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SUBJECT: Thermal Radiative Heat Transfer to Space from a Body Enclosed by a Semitransparent Body

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FROM: R. J. Hefner

Abstract

Bumpers have been proposed for protecting space radiator systems from penetration by meteoroids. The development of equations to determine the thermal energy dissipation to space by a hot body completely enclosed by a second body is presented. The particular case of heat dissipation from space radiators enclosed within thin bumpers is considered, and the criteria for selection of bumper materials for a minimum weight radiator system are discussed.

DISCLAIMER

Thermal Radiative Heat Transfer to Space from a Body Enclosed by a Semitransparent Body

Semitransparent plastic bumpers have been proposed as an effective means of protecting space radiators from penetration by meteoroids. It was the purpose of this study to determine the effect of such a bumper on the thermal performance of a space radiator. The equations developed, however, are equally valid for any system consisting of one body completely enclosed by a second body and dissipating heat to space by thermal radiation.

Consider a system consisting of a solid body R completely enclosed by a second solid body B, as illustrated in Fig. 1. The volume between body R and body B is completely devoid of any absorbing media, and the system is radiating to space. For reference, the surface of body R will be denoted 1, the inside surface of body B by 2, the outside surface of body B by 3, and the surrounding space by 4. The following postulates will apply to the system:

a) Lambert's cosine principle applies to the distribution of thermal radiation intensity from all surfaces.

b) The temperature of space surrounding the system is at zero degrees absolute with a coefficient of absorption of unity.

c) The temperature of each surface is uniform.

d) Body R is opaque to thermal radiation.

e) The emissivity of surfaces 1 and 2 and the transmissivity of body B are constant over the temperature range of the system. (This implies that the absorptivity and emissivity of a surface are numerically equal.)

f) The transmissivity of body B is independent of the thickness of the body.

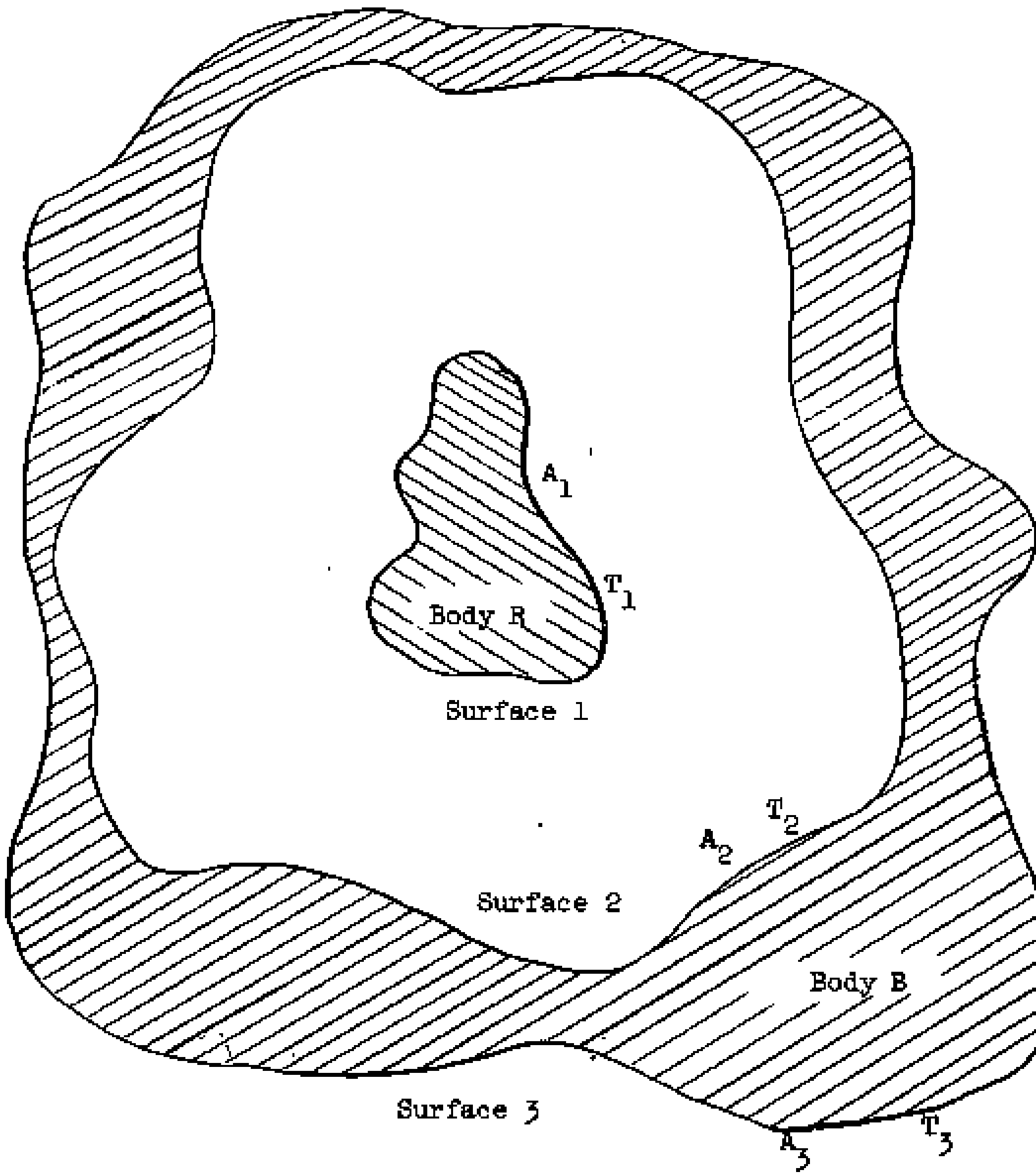

Fig. 1. Cross Section of an Opãque Body Completely Enclosed by a Semitransparent Body

