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NATIONAL BUREAU OF STANDARDS REPORT

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THE THERMAL INSULATING VALUE OF AIR SPACES

by

H. E. Robinson
F. J. Powlitch

Report to

Division of Housing Research
Housing and Home Finance Agency

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

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to

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ABSTRACT

This report presents the results of measurements of the heat transfer coefficients for air spaces of various emissivities and thicknesses. The data were obtained for five different orientations and directions of heat flow, ranging from heat flow vertically upward to heat flow vertically downward, and encompass a range of values of temperature difference across the air space.

The total coefficient for heat transfer across an air space is shown to be the sum of a coefficient for heat transfer by radiation and a coefficient for heat transfer by combined convection and conduction. These two coefficients depend upon different variables characterizing the air space. Curves are given for computation of the radiation coefficient for spaces of various emissivities and mean temperatures. Other curves give values of the coefficient for convection and conduction combined for air spaces of different orientations and directions of heat flow and for various air-space thicknesses and temperature differences. Equations are given for adjusting this coefficient for different mean temperatures.

The method of utilizing the data to calculate the insulating value of a particular air space, or of air spaces in a building construction, is illustrated by several examples of general interest. A description is given of the guarded hot box heat transfer apparatus and of the test air spaces used to obtain the data presented in this report.

I. INTRODUCTION

In 1945, at the request of the Federal Public Housing Authority, the National Bureau of Standards reviewed published data on the thermal conductance of reflective and nonreflective air spaces and submitted a summary of representative values. These served as a basis for FPHA Bulletin No. 19 "Reflective Insulation", issued in 1945, and were used for reflective insulations in Federal Housing Administration Technical Circular No. 7 "Calculation of Building Section Heat Transmission Coefficients", revised in January 1947.

It was recognized in 1945 that the increased availability of aluminum after the war would lead to greater utilization of aluminum foil and reflective surfaces to form thermally insulating air spaces in buildings. However, the review referred to indicated that the then available published data were limited and incomplete in regard to total heat transfer coefficients for air spaces of various emissivities and with various orientations and directions of heat flow, and for the ranges of air space thickness, temperature difference, and mean temperature encountered in building insulation service.

The total heat flow across a plane air space is the result of heat flow by radiation, and heat flow by convection and conduction combined, between its facing surfaces. There are theoretical reasons for regarding these heat flows as taking place independently. Consequently, the coefficient for the total heat transfer across a plane air space can be considered as the sum of two coefficients:

(a) A radiation coefficient which depends only upon the emissivities and temperatures of the surfaces, and

(b) A coefficient for convection and conduction combined which does not depend upon the emissivities of the surfaces but does vary with the thickness of the space, its orientation, and the direction of heat flow and with the temperatures of the surfaces, assuming the latter are reasonably smooth.

Thus, a flexible means of calculating the total heat transfer coefficients for a wide variety of air spaces would be provided if data as to the two component coefficients were made available.

The radiation coefficient can be calculated readily, by means of the Stefan-Boltzman law, for given surface emissivities and temperatures. However, complete and consistent data as to the convection-conduction coefficient, over a wide range of pertinent conditions, were not available at the time of the review mentioned, or later.

Accordingly, at the request of the Housing and Home Finance Agency, and with its financial support, an apparatus of the guarded hot box type was constructed during 1948, suitable for the purpose of determining the heat transfer coefficients of air spaces of the kind used in buildings. After use for other measurements desired by the supporting agency, the apparatus became available in 1950 for conducting the extensive series of tests required for the above purpose, the results of which are summarized in this report.

2. PRELIMINARY CONSIDERATIONS AND METHOD OF INVESTIGATION

Heat flows across an air space, from the warmer to the cooler surface, by radiation and by convection and conduction combined. The component due to radiation depends only upon the absolute temperatures of the facing surfaces of the space, and upon its effective emissivity. The component due to convection and conduction combined, however, depends upon a number of other factors, such as orientation, temperature difference, and dimensions of the space. There are good theoretical reasons for believing that for given air space surface temperatures, the two components are independent of each other, and that, therefore, the thermal conductance or coefficient for the total heat transfer across an air space by all modes combined, C, can be regarded as the sum of two independent coefficients, as indicated in the following equation:

$$C = Eh_r + h_c \qquad (1)$$

The two terms on the right-hand side of equation 1 represent, respectively, the radiation coefficient and the coefficient for convection and conduction combined.

The coefficient for heat transfer by radiation across an air space, per unit of temperature difference of its surfaces, is given by Eh_r, where E is the effective emissivity of the space and, in Btu/hr ft^2 (deg F),

$$h_r = \frac{0.172 \times 10^{-8} \left(T_1^{\,4} - T_2^{\,4}\right)}{(T_1 - T_2)} = 0.00686 \left(\frac{T_m}{100}\right)^3 \text{ approx.} \qquad (2)$$

T_1 and T_2 are the absolute temperatures (°F + 460°) of the warmer and cooler surfaces, respectively, of the air space, and t_m equals $1/2(T_1 + T_2)$. Values of h_r for various air space mean temperatures (expressed in Fahrenheit degrees) are given by the curve of Figure 1.

The effective emissivity, E, of an air space of uniform thickness, and of extent large compared to its thickness, is calculated by means of the equation

$$\frac{1}{E} = \frac{1}{e_1} + \frac{1}{e_2} - 1 \qquad (3)$$

where e_1 and e_2 are the total emissivities of the facing surfaces. Values of E for various values of e_1 and e_2 are given by the curves of Figure 2.

The component of heat transfer across a plane air space due to convection and conduction depends upon the orientation of the air space and the direction of heat flow, on its thickness and the temperature difference across it and, to a lesser extent, upon the mean temperature of the space.

The heat transfer by convection and conduction combined cannot be less than would take place by conduction alone through still air, but when convection occurs in the space, the heat transfer may increase materially above that value. The coefficient for heat transfer by convection and conduction combined, per unit of temperature difference of the surfaces, can be represented by h_c, the value of which depends upon the circumstances enumerated above.

Since values of h_c cannot be calculated analytically at present, it was necessary to determine them experimentally. To do this, measurements were made on a number of air spaces under various conditions of temperature and orientation to determine for each condition the total heat transfer coefficient, or thermal conductance, C, of the space, per unit of temperature difference between its facing surfaces. The value of the radiation coefficient, Eh_r, was computed for each measurement, using equations 2 and 3 and values of e_1 and e_2 determined by separate measurements made on samples of the materials which formed the surfaces of the air spaces used. The value of h_c for a particular test condition was obtained by subtracting Eh_r from the total coefficient, C, in accordance with equation 1. The dependence of the value of h_c upon the various geometric and temperature conditions was then determined by dimensional analysis of the collected data. This analysis led to a method of making adjustments of the results to correspond to one mean temperature (50° F was selected), and to equations for adjusting values at 50° F to other mean temperatures, as set forth later.

In most of the measurements for determining h_c, the air spaces used had low effective emissivities, E, so that the quantity Eh_r was a small fraction of the magnitude of the measured quantity C or the derived quantity h_c. However, tests were also made with spaces having large and intermediate values of E to test the experimental validity of equation 1.

3. TEST APPARATUS AND TEST METHOD

The heat transfer measurements were made on air spaces in test panels by means of a guarded hot box apparatus, shown schematically in the drawing of Plate 1 and in the photograph of Plate 2. The apparatus was designed for panels 5 1/2 by 8 ft in size, up to one foot thick, and as shown by the photograph, was mounted on horizontal trunnions which allowed the entire apparatus to be rotated to position the plane of the panel at any angle up to 90° from the vertical in either direction.

The hot box apparatus conformed substantially to the requirements of ASTM 236-53 "Method of Test for Thermal Conductance and Transmittance of Built-Up Sections by Means of the Guarded Hot Box", except for the additional feature of rotatability on trunnions.

As shown in Plate 1, the apparatus consisted of a cold box (3) and a warm or guard box (4), between which the test panel (1) was interposed. The two boxes were held together by means of long bolts engaging lugs around the periphery of the joint, as shown in the photograph. The guard box contained a five-sided metering box (5), the open face of which was pressed against the face of the panel by four compression springs (14). The contact between the metering box and the panel was made substantially airtight by the rubber gasket (2), the center line of which bounded the centrally located "metering area" of the panel, a rectangle 60 inches high and 32 inches wide, through which heat flow was measured during a test. The remainder of the panel area constituted a "guard", through which heat flow was not measured, but which protected the metering area against lateral heat flow. Peripheral insulation (13) was placed at the edges of the panel to minimize lateral heat flow.

During measurements, the guard and metering boxes were held at the same selected constant temperature by electric heaters (7) controlled by sensitive thermostats in each box. The cold box was held at the desired constant lower temperature by means of the refrigerating coil (6) and a small electric reheating coil (7) controlled by a thermostat. The air in each box was circulated continuously at a moderate rate by the fans (11). In the metering and cold boxes, the circulated air was constrained by the baffles (12) to pass downward and upward, respectively, along the surface of the panel. The direction of air flow on each side was the same as would result from natural convection at the panel faces. The average air velocity in the metering box baffle space was about 39 ft/min, and in the cold box baffle space was about 92 ft/min. Air temperatures in the baffle spaces and at other locations in the boxes were measured by means of copper-constantan thermocouples (10) as indicated.

All electrical input to the heaters, fan, and other devices in the metering box was measured by means of a calibrated watt-hour meter readable to one watt-hour. Under steady temperature conditions, and with the guard box temperature adjusted to correspond to the metering box temperature, so that there was no average temperature difference or heat interchange through the metering box walls, all of the heat furnished to the metering box was transmitted to the cold box through the known metering area (13.33 sq ft) of the panel. The average rate of heat flow per square foot of metering area could thus be determined.

To assure that there was little or no heat interchange between the metering box and the surrounding guard box, a compound thermocouple (9) was used, consisting of 10 series-connected differential thermocouples with the junctions of each pair cemented opposite each other in grooves in the surfaces of the metering box. The metering box, except for the hardwood nosepiece which carried the gasket (2), was made entirely of selected balsa wood, glued and doweled, forming walls 2.5 in. thick with a nearly uniform conductance of 0.16 Btu/hr ft^2(deg F). It was determined by calibration measurements that for a reading of 100 microvolts on the compound thermocouple (9), the heat flow between metering and guard boxes was approximately 3.5 Btu/hr, or 0.26 Btu/hr per square foot of panel metering area. For most tests of this investigation, the average reading of this thermocouple during a test was less than 30 microvolts, indicating that heat interchange between guard and metering boxes was a very small percentage of the total heat input; nevertheless, for each test the observed heat input was corrected for such interchange.

The guard-box temperature was made to correspond automatically to that of the metering box by means of the compound thermocouple (8) consisting of 10 differential thermocouples in series arranged similarly to those of compound thermocouple (9) but with their junctions in the air about one inch from the surfaces of the metering box. The emf from the compound thermocouple (8) controlled a relay which supplied heat to the guard box as required. This system of control was very sensitive, and automatically kept the reading of compound thermocouple (9) steadily near zero throughout the tests.

The temperatures of the surfaces of the test panels bounding the air spaces were determined by means of 10 thermocouples on each surface of the panel, as indicated in the description of the test panels (Plate 3).

