant, dependent on material properties
 k = thermal conductivity of fin (Btu/hr-ft- $^{\circ}\text{R}$)
 L = condenser length (ft)
 l = distance along condenser (ft)
 M = mass (grams)
 M_R = radiation modulus
 N = number of tubes
 n = number of meteoroid punctures
 P = pressure (lb/ft^2)
 $P(n)$ = probability that n punctures will occur in time τ' of a sensitive area A'
 $P(o)$ = probability that no punctures will occur in time τ' of a sensitive area A'
 p = penetration depth (cm)
 Q = radiative power (Btu/hr)
 Re = Reynold's number

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NOMENCLATURE (Continued)

- T_o = radiating surface temperature ($^{\circ}\text{R}$) (also tube temperature)
- T_{sink} = radiative sink temperature ($^{\circ}\text{R}$)
- T_{in} = fluid temperature into subcooler ($^{\circ}\text{R}$)
- T_{out} = fluid temperature out of subcooler ($^{\circ}\text{R}$)
- T_x = temperature at a point on the fin ($^{\circ}\text{R}$)
- t = fin thickness (ft)
- t_1 = required thickness for meteorite protection (cm)
- t' = required thickness for meteorite protection (mils)
- u = gas velocity (ft/sec)
- v = projectile, or meteoroid, velocity (km/sec)
- W = flow rate (lb/sec)
- X = dimensionless parameter of Lockhart and Martinelli
- x = distance along fin perpendicular to direction of fluid flow (ft) (also vapor quality)
- y = distance along flat plate radiator, parallel to direction of fluid flow (ft)

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- ϵ = surface emissivity
- η = fin effectiveness
- ϕ_g = dimensionless parameter of Lockhart and Martinelli
- ϕ = number of meteorite particles of mass greater than M per square meter per second
- μ = viscosity (lb-mass/ft-sec)
- σ = Stephen Boltzman constant (0.171×10^{-8} Btu/hr-ft²- $^{\circ}\text{R}^4$)

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NOMENCLATURE (Continued)

ρ = density (lb-mass/ft³)

ψ = number of penetrations of target thickness t , (cm)
per square meter per second

τ = time of exposure in years

τ' = time of exposure in seconds

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T = total

l = liquid phase

g = gas phase

TPF = two phase friction

M = momentum

o = entrance

E = exit

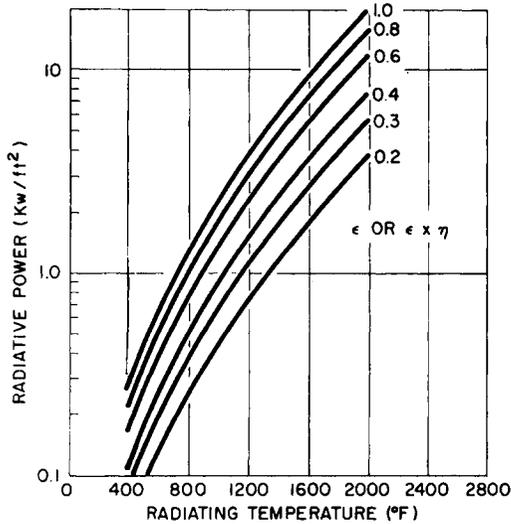


Fig. 1. Radiative Heat Flux vs Temperature.

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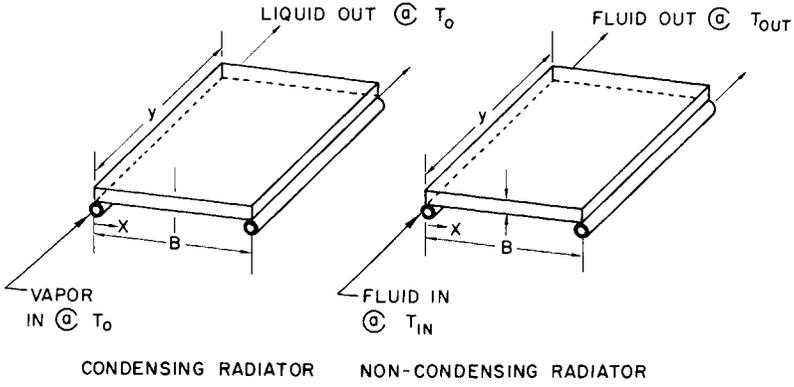


