APPENDIX 2

SEALS: SOME EXPERIMENTAL RESULTS

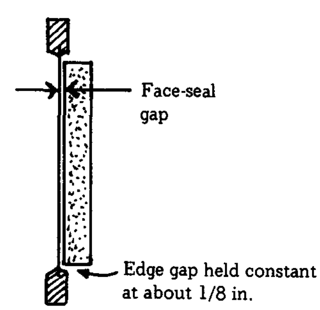
Having found no adequate accounts of how tight the seals at the edges of shutters and shades should be, or how large the penalties are if the seals are not tight, I made some crude experiments early in 1979. The results are summarized below.

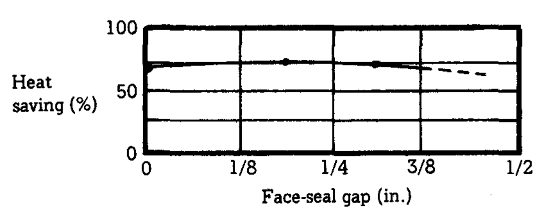
FACE SEAL

To find the tolerance on face-seal gap, I conducted several experiments on an east window of my house in Cambridge, Mass. The window is doubleglazed: it has a main window and, 31/2 in. from it, a storm window. I obtained a 1112-inch-thick sheet of Styrofoam SM and mounted it at various distances (0 in., *1/32* in., *3/16* in., and *5/16* in.) from the main window glazing. Small shims were used to define the spacings. In each case there was an edge gap of *1/8* in. between the edge of the plate and the adjacent member of the sash frame. For each position of the plate, I measured the actual percentage of heatsaving achieved by the plate. The experimental method of evaluating the saving is indicated in Appendix 1.

The results are shown in the accompanying graph. To my surprise, and contrary to rumor, the heat-saving remains very high even when the faceseal gap was increased to *5/16* in. The same general result was found using a l/2-in. plate of Thermax.

I conclude that even when the thickness of the air space between the insulating plate and the glazing is as great as *5/16* in. (and even when there is a 1/8-*in.* edge gap at each edge of the plate), practically no room air circulates into the air space. In other words, if the homeowner attempts to insulate a window by pressing an insulating sheet against the glass, he "can't miss"; even if, for some reason, the plate is *5/16* in. distant from the glass, the effectiveness of the plate is high.

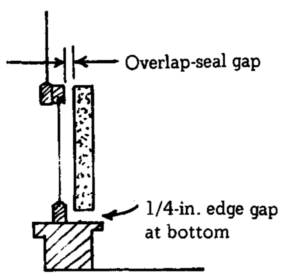




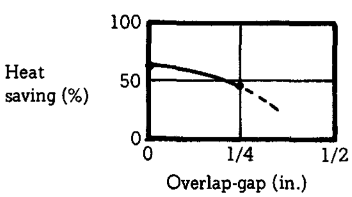
OVERLAP SEAL AGAINST SASH FRAME

In exploring the tolerance on gaps at overlap seals against the sash frame, I used a 1 1/2-in. Styrofoam plate and, in the first test, pressed it firmly against the wooden members of the sash frame. There was then a I-in. air space between the insulating plate and the glass. In a subsequent test I mounted the plate *1/4* in. from the wooden members. In both tests the vertical edges of the plate were wedged tightly between the jambs and there was a 1/4-in. edge gap at the bottom.

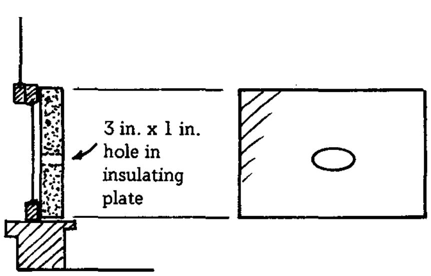
The accompanying graph shows the results. The heat-saving decreased only slightly when there were 1/4-in. gaps at top and bottom. I expect that with 1/8-in. gaps the decrease in heat-saving would have been negligible.