All test measurements were made under steady temperature conditions following a prior period during which these conditions were attained. The duration of the steady state test period, during which the periodic observations were taken from which the results were calculated, was in no case less than 16 hours, and in most tests was

more than 20 hours. All thermocouples were read manually by means of a potentiometer with which temperature changes of less than 0.05 deg F were readable. The steadiness of temperatures during almost all of the tests was such that average air or panel surface temperatures did not vary as much as 0.2 deg F during the test period.

The average test conductance of the panel space was calculated from the average net heat flow into the panel, per square foot of metering area, and the average temperature difference between the panel surfaces. Since about 5 percent of the metering area included the wood members bounding the metering area air space, a small correction was made to allow for the effect of heat conduction through the wood. The adjusted conductance thus obtained represented the average thermal conductance, C, of the panel air space, or one half the average conductance of each air space if the panel space was divided into two approximately equal spaces in tandem.

4. TEST PANELS AND AIR SPACES

The air spaces used were of various thicknesses and had various effective emissivities due to the use of different materials for their bounding surfaces. In order to vary the air-space thicknesses, several test panel frames of clear fir, substantially alike except for thickness, were used, as illustrated in Plate 3.

The test area was the 32- by 60-inch rectangle at the center of the panel, and included half of the wood bounding the test air space, which constituted 5 percent of the test area. The other air spaces of the panel constituted a guard area and were made similar to the test space in all possible respects.

The faces of the panels were made of 19-gage galvanized sheet steel, painted on the outer surface as indicated in Plate 3. The sheet covering the metering area on both sides was separated by a 1/8-inch gap from the covers of the guard air spaces to minimize lateral heat flow into or out of the metering area. To seal the air spaces, the several sheets were screwed to the frames over 1/16-inch felt strips glued to the wood.

Thermocouples for measuring the temperature of the sheet metal faces were permanently soldered to the outer surface of the sheet at ten positions on each face, as shown in Plate 3. The thermocouple leads were cemented to the face of the sheet until they reached a take-off point at the center of the sheet. The surface was painted after the thermocouples were attached.

For those tests in which the inside surface of the warm side of the air space was highly reflective, very bright and clean aluminum foil was cemented to that surface. For some later tests, where it was desired that this surface be one of high emissivity, the aluminum foil was removed and the surface was painted with black or white paint. In all of the tests, the inner surface of the cold side of the panel was painted with either black or white paint of high emissivity.

Single or double air spaces in the panel were obtained as indicated by the "sections A-A" of Plate 3. Table 1 gives particulars as to the dimensions and nature of the spaces on which measurements were made; the decimal part of the "Panel Modification Number" differentiates panels in regard to thickness.

Table 1. Description of Panels and Air Spaces Tested

Test panel	Spaces	Space No. 1		Space No. 2		Orientations	Tests
		Thickness	E*	Thickness	E*		
	No.	in.		in.		No.	No.
1.1	1	5/8	0.028 a	---	---	5	15
1.2	1	7/8	.028 a	---	---	5	17
1.3	1	1 1/2	.028 a	---	---	5	22
1.4	1	3 3/8	.028 a	---	---	5	20
2.1	2	1 11/16	.014 b	1 11/16	0.028 a	5	22
Number of tests of highly-reflective spaces, total							96
3.1	2	1 11/16	0.219 c	1 11/16	0.219 c	5	17
4.1	2	1 11/16	.195 d	1 11/16	.195 d	3	3
5.1	2	1 11/16	.764 e	1 11/16	.764 e	3	3
6.1	1	5/8	.712 f	---	---	5	11
6.2	1	1 1/2	.712 f	---	---	5	16
Number of tests of spaces of moderate and high emissivity, total							50

*E is the effective emissivity of the air space, bounded by surfaces indicated by letters as listed below:

Nature and emissivities of surfaces of air spaces

a - Aluminum foil, e = 0.028; black paint , e = 0.832
b - Aluminum foil, e = 0.028; aluminum foil, e = 0.028
c - Reflective coated paper, e = 0.229; black paint, e = 0.832
d - Reflective coated paper, e = 0.198; white paint, e = 0.927
e - Brown kraft paper, e = 0.813; white paint, e = 0.927
f - Black paint, e = 0.832; black paint, e = 0.832

The total emissivities of the surfaces of the materials used to bound the air spaces were determined by separate measurements made on samples of the materials, most of which were taken from the panels after the tests. The emissivities were determined by measuring the thermal radiation from the surface and the radiation from a black re-entrant cone at the same temperature, using a sensitive total radiation pyrometer at a different temperature. The emissivity was calculated as the ratio of the two readings times the estimated virtual emissivity of the re-entrant cones used (0.98 and 0.99).

5. RESULTS

Values of h_c, the coefficient for heat transfer across air spaces by conduction and convection combined, as determined by these measurements, are shown in a form convenient for general use by the curves of Figures 3, 4, 5, 6, and 7.

Each figure represents one orientation of the air space and direction of heat flow, as indicated, e.g., a horizontal air space with heat flow upward (Figure 3). The several solid curves on each figure give values of $(h_c)_{50}$, the convection-conduction coefficient at an air space mean temperature of 50° F, versus the thickness of the space, ℓ, in inches, for several selected values of the temperature difference across the space, θ, in degrees F. The dashed line gives the calculated heat transfer coefficient for the space with heat transfer by conduction only, with no convection. In a sense, this curve represents the limiting value of $(h_c)_{50}$ as the temperature difference, θ, approaches zero.

Examination of these figures and curves shows that except for a horizontal space with heat flow downward, the convection-conduction coefficient is considerably greater than that for conduction only, and that the effect of convection becomes greater as the temperature difference across a given space increases. However, when the thickness of the space is reduced to a small enough value, the coefficient approaches that for conduction only, regardless of the temperature difference, although the critical thickness is smaller for the larger temperature differences.

The effect of mean temperature on the value of the coefficient h_c depends upon whether convection is a factor in the heat transfer. For cases in which convection is a factor (that is, where $(h_c)_{50}$ departs from the near neighborhood of the "conduction only" curve), the value of h_c for an air space mean temperature, t (°F), is given approximately by

$$(h_c)_t = (h_c)_{50} \left[1 - 0.001\,(t - 50)\right] \qquad (4)$$

For cases in which convection is practically negligible (that is, where $(h_c)_{50}$ lies very close to the "conduction only" curve) the value of $(h_c)_t$ is given approximately by

$$(h_c)_t = (h_c)_{50} \left[1 + 0.0017 (t - 50)\right] \qquad (5)$$

For intermediate cases, where $(h_c)_{50}$ is on the order of 50 percent greater than the corresponding value indicated by the "conduction only" curve, the temperature coefficient approaches zero and mean temperature has little effect.

These adjustments for mean temperature are on the order of 1.0 to 1.7 percent of $(h_c)_{50}$ for a change of 10 degrees F in the air space mean temperature. For many purposes in estimating the total conductances of building air spaces, especially of those which are not highly reflective, the change of h_c with mean temperature may be of little practical importance.

For the vertical air space, and the spaces at 45° with heat flow upward or downward, the curves (Figures 5, 4, and 6, respectively) show that $(h_c)_{50}$ has a minimum value at a space thickness of an inch, more or less, depending upon the orientation and temperature difference, then increases slightly and finally decreases slowly, as the space thickness increases. Thus, for a given vertical or oblique orientation and temperature difference, there is an optimum thickness at which the insulating value of the space is greater than for thicknesses somewhat larger or smaller than the optimum value. However, at considerably larger thicknesses, the insulating value of the space may be even greater than at the optimum thickness occurring at a value near one inch.

The results presented as curves in Figures 3 to 7 were based upon the curves given in Figure 8, which show in a more compact way the relationship between $(h_c)_{50}$ and the variables θ and ℓ, for each orientation. The curves of Figure 8, which were derived by dimensional analysis of the test data, are plotted on bilogarithmic coordinate paper, and show the variation of $(h_c\ell)_{50}$ for each orientation versus the coordinate $\theta\ell^3$. The curves were drawn smoothly through the plotted points, which represent all of the values of h_c obtained in the 96 tests made on highly reflective air spaces (test panels 1.1 through 2.1). Only 88 of the test points are plotted, since eight practically coincided with eight others.

In general, the plotted points representing the test data conform to within a few percent to the values indicated by the smooth curves, except for the case of a horizontal space with heat flow downward for values of $\theta\ell^3$ greater than 300. The accuracy of the

results is best indicated by the congruence of the curves for a vertical space, and for spaces with heat flow downward, to the horizontal dashed line for conduction only, at low values of $\theta\ell^3$. The fact that these three curves become asymptotic to a horizontal line, at a value of $(h_c\ell)_{50}$ of about 0.175, is equivalent to a determination of the thermal conductivity of air at 50° F as equal to 0.175 Btu/hr ft^2(deg F/in.), as compared to 0.173 as given for dry air at 50° F in the NBS-NACA Tables of the Thermal Properties of Gases (Table 2.42, 1950).

The data for the spaces with heat flow upward did not extend to values of $\theta\ell^3$ small enough for them to approach the conduction only line; these curves have been extended by dashed lines to indicate their probable trend. The curves of Figures 3 to 7 are also represented by dashed lines where they are based upon this extrapolation.

The 50 tests conducted on air spaces of moderate and high emissivity (panels 3.1 through 6.2) provided information as to the practical validity of equation 1. Values of $(h_c\ell)_{50}$ were obtained from the test results in the same manner as those calculated from the results obtained with highly reflective spaces. These values were divided by values of $(h_c\ell)_{50}$ taken from the curves of Figure 8 at corresponding values of $\theta\ell^3$. The average value of the 50 quotients so obtained was 1.009. The limits of the average quotient, determined for a 95 percent confidence interval for a group of 50 values, were found to be 1.028 and 0.989. Since these limits bracket closely the ideal value 1.000, it is concluded that the practical validity of equation 1 is substantiated, and conversely, that the method and data given in this report for calculating the total conductance apply equally well for air spaces of low and of high emissivity.

6. LIMITATIONS ON APPLICABILITY OF THE DATA

The values of h_c in this report apply for air spaces of uniform thickness, with reasonably flat surfaces of moderate smoothness, and with no leakage of air into or out of the space, or between spaces where two were used. In addition, the ratio of the height (or length) of the air space to its thickness ranged from about 18 to 96 and the average temperature gradient of its surfaces in the direction of the height (or length) was not greater than about 0.2 degree F per foot for each Btu/hr ft^2 of heat transfer across the air space. A large proportion of the air spaces encountered in buildings conform reasonably well to the characteristics described above, and the results herein should be applicable and appropriate for such spaces.

However, it must be recognized that some air spaces encountered or used in building constructions may depart significantly from the characteristics for which these data are pertinent. For example, two air spaces of nonuniform thickness may be formed by a membrane of foil or paper laid over the faces of studs and bellied into the stud space. Again, the membrane dividing two air spaces may not be completely sealed at the edges or ends, or may be perforated or torn, allowing air to circulate between them. Further, a reflective dividing membrane may be inappropriately installed in such a situation that vapor may condense upon it under certain conditions, in which case its reflectivity might be impaired by an indeterminate amount. For cases such as these, there would be uncertainty in applying the results given in this report: in the first two cases because appropriate values of h_c would not be known, and in the last case because the appropriate value of E would not be known. It is believed that, in general, the heat transfer across such off-characteristic air spaces would be significantly greater than for the spaces considered in this investigation. Evaluation, even approximately, of the effect of such variables on the space conductance would be desirable but would entail further experimental study.