Since body B completely encloses body R, and the transmissivity of body R is zero, the following relations apply:

$$F_{12} + F_{11} = 1 \quad (1\text{-a})$$

$$F_{21} + F_{22} = 1 \quad (1\text{-b})$$

$$F_{34} + F_{33} = 1 \quad (1\text{-c})$$

$$\epsilon_1 + \rho_1 = 1 \quad (1\text{-d})$$

$$\epsilon_2 + \rho_2 + \tau_2 = 1 \quad (1\text{-e})$$

where

F_{ij} = radiation interchange configuration factor from surface i to surface j

ϵ = emissivity

ρ = reflectivity

τ = transmissivity

subscripts 1 and 2 refer to surfaces 1 and 2 respectively

Consider first the total thermal radiation from surface 1 as illustrated in Fig. 2. Of any quantity of heat X leaving surface 1, an amount $F_{12}X$ reaches surface 2, and an amount $F_{11}X$ is intercepted at surface 1. Following for the moment only the quantity $F_{11}X$ which is intercepted at surface 1, $\epsilon_1 F_{11}X$ is absorbed and $\rho_1 F_{11}X$ is reflected. This reflection results in a quantity $F_{12}\rho_1 F_{11}X$ reaching surface 2, $\epsilon_1 F_{11}\rho_1 F_{11}X$ absorbed at surface 1, and a reflection of $\rho_1 F_{11}\rho_1 F_{11}X$ at surface 1.

Continuing in this manner it can be seen that of the total quantity of radiation X leaving surface 1, only:

$$F_{12}\,(1 + \rho_1 F_{11} + \rho_1{}^2 F_{11}{}^2 + \rho_1{}^3 F_{11}{}^3 \ldots)\,X \quad (2)$$

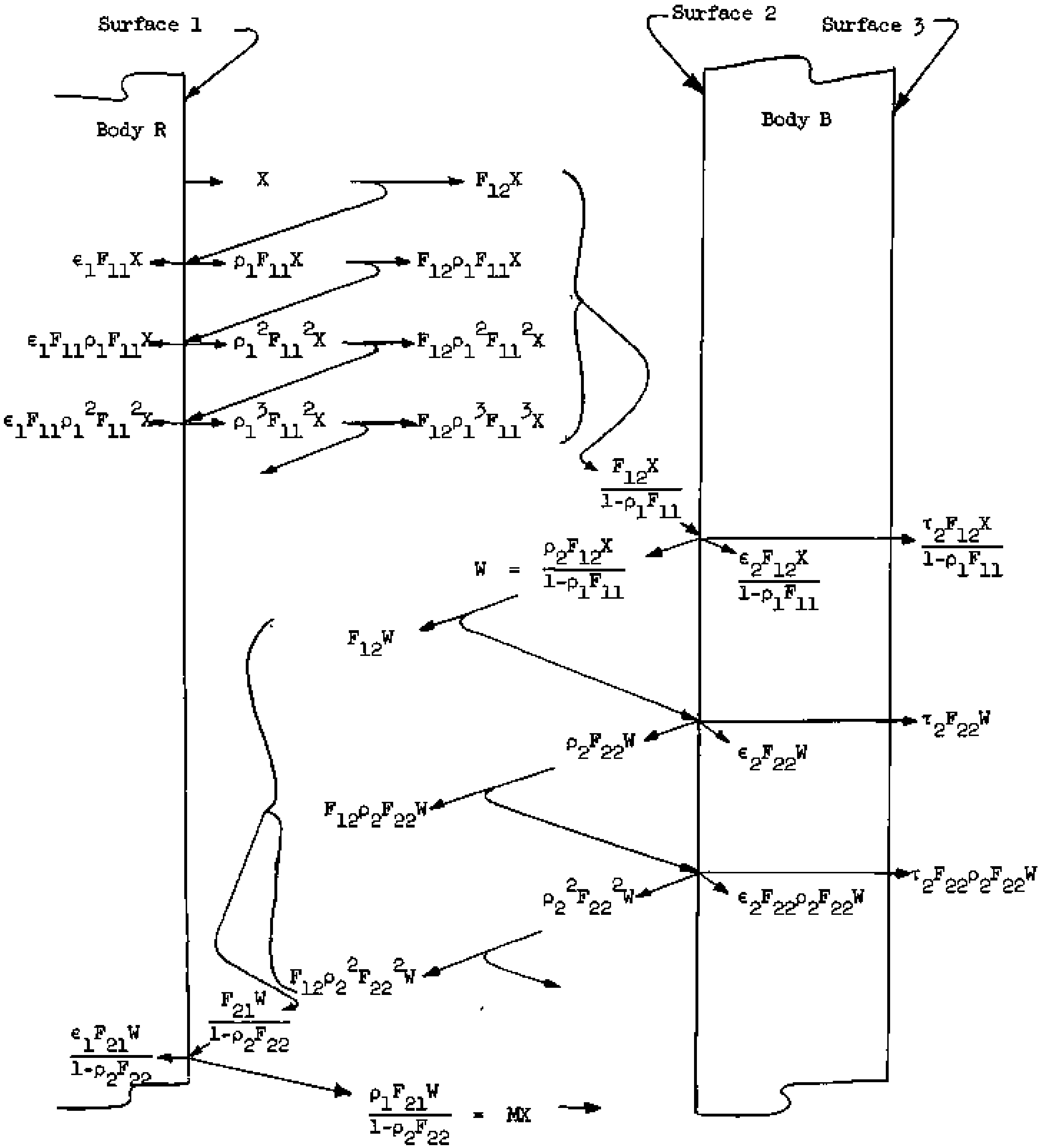

Fig. 2. Thermal Radiation from Surface 1

is actually received at surface 2, while:

$$\epsilon_1 F_{11} (1 + \rho_1 F_{11} + \rho_1^2 F_{11}^2 + \rho_1^3 F_{11}^3 + \ldots) X \tag{3}$$

is absorbed by surface 1 without ever having reached surface 2.

