HEAT INSULATION

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Preface

This book is designed to bring together in one volume for the engineer, architect, and student some of the miscellaneous information on heat insulation. No attempt is made to produce a textbook, although the material should prove valuable to students of heat insulation.

Since 1910, I have been closely connected with the development and application of equipment to determine the effectiveness of heat insulation over a wide range of temperature. One chapter is devoted to a brief discussion of various types of test equipment, some of which have never been publicized. This section should prove to be of considerable interest to the laboratory man dealing with heat insulation.

In Chapter Four, the various factors that may affect the value of the coefficient of thermal conductivity are discussed in detail. Many of them are frequently neglected in calculation, although they might have an important effect on the result.

The chapter on reflective insulation is based chiefly on work carried out in the Heat Measurements Laboratory at the Massachusetts Institute of Technology since 1929. The information presented concerning this extremely interesting and controversial type of insulation should help clarify the situation.

Although no attempt has been made to cover the "unsteady" state of heat flow, the chapter on specific heat and thermal diffusivity, as well as the tables in the appendix, was added because many of the values were determined under my supervision. The tables on coefficients of thermal conductivity in the appendix are up-to-date and include most of the materials in common use.

It is impossible to present here the names of all who have aided me in collecting the material for this book. Many manufacturers of insulating material have been very kind in giving me whatever information they have on the subject. I wish to express my deep appreciation to Prof. W. H. McAdams, who

advised me to proceed with this work, to Prof. L. R. Vianey, who has given considerable time in reviewing the manuscript, and to Mr. S. W. Sennett, who is responsible for all the drawings.

GORDON B. WILKES

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CHAPTER ONE

Purposes of Heat Insulation

Thermal or heat insulation may be defined as those materials or combination of materials, with air or evacuated spaces, which will retard the transfer of heat with reasonable effectiveness under ordinary conditions. It does not seem accurate to define thermal insulation as those materials that have a coefficient of thermal conductivity less than some specified value. For example, aluminum foil has a coefficient of thermal conductivity roughly five thousand times that of conventional low-temperature insulation, but when installed correctly with air spaces it may be classed as an equally good insulator. Furthermore, with the exception of reflective insulation and the Dewar flask (Thermos bottle), the insulating value of most materials is approximately proportional to their thickness. Consequently, heat flow may be retarded effectively with materials having relatively high coefficients of thermal conductivity if sufficient thickness is provided. A common brick wall 36 in. thick would have about the same rate of heat transfer as a reasonably well-insulated frame wall 6 in. thick.

Heat may be transferred through insulation by conduction, convection, or radiation, or any combination of these three methods. No known heat insulation is 100% effective in the prevention of heat transfer as long as there is a temperature difference between the two sides of the insulation, but a Dewar flask approaches this ideal insulation since conduction and convection are practically eliminated by the high vacuum and since radiation is minimized by the silvering of the glass. The chief loss or gain of heat from a Thermos bottle occurs by means of conduction along the glass at the neck of the bottle.

The main purpose of insulation is the conservation of heat. Industry, as well as the home owner, appreciated at an early date the advantages gained by insulation. Low-pressure steam boilers and engines were usually insulated with wood from the time of James Watt until the advent of manufactured insulation late in the nineteenth century. Years ago, farmers along the New England coast and in Nova Scotia banked dried eelgrass, a salt-water plant, around their houses in the fall in order to conserve the heat from their fireplaces and stoves during cold weather. This is still done today to a limited extent in some rural communities. Close (22) states that "The old Pierce House in Dorchester, Mass., built in 1635 and one of the oldest houses in America, was insulated with Zostera Marina stuffed between the studding." *Zostera marina* is simply the scientific name for eelgrass, which when dried becomes an excellent insulator.

The snowhouse of the Eskimo is a good example of the use of natural materials for insulation under severe conditions. Stefansson (94) describes the temperature conditions within one of these houses as follows:

−50°F	outdoor temperature.
−45	on the floor of alleyway just outside of door.
0	level with top of door.
20	on the bed platform.
40	at the level of one's shoulders.
60	just above one's head and about 1 ft below the highest point of dome.

Figure 1–1 shows a cross-sectional view of such a snowhouse. If the ice on the inside wall of these snowhouses begins to melt, the insulation is too good and the Eskimo goes outside and, with the aid of a knife, makes the wall thinner, thus increasing the rate of heat flow sufficiently to keep the inside wall surface just below the freezing point. The heat is supplied from an oil or blubber stove. The Eskimo is thus able to live through the severe arctic winters using only materials at hand for building and heating his home.

Since 1900, the manufacture of insulating materials has become a major industry in this country, supplying hundreds of different types of insulating materials for all ranges of temperature and other conditions.

It must not be taken for granted that insulation invariably retards the rate of heat flow. The addition of small amounts of insulation to small-diameter wires or tubes, such as a layer or two of asbestos paper or string, frequently increases the rate of heat transfer from the wire or tube to the ambient air. A United States patent issued in 1938 states that a small-diameter wire is insulated by wrapping asbestos string around the wire, thus improving the operation of a device because of the decreased heat loss. Actually, by laboratory test, it was shown that the rate of heat loss was increased by the addition of the

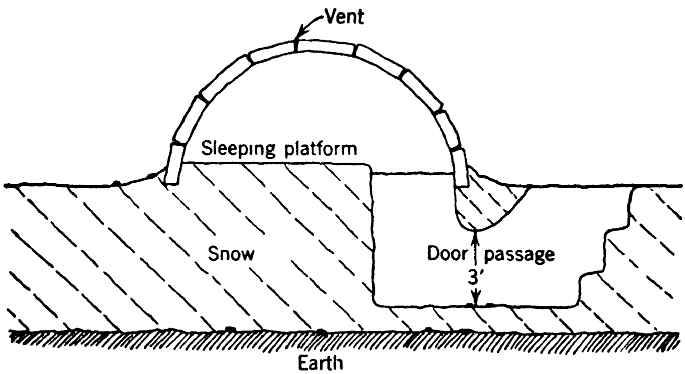


Fig. 1-1. Eskimo snowhouse.

asbestos string (see Chapter Two). The covering of bright tin warm-air ducts with several layers of asbestos paper generally increases the rate of heat loss from the ducts (see Chapter Eight).

Frequently the primary purpose of insulation is preventing water-vapor condensation on the inside of factory roofs where the manufacturing process requires or develops a high relative humidity, as in cotton or paper mills. Cold-water pipes in buildings are often insulated solely to prevent condensation and consequent dripping of water on warm, humid days.

The creation of more comfortable living or working conditions is another important reason for insulation. Research by the American Society of Heating and Ventilating Engineers (43*a*) proved that the temperature of a horizontal black roof reached 165°F on clear summer days. This temperature produces extreme discomfort for those in the area below such a roof. Apartment houses, hospitals, and homes would provide far more comfortable conditions on the top floor if the roofs were properly insulated. The maternity ward of a hospital located not far from Boston, Mass., was on the top floor beneath a flat tar and gravel roof. There was an unventilated air space 3 ft high between the roof and the ceiling. During heat waves in the summer months, the temperatures in this ward became almost unbearable. A single sheet of reflective insulation was attached to the ceiling joists, and the air space under the roof was ventilated. Upon the completion of these changes, the top floor of the hospital was just as desirable from a comfort standpoint during the summer as the lower floors. The expense was small since the insulation was applied by the hospital engineer and a helper.

Flues are sometimes insulated to improve working conditions and not with any intention of saving fuel.

For bake ovens and most furnaces, the addition of insulation generally produces a more uniform distribution of temperature. This holds true also for house insulation.

The low heat capacity of many insulators is a very important factor for intermittently heated furnaces, ovens, and buildings. In the Heat Measurements Laboratory, Massachusetts Institute of Technology, there were two experimental kilns: one was constructed of firebrick with insulating brick outside, and the other, of the same inside dimensions, was built only of insulating brick. When the two kilns were operating at a temperature of 2700°F continuously, their rates of fuel consumption were practically the same, but, with a cold start and the same rate of fuel consumption in each, the kiln of insulating brick only could be raised to a temperature of 2700°F in 1 hr, whereas it took 4 hr to bring the kiln lined with firebrick to the same temperature. The representative of a prominent manufacturer of insulating refractories once stated that he believed that his firm sold more insulating brick on account of their low heat capacity than on account of their insulating value.

In a similar manner, low heat capacity is important to refrigerated trucks, cars, or rooms that have to be chilled quickly. During World War II, several low-temperature chambers were constructed for the testing of airplanes under rapid temperature changes. These changes ranged from 60°F below zero to 100°F above zero, corresponding to the temperatures that an airplane might encounter as it came from a high altitude to the surface of the earth. Low heat capacity of the insulated walls of such test chambers was far more significant than the actual insulating value of the wall itself.

Exposed structural steel in buildings is considered a severe fire hazard, especially if the building contains combustible material. It has been common practice for many years to protect the steel with concrete, tile, brick, etc. The National Board of Fire Underwriters (68) made a large number of tests on loaded columns of different construction by exposing the columns to a temperature of 1700°F until they collapsed. The unprotected steel columns withstood these conditions for about 10 min, but similar steel columns protected with concrete withstood this severe test for 8 hr or more.

In 1915 a large theater was being designed in Boston, Mass., and, in order to avoid having any more columns than necessary, the structural-steel roof trusses had to be of considerable span. If these trusses had been protected with concrete, the weight would have increased materially, which would have meant heavier steel and more expense. It was suggested that the steel could be protected with 85% magnesia block insulation with considerable saving in cost and weight and at the same time be safe from fire. Tests were made on steel members insulated with 85% magnesia blocks by placing them in a furnace at 1700°F. Since approximately 2 hr were required for the temperature of the steel to reach 212°F, this type of construction was considered safe by the Building Commissioner and the theater roof trusses were installed accordingly. It is believed that this was the first time that lightweight insulation was employed for such a purpose.