Fig. 2. Tube and Fin Radiator

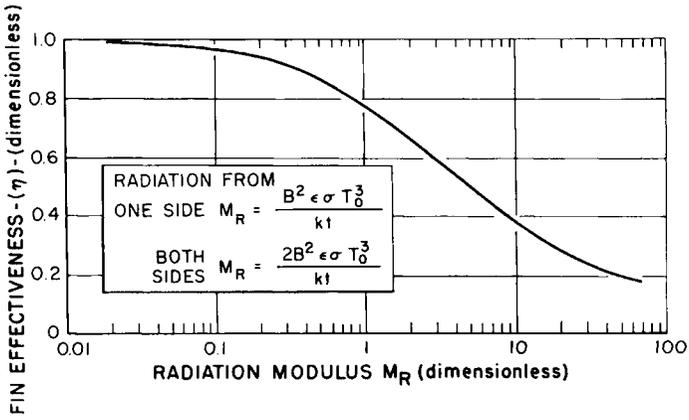


Fig. 3. Fin Effectiveness vs Radiation Modulus

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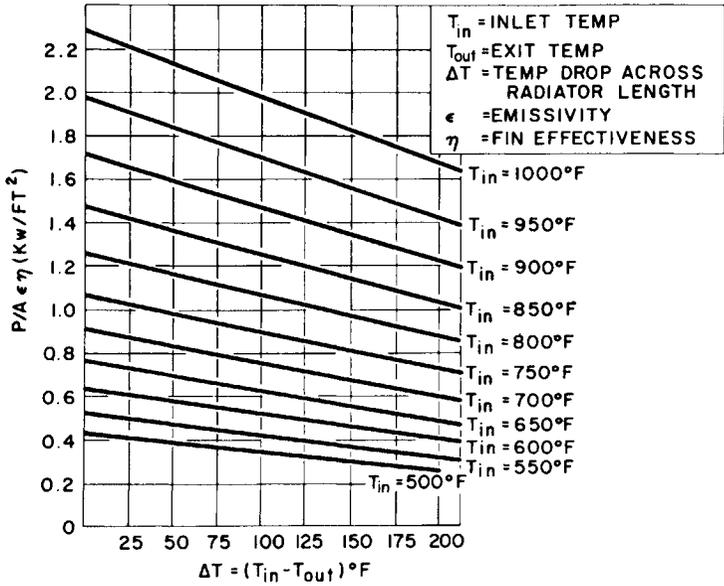


Fig. 4. Radiator Area Requirements for Subcooler Section

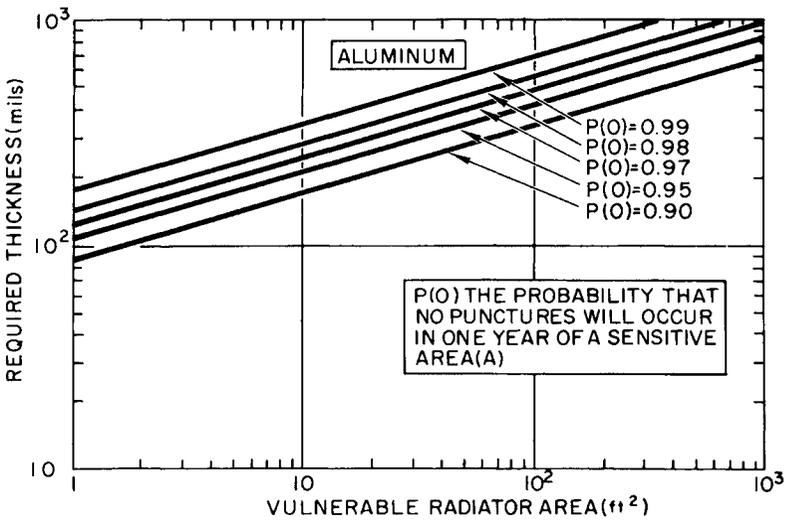


Fig. 5. Material Thickness vs Radiator Area (Aluminum)