7. UTILIZATION OF THE DATA

The total thermal conductance of a particular air space can be readily computed if the air space thickness, orientation, effective emissivity, temperature difference, and mean temperature are known. For example, consider a vertical air space 1 1/2 inches thick, with surfaces such that its effective emissivity is 0.05, and having a temperature difference across it of 20 deg F and a mean temperature of 50° F. From Figure 5 for vertical spaces (or Figure 8), $(h_c)_{50}$ is 0.308. From Figure 1, $(h_r)_{50}$ is 0.91. The conductance of the space, C_{50}, in accordance with equation 1, is therefore 0.308 + 0.05 (0.91) = 0.354 Btu/hr ft^2 (deg F), and its thermal resistance, which equals the reciprocal of the conductance, is 1/0.354 or 2.8 deg F per Btu/hr ft^2. The computation for an air space mean temperature of 30° F, instead of 50° F as above, is also straightforward. Using equation 4, $(h_c)_{30} = (h_c)_{50} \times [1 - 0.001 (30 - 50)] = 0.308 \times 1.02 = 0.314$. From Figure 1, $(h_r)_{30} = 0.81$. Hence, the conductance of the space at 30° F mean temperature, C_{30}, is 0.314 + 0.05 (.81) = 0.354 Btu/hr ft^2 (deg F). It happens that in this instance C_{30} equals C_{50}, but this would not necessarily be true in other cases although the difference may be practically negligible.

Values of the thermal resistance of 1 1/2-inch air spaces at a mean temperature of 50° F and for a temperature difference of 20 deg F are given in Table 2 for various orientations and effective emissivities E. The tabulated resistance values indicate in an

approximate way the relative insulating values of air spaces in different orientations and with different emissivities, and will be useful in making calculations for particular building constructions involving air spaces, as indicated later.

Table 2. Thermal Resistances* of 1 1/2-inch Air Spaces

Temp. diff. = 20 deg F Mean temp. = 50° F

Orientation of space	Direction of heat flow	Effective emissivity (E) of space			
		0.05	0.20	0.50	0.82
Horizontal	Down	5.7	3.2	1.7	1.1
45° angle	Down	3.9	2.6	1.5	1.0
Vertical	Horizontal	2.8	2.0	1.3	0.9
45° angle	Up	2.3	1.8	1.2	.9
Horizontal	Up	2.0	1.6	1.1	.8

*Degrees F per Btu/hr ft^2

Calculation of the conductance (or thermal resistance) of an air space in a building construction such as a wall is a little more complicated because the temperature difference across such a space depends upon the temperature difference across the construction, and upon the ratio of the thermal resistance of the space to the total thermal resistance of the construction including that of the air space (or spaces). In other words, it is necessary to know or assume the conductance or resistance of the space (or spaces) before a close computation can be made. The "trial and error" procedure required leads quickly to a correct result, as shown below.

Consider the building construction (wall) shown in Figure 9, with two air spaces formed by a central membrane dividing the 2 by 4 stud space. Using the quantities indicated in Figure 9, the temperature drop, $t_2 - t_3$, between the gyplath and sheathing, is given by

$$t_2 - t_3 = (t_1 - t_4) \times \frac{R_{2-3}}{R_{2-3} + (R_{1-2} + R_{3-4})} \qquad (6)$$

The term $(R_{1-2} + R_{3-4})$ is seen to be the thermal resistance of the construction and the surface air films, exclusive of the resistance of the air space or spaces, and is readily calculated for a given construction using data given in references such as FHA Technical Circular No. 7 or the ASHVE "Guide".

Similarly, the average mean temperature of the air space or spaces is given by

$$t_m = t_4 + (t_1 - t_4) \times \frac{R_{3-4} + \frac{1}{2}(R_{2-3})}{R_{2-3} + (R_{1-2} + R_{3-4})} \qquad (7)$$

These equations can be solved to obtain approximate values of $t_2 - t_3$ and t_m if appropriate values of R_{2-3} are assumed. In general, it will be found that the thermal resistances tabulated in Table 2 will serve to provide good first approximations to R_{2-3}. To obtain a value of R_{2-3}, an approximate value of the resistance per space is taken from Table 2, using an appropriate value of E, which can be taken as the average of the effective emissivities of the spaces, if there are more than one, as determined by the values of the surface emissivities e_2, e_2', e'_3, etc. and by means of Figure 2. For values of E not given in the table, interpolation is permissible.

For example, assume $t_1 = 70°$ F, $t_4 = 0°$ F, e_2 and $e_3 = 0.9$ (ordinary building surfaces), and e_2' and $e'_3 = 0.05$ (reflective foil surfaces). Then each of the air spaces will have an effective emissivity E of 0.05 (Figure 2) and the average value is 0.05. Referring to Table 2, the resistance of one space for this case is given as 2.8, and R_{2-3} for two spaces is, therefore, 5.6. For the construction indicated in Figure 9, $R_{1-2} = 1.03$ and $R_{3-4} = 2.11$. Accordingly

$$t_2 - t_3 = (70) \times \frac{5.6}{5.6 + 3.14} = 70 \times \frac{5.6}{8.74} = 45 \text{ deg F}$$

$$t_m = 0 + 70 \times \frac{2.11 + \frac{1}{2}(5.6)}{8.74} = 70 \times \frac{4.91}{8.74} = 39° \text{ F}$$

The temperature difference per space, since there are two spaces, is $\frac{1}{2}(t_2 - t_3)$ or 22.5 deg F. Considered rigorously, the two spaces have different mean temperatures, given approximately by 39° F ± 22.5/2, or 50° F and 28° F, but for the purposes of this calculation, the average value of 39° F is satisfactory. Actually, as indicated in the earlier example, the effect of mean temperature for this case is negligible, and this example is therefore worked out using values for a 50° F mean temperature.

Using Figure 5, $(h_c)_{50}$ for a vertical space 1.8 inches wide with a temperature difference of 22.5 deg F is 0.32, and adding the radiation coefficient $(Eh_r)_{50}$, which equals 0.046, it is found that C_{50} is 0.366 and the air-space resistance is 2.73. This is so close to 2.8, the trial resistance, that recomputation is not necessary, and using the value 2.73, the U-value of the construction is given by $\frac{1}{U} = (R_{1-2} + R_{3-4}) + 2(2.73)$, whence $U = \frac{1}{8.6} = 0.116$ Btu/hr ft^2(deg F). If a recomputation had been made, using 2.73 as a trial resistance for each air space, the results would have been $t_2 - t_3 = 44.5$, $t_m = 39.4$, and an average air space resistance equal to 2.73, as before, indicating that the first results were satisfactorily correct.

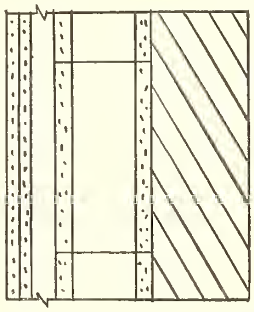
Before leaving this example, it is pertinent to notice that the membrane temperature in this case is approximately the same as the mean temperature of the 2 by 4 stud space, or 39° F. If the membrane consisted of a metallic foil impermeable to water vapor, the vapor pressure on its warm side would be substantially equal to that in the building, and condensation might occur on the foil when the dewpoint temperature indoors exceeded 39° F.

The temperature of the inner surface of single glass windows, with 70° F indoors and still air at 0° F outdoors, would not ordinarily exceed 35° F, and an indoor dewpoint of 39° F could not be maintained without considerable condensation on the windows. Accordingly, condensation on a foil membrane would, in this case, be unlikely if single glass windows in the house were free of excessive condensation. However, if storm sash or double glass were used in all the windows, the indoor dewpoint might rise to about 50° F before window condensation occurred, and there would be considerable risk of condensation on the foil membrane. This risk could be eliminated by applying a vapor barrier such as another sheet of foil, or a membrane of equal vapor resistance, at some position on the warm side of the stud space, such as position 2 in Figure 9.

The U-values of several typical wall, ceiling, and floor constructions used in houses, with various insulation applications, are given in the following pages. The U-values are expressed in Btu/hr ft^2 per degree F of air temperature difference on the two sides of the construction.

These values were calculated using the methods and data presented in this report, insofar as air spaces are concerned, and using for other components of the construction coefficients taken from FHA Technical Circular No. 7, revised January 1947. In accordance with the U-value calculations given in T.C. No. 7, no allowance was made for the effect of wood studs or joists on the average U-value of the construction. In most cases, where the calculated U-value for the between-members area is of the magnitude of 0.08 to 0.14, the effect of

the wood members is small. If it is desired to take into account the effects of heat flow through studs or joists, the procedure for parallel heat flow given in the section "Computed Heat Transmission Coefficients" on pages 182-4 of the 1953 ASHVE "Guide" may be used.

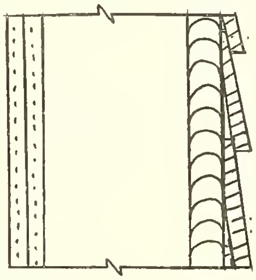


WALLS

1. 4-in. brick, 4-in. cinder block, 25/32-in. furring space, 1/2-in. plaster on 3/8-in. gyplath.

   Assumed air temperatures: 70° and 0° F

	U
(a) Ordinary furring space, E of space = 0.82	0.25
(b) Reflective coated paper on gyplath, E of space = 0.20	.20
(c) Aluminum foil backing on gyplath, E of space = 0.05	.18
(d) Space filled with fibrous insulation (if dry)	.17



2. Yellow pine lap siding, 25/32-in. wood sheathing, 2 x 4 stud space, 1/2-in. plaster on 3/8-in. gyplath.

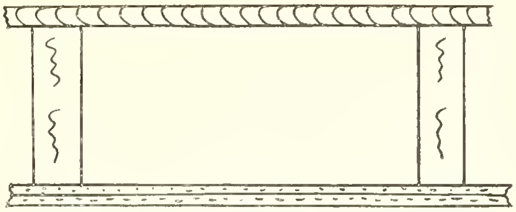
   Assumed air temperatures: 70° and 0° F

(a) Stud space uninsulated, one space, E = 0.82	.24
(b) Aluminum foil backing on gyplath, one space, E = 0.05	.18
(c) Stud space divided equally by ordinary kraft paper or cardboard, 2 spaces, E = 0.82 for each	.19
(d) Stud space divided equally by vapor-permeable reflective paper sheet, 2 spaces, E = 0.20 for each	.138
(e) Stud space divided equally by aluminum foil sheet, 2 spaces, E = 0.05 for each	.116*
(f) Stud space divided equally by 1-in. fibrous blanket insulation with aluminum foil on warm side, 2 spaces, E = 0.05 and 0.82	.091
(g) Two-inch fibrous blanket insulation against gyplath, with vapor-permeable reflective paper on cold side, one space, E = 0.20	.078
(h) Same as (f), with ordinary paper on cold side of blanket, one space, E = 0.82	.085
(i) Stud space divided into 3 equal spaces by two sheets of aluminum foil, E (avg.) = 0.04	.077

*There may be danger of condensation of moisture on the foil surface, depending upon temperature conditions and the relative humidity inside the building.