Overlap seal against sash frame



Effect of cutting central hole in the plate Tests made in April 1979 showed that cutting a 3-in. by



Hole that does not affect the heat-saving

l-in. hole in the center of an insulating plate mounted in the above-specified manner (with or without the 1/4-in. gap mentioned) has no detectable effect on the heat-saving if the window is double glazed and reasonably airtight.

OVERLAP SEAL AGAINST FIXED FRAME OF WINDOW

I made no reliable tests on the tolerance that applies here. Presumably the tolerance is smaller than in the above-discussed cases, because, with the insulating plate applied to the face of the fixed frame of the window, the thickness of the air space between the plate and the glazing is several inches, Le., enough to allow greater freedom for circulation of the more-or-less trapped air. Guess: the gap should be kept less than *3/16* in.

EDGE SEAL

Here also I have no reliable data. Guess: the edge-seal gap should be kept less than *1/8* in.

INDEPENDENCE OF THE UPPER AND LOWER HALVES OF THE WINDOW

I found that when just the lower half of the window was provided with a 1/2-in. Styrofoam plate (or with a l/2-in. Thermax plate), the upper half being left entirely without insulation, the heat saving at the lower half was the same as if both halves had

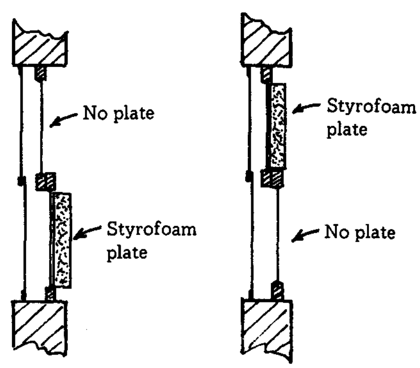


Plate on lower half of window only (at left) and on upper half only (at right). In such case the plate functions with full effectiveness.

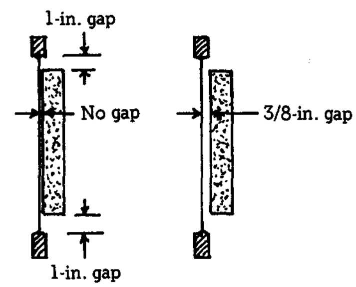
been insulated equally. The converse was true also. In summary, the two halves were found to be independent with respect to use or non-use of an insulating plate. This result was obtained using a fairly airtight window that included main window and storm window wi th a 3 1/2- in. air space between them.

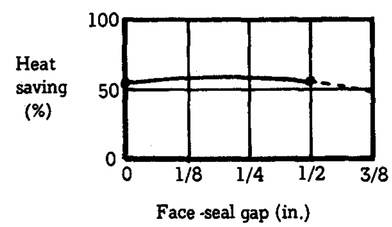
UNDER SIZED PLATE PRESSED AGAINST GLASS

How much is the heat-saving reduced if an insulating plate that is to be pressed against the glazing is somewhat too small, Le., not quite wide enough and not quite high enough?

To answer this question, I made a test in which the 1 1/2-in. Styrofoam plate was considerably undersized: it lacked 2 in. in width and 2 in. in height. When it was pressed against the glass, there was a i-inch -wide area of exposed glass at the top and bottom and also at the left and right. In a second test, this plate was mounted 3/8 in. from the glass.

As shown by the curve in the accompanying graph, the heat-saving was almost identical in the two tests and was only about 15 or 20% less than would be expected if a full-area plate had been used. The conclusion is that no noteworthy harm is done if a plate (pressed more or less closely against the glass) is a few percent undersized. The harm is merely proportional to the shortfall in plate area.





APPENDIX 3

FLOW OF ENERGY FROM A HOTTER FLAT SURFACE TO A NEARBY COOLER FLAT SURF ACE

Designers of thermal shades make much use of thin ~regions of trapped air. Often they employ thin sheets (of aluminum, e.g.) that, with respect to far-IR radiation, have high reflectance and low emittance.