The scope of this book does not cover acoustics, but it certainly should be mentioned that most heat insulators have excellent acoustical properties and they often serve to retard the transmission of sound through walls, ceilings and floors, and to absorb sound in auditoriums, class rooms, and offices.

Govan (32*a*) mentions the use of heavily insulated walls and roofs on fruit and vegetable storage buildings in Canada to prevent freezing under winter conditions without supplying heat to the building. Because of the large heat capacity of the fruits and vegetables and the low rate of heat transfer through the thick insulated walls, the temperature within the buildings could be kept above the freezing point throughout the winter. At times the outdoor temperatures went as low as 0°F, but cold spells were followed by warmer weather and the temperature inside fluctuated very little. Some heat was undoubtedly supplied from the ground under these structures as the floor was not insulated. The performance of these buildings is not extraordinary if one considers the fact that the ground does not freeze more than a few feet below the surface although the air temperature may go well below zero several times during the winter months. Modern low-temperature insulation is a much better insulator than frozen soil, so it requires only a reasonably well-insulated wall to prevent a building of this kind from reaching the freezing point.

White (105) mentions the very great gain in storage space in a small refrigerator or deep freeze unit which is achieved through better insulation with no change in the exterior dimensions. If the external dimensions of a box were 3 ft by 3 ft by 3 ft and if it were insulated with 4 in. of a material having a coefficient of thermal conductivity (K) of 0.34 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, the internal volume would be 12.4 cu ft. If an insulator had a K value of 0.15 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, and were only 2 in. thick, the internal volume would be 19.0 cu ft. This figure represents a gain of 52% in the internal volume with the same external dimensions and essentially the same rate of heat flow through the walls. The calculation is based on the assumption that there is no "through" metal, such as copper tubes, wires, etc., and that there is no greater heat loss around the doors than through the walls.

Close (22) mentions roof insulation to reduce the thermal expansion due to temperature changes. He calculates that, with the outside surface of the roof varying from −10°F to 140°F and the inside air temperature remaining constant at 70°F, the outer surface of a 6-in. concrete slab roof covered with a built-up roofing would vary from 3.9°F to 127.8°F, a change in temperature of 123.9°F. If 1 in. of insulation with a K value of 0.33 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, were placed between the concrete and the built-up roofing, Close figures that the outer surface of the concrete, under the same conditions as stated above, would vary from 47°F to 90.2°F, a change in temperature of 43.2°F. Since the temperature variation is only about one third as much for the insulated roof as for the uninsulated roof, the thermal expansion would be only about one third as much. The troubles from cracking and possibly pushing out of walls by the expansion of the roof would certainly be reduced by the insulation.

Nielsen (69) shows rather conclusively in his excellent paper that the familiar lath patterns caused by dirt on ceilings and walls of uninsulated buildings can be minimized by the proper insulation. Very fine particles of dirt tend to deposit on relatively cold surfaces if the adjoining surfaces are warm. Plaster is a much better conductor of heat than wood; consequently, on an uninsulated wall with an interior finish of wood lath and plaster, the wall surface between the laths is colder than that over the laths, during cold weather. This temperature difference tends to produce dirt streaks on the surface between the laths, thus making typical lath patterns. If insulation is placed back of the plaster, the rate of heat flow through the wall or ceiling is much reduced; therefore, the temperature difference between points on the surface over the laths and the space between is much reduced, and consequently the tendency to deposit dust in patterns is also reduced.

There are many other instances in which insulation has been installed for purposes other than saving heat, and it is hoped that these few examples have illustrated the possibilities.

CHAPTER TWO

Fundamental Formulas

Most of the formulas commonly employed for calculating the rate of heat flow through insulation under equilibrium or steady-state conditions are given in this chapter. The symbols in these formulas together with some customary units are shown in Table 2–1.

TABLE 2–1

SYMBOLS

A = area, sq ft.
A_{eff} = effective area, sq ft.
C = coefficient of thermal conductance, Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
C' = convection constant, Btu, hr^{-1}, ft^{-2}, $°F^{-5/4}$.
e = emissivity, dimensionless.
e_{eff} = effective emissivity, dimensionless.
E = combined radiation and convection coefficient for heat transfer from surface to air ($h_r + h_{cv}$), Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
h = coefficient of heat transfer, Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
h_{cv} = coefficient of heat transfer by convection, Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
h_r = coefficient of heat transfer by radiation, Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
K = coefficient of thermal conductivity, Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, or Btu, hr^{-1}, ft^{-2}, ft, $°F^{-1}$.
l = thickness, in. or ft.
q = quantity of heat per unit time, Btu, hr^{-1}.
q_{cd} = quantity of heat by conduction per unit time, Btu, hr^{-1}.
q_{cv} = quantity of heat by convection per unit time, Btu, hr^{-1}.
q_r = quantity of heat by radiation per unit time, Btu, hr^{-1}.
T = absolute temperature, $°F_{abs}$ or °R.
t = temperature, °F.
U = coefficient of thermal transmittance, Btu, hr^{-1}, ft^{-2}, $°F^{-1}$. Overall or air-to-air coefficient.
σ = Stefan-Boltzmann constant, 17.3×10^{-10} Btu, hr^{-1}, ft^{-2}, $°F_{abs}^{-4}$.

CONDUCTION

The basic formula for calculating the rate of heat flow by conduction under steady-state conditions was developed by Fourier in 1822 and may be expressed as follows:

$$q_{cd} = \frac{KA(t_2 - t_1)}{l} \tag{2-1}$$

In other words, the rate of heat flow by conduction through a flat slab is proportional to the area and the temperature difference between the two surfaces. It is inversely proportional to the thickness. K is a factor called the coefficient of thermal conductivity, its numerical value depending upon the particular material in question.

Obviously the units of K depend upon the units used in the other terms of Eq. 2–1. Unfortunately, engineers and scientists have reached no general agreement as to the proper units for K. Table 2–2 gives some of the units that are most common at present.

TABLE 2–2

UNITS OF K

q	A	$t_2 - t_1$	l	*Used by*
1. Joules, sec^{-1}	Cm^2	°C	Cm	Electrical engineers and *International Critical Tables*
2. Gm cal, sec^{-1}	Cm^2	°C	Cm	Physicists
3. Kg cal, hr^{-1}	$Meter^2$	°C	Meter	Foreign engineers
4. Btu, hr^{-1}	Ft^2	°F	In.	Mechanical and refrigerating engineers
5. Btu, hr^{-1}	Ft^2	°F	Ft	Chemical engineers
6. Btu, hr^{-1}	$In.^2$	°F	In.	
7. Watts	$Meter^2$	°C	Meter	Mks system (67)

K may be expressed as follows:

1. Joules, sec^{-1}, cm^{-2}, $°C^{-1}$, cm
2. Gm cal, sec^{-1}, cm^{-2}, $°C^{-1}$, cm
3. Kg cal, hr^{-1}, $meter^{-2}$, $°C^{-1}$, meter
4. Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, in.
5. Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, ft
6. Btu, hr^{-1}, $in.^{-2}$, $°F^{-1}$, in.
7. Watts, $meter^{-2}$, $°C^{-1}$, meter

There are many other ways of expressing the units of K but there is no general agreement as to the best method. One common way is Btu/hr/sq ft/°F/in. This expression, however, is meaningless as regards the location of the units in the denominator or numerator of the equation and should be discouraged.

TABLE 2-3

Conversion Factors for K

Joules, sec^{-1}, cm^{-2}, $°C^{-1}$, cm	Gm cal, sec^{-1}, cm^{-2}, $°C^{-1}$, cm	Kg cal, hr^{-1}, m^{-2}, $°C^{-1}$, m	Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, in.	Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, ft	Btu, hr^{-1}, $in.^{-2}$, $°F^{-1}$, in.	Watts, m^{-2}, $°C^{-1}$, m
1.000	0.239	86.1	693	57.7	4.81	100
4.185	1.000	360	2903	242	20.2	419
0.0116	0.00278	1.000	8.06	0.672	0.0560	1.16
0.00144	0.000345	0.124	1.000	0.0833	0.00695	0.144
0.0173	0.00414	1.49	12.0	1.000	0.0835	1.73
0.208	0.0497	17.9	144	12.0	1.000	20.8
0.0100	0.00239	0.861	6.93	0.577	0.0481	1.000

Example. The K value for an insulating material of foreign manufacture is given as 0.041 kg cal, hr^{-1}, $meter^{-2}$, $°C^{-1}$, meter. To convert this value into other units, run down the column headed Kg cal, hr^{-1}, m^{-2}, $°C^{-1}$, m, until the value of 1.000 is found, and then move horizontally to the column headed by the desired units and multiply the original K value by the factor shown. Thus a K value of 0.041 kg cal, hr^{-1}, $meter^{-2}$, $°C^{-1}$, meter is equal to $0.041 \times 0.672 = 0.0275$ Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, ft or to $0.041 \times 0.00278 = 0.000114$ gm cal, sec^{-1}, cm^{-2}, $°C^{-1}$, cm.

For mathematical calculations involving calculus, the area and thickness should be expressed in consistent units. For example, one cannot give the area in square feet and the thickness in inches.

Cylindrical shells

The fundamental formula, Eq. 2-1, applies to all problems involving the determination of the rate of heat flow through insulation under steady-state conditions, provided that the proper area is used. In many situations, however, such as in pipe insulation, the area across which the heat is flowing becomes greater as the thickness increases, and one requires special formulas.