CEILINGS

3. 1/2-in. plaster on 3/8-in. gyp-lath, 2 x 8 joist space, 25/32-in. yellow pine flooring.



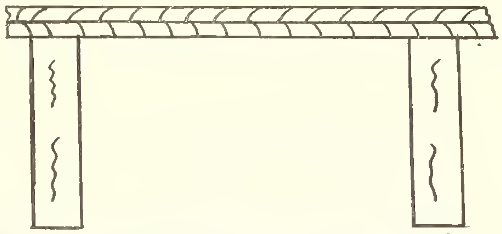
U_w is the U-value for winter conditions, with heat flow upward and assumed air temperatures of 80° (room) and 10° F (attic).

U_s is the U-value for summer conditions, with heat flow downward and assumed air temperatures of 120° (attic) and 90° F (room)

	U_w	U_s
(a) No insulation in joist space, one space, E = 0.82	0.30	0.25
(b) Aluminum foil backing on gyplath, one space, E = 0.05	.22	.088
(c) Reflective paper backing on gyplath, one space, E = 0.20	.24	.156
(d) Aluminum foil sheet dividing joist space equally, 2 spaces, E = 0.05 for each	.146*	.052
(e) Vapor-permeable reflective paper sheet dividing joist space equally, 2 spaces, E = 0.20 for each	.170	.105
(f) Two-inch blanket of fibrous insulation resting on gyplath, with vapor-permeable reflective paper on top side of blanket, one space, E = 0.20	.084	.073
(g) One-inch blanket of fibrous insulation dividing joist space equally, with aluminum foil membrane on underside of blanket, 2 spaces, E = 0.05 and 0.82	.105	.065
(h) Four inches of fibrous insulation resting on gyplath, one space, E = 0.82	.055	.053
(i) Construction as shown, but with vapor-permeable reflective paper laid over joists in place of the wood flooring, one space, E = 0.20, and top-surface film coefficient for reflective paper taken as 1.30 Btu/hr ft^2 (deg F) for U_w and 1.70 for U_s	.30	.20

*There may be danger of condensation of moisture on the foil surface depending upon temperature conditions and the relative humidity inside the building.

FLOORS



4. 13/16-in. hardwood flooring, building paper, 25/32-in. yellow pine subflooring, 2 x 10 joists.

Assumed conditions: heat flow downward, from air at 65° (room) to 30° F (under floor).

	U
(a) Uninsulated floor	0.28
(b) Sheet of aluminum foil fastened to bottom of joists, one space, E = 0.05	.070
(c) Sheet of reflective paper fastened to bottom of joists, one space, E = 0.20	.114
(d) Sheet of 1/2-inch rigid insulation board fastened to bottom of joists, one space, E = 0.82	.160
(e) Same as (d) but with sheet of aluminum foil dividing joist space equally, 2 spaces, E = 0.05 for each	.044
(f) Same as (d) but with sheet of reflective paper dividing joist space equally, 2 spaces, E = 0.20 for each	.075
(g) Two-inch fibrous blanket insulation installed at midheight of joist space	.082

8. DISCUSSION AND REMARKS

Study of the U-values for the various constructions indicates, among others, the following practical conclusions:

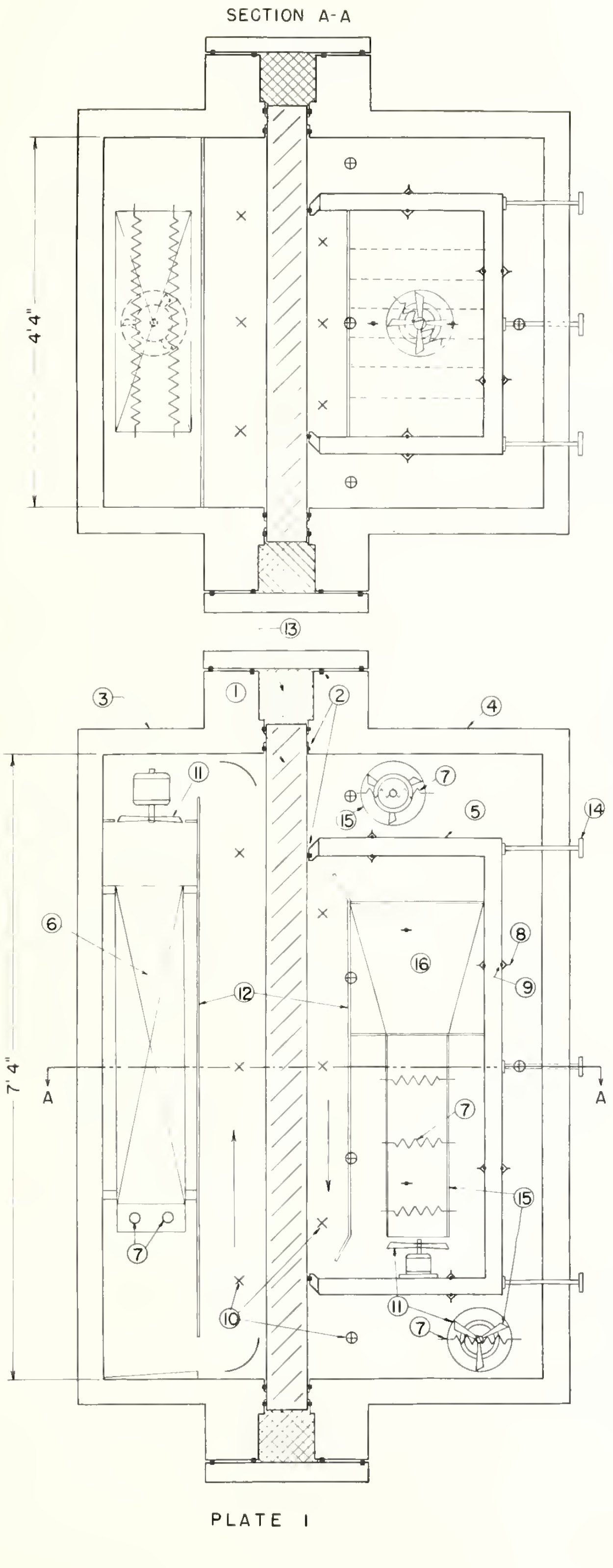
(1) The insulating effect of a furring or stud space can be markedly increased at low cost by use of a reflective surface on the back of the gyplath or other plaster base (compare Walls 1b and 1c with 1a and 1d, and Wall 2b with 2a).

(2) Well-insulated walls can be made using reflective sheets to subdivide the stud space. However, circulation of air between spaces must be prevented by adequate sealing of each dividing membrane, and in the case of vapor-impermeable membranes, consideration must be given to the possibility of moisture condensation on their surfaces.

(3) The great value of reflective air spaces in reducing heat flow downward is indicated by comparisons of the U-values for summer and winter conditions for the ceiling constructions, and by the very considerable insulating effect of reflective spaces used under floors.

One ceiling insulation application, not covered directly by the data of this report, is worth consideration. In the case of Ceiling 3i, the U-value for summer conditions (U_s = 0.20) does not tell the whole story for the temperature conditions existing in an attic heated by the sun--actually, a considerably better insulating effect is achieved. The computations for the ceilings are based upon an assumed temperature of 120° F for the attic air near the floor and approximately that temperature for the attic surfaces. In actual fact, during the hours of solar heating, the underside of the roof may reach temperatures of 140° F or more. Radiation from these hot surfaces to floor or insulation surfaces of low reflective quality (e = 0.9) may heat them to temperatures as high as 120° F, and for such surfaces the temperature conditions assumed for the calculated U-values are appropriate. However, for a reflective surface such as a reflectively-coated paper (e = 0.2), the roof radiation is largely reflected and the reflective surface temperature would not exceed about 100° F for similar attic conditions. Consequently, the effect of high reflectivity of the top surface of a ceiling construction (the part that "sees" the roof) is to reduce markedly the temperature difference acting to cause heat to flow through the construction. For this reason, ceiling constructions such as 3i, and others having reflective exposed top surfaces, are more advantageous for summer conditions than the calculated U-value indicates. Because the reflective surface in effect reduces the temperature difference acting on the ceiling construction, its effect is, approximately, proportionately the same regardless of the insulating value of the rest of the construction.

Since many attics are unfloored, at least in part, the advantages of reflective top surfaces can be provided at low cost, yet with marked benefit under summer conditions, especially in hot sunny climates. This may be accomplished by use of a blanket insulation with a reflective surface on its top (or cold in winter) side, placed between the joists, or by stretching a reflective sheet material over the top of the joists, either parallel or perpendicular to them, whichever is more convenient. It should be pointed out, however, that the reflective surface must be quite permeable to water vapor to avoid the possibility of moisture condensation on its underside in winter, unless a better vapor barrier is also used near the warm side of the construction.



① TEST PANEL
② RUBBER GASKETS
③ COLD BOX
④ GUARD BOX
⑤ METER BOX
⑥ REFRIGERATING COIL
⑦ ELECTRIC HEATERS
⑧ AIR COMPOUND THERMOCOUPLES
⑨ SURFACE COMPOUND THERMOCOUPLES
⑩ AIR TEMPERATURE THERMOCOUPLES
⑪ AIR CIRCULATING FANS
⑫ BAFFLES
⑬ EDGE INSULATION
⑭ COMPRESSION SPRING HANDWHEELS
⑮ HEATER DUCTS
⑯ METER HEAT DIFFUSER