In this appendix I derive the basic equation for flow of radiant energy from one large flat sheet to a nearby parallel sheet, with a thin region of trapped air between. Also I discuss combined flow: simultaneous flow by radiation and other processes.

The subject is complicated, and not enough reliable information is available. The only bright spot is that the laws governing the flow of radiation are highly accurate and fairly easy to understand.

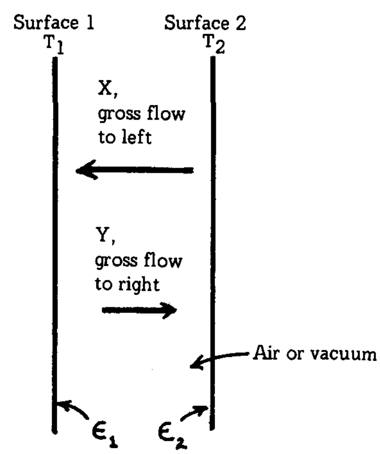
FLOW BY RADIA TION

Derivation ofBasic Equation

Here I derive the basic equation for flow of radiant energy across the gap between two parallel flat surfaces, for example between two flat shades 1 in. apart, or between a flat shade and a flat sheet of glass 2 in. away. The equation applies to any two flat homogeneous surfaces that are parallel to one another provided (1) the material between them (air, ordinarily) has 100% transmittance for far-IR radiation, and (2) the distance between the sheets is small compared to their width and height.

I call the cooler surface Surface 1 and the hotter one (assumed just 1 F degree hotter) Surface 2. The respective emittances for far-IR are e_1 and 22 and the respective reflectances for far-IR are (1 ~ ed and (1 - e2)'

The basic physical fact is that each square foot of such surface emits, per hour, eCT^4 of radiant energy. C is 1.71 X 1O-9. T^4 is the fourth power of the absolute temperature; for example, if the temperature is 70°F, i.e., 529.6 on the absolute (Rankine) scale, then T^4 is (529.6)4, that is, 7.87×10^{10}• First I deal with the gross flows X and Y from each surface toward the other. Then I find the difference: the net flow.



Symbols used for gross flows of energy

Common sense indicates that the gross flow from Surface 2 to Surface 1 is:

$$\begin{pmatrix}\text{Amount emitted by}\\ \text{Surface 2}\end{pmatrix} + \begin{pmatrix}\text{Gross}\\ \text{counterflow}\end{pmatrix} \times \begin{pmatrix}\text{Reflectance of}\\ \text{Surface 2}\end{pmatrix}$$

i.e.: $X = \epsilon_2 C T_2^4 + Y(1 - \epsilon_2)$

It is obvious also that the gross flow from Surface 1 to Surface 2 is:

$$\begin{pmatrix}\text{Amount emitted by}\\ \text{Surface 1}\end{pmatrix} + \begin{pmatrix}\text{Gross}\\ \text{flow X}\end{pmatrix} \times \begin{pmatrix}\text{Reflectance of}\\ \text{Surface 1}\end{pmatrix}$$

i.e.: $Y = \epsilon_1 C T_1^4 + X(1 - \epsilon_1)$

To simplify this pair of simultaneous equations, I eliminate T_2 by expressing it as $(T_1 + 1)$ and I express T_2^4 as $(T_1 + 1)^4$—which, if T_1 is extremely large compared to 1, may be reduced to $(T_1^4 + 4T_1^3)$.