Figure 2-1 illustrates a cross-section of a cylindrical shell of insulation corresponding to a pipe covering. Taking a thin cylindrical shell of radius ρ, thickness $d\rho$, and unit length, and representing the temperature drop across this shell by dt, we find that Eq. 2-1 gives

$$q = \frac{K 2\pi\rho\, dt}{d\rho}$$

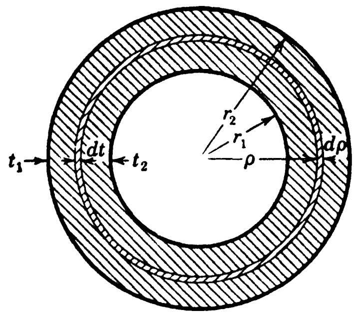


FIG. 2-1. Cylindrical shell. Derivation of conduction formula.

Setting up the proper limits for the shell shown in Fig. 2-1 and transposing, we have

$$q \int_{r_1}^{r_2} \frac{d\rho}{\rho} = 2\pi K \int_{t_1}^{t_2} dt$$

Integrating and transposing, we obtain the formula to determine the rate of heat flow through a cylindrical shell of *unit length.*

$$q = \frac{2\pi K(t_2 - t_1)}{\ln (r_2/r_1)} \tag{2-2}$$

where ln is the natural or Naperian logarithm.

Since Eq. 2-1 applies to this problem if the proper area is used,

$$\frac{A}{l} = \frac{2\pi}{\ln (r_2/r_1)}$$

and

$$A = \frac{2\pi(r_2 - r_1)}{\ln (r_2/r_1)} \tag{2-3}$$

which is called the *log mean area* (unit length).

The arithmetic mean area per unit length is equal to $\pi(r_2 + r_1)$, and considerable error would be involved if this were used in place of the log mean area when the ratio of r_2/r_1 is large. Table 2-4 shows how the error increases as the ratio of r_2/r_1 becomes larger.

TABLE 2-4

UNIT LENGTH (INCHES) OF TUBE, $r_1 = 1$ INCH (FIG. 2-2)

$r_2 - r_1$	Arithmetic Mean A	Log Mean A	Ratio r_2/r_1	Percentage of Error When Arithmetic Mean A Replaces Log Mean A
0.5 in.	7.85 in.^{2}	7.75 in.^{2}	1.5	1.3
1.0	9.43	9.07	2.0	4.0
2.0	12.6	11.5	3.0	10.0
4.0	18.8	15.6	5.0	20.5
10.0	37.7	26.2	11.0	43.9

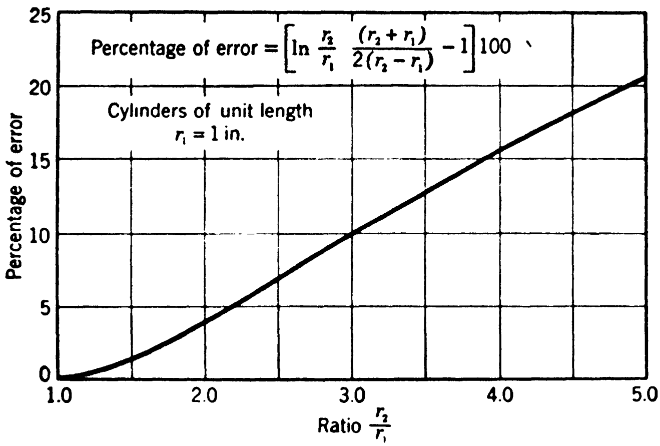


FIG. 2-2. Error caused when arithmetic mean area replaces log mean area.

With relatively large cylindrical tanks, the ratio r_2/r_1 approaches unity, and no appreciable error is caused by employing the arithmetic mean area.

In comparing the insulating value of pipe coverings, the engineer usually makes a comparison of the rate of heat loss per square foot of pipe surface. Equation 2-2 may be readily converted to give the rate of heat flow per unit area of pipe surface by dividing the log mean area per unit length by $2\pi r_1$.

$$q = \frac{K(t_2 - t_1)}{r_1 \ln (r_2/r_1)} \quad \text{(unit area pipe surface)} \qquad (2\text{-}4)$$

Spherical shells

The equation for determining the rate of heat flow through the insulation of spheres is derived in a similar manner to that for cylindrical shells.

Let Fig. 2–1 represent the cross-section, passing through the center, of an insulated sphere. Taking a thin spherical shell of radius ρ, thickness $d\rho$, and with a temperature drop across the shell of dt, we find by Eq. 2–1 that the rate of heat flow by conduction is

$$q = \frac{K 4\pi\rho^2\, dt}{d\rho}$$

Using the proper limits as shown in Fig. 2–1 and transposing, we have

$$q \int_{r_1}^{r_2} \frac{d\rho}{\rho^2} = 4\pi K \int_{t_1}^{t_2} dt$$

Completing the integration and transposing, we obtain the equation for determining the rate of heat flow through spherical shells.

$$q = \frac{4\pi r_1 r_2 K(t_2 - t_1)}{r_2 - r_1} \tag{2–5}$$

Comparing Eq. 2–5 with Eq. 2–1 shows that

$$\frac{A}{l} = \frac{4\pi r_2 r_1}{r_2 - r_1}$$

and

$$A = 4\pi r_2 r_1 = \sqrt{A_1 A_2} \tag{2–6}$$

where A_1 and A_2 are the inside and outside areas of the insulation. This value $\sqrt{A_1 A_2}$ is known as the *geometric mean area*.

The error caused by taking the arithmetic mean of the inside and outside areas of a spherical shell instead of the geometric mean area is indicated in Table 2–5.

TABLE 2-5

SPHERICAL SHELL, $r_1 = 2$ IN. (FIG. 2-3)

$r_2 - r_1$	Arithmetic Mean A	Geometric Mean A	Ratio r_2/r_1	Percentage of Error When Arithmetic Mean A Replaces Geometric Mean A
0.2 in.	55.6 in.^{2}	55.3 in.^{2}	1.1	0.45
0.5	64.4	62.8	1.25	2.50
1.0	81.6	75.3	1.50	8.33
2.0	125.5	100.3	2.0	25
4.0	251	151	3.0	66.7

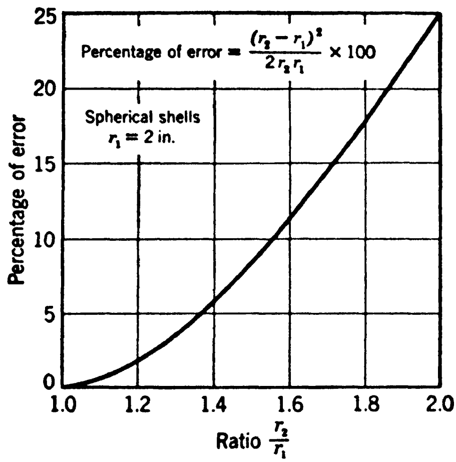


FIG. 2-3. Error caused when arithmetic mean area replaces geometric mean area.

Rectangular parallelepipeds

The formulas for calculating the rate of heat flow through slabs or plates with infinite parallel surfaces, and through the walls of hollow cylinders or spheres, are rigidly correct, provided that K does not vary with temperature for the range in question.

Many furnaces and ovens have a rectangular shape. If the walls are thin compared to the linear dimensions of the furnace, the rate of heat flow may be computed by Eq. 2-1, with A equal to the arithmetic mean area, $(A_1 + A_2)/2$, of the inside and outside area. The geometric mean area, $\sqrt{A_1 A_2}$, would be more accurate. If, however, the walls are relatively thick compared to the linear furnace dimensions, the above method introduces serious error because, although the area through which the heat is flowing increases with the square of the distance from the center of the furnace, the heat flow is not perpendicular to these areas, particularly at the edges and corners.

The mathematical calculation for this latter problem is quite involved. Carlslaw and Jaeger (20), Langmuir, Adams, and Meikle (54), and McAdams (60), have suggested the following equations, which approximate the mean area to be taken in Eq. 2–1 for rectangular parallelepipeds with uniform wall thickness.

Case 1. All inside edges, e, are greater than $\frac{1}{5}l$ and less than $2l$.

$$A_{eff} = A_1 + 0.54 \times l \times \Sigma e + 1.2l^2 \tag{2–7}$$

where A_1 = inside area.
Σe = sum of the twelve inside edges.
l = thickness of wall.

Case 2. Two inside dimensions greater than $\frac{1}{5}l$, and one inside dimension less than $\frac{1}{5}l$.

$$A_{eff} = A_1 + 0.465 \times l \times \Sigma e \tag{2–8}$$

Case 3. One inside dimension greater than $\frac{1}{5}l$, and two inside dimensions less than $\frac{1}{5}l$.

$$A_{eff} = \frac{2.78El}{\log (A_2/A_1)} \tag{2–9}$$

where E = longest inside dimension.

Case 4. All three inside dimensions less than $\frac{1}{5}l$.

$$A_{eff} = 0.79\sqrt{A_1A_2} \tag{2–10}$$

where A_1 = inside area.
A_2 = outside area.

Example. A rectangular oven with inside dimensions of 1 ft, 1.5 ft, and 1.5 ft is insulated with a wall 1 ft thick. Assuming that the inside and outside surfaces are each isothermal, the conditions correspond to Case 1 (Eq. 2–7).

$$A = 10.5 + 0.54 \times 1 \times 16 + 1.2 \times 1^2 = 20.4 \text{ sq ft}$$

The geometric mean area would be 26.4 sq ft or 29% greater, and the arithmetic mean area would be 38.5 sq ft or nearly 100% greater.