Equation (2), the quantity of radiation actually reaching surface 2 due to an amount X leaving surface 1, can be more conveniently expressed as $F_{12}X/(1 - \rho_1 F_{11})$. Of this quantity $\tau_2 F_{12}X/(1 - \rho_1 F_{11})$ is transmitted through body B to space, $\epsilon_2 F_{12}X/(1 - \rho_1 F_{11})$ is absorbed at surface 2, and $\rho_2 F_{12}X/(1 - \rho_1 F_{11})$ is reflected. Letting:

$$w = \rho_2 F_{12}X/(1 - \rho_1 F_{11}) \tag{4}$$

$F_{21}w$ reaches surface 1, and $F_{22}w$ is intercepted by surface 2. Considering the quantity intercepted at surface 2, $\tau_2 F_{22}w$ is transmitted to space, $\epsilon_2 F_{22}w$ is absorbed at surface 2, and $\rho_2 F_{22}w$ is reflected. This reflection results in an amount $F_{21}\rho_2 F_{22}w$ reaching surface 2 and $F_{22}\rho_2 F_{22}w$ being intercepted at surface 2. Continuing in this manner it can be seen that of the quantity w leaving surface 2:

$$F_{21} (1 + \rho_2 F_{22} + \rho_2^2 F_{22}^2 + \rho_2^2 F_{22}^3 + \ldots) w \tag{5}$$

reaches surface 1:

$$\epsilon_2 F_{22} (1 + \rho_2 F_{22} + \rho_2^2 F_{22}^2 + \ldots) w \tag{6}$$

is absorbed at surface 2; and:

$$\tau_2 F_{22} (1 + \rho_2 F_{22} + \rho_2^2 F_{22}^2 + \ldots) w \tag{7}$$

is transmitted to space.

Replacing w in Eq. (5) by its equivalent, expressed by Eq. (4), and simplifying, the total amount of radiation reaching surface 1 from surface 2 due to a quantity X emitted at surface 1 is given by $\rho_2 F_{12} F_{21} X/(1 - \rho_1 F_{11})(1 - \rho_2 F_{22})$. Of this,

$\epsilon_1\rho_2F_{12}F_{21}X/(1 - \rho_1F_{11})(1 - \rho_2F_{22})$ is absorbed at surface 1, and $\rho_1\rho_2F_{12}F_{21}X/(1 - \rho_1F_{11})(1 - \rho_2F_{22})$ is reflected. This fraction reflected at surface 1 progresses in exactly the same manner as the quantity X, and thus the pattern repeats itself.

Defining:

$$M = \frac{\rho_1\rho_2F_{12}F_{21}}{(1 - \rho_1F_{11})(1 - \rho_2F_{22})} \qquad (8)$$

the fraction of X which is initially reflected at surface 1 after having been reflected once at surface 2 is then MX. If for the moment it is considered that MX progresses no farther, the total absorption at surface 1 and surface 2 and the total transmission to space due to an emission X at surface 1 would be those quantities given by Fig. 3.

The total thermal energy emitted by surface 1 per unit time is $\sigma\epsilon_1A_1T_1^4$. Inserting this quantity for X in Fig. 3, the total absorption at surface 2 resulting from primary emission at surface 1 and all subsequent reflections is given by:

$$\frac{\sigma\epsilon_1\epsilon_2F_{12}A_1T_1^4\ (1 + M + M^2 + \ldots)}{(1 - \rho_1F_{11})(1 - \rho_2F_{22})} \qquad (9)$$

Replacing M by its equivalent expressed by Eq. (8) and simplifying, the total heat transfer by radiation from surface 1 to surface 2 is:

$$q_{1\rightarrow 2} = \frac{\sigma\epsilon_1\epsilon_2F_{12}A_1T_1^4}{(1 - \rho_1F_{11})(1 - \rho_2F_{22}) - \rho_1\rho_2F_{12}F_{21}} \qquad (10)$$

In a like manner the total energy transmitted to space by thermal radiation from surface 1 is given by:

$$q_{1\rightarrow 4} = \frac{\sigma\epsilon_1\tau_2F_{12}A_1T_1^4}{(1 - \rho_1F_{11})(1 - \rho_2F_{22}) - \rho_1\rho_2F_{12}F_{21}} \qquad (11)$$