The equations become:

$$X = \epsilon_2 C T_1^4 + \epsilon_2 C 4 T_1^3 + Y(1 - \epsilon_2)$$
$$Y = \epsilon_1 C T_1^4 + X(1 - \epsilon_1)$$

Solving this pair of equations for the gross flows, I obtain:

$$\text{Gross flow to left} = X = CT_1^4 + \frac{\left(\frac{1}{\epsilon_1}\right) 4CT_1^3}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}$$

$$\text{Gross flow to right} = Y = CT_1^4 + \frac{\left(\frac{1}{\epsilon_1} - 1\right) 4CT_1^3}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}$$

Subtracting, to obtain the *net flow*, I arrive at this famous equation:

$$\text{Net flow} = (X - Y) = \frac{4CT_1^3}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}$$

Btu per square foot per hour,
with temperature difference of 1 F degree

Note concerning the thickness of the gap between the two surfaces: The gap thickness is irrelevant to the flow of radiation. Whether the gap is 1/2 in., or 3 in., the flow is identical, provided that the widths and heights of the surfaces are very much greater than 3 in. The only relevant quantities are· the temperatures and *emittances* of the surfaces. (Strictly speaking, the reflectances too are important; but a reflectance can be expressed as (1 - emittance) and I have written the equation in such a way that emittances, but not reflectances, appear explicitly.) Of course, if the space between the surlaces were filled with black smoke, or black lquid, the situation would be entirely different: the flow would then depend strongly on the gap thickness.

The quantity

$$\frac{1}{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}$$

is often called the *effective* emittance, E, of the pair of surfaces. The main equation may be rewritten thus:

Net Flow by Radiation ~ 4ECT3.

or, in general, delEcT^3 (aT).

Illustrative Examples

Example 1 What is the net flow when $\epsilon_1 = \epsilon_2 = 0$, i.e., when both surfaces are perfect reflectors?

Answer Setting ϵ_1 and ϵ_2 equal to zero in the main equation, one finds that the denominator is infinitely large. Thus the quantity as a whole is zero.

Net flow = 0.

Example 2 What is the net flow when $\epsilon_1 = \epsilon_2 = 1$, i.e., when both surfaces are perfect emitters and absorbers, and $\Delta T = 1$?

Answer Setting ϵ_1 and ϵ_2 equal to 1, one finds that the denominator has the value 1. Thus the quantity as a whole is simply $4CT_1^3$.

Net flow = $4CT_1^3$.

If $T_1 = 70°F$, or 529.6 absolute, the net flow is

$$4(1.71 \times 10^{-9})(529.6)^3 = 1.01 \text{ Btu/(ft}^2 \text{ hr °F)}.$$

Example 3 Consider the case where $\Delta T = 1$ and $\epsilon_1 = \epsilon_2 = 0.5$. What is the net flow in this case? Here the denominator becomes:

$$\left(\frac{1}{0.5} + \frac{1}{0.5} - 1\right) = (2 + 2 - 1) = 3$$

and the quantity as a whole is:

$$\text{Net flow} = \frac{4CT_1^3}{3}.$$

If Tl is 70°F, the net flow is 33 BtU/(ft2 hr OF). In other words, cutting the emittances in half reduces the net flow to one third.

Tabulations

The accompanying tables present illustrative values of the net flow of radiant energy from a flat vertical surface at 70°F to a nearby parallel surface at 69°F. Various values of emittances are used. Radiation resistances values, discussed in a later paragraph, are included also.

Emittances (pure no.)		Net flow of Radiant energy (Btu/(ft^2 hr °F))	Radiation resistance ((ft^2 hr °F)/Btu)
Emittances are the same, i.e., $\epsilon_1 = \epsilon_2$			
0.0		0.000	Infinity
0.1		0.053	19
0.2		0.11	9.1
0.3		0.18	5.5
0.4		0.25	4.0
0.5		0.33	3.0
0.6		0.43	2.3
0.7		0.54	1.9
0.8		0.67	1.5
0.9		0.82	1.2
1.0		1.01*	0.99*
ϵ_1	ϵ_2		
0.2	0.2	0.11	9.1
0.2	0.4	0.15	6.5
0.2	0.6	0.18	5.7
0.2	0.8	0.19	5.3