If all the inside dimensions are more than twice the wall thickness, the geometric mean area is sufficiently precise. It is suitable in most industrial applications of insulation, and the rate of heat flow is

$$q = \frac{K\sqrt{A_1A_2}(t_2 - t_1)}{l} \tag{2-11}$$

For large furnaces, ovens, etc., the arithmetic mean area, $(A_1 + A_2)/2$, can be employed without any significant error. If the lengths of all inside edges are at least ten times the wall thickness, the arithmetic mean area can be used with an error of less than 2%.

RADIATION

A knowledge of the laws of radiation is essential to one who would be well versed in the subject of heat insulation. Radiation is of considerable significance as a means of heat transfer at room temperature and even lower, although frequently it is erroneously assumed to be important only in dealing with high temperatures. Experiment has determined that radiation is responsible for roughly two thirds of the heat transferred across a vertical air space, bounded by wood or plaster, such as found in the walls of typical frame construction. Reflective insulation (Chapter Six) is primarily based on a means of minimizing the transfer of heat by radiation across air spaces. Radiation between the inside surface of the outer walls of a house and the interior is roughly equal in value to the transfer of heat by natural convection.

The propagation of heat by radiation involves the same laws that apply to visible radiation or light. The visible radiation spectrum represents merely a small portion of the heat radiation spectrum, which extends to much longer wave lengths than are visible to the human eye. Radiation is the fastest known means of transferring heat since it travels with the speed of light, or about 180,000 miles per sec.

The fourth-power law for radiation from black bodies, determined experimentally by Stefan, 1879, and theoretically by Boltzmann, 1884, states that

$$q_r = \sigma T^4 \quad \text{(unit area)} \tag{2-12}$$

A *black body*, or perfect radiator, is a theoretical body that would absorb all radiation falling on it, reflecting and transmitting none. No known material has such properties, but some opaque bodies approach the absorbing power of a black body within a few per cent whereas others absorb only a small fraction of the incident radiation. If a nonblack body were placed in an enclosure, the walls of which were maintained at a uniform temperature, the temperature of the nonblack body would eventually be the same as that of the walls, and thermal equilibrium would be established. The nonblack body would still be absorbing radiation from the walls and, with the temperature constant, it would have to emit or radiate heat at the same rate as it absorbs. Any radiant heat from the walls striking the nonblack body which was not absorbed would be reflected, if we assume the nonblack body to be opaque.

$$e = a = 1 - r \tag{2-13}$$

where e = emissivity, or the ratio of heat radiated by a body to that of a black body under the same conditions.
a = absorptivity, or the ratio of radiant heat absorbed by a body to that of a black body under similar conditions.
r = reflectivity, or the ratio of radiant heat reflected by a body to that of a black body under similar conditions.

Generally we are concerned with the *net* transfer of heat between a surface and its surroundings. The surface is receiving radiant heat from the surroundings and at the same time is emitting radiation to the surroundings. The difference between what is absorbed and what is emitted is the *net* transfer of heat by radiation. It may be expressed as follows:

$$q_r = A_{eff} e_{eff} \sigma (T_2^4 - T_1^4) \tag{2-14}$$

where e_{eff} = the effective emissivity.
A_{eff} = the effective area.

Parallel planes

For infinite parallel planes at absolute temperatures T_2 and T_1 and having emissivities of e_2 and e_1, it will be shown that the effective emissivity is equal to

$$e_{eff} = \frac{1}{(1/e_1) + (1/e_2) - 1} \tag{2-15}$$

To derive Eq. 2–15, consider the two large parallel planes 1 and 2 (Fig. 2–4) that have temperatures T_1 and T_2 and emissivities e_1 and e_2, respectively. If we determine all the

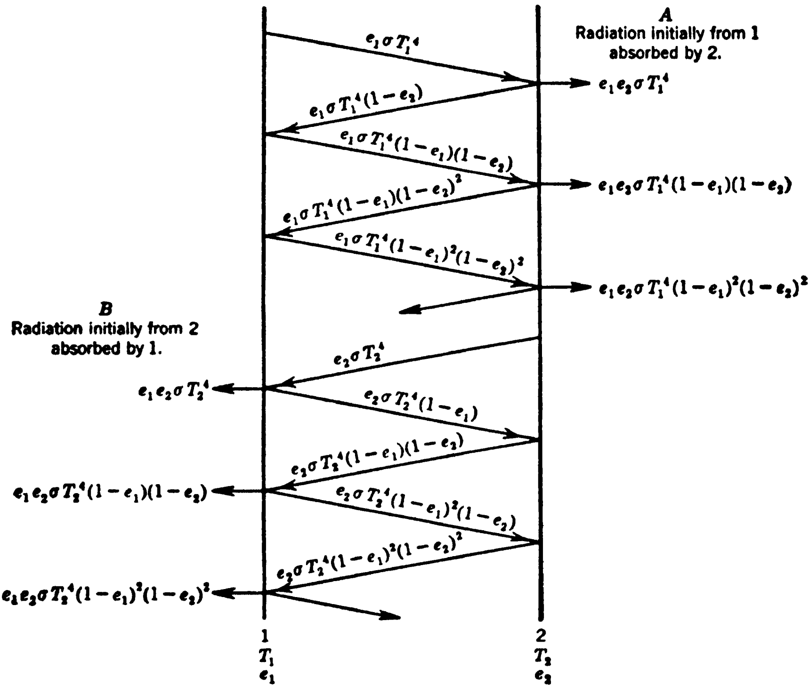


FIG. 2–4. Derivation of effective emissivity formula. Infinite parallel planes.

radiant heat, initially from plane 1, that is absorbed by plane 2 and subtract from it all the radiant heat, initially from plane 2, that is absorbed by plane 1, we can determine the net exchange of heat between the two planes.

The net transfer of heat by radiation per unit area is equal to the total of column A minus the total of column B (Fig. 2–4).

Let $z = (1 - e_1)(1 - e_2)$. Then

$$A - B = e_1e_2\sigma T_1^4(1 + z + z^2 + \cdots) - e_1e_2\sigma T_2^4(1 + z + z^2 + \cdots)$$

The limit of the series $(1 + z + z^2 + \cdots)$, if $z < 1$, is $1/(1 - z)$.

$$A - B = \frac{e_1 e_2 \sigma T_1^4}{e_1 + e_2 - e_1 e_2} - \frac{e_1 e_2 \sigma T_2^4}{e_1 + e_2 - e_1 e_2}$$

$$= \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} \sigma (T_1^4 - T_2^4)$$

Comparing this equation with Eq. 2–14, keeping in mind that this derivation is for unit area, we can readily see that

$$e_{eff} = \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} = \frac{1}{(1/e_1) + (1/e_2) - 1}$$

Equation 2–15 can serve to determine the effective emissivity for finite parallel planes as long as their area is large compared to the distance between them. The effective emissivity for reflective insulation placed between the studs, joists, or rafters of a frame dwelling can be calculated by it without serious error.

Small body in relatively large enclosure

The walls of a large enclosure, assuming that they all are at the same temperature, absorb the radiation from a small body, like a black body, because of multiple reflection from one portion of the wall to another with some absorption (the amount depending upon the emissivity of the wall surface). The emissivity of the walls in such a situation has no appreciable effect on radiant-heat transfer. In Eq. 2–14, e_{eff} = emissivity of small body, A_{eff} = area of the small body.

Enclosed body (convex surface) and enclosing body (concave surface)

Christiansen in 1883 (21) derived the following formula for a problem of this type:

$$q_r = \frac{1}{(1/e_1) + (A_1/A_2)[(1/e_2) - 1]} \sigma A_1 (T_1^4 - T_2^4) \quad (2\text{–}16)$$

where A_1 = area of the enclosed body.
A_2 = area of the enclosing body.
e_1 = emissivity of the enclosed body.
e_2 = emissivity of the enclosing body.

If the enclosing surface has an emissivity of 0.9 or greater, the ratio of A_1/A_2 may be as great as $\frac{1}{5}$, with no important error involved if e_{eff} is taken as the emissivity of the enclosed body. If the enclosing body has an emissivity of 0.05, the ratio A_1/A_2 must be less than $\frac{1}{20}$ if the error, due to the use of e_{eff} as the emissivity of the enclosed body, is less than 4%.

Equal and opposite parallel squares

For equal and opposite parallel squares, Hottel (43) recommends the effective areas shown in Fig. 2–5.

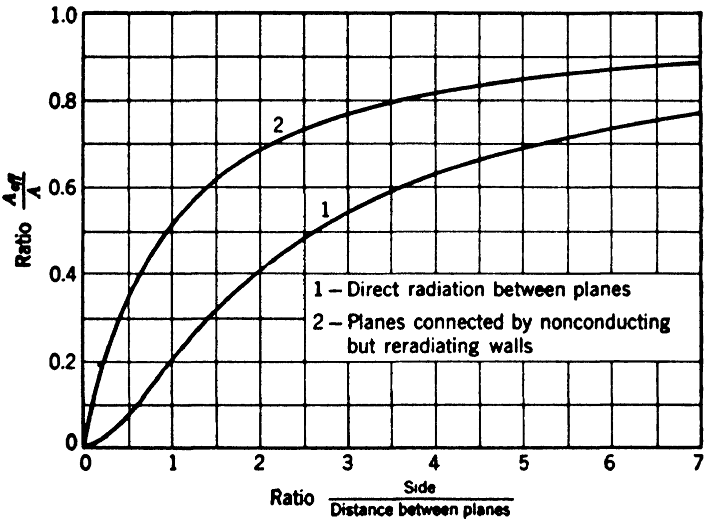


FIG. 2–5. Effective area. Radiation between equal and opposite squares.