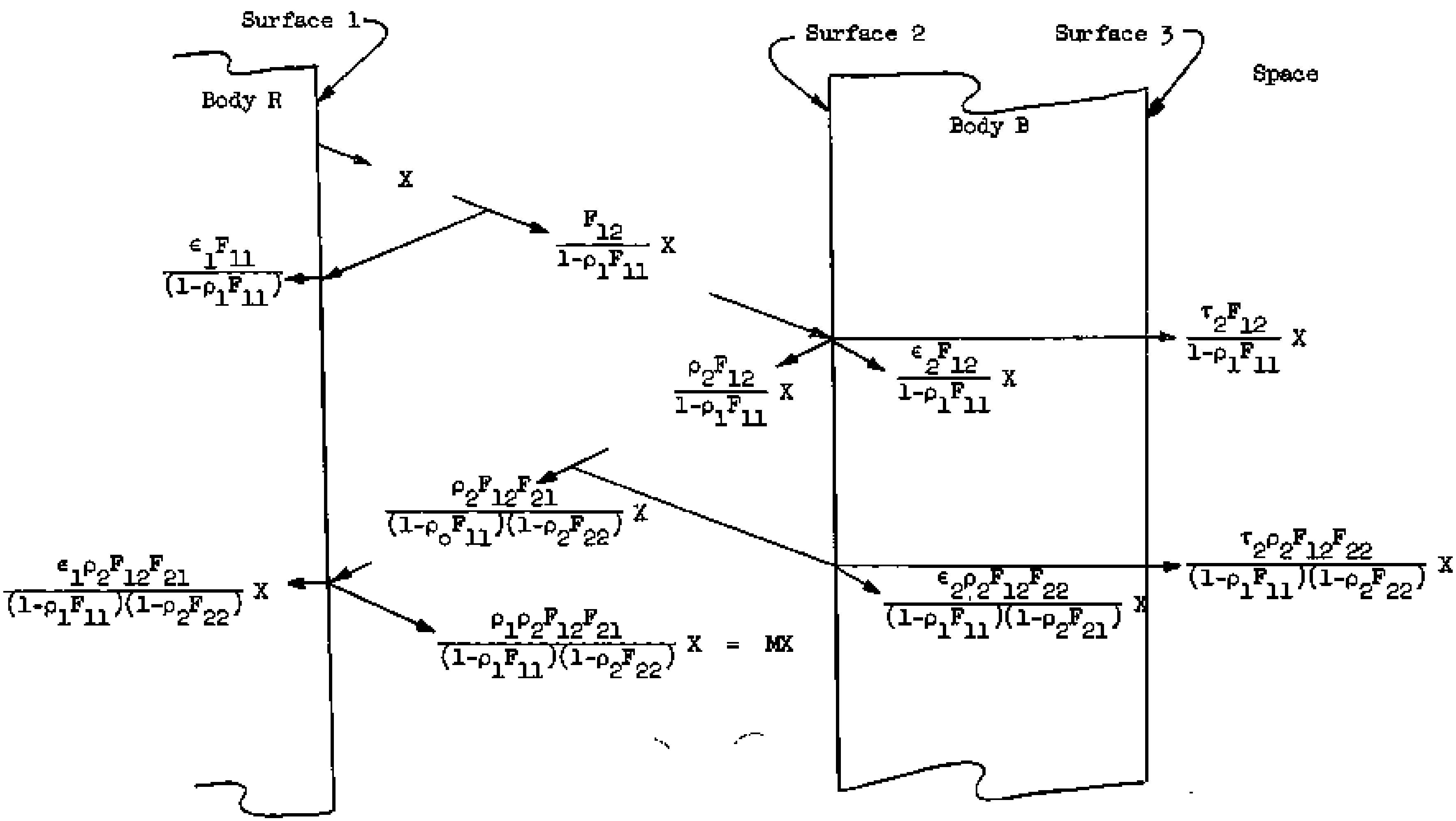

Fig. 3. Distribution of Thermal Radiation from Surface 1

By performing a similar analysis for the total thermal radiation from surface 2 it can be shown that the heat transferred to surface 1 can be expressed as:

$$q_{2 \rightarrow 1} = \frac{\sigma \epsilon_1 \epsilon_2 F_{21} A_2 T_2^{\ 4}}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} \qquad (12)$$

and that the energy transmission to space from surface 2 is given by:

$$q_{2 \rightarrow 4} = \frac{\sigma \epsilon_2 \tau_2 A_2 T_2^{\ 4} \ [F_{22} + \rho_1 F_{12} F_{21} - \rho_1 F_{11} F_{22}]}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} \qquad (13)$$

An analysis of the total thermal radiation from surface 3 reveals for the transfer of energy to space:

$$q_{3 \rightarrow 4} = \frac{\sigma \epsilon_3 A_3 T_3^{\ 4}}{1 - \rho_3 F_{33}} \left[F_{34} + \frac{\tau_2^{\ 2} F_{33} \ (F_{22} + \rho_1 F_{12} F_{21} - \rho_1 F_{11} F_{22})}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} \right] \qquad (14)$$

and for the radiative transfer of heat from surface 3 to surface 1:

$$q_{3 \rightarrow 1} = \frac{\sigma \epsilon_1 \tau_2 \epsilon_3 F_{31} A_3 T_3^{\ 4}}{(1 - \rho_3 F_{33})[(1 - \rho_1 F_{11})(1 - \rho_2 F_{22} - \rho_1 \rho_2 F_{12} F_{21}]} \qquad (15)$$

At steady state the net heat transfer from body R to body B is:

$$q_{RB} = q_{1 \rightarrow 2} - q_{2 \rightarrow 1} - q_{3 \rightarrow 1} \qquad (16)$$

$$q_{RB} = \frac{\sigma \epsilon_1 \epsilon_2 \ (F_{12} A_1 T_1^{\ 4} - F_{21} A_2 T_2^{\ 4}) - \sigma \epsilon_1 \tau_2 \epsilon_3 F_{31} A_3 T_3^{\ 4} / (1 - \rho_3 F_{33})}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{11} F_{22}} \qquad (17)$$

The steady state net heat transfer from body R to space by thermal transmission through body B is simply the radiative heat transfer from surface 1 to space since there is no return radiation from space. Thus:

$$q_{RS} = \frac{\sigma \epsilon_1 \tau_2 F_{12} A_1 T_1^{\ 4}}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} \qquad (18)$$