Emittances (pure no.)		Net flow of Radiant energy (Btu/(ft^2 hr °F))	Radiation resistance ((ft^2 hr °F)/Btu)
Emittances are the same, i.e.,			
ϵ_1	ϵ_2		
0.4	0.2	0.15	6.5
0.4	0.4	0.25	4.0
0.4	0.6	0.32	3.2
0.4	0.8	0.36	2.8
0.6	0.2	0.18	5.7
0.6	0.4	0.32	3.2
0.6	0.6	0.43	2.3
0.6	0.8	0.52	1.9
0.8	0.2	0.19	5.3
0.8	0.4	0.36	2.8
0.8	0.6	0.52	1.9
0.8	0.8	0.67	1.5

* The fact that this number is almost (but not quite) 1.00 is a coincidence stemming from the definitions of Btu, foot, and hour. The numbers are not ratios but absolute physical amounts.

Inspection of the tables reveals these interesting facts:

When both emittances are small, the relative amount of radiant energy flowing is much smaller yet. For example, if each emittance is 0.10, the energy flow is only 0.05 times the maximum flow.

When one emittance is large and one is very small, the latter governs. Thus if the emittances are 0.1 and 0.9, the net flow is only about 0.1 times the maximum flow.

When both emittances are very large, the relative flow is only slightly less than the smaller of the emittances. Thus if both emittances are 0.8, the net flow is 0.67 times the maximum amount.

Actual Values of F ar- **IR** Emittance

The amount of reliable information readily available on the actual far-IR (4-to-40 microns) emittances of materials used in windows, shutters, shades, and room furnishings is very small. Some information may be found in the ASHRAE 1977 Handbook *of Fundamentals,* p. 22.11 and in Infrared Systems Engineering by R. D. Hudson, Wiley Co. (1969).

Some representative values are:

	Far-IR emittance
Aluminum	
Foil, dull side	0.030
Foil, shiny side	0.036
Sheet, regular	0.09 to 0.12
Sandblasted	0.21
Anodized	0.77
Steel	
Stainless (18-8)	0.44
Stainless (18-8) buffed	0.16
Galvanized, bright	0.25
Silver, polished	0.03
Tin plated onto steel	0.07
Copper	
Polished	0.05
Heavily oxidized	0.78
Glass	
Ordinary	0.84
Polished plate	0.84
Fiberglass batt	0.75
Common building materials, e.g., wood, paper, concrete, brick, plaster, ordinary paint	0.9
Miscellaneous: sand, soil, wet soil, water, human skin	0.90 to 0.95
Foam-type insulating materials	0.90 (per my guess)

Concept of Resistance to Flow of Radiation

Engineers normally draw an analogy between (1) flow of radiant energy from one large flat surface, via a region of air, to a nearby parallel flat surface, and (2) flow of thermal energy through a slab of solid opaque material. Consider first a 2inch-thick slab of Styrofoam. Suppose that the two surfaces differ in temperature by 1 F degree. Then the amount of heat that flows is about 0.1 Btu/(ft^2 hr ~F). The reciprocal of this, i.e., 10 (ft^2 hr °F)/Btu, is called the conductive *resistance.*

Consider now two parallel flat surfaces (at 70°F and 69°F) with an air gap between them. Suppose that the emittances are 1.0. Then, as explained in previous paragraphs, the flow by radiation is 1.01 Btu/(ft2 hr OF). The reciprocal of this is 0.99 (ft^2 hr °F)/Btu. Engineers like to call this the *radiation resistance* Tt of the pair of surfaces and the intervening gap.

In the general case, the reciprocal of the abovederived main equation is called the radiation resistance of the pair of surfaces and intervening gap. That is, whatever the surfaces consist of-whatever the emittances-the term radiation resistance is applied to the reciprocal quantity:

$$\text{Radiation resistance} = R_r = \frac{\left(\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1\right)}{4CT_1^3}.$$

To a physicist, such terminology may be offensive, because radiation traveling through a vacuum or air encounters virtually no resistance. If, in flow-by-radiation situations, there is anything truly analogous to resistance, it resides in the surfaces themselves, i.e., in the detailed process of emitting and absorbing radiation.