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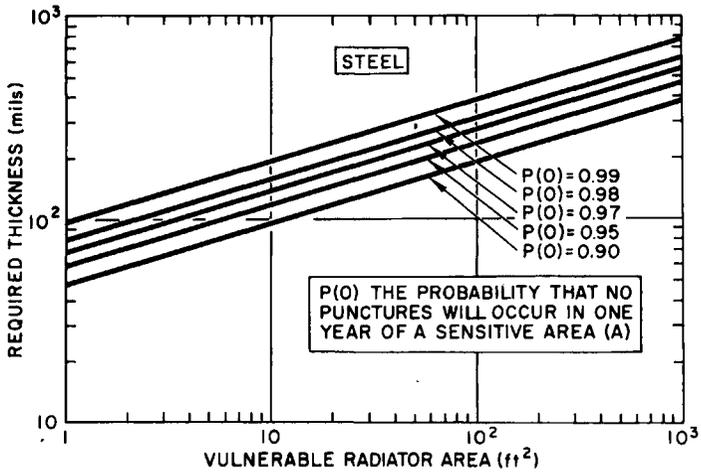


Fig. 6. Material Thickness vs Radiator Area (Steel)

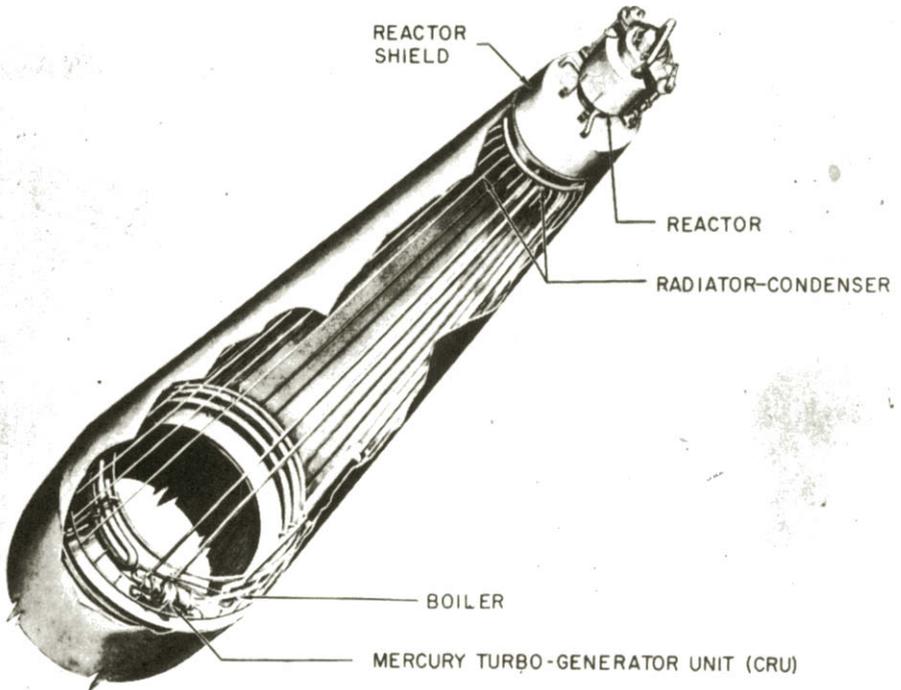


Fig. 7. SNAP-2 Radiator Configuration

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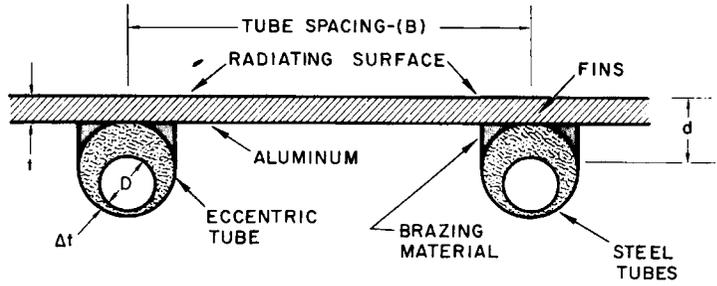