The net heat transfer from body B to space at steady state is:

$$q_{BS} = q_{2 \to 4} + q_{3 \to 4}$$

$$q_{BS} = \frac{\sigma \epsilon_2 \tau_2 A_2 T_2^{\,4} \, (F_{22} + \rho_1 F_{12} F_{21} - \rho_1 F_{11} F_{22})}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} + \frac{\sigma \epsilon_3 A_3 T_3^{\,4}}{1 - \rho_3 F_{33}}$$

$$\left[F_{34} + \frac{\tau_2^{\,2} F_{33} \, (F_{22} + \rho_1 F_{12} F_{21} - \rho_2 F_{11} F_{22})}{(1 - \rho_1 F_{11})(1 - \rho_2 F_{22}) - \rho_1 \rho_2 F_{12} F_{21}} \right] \qquad (19)$$

The equations developed above apply to any solid body completely enclosed by a second body if the medium outside the outer body is a perfect absorber. For the particular system in question in this study, that of a space radiator enclosed by a semitransparent plastic bumper, the bumper is quite thin and is generally in the shape of an ellipsoid. Thus the following simplifications can be made with little or no error being introduced:

$$A_2 = A_3 \qquad (20\text{-a})$$

$$T_2 = T_3 \qquad (20\text{-b})$$

$$F_{34} = 1.0, \quad (F_{33} = 0, \; F_{31} = 0) \qquad (20\text{-c})$$

$$\epsilon_2 = \epsilon_3 \qquad (20\text{-d})$$

The general equations developed above then reduce to:

$$q_{RB} = \frac{\sigma \epsilon_R \epsilon_B \, (F_{RB} A_R T_R^{\,4} - F_{BR} A_B T_B^{\,4})}{(1-\rho_R F_{RR})(1-\rho_B F_{BB}) - \rho_R \rho_B F_{RB} F_{BR}} \qquad (21)$$

$$q_{RS} = \frac{\sigma \epsilon_R \tau_B F_{RB} A_R T_R^{\,4}}{(1 - \rho_R F_{RR})(1 - \rho_B F_{BB}) - \rho_R \rho_B F_{RB} F_{BR}} \qquad (22)$$

$$q_{BS} = \sigma \epsilon_B A_B T_B^{\,4} \left[1 + \frac{\tau_B \, (F_{BB} + \rho_R F_{RB} F_{BR} - \rho_R F_{RR} F_{BB})}{(1 - \rho_R F_{RR})(1 - \rho_B F_{BB}) - \rho_R \rho_B F_{RB} F_{BR}} \right] \qquad (23)$$

where the subscripts R, B, and S refer to the radiator, bumper, and space respectively.

The total heat that would be dissipated to space by the radiator if the bumper were not present can be shown to be:

$$q'_{RS} = \frac{\sigma\epsilon_R F_{RS} A_R T_R^4}{1 - \rho_R F_{RR}} \tag{24}$$

As a criterion for determining the thermal performance of the bumper, define the thermal efficiency, η, as the total heat dissipated by the radiator with the bumper in place divided by the total heat dissipated by the radiator without a bumper. With the bumper in place the total heat dissipated by the radiator is $q_{RB} + q_{RS}$; thus the thermal efficiency of the bumper is:

$$\eta = \frac{q_{RB} + q_{RS}}{q'_{RS}} \tag{25}$$

$$\eta = \frac{[\epsilon_B (F_{RB}A_R T_R^4 - F_{BR}A_B T_B^4) + \tau_B F_{RB} A_R T_R^4][1 - \rho_R F_{RR}]}{[(1 - \rho_R F_{RR})(1 - \rho_B F_{BR}) - \rho_R \rho_B F_{RB} F_{BR}][F_{RS} A_R T_R^4]} \tag{26}$$

It has been postulated that Lambert's cosine principle applies to the radiation from both the radiator and the bumper; consequently it follows that:

$$F_{RB}A_R = F_{BR}A_B \tag{27}$$

and from the physical geometry of the system it must follow that:

$$F_{RS} = F_{RB} \tag{28}$$

Therefore Eq. (26) can be reduced to the following:

$$\eta = \frac{[1 - (1 - \epsilon_R)(1 - F_{RB})]\left\{\tau_B + \epsilon_B\left[1 - \left(\frac{T_B}{T_R}\right)^4\right]\right\}}{[1 - (1 - \epsilon_R)(1 - F_{RB})](\tau_B + \epsilon_B) + \epsilon_R F_{RB}(1 - \tau_B - \epsilon_B)\frac{A_R}{A_B}} \tag{29}$$

At steady state the net heat transferred to the bumper from the radiator must equal the energy dissipated to space by the bumper. Thus equating Eqs. (21) and (23), an expression can be obtained for the temperature of the bumper at steady state. Expressed as the ratio of T_B/T_R, this expression can be written as:

$$\frac{T_B}{T_R} = \left\{2 + (2\tau_B + \epsilon_B)\left[\frac{A_B}{A_R}\left(\frac{1}{F_{RB}} + \frac{1}{\epsilon_R} - 1\right) - 1\right]\right\}^{-1/4} \qquad (30)$$

Equations (29) and (30) can be used to determine the temperature and thermal efficiency of a bumper enclosing a space radiator for either an opaque or semitransparent bumper provided the emissivity of the bumper is the same on the outer surface and inner surface. It should be noted, however, that Eq. (3) is not valid if the emissivity of the bumper is zero, but such a situation never occurs in reality.