Note that the radiation resistance varies with the temperatures of the surfaces. The hotter they are, the more energy flows and the lower the radiation resistance.

General equation In general, two parallel surfaces may differ in temperature by an amount aT that may be much larger than 1 F degree. The general (approximate) equation that applies is:

$$\begin{aligned}\text{Net flow by radiation} &= 4ECT^3(\Delta T)\\ &= 6.85 \times 10^{-9}ET^3(\Delta T)\ \text{Btu/(ft}^2\text{ hr)}\end{aligned}$$

where T is the *average* Rankine temperature of the two surfaces and aT is the temperature difference.

FLOW BY RADIATION, CONVECTION, ETC., IN PARALLEL

If there is air in the space between the two parallel surfaces, two kinds of flow occur simultaneously: flow by radiation and flow by ordinary convection. They occur independently. If there were no air in the intervening space (i.e., if there were a vacuum there), the radiant flow would continue as before, but there would be no convective flow. If there were air between the two surfaces but the emittances of the surfaces were somehow made to be zero, as by some ideal silvering, the radiative flow would cease but the convective flow through the intervening air would continue.

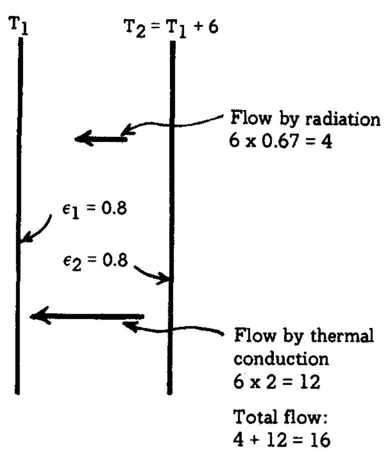
The simultaneous flows by radiation and convection are called parallel flows because each starts at the same surface (Surface 2) and ends at the same surface (Surface 1) and the flow mechanisms are independent.

Because the flows are in parallel, the combined conductance is easily found, being simply the sum of the individual conductances. The actual total energy flow with any given temperature difference across the system is the product of the total conductance and the temperature difference.

To find the combined resistance, one merely obtains the reciprocal of the combined conductance.

Example Consider two parallel surfaces, at 70°F and 76°F, with an intervening region of trapped air. Assume that each surface h~as an emittance of 0.8. Assume that the intervening region of air is thin enough to have a purely thermal conductance of 2. How much energy will flow?

Answer The radiation conductance (with emittances of 0.8) is 0.67. The thermal conductance is 2.0. Thus the total conductance is 2.67. The temperature difference is 6°F. Thus the total rate of energy flow is 6 x 2.67 = 16 Btu/(ft^2 hr).



Example of parallel energy flow between two surfaces

The accompanying table shows the combined conductance values (and combined resistance values) of a pair of parallel vertical surfaces with an intervening region of air-for various values of effective emittance and various thicknesses of air space. In each case the average temperature of the system is 50°F and the temperature difference across the system is 30 F degrees; thus T ı and T z are 35°F and 65°F. (Note: When Tl and Tz are much lower, say 0 and 30°F respectively. the combined resistance is about 10 to 25% greater because the thermal conductance of the air is less.)

Effective emittance E	Thickness of air space			
	0.5 in.	0.75 in.	1.5 in.	3.5 in.
	Combined conductance, Btu/(ft^2 hr °F)			
0.05	0.41	0.36	0.41	0.39
0.20	0.54	0.50	0.54	0.53
0.50	0.81	0.77	0.81	0.80
0.82	1.11	1.11	1.11	1.11
	Combined resistance, (ft^2 hr °F)/Btu			
0.05	2.46	2.77	2.46	2.55
0.20	1.84	2.01	1.84	1.89
0.50	1.23	1.30	1.23	1.25
0.82	0.90	0.90	0.90	0.91

Source: ASHRAE *Handbook of Fundamentals* 1977, p. 22.12.