Fig. 8. Detail of Tube and Fin Arrangement

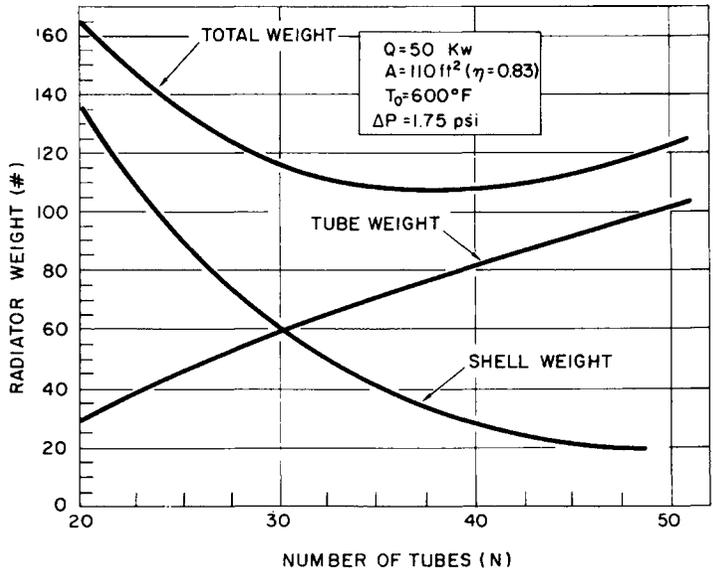


Fig. 9. Radiator Weight vs Number of Tubes

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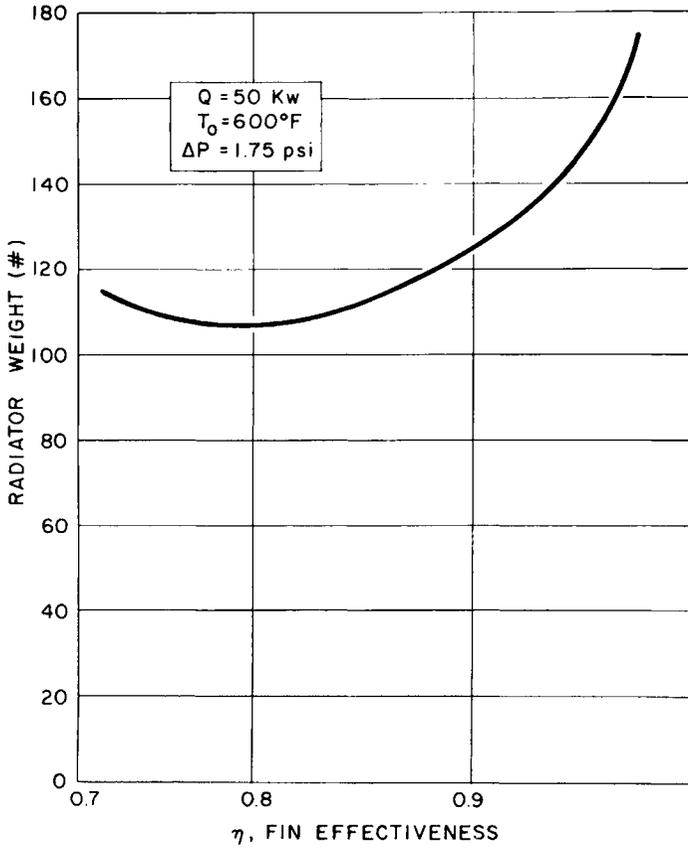


Fig. 10. Minimum Radiator Weight vs Fin Effectiveness