The exact effective emissivity for this case is unknown but lies between e_1e_2 and $1/[(1/e_1) + (1/e_2) - 1]$, nearer e_1e_2 when the areas are small compared to the distance between them, and nearer $1/[(1/e_1) + (1/e_2) - 1]$ when the areas are large compared to the distance between them. If the emissivity of either square is 0.9 or greater, the difference between the two limiting values given above is slight.

Example. **Find the rate of radiant-heat transfer from the ceiling to the floor of a room 15 ft by 15 ft by 10 ft high. The ceiling temperature is 100°F and the floor 68°F. The emissivity of both floor and ceiling is**

0.90. The walls of the room will be considered nonconducting and reradiating.

$$A_1 = A_2 = 15 \times 15 = 225 \text{ ft}^2$$

$$T_1 = 560°\text{R} \quad \text{and} \quad T_2 = 528°\text{R}$$

$$e_{eff} = e_1 e_2 = 0.81$$

Ratio of side of square to distance apart = 1.5

$$A_{eff} = 0\ 61 \times 225 = 137 \text{ ft}^2$$

$$\begin{aligned} q_r &= e_{eff} A_{eff} \sigma (T_1^4 - T_2^4) \\ &= 0.81 \times 137 \times 17.2 \times 10^{-10} (560^4 - 528^4) \\ &= 4100 \text{ Btu, hr}^{-1} \end{aligned}$$

CONVECTION

The transfer of heat by convection is usually treated either as forced convection or natural (free) convection. The subject of forced convection is extremely important to those interested in the promotion of heat transfer, and information concerning the methods of calculation may be found in McAdams (60), Fishenden and Saunders (31), Jakob and Hawkins (46), and many others. Since forced convection is usually of minor importance where insulation is effective, the subject will not be treated here.

The fundamental laws governing heat transfer by radiation and conduction have been determined both experimentally and theoretically and have a sound basis. The laws applying to heat transfer by natural convection from a surface to the ambient air or between two surfaces separated by an air space are based upon experiment only and are empirical. The factors affecting the rate of heat transfer by natural convection are more numerous and more difficult to determine than those affecting conduction and radiation. The rate of heat transfer per unit area from a vertical surface at a constant temperature above that of the ambient air varies with the height up to approximately 2 ft and then it remains essentially constant for greater heights. Experimental values determined on vertical surfaces with heights less than 2 ft cannot be safely assumed to apply to heights over 2 ft. If a vertical surface is oriented to a horizontal position with the heat flowing upward, the convection loss is increased materially, and again it varies with the dimensions of the plate. Experimental data are meager for natural convection, and, consequently, calculated results are only approximate values.

Fortunately, natural convection from the surface of a well-insulated wall usually has but a slight effect on the rate of heat transfer through the wall, and rough approximations of the convection loss are sufficient. The effect becomes increasingly important as the amount of insulation diminishes. Of the total resistance of a 6-in. corkboard wall exposed to still air on both sides, the resistance offered to heat transfer by natural convection at the two faces amounts to about 4%. Consequently, if the convection resistance were known to within 25% of its true value, the rate of heat transfer through the wall could be calculated to within 1%, which is more precise than the conditions usually warrant. With a ½-in. fiberboard partition, convection is responsible for roughly 33% of the total resistance to heat transfer from the air on one side to the air on the other. If the convection is known only to 25% of its true value, the error in calculating the total heat transfer may be nearly 10%.

Convection from horizontal cylinders to air

Experimental data for the rate of heat loss by natural convection from horizontal cylinders to air at room temperature are fairly abundant. McAdams (60) has correlated these data and recommends the following formula:

$$h_{cv} = 0.415\left(\frac{\Delta t}{D}\right)^{1/4} \qquad (2\text{-}17)$$

where h_{cv} = Btu, hr^{-1}, ft^{-2}, °F^{-1}.

Δt = average temperature difference between cylinder surface and ambient air, °F.

D = diameter of cylinder, in.

Range of 0.55-in. to 10.75-in. diameter, Δt from 28°F to 700°F.

The available data for convection losses from large plates in various positions are not of sufficient quantity to give anything but approximate formulas. The bulk of the experimental evidence indicates that natural convection varies approximately with the ⁵⁄₄ power of the temperature difference except for horizontal surfaces, heat flowing downward.

Vertical surfaces

The coefficient of heat transfer by natural convection, h_{cv}, for large vertical plates in air is shown in Fig. 2-6 from data by

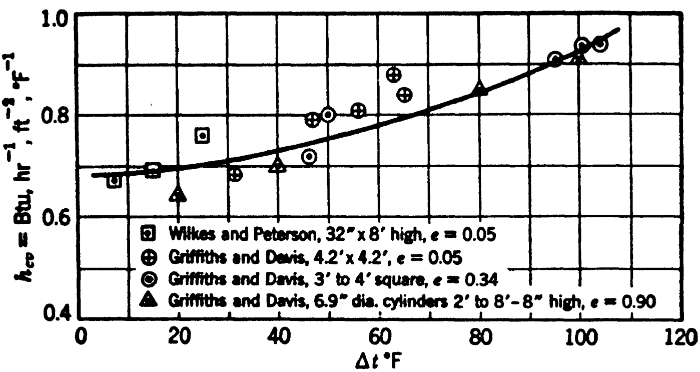


FIG. 2-6. Natural convection. Vertical surfaces >2 ft high.

Griffiths and Davis (35) and Wilkes and Peterson (115*b*). These experiments covered heights from 2 ft to 8 ft, and it is apparent that the coefficient, h_{cv}, for this range is not affected by height. Vertical cylinders 6.9 in. in diameter indicated very little difference from vertical plates of the same height with reference to heat loss by natural convection.

For heights less than 2 ft, the value of h_{cv} becomes greater as the height diminishes. Griffiths and Davis made tests on 6.9-in.-diameter cylinders of heights varying from 1.83 in. to 23 in. For heights less than 2 ft, their value of h_{cv} varies with the height as well as the ⁵⁄₄ power of the temperature difference, and so in Fig. 2-7 a convection constant, $C' = h_{cv}/(\Delta t^{1/4}) = q_{cv}/(\Delta t^{5/4})$, is plotted against height. Above 2 ft, the value of C' is 0.30 Btu, hr^{-1}, ft^{-2}. $°F^{-1}$.

Griffiths and Davis offer the following suggestion as to the mechanism of the transfer of heat by natural convection from a vertical plane to the ambient air. The air close to the surface of the plane becomes heated and rises, inducing a draft of cool air to flow upward close to the lower portion of the plane. Heat

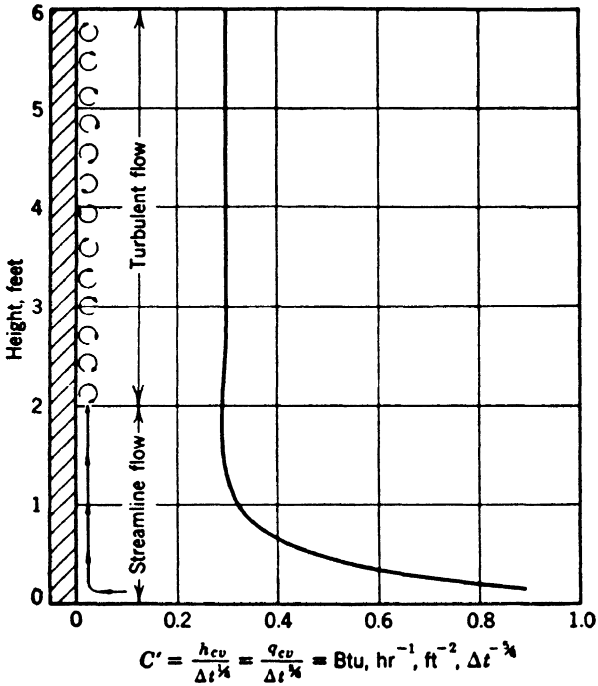


FIG. 2-7. Natural convection. Vertical surfaces, variation with height.

transfer is rapid, as the temperature difference between the air close to the surface and the surface is maximum. The air as it rises becomes warmer, and, consequently, the rate of heat transfer diminishes even though the velocity is increasing. At a height of approximately 2 ft, the velocity increases sufficiently to produce turbulent flow instead of streamline flow. The temperature difference between the air and the plate then becomes nearly constant, as well as the velocity, and consequently the value of h_{cv} is constant with respect to height above 2 ft. Figure 2-7 illustrates this explanation diagrammatically.

Horizontal surfaces

The values of h_{cv} for horizontal plates with heat flow upward or downward is based on rather meager data. Figure 2–8 shows the results found by Griffiths and Davis (35) and Wilkes and Peterson (115*b*). These results agree fairly well for vertical plates but they do not agree for the horizontal position with heat flow up or down. The larger plates, 32 in. by 8 ft, of

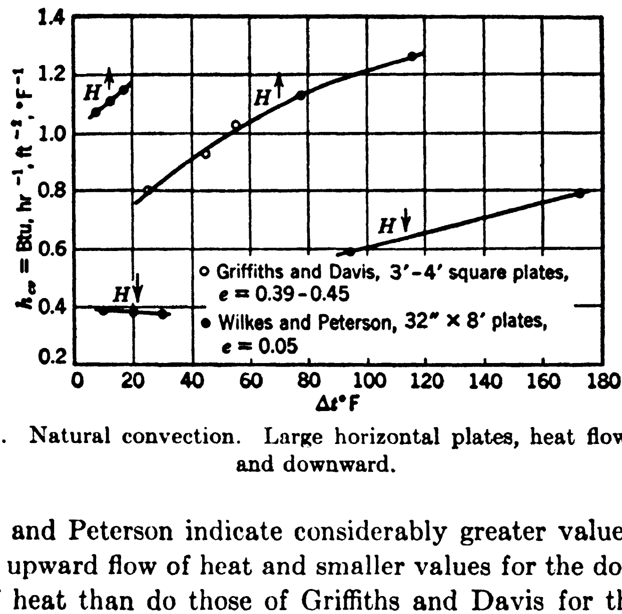


FIG. 2–8. Natural convection. Large horizontal plates, heat flow upward and downward.