Inspection of the data suggests that:

Decreasing the effective emittance from 0.82 to 0.05 increases the combined resistance greatly and reduces the combined conductance greatly.

Changing the airfilm thickness over a wide rane 0.5 to 3.5 in.) has ractically no effect on the combined resistance or conductance.

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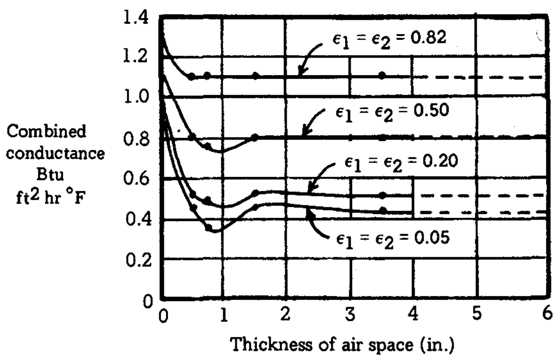
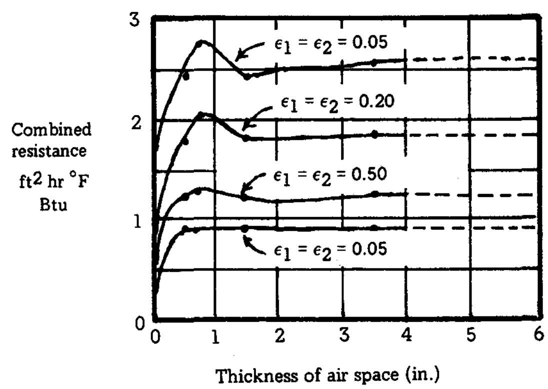
The accompanying graphs make the tabulated data easier to grasp. In preparing the graphs I have assumed (guessed) that the thermal conductance (and combined conductance) increases rapidly when the distance between the surfaces is reduced from 1/4 in. to smaller values.





FLOW WHEN A THICK OPAQUE PLATE IS INVOLVED ALSO

If a pair of parallel surfaces and intervening air gap is in series with an ordinary insulating plate, one finds the total resistance merely by adding the two resistances. The total conductance is the reciprocal of this. The energy flow is the product of the temperature difference and the total conductance.

Example Consider two parallel surfaces (with 1.5 in. of air between) and, immediately adjacent to them, an R-10 Styrofoam plate. Suppose that the individual emittances of the surfaces are 0.67, with the consequence that the effective emittance of the pair is:

1
E = 1/067 + 1/067 ~ 1 0.50.

Then one finds from the table that the combined resistance of the pair of surfaces and intervening air is 1.23. Inasmuch as the resistance of the Styrofoam plate is 10, the overall resistance is 11.23. The reciprocal of this, Le., the overall conductance, is about 0.09. Therefore the energy flow is merely the product of the overall temperature difference and 0.09 Btu/(ft2 hr OF).

FLOW PERTINENT TO A SINGLE SURFACE

This subject is discussed in Chapter 2.

A further fact concerning an outdoor airfilm flanking a vertical wall or vertical sheet of glass is provided by a National Bureau of Standards report "Retrofitting Existing Housing for Energy Conservation: an Economic Analysis," by S. R. Peterson, Dec. 1974. 70 p. SD Cat. No. C13-29/2 :64. $1.35. R for surface and airfilm depends on the outdoor windspeed approximately according to this formula:

$$R = \frac{4}{8 + \left(\frac{\text{actual windspeed}}{\text{1 mph windspeed}}\right)}$$

This implies that for windspeeds of 0, 5, 10, 15, 20, and 40 mph the R-values are 0.5, 0.31, 0.22, 0.17, 0.14, and 0.08 respectively.