Most space radiator configurations which have been proposed have irregular surfaces and nonuniform surface temperatures. Such a situation is in contrast to the postulated system in the derivation of Eqs. (29) and (30). This difficulty can be eliminated by considering that the total heat which would be dissipated by the actual radiator by radiation to a perfect absorbing media:

$$\sigma\, \epsilon_R \int_{A_R} F_{RS} T_R^{\,4} d\, A_R \qquad (31)$$

is being uniformly emitted from an elastic envelope stretched tightly over the radiator. Such an assumed situation results in a radiator with a pseudoarea A_R' which is the minimum surface formed by replacing all dimples and areas of positive curvature by plane areas. The configuration factor from such a pseudosurface to the bumper is unity, and since the reciprocity theorem must hold for the postulated Lambertian cosine distribution, this pseudoarea can be obtained by:

$$A'_R = \int_{A_R} F_{RS} d\, A_R \tag{32}$$

The uniform pseudotemperature of this surface is obtained by:

$$T'_R = \left[\frac{1}{A_R'} \int_{A_R} F_{RS} T_R{}^4 d\, A_R\right]^{1/4} \tag{33}$$

Employing such an assumed situation, Eqs. (29) and (30) reduce to:

$$\eta = \frac{\tau_B + \epsilon_B \left[1 - \left(\frac{T_B}{T_R'}\right)^4\right]}{\tau_B + \epsilon_B + \epsilon_R \left(1 - \tau_B - \epsilon_B\right) \frac{A_R'}{A_B}} \tag{34}$$

$$\frac{T_B}{T_R'} = \left[2 + (2\tau_B + \epsilon_B)\left(\frac{1}{\epsilon_R} \frac{A_B}{A_R'} - 1\right)\right]^{-1/4} \tag{35}$$

Equations (34) and (35) can be used for any radiator configuration whether the radiator surface temperature is uniform or not, and it is usually more expedient to use this technique.

From Eq. (35) it can be seen that, other quantities being equal, the temperature of the bumper decreases with an incrase in bumper area relative to the radiator area. Since plastic bumpers normally cannot withstand very high temperatures, this implies that such a bumper must be quite large, and even though it may be quite thin, imposes a considerable weight penalty on a space radiator system. Consequently it has been suggested that metallic bumpers which can withstand much higher temperatures be placed in close proximity of the raditor, thus reducing the weight of bumper material required.

The question as to whether it is better from an over-all weight standpoint to use a metallic bumper or semitransparent plastic bumper cannot be answered in completely general terms since it depends upon the thermal properties of the materials involved and the size of the radiator system under consideration. The following analysis is typical and illustrates the criteria governing the choice of bumper material.

Consider a radiator with a heat dissipation requirement of 400 kw. Assume that the configuration, weight, surface area, and mean surface temperature of the radiator needed to dissipate the required heat is known if there were no bumper. Then it follows that:

$$W_R = k_1 A'_R \tag{36}$$

where

W_R = weight of radiator needed to dissipate the required heat with no bumper

A'_R = minimum pseudoarea determined from the known radiation surface area as previously discussed

k_1 = constant of proportionality

The weight of the radiator system required with a bumper covering the radiator then is:

$$W_{RS} = (W_R + W_B)/\eta_B \tag{37}$$

where

W_{RS} = weight of radiator and bumper required

W_R = weight of radiator required when no bumper is used

W_B = weight of bumper for radiator with weight W_R and pseudo-area A'_R

η_B = thermal efficiency of bumper

The bumper will have a uniform thickness and density so it follows that:

$$W_B/A_B = k_2 \tag{38}$$

where A_B is the area of the inner surface of the bumper. (A_B also equals the outer surface of the bumper since A_B is large compared to the bumper thickness.)

Introducing Eqs. (36) and (38) into (37) and rearranging, there is obtained:

$$\frac{W_{RS}}{W_R} = \frac{1}{\eta_B}\left(1 + \frac{k_1}{k_2}\frac{A_B}{A'_R}\right) \tag{39}$$

The thickness of bumper required for effective meteoroid protection is a function of the area of the bumper and the density of the bumper material; also the weight per unit pseudoarea of the radiator is a function of the actual radiator area, but it has been shown that the ratio k_1/k_2 can be assumed to be a constant over fairly wide ranges of radiator and bumper areas and bumper density.[1] It has also been found that this ratio is approximately equal to 0.04 for a radiator system designed to dissipate 400 kw of heat. Thus Eq. (39) can be written:

$$\frac{W_{RS}}{W_R} = \frac{1}{\eta_B}\left(1 + 0.04\,\frac{A_B}{A'_R}\right) \tag{40}$$

Since the bumper is not designed to stop a meteoroid but merely to disperse it either by vaporizing it or shattering it, there must be sufficient distance between the bumper and the radiator to allow for enough dispersion to ensure protection of the radiator wall. Thus there is a lower limit to the ratio A_B/A'_R for the bumper to be effective. Also, since η_B increases with an increase of A_B/A'_R, it can be shown that there is an optimum value of A_B/A'_R for which the weight of the radiator system will be a minimum. Again, it has been found that for the radiator under consideration the optimum value of A_B/A'_R is always greater than the minimum value required for effective meteoroid protection.[1]

The optimum weight of a radiator system employing a bumper which is not temperature-controlled can be determined from Eqs. (34), (35),

[1] R. J. Hefner and P. G. Lafyatis, Protection of Space Vehicles from Meteorite Penetration, ORNL-CF-60-1-67, January 20, 1960.

and (40) if the thermal properties of the radiator and bumper material are known. Table I has been prepared to show the weight ratio of a radiator system with an opaque ($\tau_B = 0$) metallic bumper to the weight of the radiator with no bumper as a function of the emissivity of the bumper material. The emissivity of the radiator has been taken to be 0.95 in this analysis. (It is interesting to note that the thermal efficiency of the bumper increases and consequently W_{RS}/W_R decreases as the emissivity of the radiator decreases if the thermal properties of the bumper remain constant. This should not be interpreted as implying that a radiator with a low emissivity is desirable, because for such a radiator W_R would have to be increased, and the net effect would require a heavier radiator system.)