Wilkes and Peterson indicate considerably greater values of h_{cv} for the upward flow of heat and smaller values for the downward flow of heat than do those of Griffiths and Davis for the same positions. This difference may be entirely due to the larger plates and is an indication, at least, that h_{cv} for horizontal plates with heat flow upward increases with the size of the plate whereas h_{cv} for horizontal plates with heat flowing downward diminishes with the size of the plate. More experimental work is necessary before definite conclusions can be drawn as to how h_{cv} varies with size and shape of horizontal plates.

Figure 2–9 shows the results of Wilkes and Peterson on h_{cv} for plates 32 in. by 8 ft for intermediate positions between the horizontal and vertical positions.

Figure 2–10 illustrates diagrammatically a suggestion as to why h_{cv} for horizontal plates of the same size is larger for heat

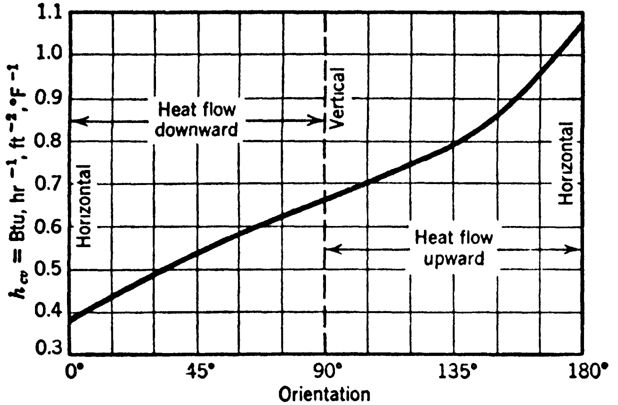
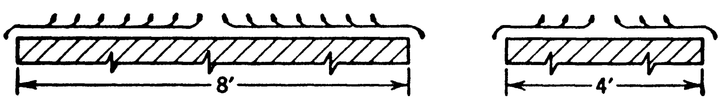
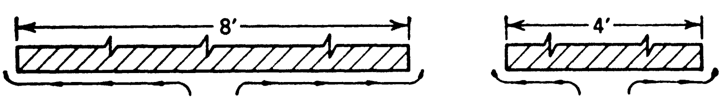


FIG. 2-9. Natural convection. Large plates, h_{cv} vs. orientation.



Air tends to leave surface as soon as heated. The larger the area, the greater the rate of heat transfer, owing to greater velocity of air over surface and to the larger Δt between surface and air.

FIG. 2-10*a*. Heat flow upward from horizontal plates.



Air tends to cling to heated surface. The larger the area, the less the rate of heat transfer, owing to smaller Δt between surface and air.

FIG. 2-10*b*. Heat flow downward from horizontal plates.

flow upward than for heat flow downward and also why h_{cv} tends to become greater with increasing size of plates for heat flow upward and tends to become smaller for heat flow downward with increasing size of plates.

The transfer of heat by natural convection from one surface to another will be covered in Chapter Six.

SURFACE COEFFICIENTS COMBINING RADIATION AND CONVECTION

It has been shown that the law governing the rate of heat flow by natural convection from a heated plate to the ambient air is different from that controlling the rate of heat transfer by radiation from the plate to its surroundings. When this surface resistance to heat flow is an important factor in determining the rate of heat transmission through a structure, radiation and convection must be treated independently, as shown earlier in this chapter.

For well-insulated walls, resistance to heat flow from the surface to the surroundings can be shown to be a relatively small part of the total, and, although it must be considered, it need not be calculated accurately. In such cases the calculation of the rate of heat transmission is much simplified if one employs a combined radiation and convection coefficient, E, that can be expressed in Btu, hr^{-1}, ft^{-2}, $°F^{-1}$. This combined coefficient is applicable only when the temperature difference between the surface and the air is slight and when the surroundings that receive the radiation are approximately at the same temperature as the air.

The value of E varies with the orientation of the surface since convection is affected to a marked extent, and it varies also with the emissivity of the surface since radiation is directly proportional to the emissivity. For temperature differences up to 20°F, both radiation and convection are roughly proportional to the temperature difference.

Figure 2–11 shows approximate values of E for flat surfaces over 2 ft square for various positions and emissivities if the temperature difference is less than 20°F. The values are calculated from Wilkes and Peterson (115*b*) and are based on a surface 32 in. by 8 ft.

It has been the practice for years by many engineers to employ a surface coefficient of $E = 1.65$ Btu, hr^{-1}, ft^{-2}, $°F^{-1}$ for calculating the combined heat loss by radiation and convection from ordinary surfaces of high emissivity, such as wood, stone, plaster, and glass, for any position and any size of surface. The

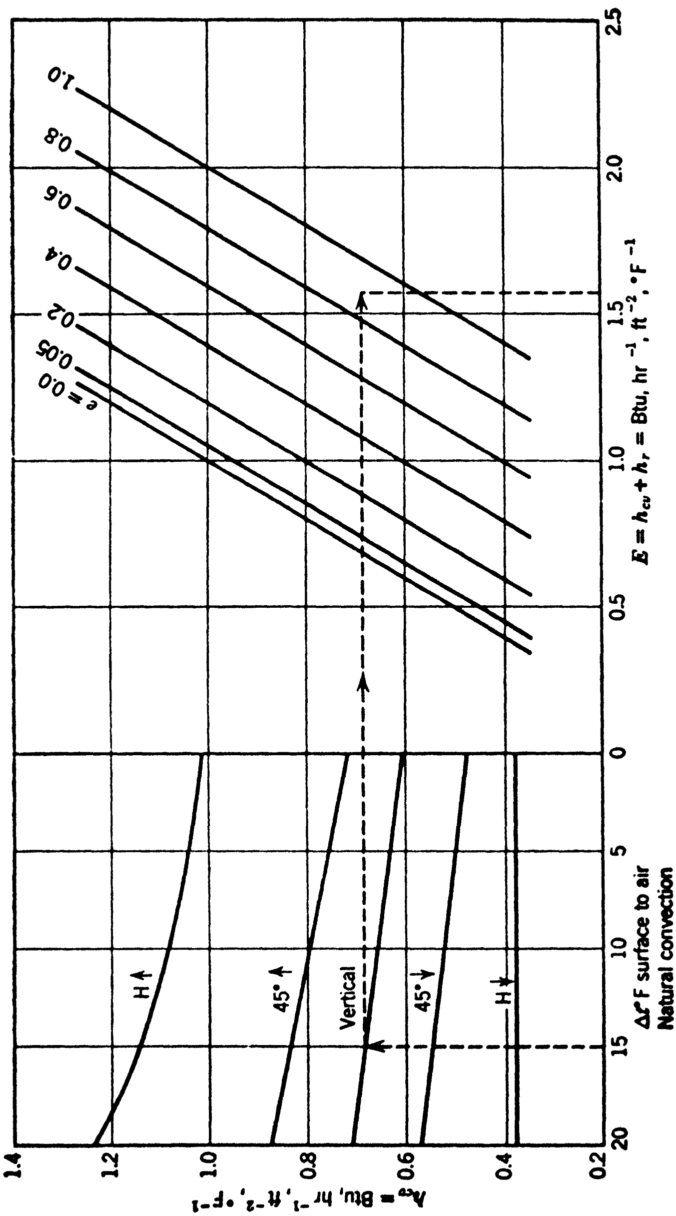


FIG. 2–11. Surface coefficient, E, vs. Δt, orientation and emissivity.

accompanying table derived from Fig. 2–11 shows the variation in E for surfaces in various positions and for two emissivities.

VALUES OF E OR $h_{cv} + h_r$ WITH 15°F, Δt

	Ordinary Surfaces, $e = 0.90$	Aluminum Foil, $e = 0.05$
Horizontal, heat upward	2.05	1.21
45° slope, heat upward	1.75	0.89
Vertical	1.59	0.74
45° slope, heat downward	1.46	0.60
Horizontal, heat downward	1.29	0.43

If the conditions of a problem permit a combined surface coefficient E, one should certainly take into account the orientation of the surface and the approximate temperature difference between the surface and its surroundings.

Compound walls

In heat transfer, the resistance to heat transmission generally consists of more than one factor. A fiberboard partition offers three resistances to heat transfer from the air on one side to the air on the other side, namely (1) the resistance to heat transfer from the air to the surface on one side, (2) the resistance to heat transfer by conduction through the fiberboard, and (3) the resistance to heat transfer from the surface to the air on the other side.

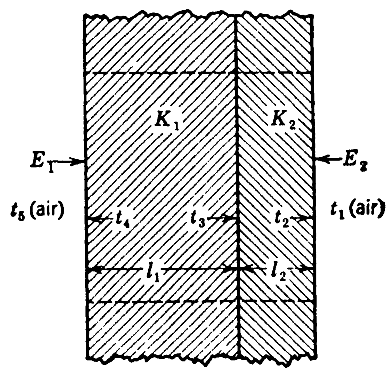


FIG. 2–12. Compound wall. Constant area.