PLATE I

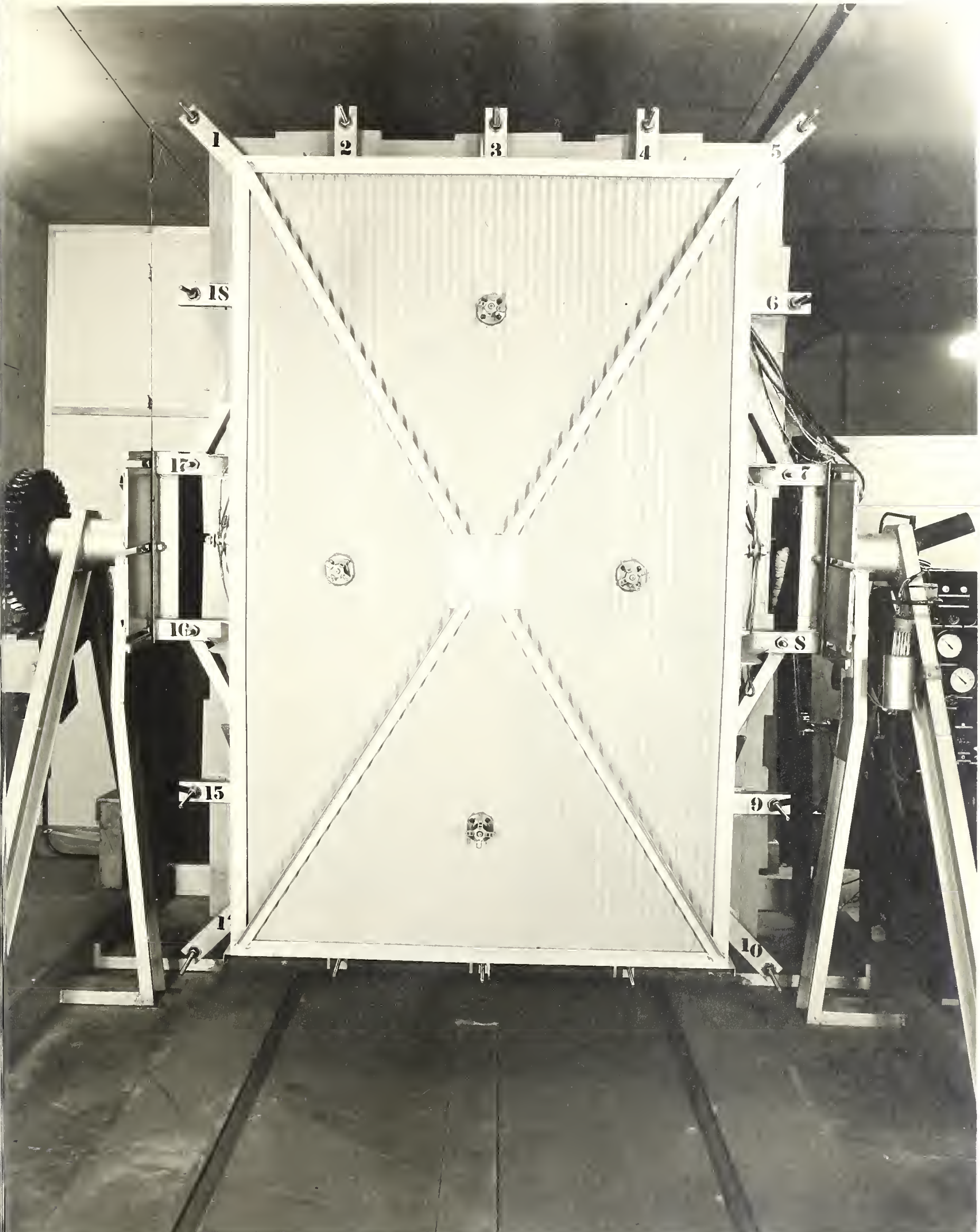
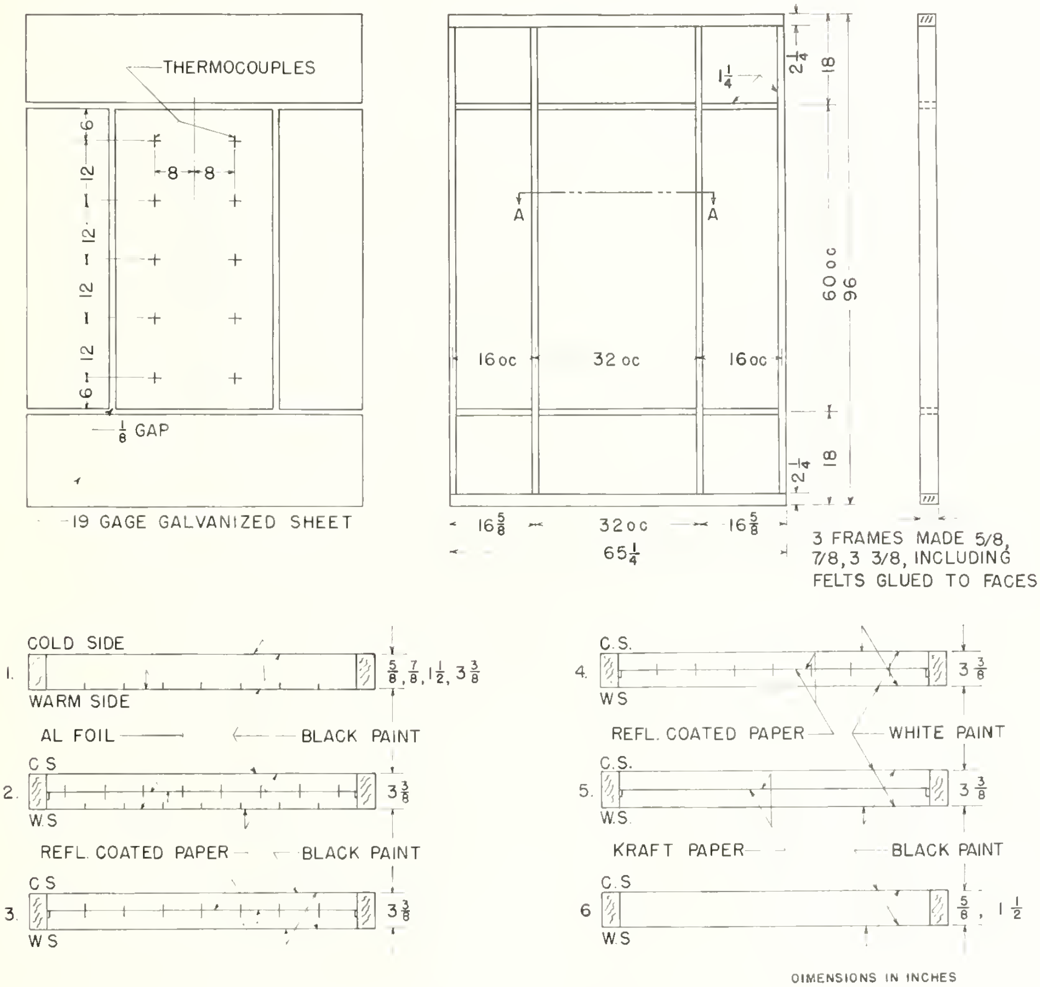
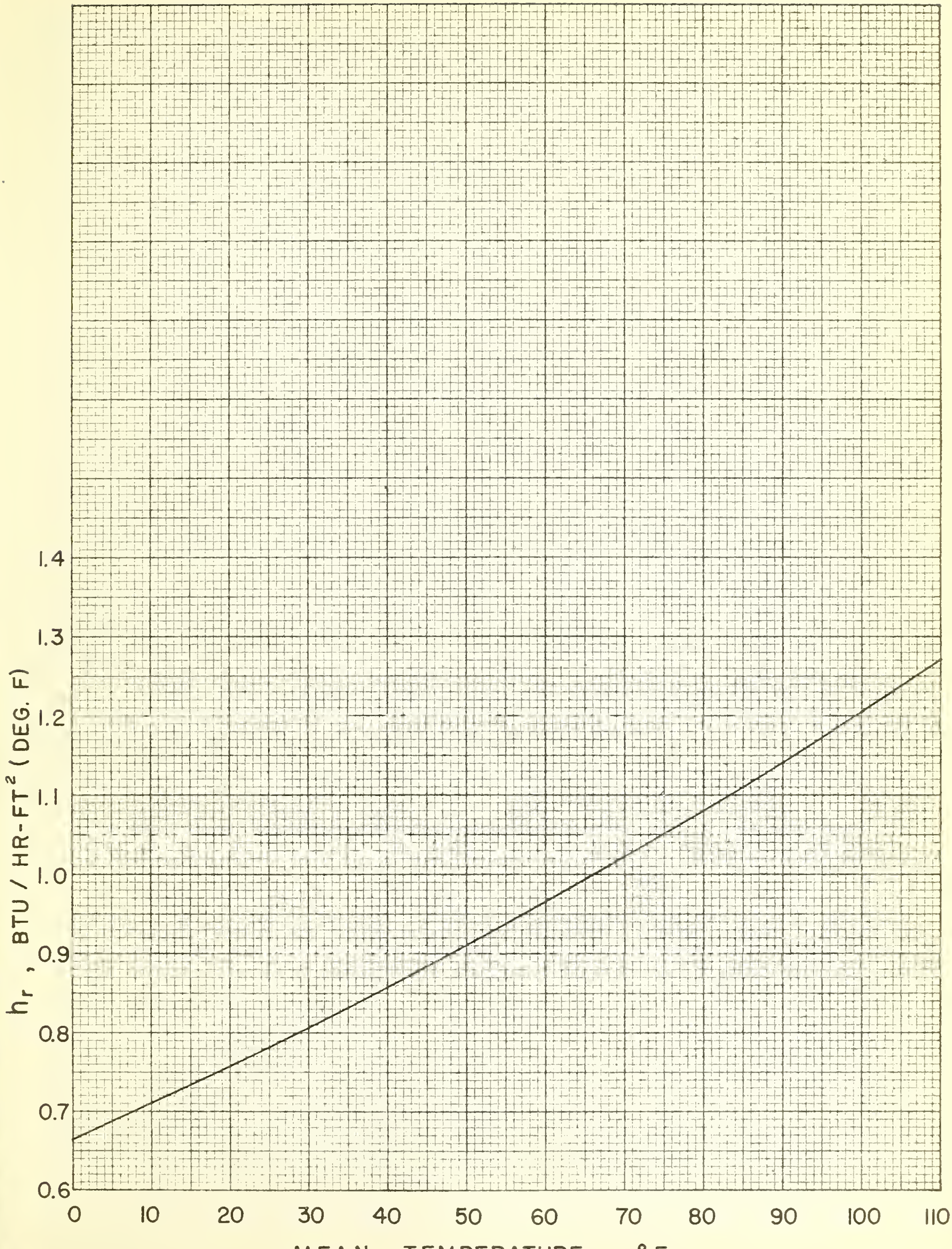


Plate 2



TEST PANELS-SECTION A-A

PLATE 3



h_r , BTU / HR-FT^2 (DEG. F)
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0
10
20
30
40
50
60
70
80
90
100
110
MEAN TEMPERATURE , °F