For a semitransparent plastic bumper Eq. (40) also applies, but A_B/A'_R is restricted by the maximum temperature the plastic can withstand. For most radiator systems this maximum bumper temperature is the controlling factor in determining the optimum value of A_B/A'_R. Figure 4 has been prepared to show the optimum ratio of the weight of the radiator system to the weight of the radiator without a bumper as a function of the transmissivity and emissivity of the bumper, assuming it is required that $T_B \leq 0.5\,\bar{T}_R$. Again, the emissivity of the radiator is assumed to be 0.95.

From Table I and Fig. 4 the choice of whether to use a metallic or plastic bumper can be made if the thermal properties of the materials in question are known. As a basis of comparison assume that the reflectivity of the bumper materials in question are equal. Figure 5 has been prepared to show the range of transmissivity and emissivity for which a radiator system with a plastic bumper would be lighter. If the transmissivity and emissivity of the plastic are within the shaded area of Fig. 5, the plastic bumper would be preferred over a metallic bumper with the same reflectivity. If they are in the

TABLE I

Optimum Weight of Radiator System with Metallic Bumper

($\tau_B = 0$)

(temperature not controlling)

Emissivity ϵ_B	Reflectivity ρ_B	W_{RS}/W_R
1.0	0	1.43
0.9	0.1	1.48
0.8	0.2	1.53
0.7	0.3	1.61
0.6	0.4	1.69
0.5	0.5	1.80
0.4	0.6	1.94

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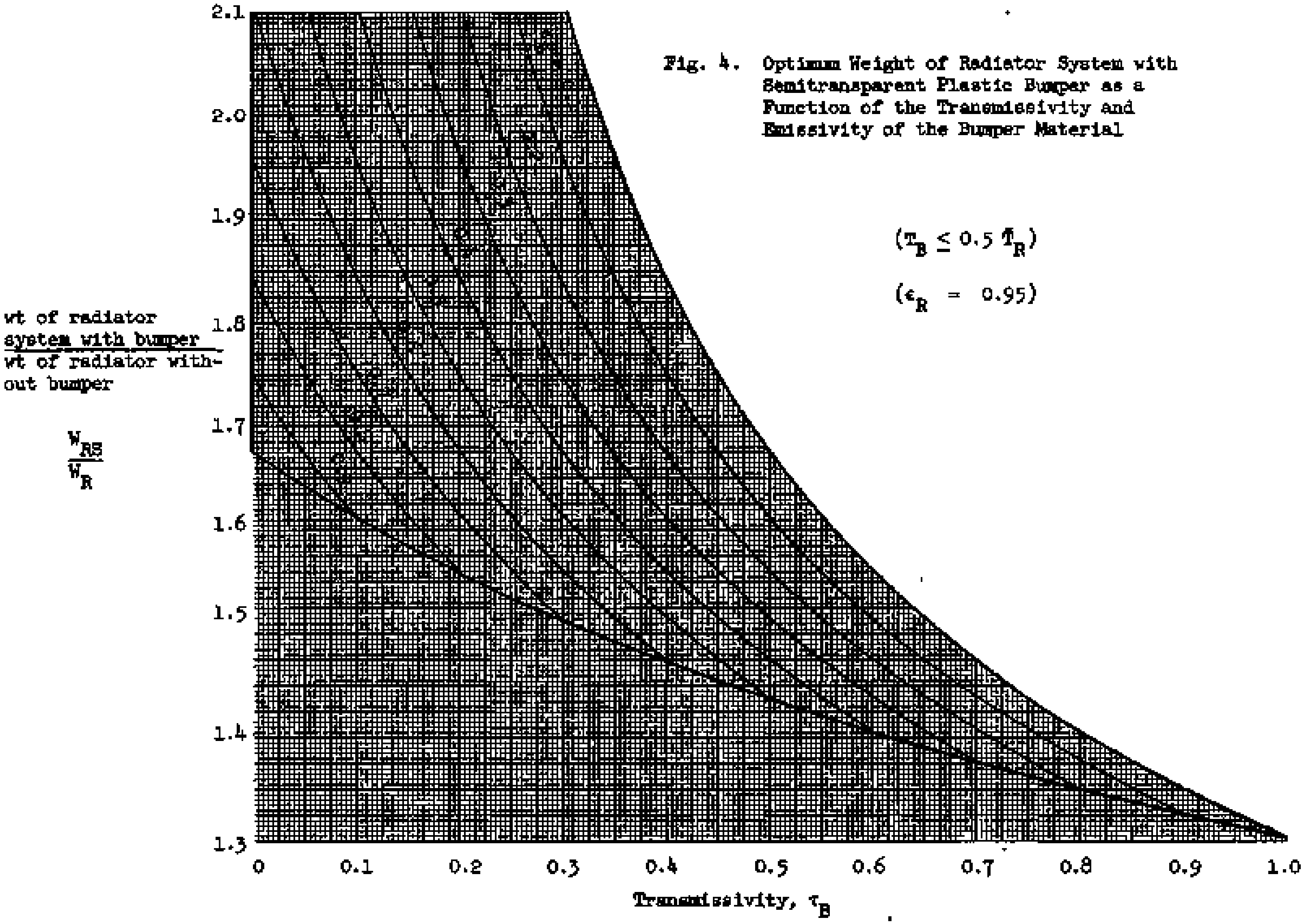