Figure 2–12 illustrates the cross-section of a wall consisting of two different materials. The rate of heat flow per unit area through this wall can be calculated as follows:

Under equilibrium conditions, the rate of heat flow per unit area entering the surface on one side must be the same as that passing through each of the two insulators and also that leaving the other surface, and so

$$q = E_1(t_5 - t_4) = \frac{K_1(t_4 - t_3)}{l_1} = \frac{K_2(t_3 - t_2)}{l_2} = E_2(t_2 - t_1)$$

or

$$q = \frac{t_5 - t_4}{1/E_1} = \frac{t_4 - t_3}{l_1/K_1} = \frac{t_3 - t_2}{l_2/K_2} = \frac{t_2 - t_1}{1/E_2}$$

Combining these equalities, we have

$$q = \frac{t_5 - t_1}{(1/E_1) + (l_1/K_1) + (l_2/K_2) + (1/E_2)} \tag{2-18}$$

If the area varies from one side to the other (Fig. 2-13), the rate of heat transmission is

$$q = \frac{t_5 - t_1}{(1/A_1E_1) + (l_1/A_{2eff}K_1) + (l_2/A_{3eff}K_2) + (1/A_4E_2)} \tag{2-19}$$

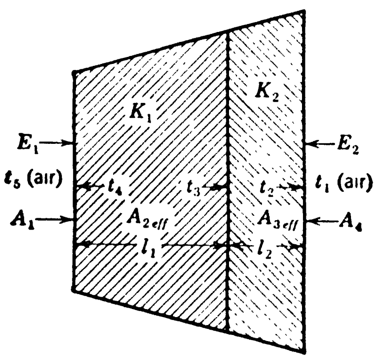


FIG. 2-13. Compound wall. Increasing area.

Frequently, it is important to determine the temperature of the wall surface or some intermediate point within the wall. The terms in the denominator of Eq. 2-18 represent the relative resistance to heat transmission for each portion, and so they are a measure of the relative drop in temperature across each portion. The temperature of the warmer surface in Fig. 2-12 would be

$$t_4 = t_5 - \frac{1/E_1}{(1/E_1) + (l_1/K_1) + (l_2/K_2) + (1/E_2)}(t_5 - t_1) \tag{2-20}$$

Assuming that both surfaces of the wall (Fig. 2-13) have an emissivity of 0.90 and that the temperature drop from the surfaces to the air is small, say 3°F for a first approximation, we find by Fig. 2-11 that $E_1 = E_2 = 1.53$ Btu, hr^{-1}, ft^{-2}, $°F^{-1}$. If $t_5 = 70°F$ and if $t_1 = 10°F$,

$K_1 = 0.27$ Btu, hr^{-1}, ft^{-2}, in., °F^{-1}

$K_2 = 0.90$ Btu, hr^{-1}, ft^{-2}, in., °F^{-1}

$l_1 = 4.0$ in.

$l_2 = 2.0$ in.

The temperature of the warmer surface would be

$$t_4 = 70 - \frac{1/1.53}{(1/1.53) + (4/0.27) + (2/0.90) + (1/1.53)}(70 - 10)$$

$$= 70 - 2.1 = 67.9°\text{F}$$

Reference to Fig. 2-11 shows that the surface coefficient for a Δt of 2.1°F is essentially the same as for the assumed 3°F, Δt. Consequently, the value obtained by the first approximation is sufficiently precise.

For compound cylindrical shells, the rate of heat transmission per unit length can be found with the following equation derived in the same way as Eq. 2-19. See Fig. 2-14.

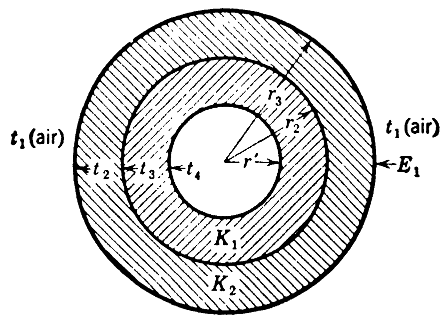


FIG. 2-14. Compound wall. Cylindrical shell.

$$q = \frac{t_4 - t_1}{\frac{\ln (r_2/r_1)}{2\pi K_1} + \frac{\ln (r_3/r_2)}{2\pi K_2} + \frac{1}{2\pi r_3 E_1}} \quad \text{(unit length)} \qquad (2\text{–}21)$$

where t_4 = 700°F.

t_1 = 100°F.

r_1 = 2 in. or 0.167 ft.

r_2 = 4 in. or 0.333 ft.

r_3 = 6 in. or 0.500 ft.
K_1 = 0.050 Btu, hr^{-1}, ft^{-2}, °F^{-1}, ft.
K_2 = 0.033 Btu, hr^{-1}, ft^{-2}, °F^{-1}, ft.
E_1 = 2.0 Btu, hr^{-1}, ft^{-2}, °F^{-1}.

Taking consistent units, we find the rate of heat transmission substituting in Eq. 2–21 as follows:

$$q = \frac{700 - 100}{\dfrac{\ln (0.333/0.167)}{2\pi 0.050} + \dfrac{\ln (0.500/0.333)}{2\pi 0.033} + \dfrac{1}{2\pi 0.500 \times 2}}$$

$$= \frac{600}{2.20 + 1.95 + 0.16} = 139 \text{ Btu, hr}^{-1}\text{, ft length}^{-1}$$

The rate of heat transmission per square foot of pipe surface is found by dividing Eq. 2–21 by $2\pi r_1$.

$$q = \frac{t_4 - t_1}{r_1 \dfrac{\ln (r_2/r_1)}{K_1} + r_1 \dfrac{\ln (r_3/r_2)}{K_2} + \dfrac{r_1}{r_3 E_1}} \quad \text{(unit area of pipe surface)} \qquad (2\text{–}22)$$

The temperature of the interface, Fig. 2–14, can be found readily since each term in the denominator of Eq. 2–22 represents the resistance to heat transmission by each element. Following the same method by which we determined the wall surface temperature, we can prove the temperature of the interface to be

$$t_3 = 700 - \frac{\dfrac{\ln (0.333/0.167)}{2\pi 0.050}}{\dfrac{\ln (0.333/0.167)}{2\pi 0.050} + \dfrac{\ln (0.500/0.333)}{2\pi 0.033} + \dfrac{1}{2\pi 0.500 \times 2.0}} \times (700 - 100)$$

$$= 700 - \frac{2.20}{2.20 + 1.95 + 0.16} \times 600 = 394°\text{F}$$

With cylinders of small diameter, such as wires and tubing, the addition of insulation may increase the rate of heat transmission from the cylinder.

Figure 2–15 represents the cross-sectional view of a cylinder of unit length and radius r_1, insulated with a material of conductivity K and thickness r_2–r_1. If E is the combined radiation

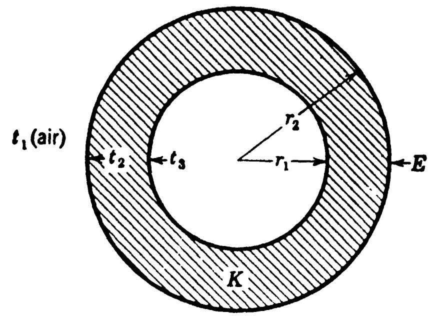


FIG. 2-15. Cylindrical shell. Unit length.

and convection coefficient for heat transfer from the surface to the air, the rate of heat transfer per unit length may be calculated by Eq. 2-21.

$$q = \frac{t_3 - t_1}{\dfrac{\ln (r_2/r_1)}{2\pi K} + \dfrac{1}{2\pi r_2 E}} \tag{2-23}$$

If we assume fixed values for t_3, t_1, r_1, K, and E, the rate of heat flow per unit length can be calculated for various values of r_2. If r_1 is sufficiently small, it will be seen that the value of q increases with increasing thickness of insulation up to a certain value of r_2 and then it diminishes as further thickness of insulation is added. Figure 2-16 illustrates this point.

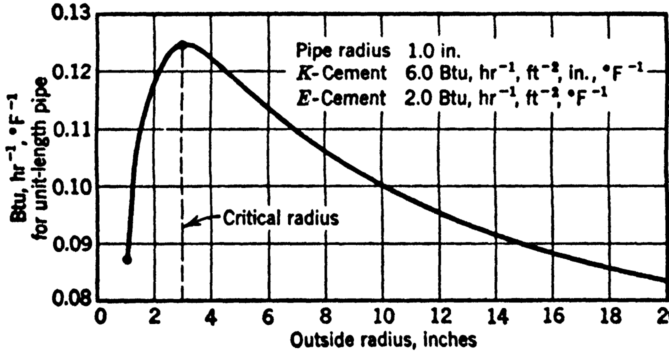


FIG. 2-16. Rate of heat flow vs. thickness of insulation. Cylinders.

In order to make q a maximum in Eq. 2–23, the denominator must be a minimum for any given temperature difference between the cylinder and the air.

$$\frac{d\left(\dfrac{\ln (r_2/r_1)}{K} + \dfrac{1}{r_2E}\right)}{dr_2} = 0$$

$$\frac{1}{K} \times \frac{1}{r_2} + \frac{-1}{Er_2^2} = 0$$

For maximum rate of heat loss from the cylinder,

$$r_2 = \frac{K}{E} \tag{2–24}$$

This value of r_2 that gives a maximum rate of heat transmission is known as the critical radius (Fig. 2–16).

With good insulation, the critical radius is small but is sometimes of considerable importance. Asbestos string wound on small-diameter glass tubes or wires generally increases the rate of heat loss. Rubber-covered wires transmit more heat radially outward than bare wires, even if the bare wire has the same emissivity as rubber. This means that a rubber-covered wire can carry more current than a bare wire for the same temperature rise in the wire.