FIGURE 1

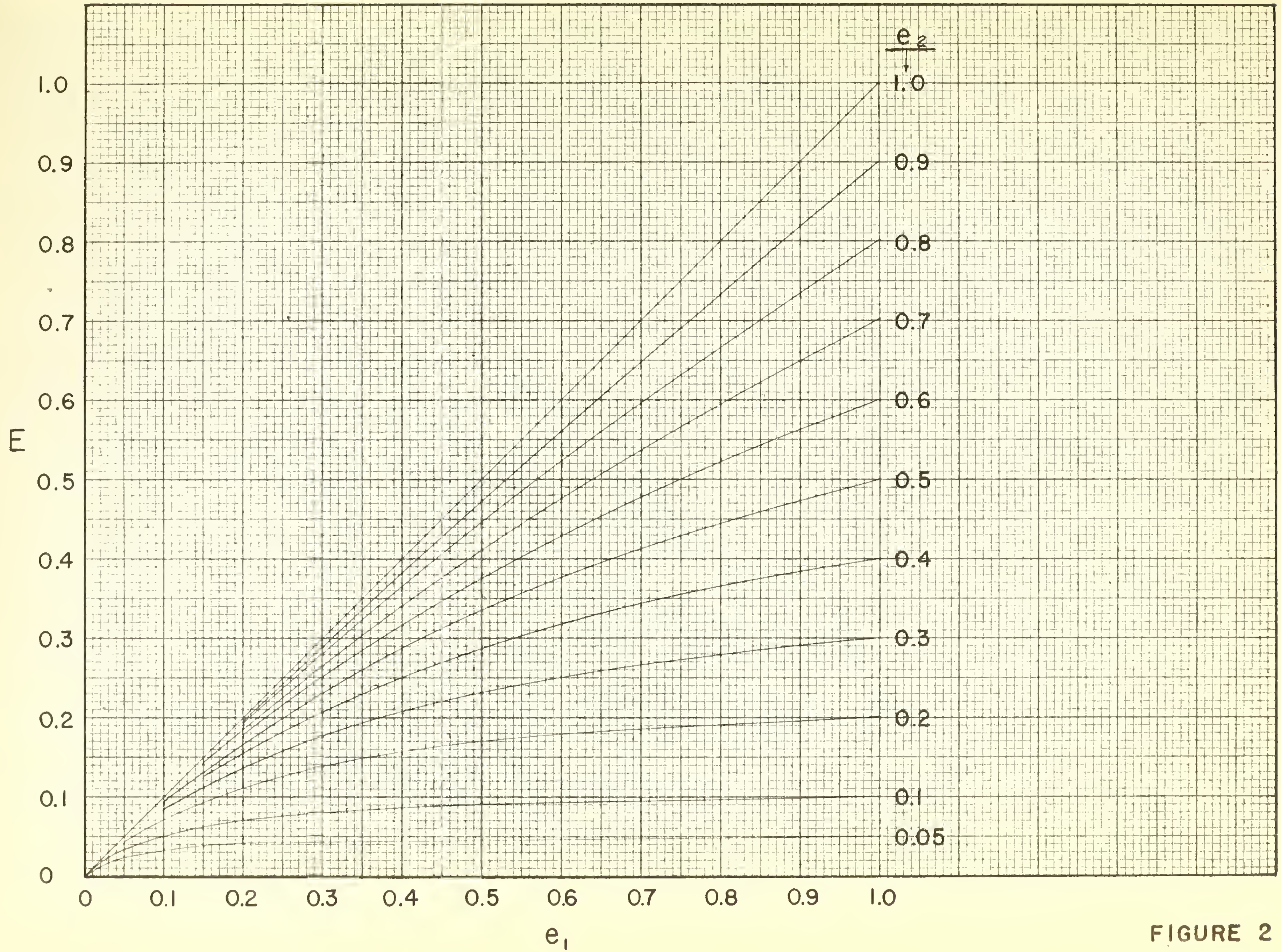
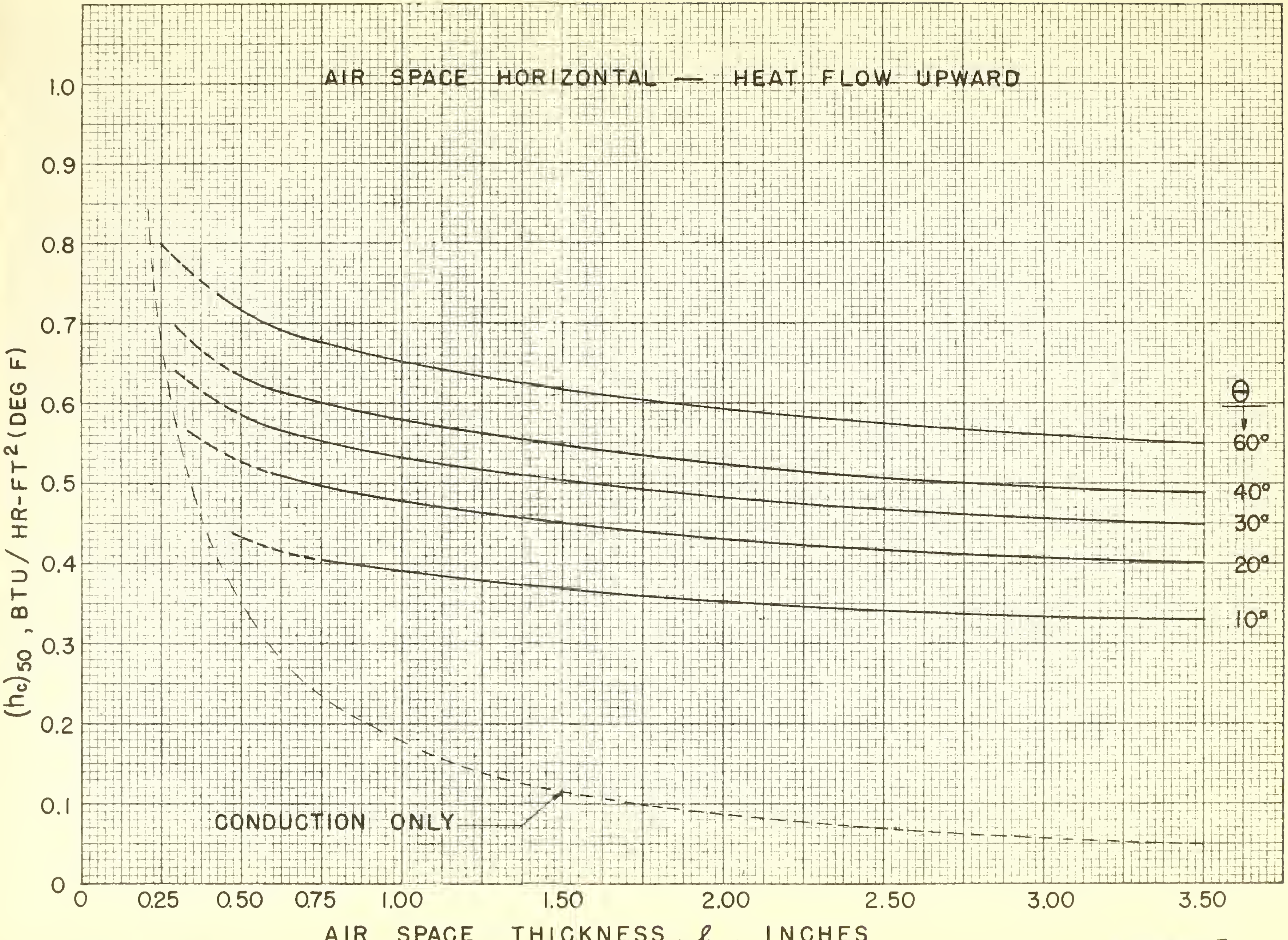
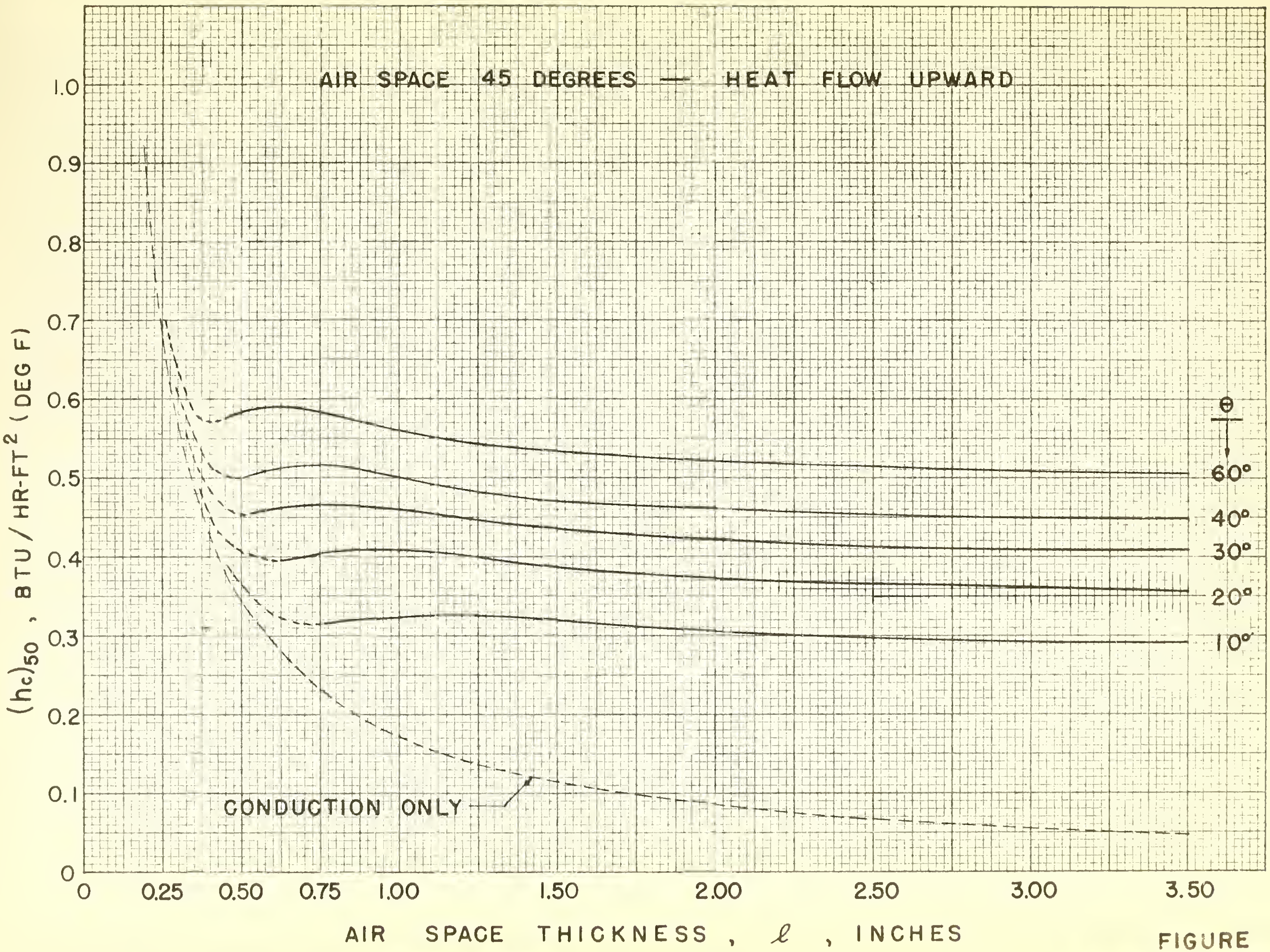

E
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
e_1
e_2
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.05

FIGURE 2



AIR SPACE HORIZONTAL — HEAT FLOW UPWARD
$(h_c)_{50}$, BTU / HR-FT^2 (DEG F)
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.25
0.50
0.75
1.00
1.50
2.00
2.50
3.00
3.50
AIR SPACE THICKNESS, ℓ, INCHES
Θ
60°
40°
30°
20°
10°
CONDUCTION ONLY

FIGURE 3



AIR SPACE 45 DEGREES — HEAT FLOW UPWARD
θ
60°
40°
30°
20°
10°
CONDUCTION ONLY
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.25
0.50
0.75
1.00
1.50
2.00
2.50
3.00
3.50
(hc)50 , BTU / HR-FT2 (DEG F)
AIR SPACE THICKNESS , ℓ , INCHES