Fig. 4. Optimum Weight of Radiator System with Semitransparent Plastic Bumper as a Function of the Transmissivity and Emissivity of the Bumper Material

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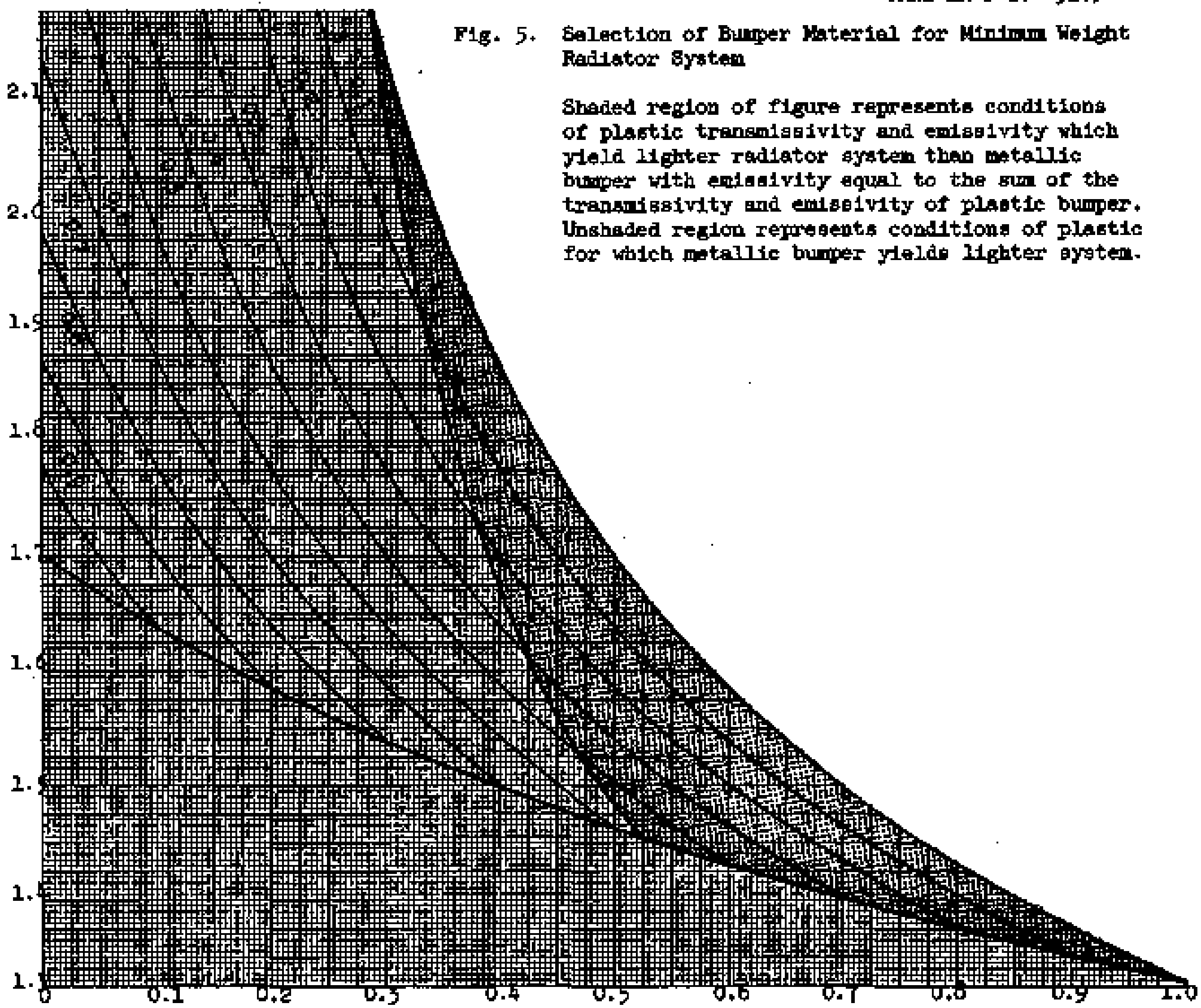

wt of radiator system with bumper / wt of radiator without bumper

$\frac{W_{RS}}{W_R}$

Transmissivity, τ_B

Fig. 5. Selection of Bumper Material for Minimum Weight Radiator System

Shaded region of figure represents conditions of plastic transmissivity and emissivity which yield lighter radiator system than metallic bumper with emissivity equal to the sum of the transmissivity and emissivity of plastic bumper. Unshaded region represents conditions of plastic for which metallic bumper yields lighter system.

unshaded area, the metallic bumper would require the lighter system. Thus, if the transmissivity of the plastic is greater than 0.54 the plastic bumper would yield the lighter system regardless of the emissivity of the metallic bumper or the emissivity of the plastic.

If the transmissivity of the plastic is less than about 0.3, the optimum value of A_B/A'_R is so great that the bumper temperature is no longer controlling, and hence both the metallic and plastic bumpers would yield the same weight system if their reflectivities are the same.

Since there are many plastic materials which have a thermal transmissivity greater than 0.6 for the thicknesses required, it can be concluded that normally a plastic bumper would be preferred over a metallic bumper if only the weight of the required radiator system is the determining factor. This analysis has not considered factors such as fabrication, durability, radiation damage, size, and cost which might also influence the selection of bumper material.

Distribution

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