FIGURE 4

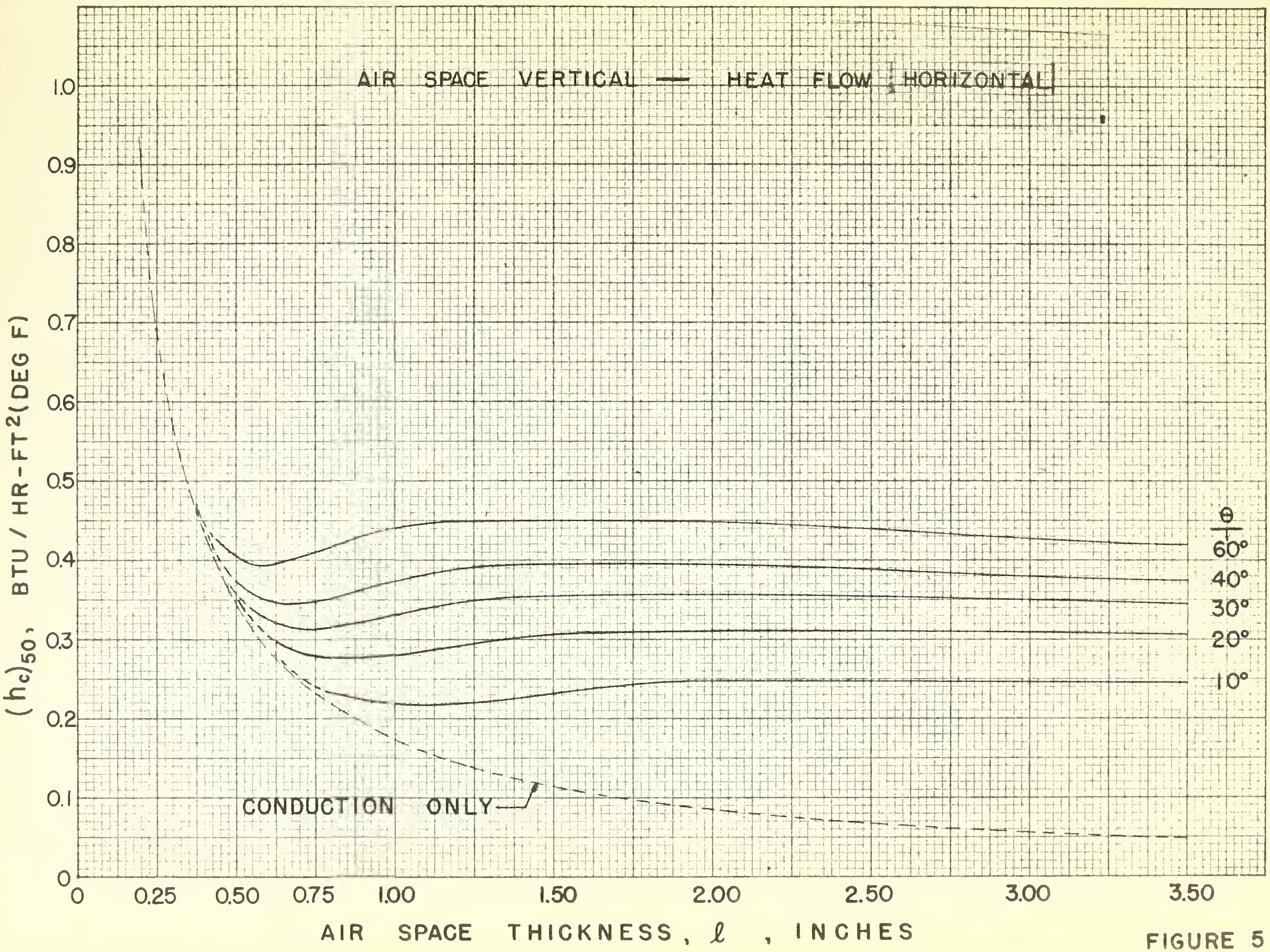

AIR SPACE VERTICAL — HEAT FLOW HORIZONTAL
$(h_c)_{50}$, BTU / HR-FT^2(DEG F)
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
0.25
0.50
0.75
1.00
1.50
2.00
2.50
3.00
3.50
AIR SPACE THICKNESS, ℓ, INCHES
θ
60°
40°
30°
20°
10°
CONDUCTION ONLY

FIGURE 5

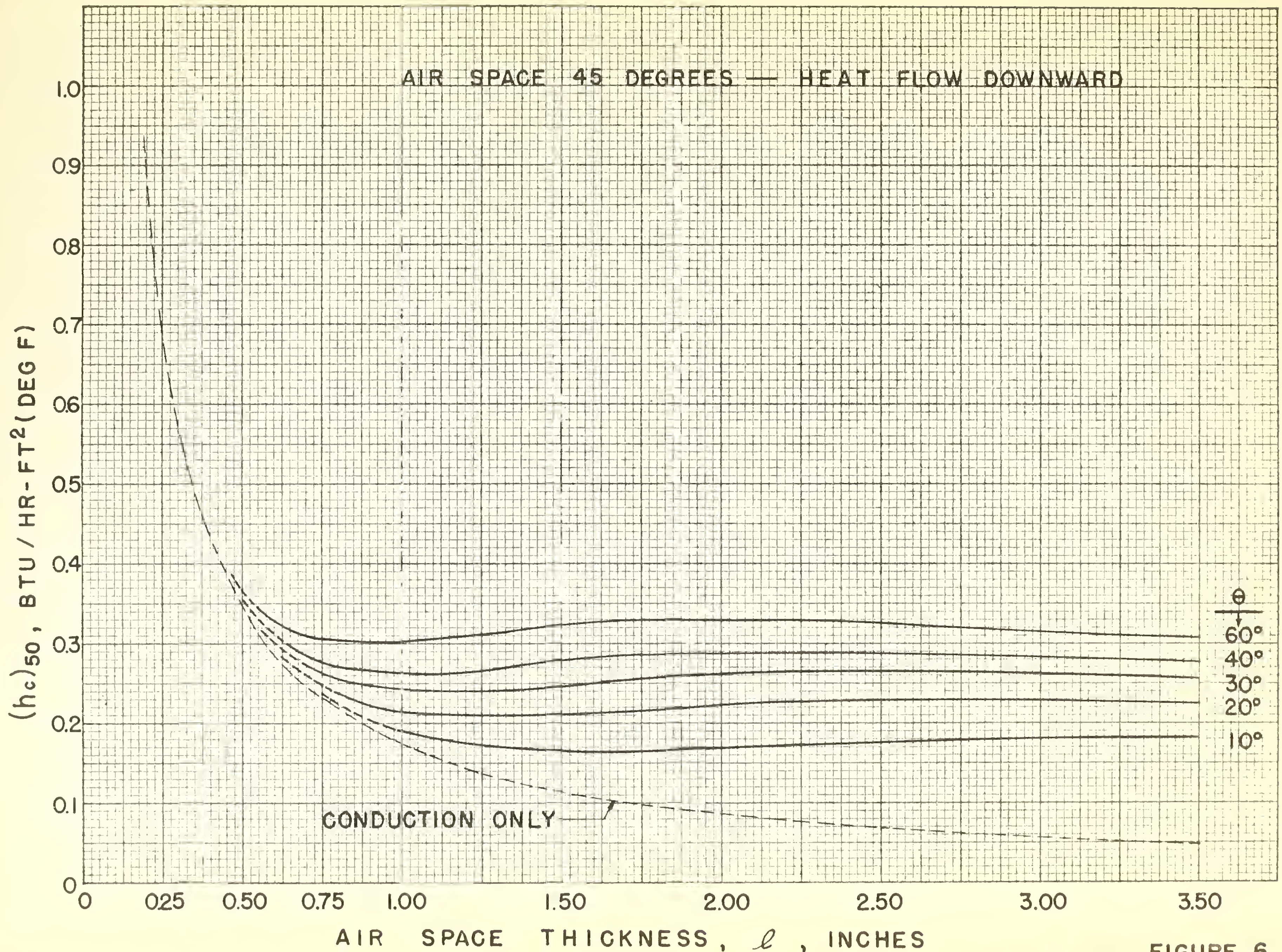

AIR SPACE 45 DEGREES — HEAT FLOW DOWNWARD
(hc)50, BTU / HR - FT2 (DEG F)
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.25
0.50
0.75
1.00
1.50
2.00
2.50
3.00
3.50
AIR SPACE THICKNESS, ℓ, INCHES
θ
60°
40°
30°
20°
10°
CONDUCTION ONLY

FIGURE 6

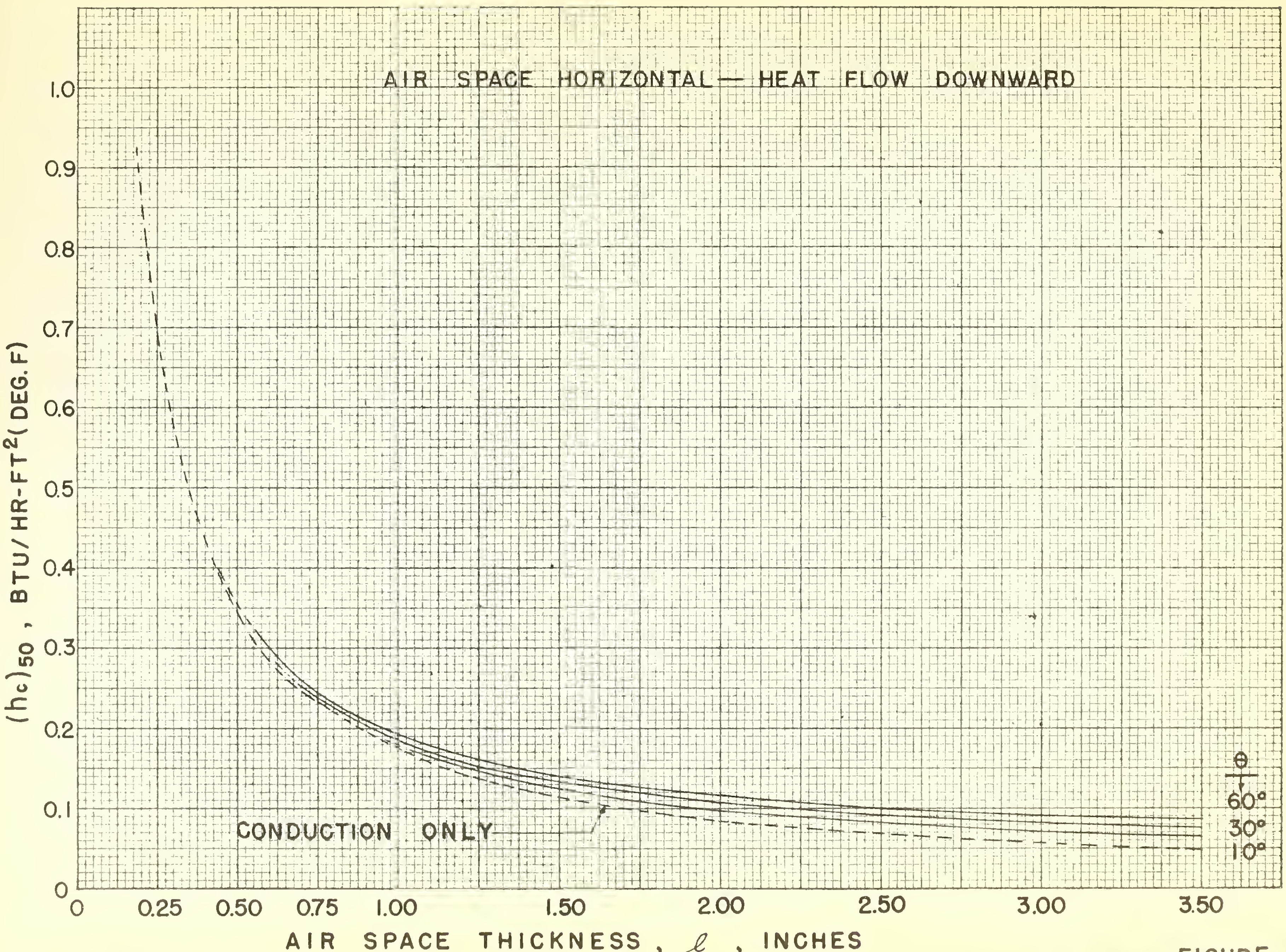

AIR SPACE HORIZONTAL — HEAT FLOW DOWNWARD
$(h_c)_{50}$, BTU/HR-FT^2(DEG. F)
AIR SPACE THICKNESS , ℓ , INCHES
CONDUCTION ONLY
θ
60°
30°
10°
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
0.25
0.50
0.75
1.00
1.50
2.00
2.50
3.00
3.50

FIGURE 7

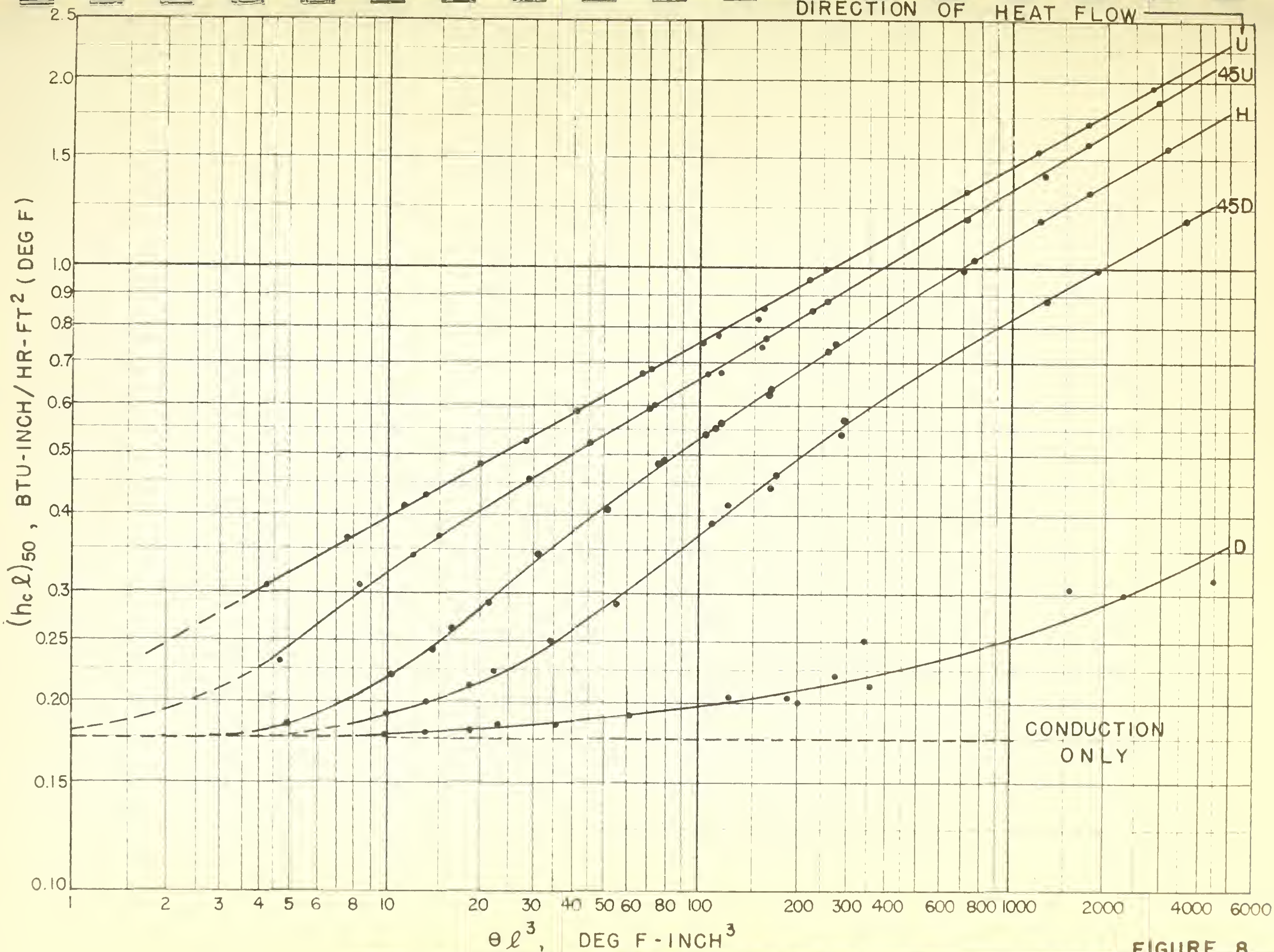

DIRECTION OF HEAT FLOW
U
45U
H
45D
D
CONDUCTION ONLY
2.5
2.0
1.5
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.25
0.20
0.15
0.10
1
2
3
4
5
6
8
10
20
30
40
50
60
80
100
200
300
400
600
800
1000
2000
4000
6000
$(h_c \ell)_{50}$, BTU-INCH/HR-FT^2 (DEG F)
$\theta \ell^3$, DEG F-INCH^3

FIGURE 8

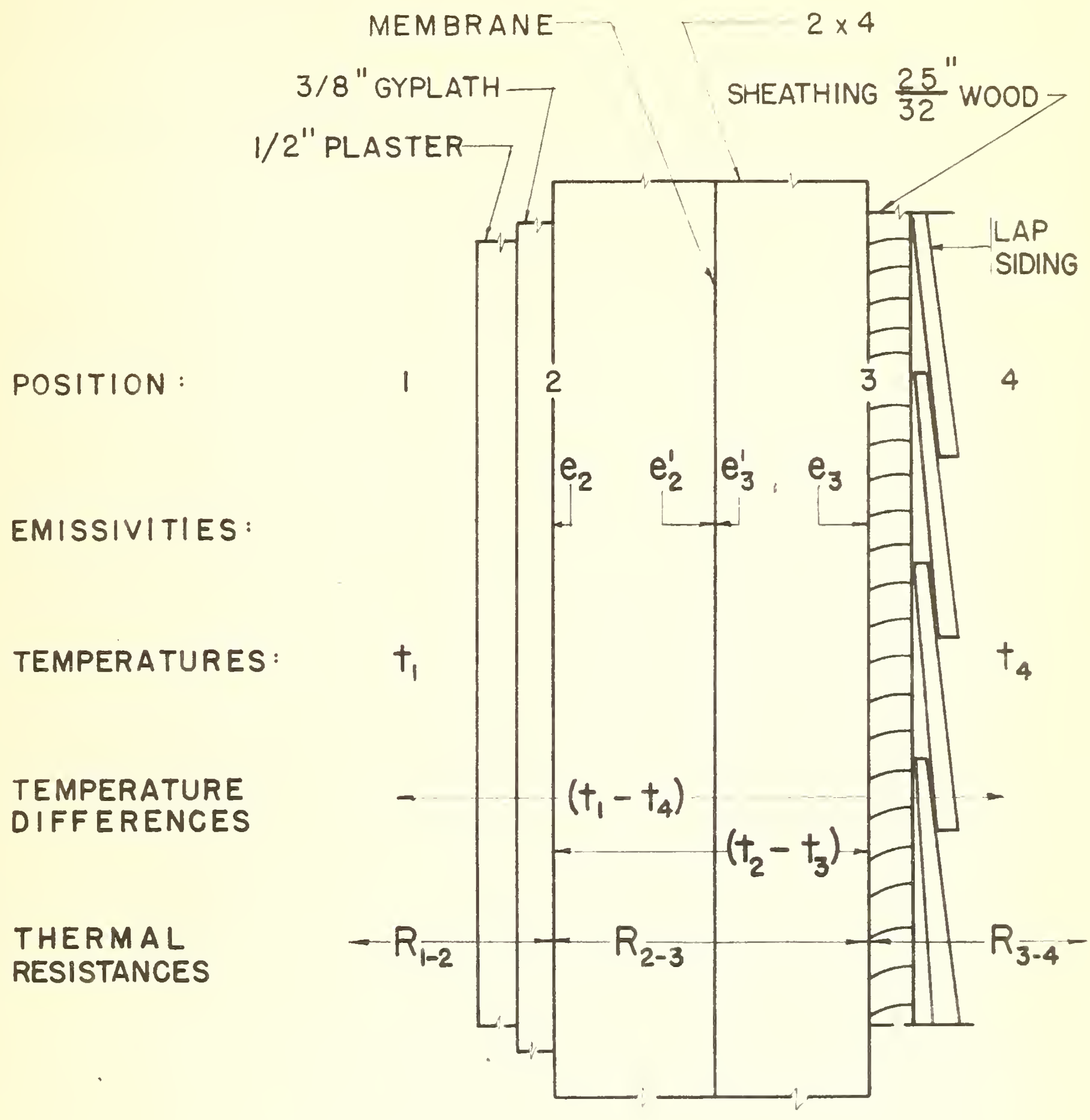

MEMBRANE
2 x 4
3/8" GYPLATH
SHEATHING $\frac{25}{32}$" WOOD
1/2" PLASTER
LAP SIDING
POSITION:
1
2
3
4
e_2
e_2'
e_3'
e_3
EMISSIVITIES:
TEMPERATURES:
t_1
t_4
TEMPERATURE DIFFERENCES
$(t_1 - t_4)$
$(t_2 - t_3)$
THERMAL RESISTANCES
R_{1-2}
R_{2-3}
R_{3-4}

FIGURE 9

THE NATIONAL BUREAU OF STANDARDS

Functions and Activities

The functions of the National Bureau of Standards are set forth in the Act of Congress, March 3, 1901, as amended by Congress in Public Law 619, 1950. These include the development and maintenance of the national standards of measurement and the provision of means and methods for making measurements consistent with these standards; the determination of physical constants and properties of materials; the development of methods and instruments for testing materials, devices, and structures; advisory services to Government Agencies on scientific and technical problems; invention and development of devices to serve special needs of the Government; and the development of standard practices, codes, and specifications. The work includes basic and applied research, development, engineering, instrumentation, testing, evaluation, calibration services, and various consultation and information services. A major portion of the Bureau's work is performed for other Government Agencies, particularly the Department of Defense and the Atomic Energy Commission. The scope of activities is suggested by the listing of divisions and sections on the inside of the front cover.

Reports and Publications

The results of the Bureau's work take the form of either actual equipment and devices or published papers and reports. Reports are issued to the sponsoring agency of a particular project or program. Published papers appear either in the Bureau's own series of publications or in the journals of professional and scientific societies. The Bureau itself publishes three monthly periodicals, available from the Government Printing Office: The Journal of Research, which presents complete papers reporting technical investigations; the Technical News Bulletin, which presents summary and preliminary reports on work in progress; and Basic Radio Propagation Predictions, which provides data for determining the best frequencies to use for radio communications throughout the world. There are also five series of nonperiodical publications: The Applied Mathematics Series, Circulars, Handbooks, Building Materials and Structures Reports, and Miscellaneous Publications.

Information on the Bureau's publications can be found in NBS Circular 460, Publications of the National Bureau of Standards ($1.00). Information on calibration services and fees can be found in NBS Circular 483, Testing by the National Bureau of Standards (25 cents). Both are available from the Government Printing Office. Inquiries regarding the Bureau's reports and publications should be addressed to the Office of Scientific Publications, National Bureau of Standards, Washington 25, D. C.

NBS