If we have relatively good conductors, such as concrete with a K value of 8 Btu, hr^{-1}, ft^{-2}, in., $°\text{F}^{-1}$, the critical radius might be $K/E = 8/2 = 4$ in. This partly explains why, in artificial skating rinks, the brine pipes can be embedded in concrete with an actual gain in efficiency of heat transfer from the pipes to the ice covering the concrete. The same reasoning applies to pipes buried in concrete floors for radiant heating of the building.

The critical radius for metals is very large on account of their high K values. For iron we find that $K/E = 300/2 = 150$ in. For any iron pipe in air, the greater the wall thickness, the greater the heat loss, until the outside radius becomes equal to the critical radius.

Compound spherical shells (Fig. 2–15)

$$q = \frac{4\pi r_1 r_2 K_1 (t_4 - t_3)}{r_2 - r_1} = \frac{4\pi r_2 r_3 K_2 (t_3 - t_2)}{r_3 - r_2} = 4\pi r_3^2 E(t_2 - t_1)$$

$$= \frac{t_4 - t_3}{(r_2 - r_1)/4\pi r_1 r_2 K_1} = \frac{t_3 - t_2}{(r_3 - r_2)/4\pi r_2 r_3 K_2} = \frac{t_2 - t_1}{1/4\pi r_3^2 E}$$

$$= \frac{t_4 - t_1}{(r_2 - r_1)/4\pi r_1 r_2 K_1 + (r_3 - r_2)/4\pi r_2 r_3 K_2 + 1/4\pi r_3^2 E} \qquad (2\text{–}25)$$

The terms in the denominator represent the relative resistance to heat flow for each element, as in the compound wall, and the temperature of the surface or of any interface can be calculated in a similar manner.

It is possible that the addition of insulation to a small sphere may actually increase the rate of heat transmission.

For a sphere as shown in Fig. 2–15,

$$q = \frac{t_3 - t_1}{(r_2 - r_1)/4\pi r_1 r_2 K + 1/4\pi r_2^2 E} \qquad (2\text{–}26)$$

To make q a maximum, the denominator in Eq. 2–26 must be a minimum, and so

$$\frac{d\left(\dfrac{r_2 - r_1}{4\pi r_1 r_2 K} + \dfrac{1}{4\pi r_2^2 E}\right)}{dr_2} = 0$$

or

$$\frac{d\left(\dfrac{1}{r_1 K} - \dfrac{1}{r_2 K} + \dfrac{1}{r_2^2 E}\right)}{dr_2} = 0$$

$$r_2 = \frac{2K}{E} \qquad (2\text{–}27)$$

which is the critical radius for a spherical shell.

CHAPTER THREE

Methods of Determining Heat Transfer Coefficients

The determination of the coefficient of thermal conductivity is accomplished by various methods, depending upon:

(1) *The conductivity of the material.* The K value for relatively good conductors is not determined with the same apparatus as that for good insulators.
(2) *The temperature range.* Extremely low temperatures (−300°F to 0°F), medium temperatures (70°F to 1000°F), and high temperatures (1000°F to 2500°F) require entirely different equipment.
(3) *Physical nature of the material.* Pipe covering, powders, reflective insulation, small crystals, etc., all require apparatus that is adapted to the nature of the material.

The other heat transfer coefficients, such as emissivity, e, surface coefficients to air, $h_{cv} + h_r$, and coefficient of heat transfer by convection, h_{cv}, are usually determined by special test equipment.

Relatively few test methods have been standardized, but, on account of the growing demand for information concerning the determination of various coefficients, a brief description of a considerable number of test methods will be given, many of which have been developed and are in use in the Heat Measurements Laboratory at M.I.T.

Strictly speaking, the coefficient of thermal conductivity should apply only to homogeneous materials in which the heat transfer is by conduction alone, as in metals. Most heat insulators consist of solids containing pores, such as insulating bricks and corkboard, or of highly porous materials having a continuous-air type of arrangement, like powders and mineral wool. Heat is transferred across an air space, such as a pore, by radiation, conduction, and (or) convection. If the pore is small and not connected to other pores, the transfer of heat is by radiation across the pore and by conduction of the gas in the pore. In this process, with relatively small temperature drops across the pore, the transfer of heat through the whole mass may be considered as conduction if a large number of pores are uniformly distributed through the material. Many of our common insulators are similar to this structure, and a K value represents their relative conductivity fairly well.

In a material having large pores, radiation becomes an important factor, and, with greater temperature differences across the pores, the heat transferred by radiation is not proportional to the temperature difference or inversely proportional to the thickness. In the lightweight continuous-air type of insulation, such as mineral wool, there is good evidence that convection within the insulator takes place under certain conditions (Chapter Four). Furthermore, some insulators are somewhat transparent to infrared radiation so that some of the heat may be transferred directly through the material by radiation.

If radiation or convection becomes an important factor in the rate of heat transmission through an insulating material, a coefficient of thermal conductivity cannot correctly represent its relative conductivity. In this event it would be better to call the coefficient of heat transfer an apparent K or K_{app}, and the conditions under which the value was found must be definitely stated, such as thickness, orientation, temperature difference, and emissivity of bounding surfaces.

It has been common practice among engineers to employ a K value for all sorts of materials, although under certain conditions, as mentioned previously, this may lead to appreciable error in the calculation of the rate of heat flow.

Guarded-plate method

The guarded-plate method of determining K values is generally accepted as our most precise procedure, and it has been standardized by the American Society for Testing Materials.[1] A rather detailed description of this method will be given since so many of the precautions mentioned and principles involved apply to all thermal conductivity determinations.

As shown in Fig. 3–1, the guarded-plate equipment consists usually of a circular or square heater made of two aluminum or copper plates (1/8 in. to 1/4 in. thick) heated by a uniformly distributed resistance wire wound on asbestos board placed between the plates and electrically insulated from them with asbestos paper or similar material. In order to minimize any edgewise flow of heat from the heater, a guard ring is placed around the heater and constructed in exactly the same fashion. The asbestos wood, on which the heating wires are attached, may be one piece extending over the whole area so that the guard and heater may be handled as a single unit. With heater and guard-ring plates of metal, the heater must be separated by about 1/16 in. from the guard so as to prevent any through metallic conduction of heat from one to the other. This gap can be assured by the insertion of a piece of asbestos paper or mica.

Two pieces of the sample of insulation under test are made approximately the same size as the heater and guard combined, and one is placed on each side of the heater. Two water-cooled plates of aluminum or brass are placed as shown in Fig. 3–1, so as to maintain a lower temperature on one side of each sample. The entire apparatus is insulated on the edges with loose or blanket insulation so that the resistance to heat flow radially outward is two to three times that of the sample in the direction of normal heat flow.

In operation, the guard must be kept at essentially the same temperature as that of the heater to prevent any appreciable flow of heat from one to the other. This can be accomplished manually, but automatic control is far better. A multiple-differential copper-constantan thermocouple may be attached to asbestos paper placed directly over the metal heater plates, and the leads from this thermocouple may be connected to a mirror galvanometer. The electric-wiring diagram for this con-

[1] Thermal Conductivity of Materials by Means of the Guarded Hot Plate, A.S.T.M. designation *C 177-45*, American Society for Testing Materials, 1916 Race Street, Philadelphia 3, Pa.

trol with the aid of a photoelectric cell is shown in Fig. 3–2. Only the current through the guard is varied by a small amount in order to balance the temperature of the guard to that of the heater. A rough balance must be made manually and then the

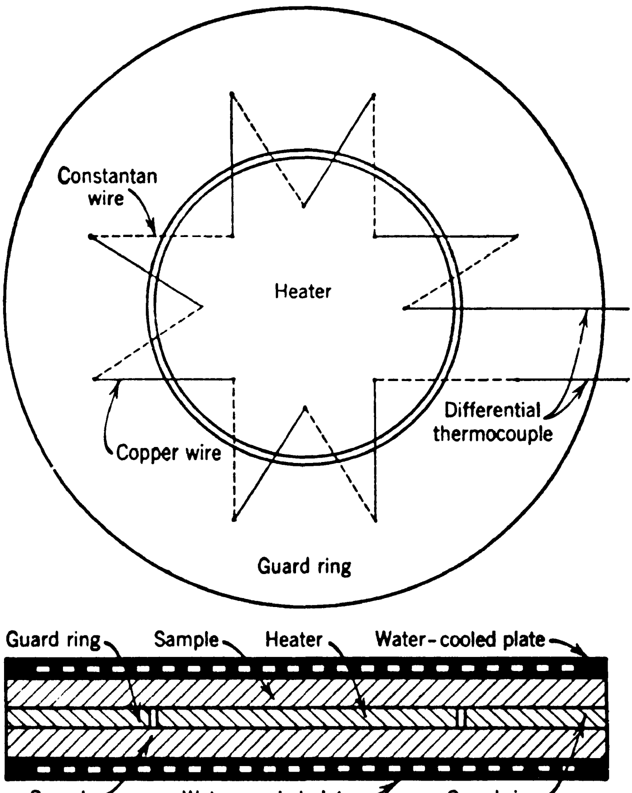


FIG. 3–1. Thermal conductivity. Guarded plate.

automatic control can be operated. By adjusting the on and off current to a small amount, a balance of ±0.02°F can be maintained for days.

The emissivity of the heater, guard, and water-cooled plates must be high (approximately 0.9). It can be made so with a high-emissivity paint or, better still, by attaching asbestos paper or similar material to the metal surfaces. There is always a resistance to heat transfer from one surface to another unless the thermal contact is perfect. This perfection is never realized in thermal contact between a metal plate and a rigid insulator, and, consequently, air pockets occur between the two surfaces. The resistance to heat flow across these pockets depends upon the conductivity of the air and radiation. The radiation is minimized by means of low-emissivity surfaces, such as copper

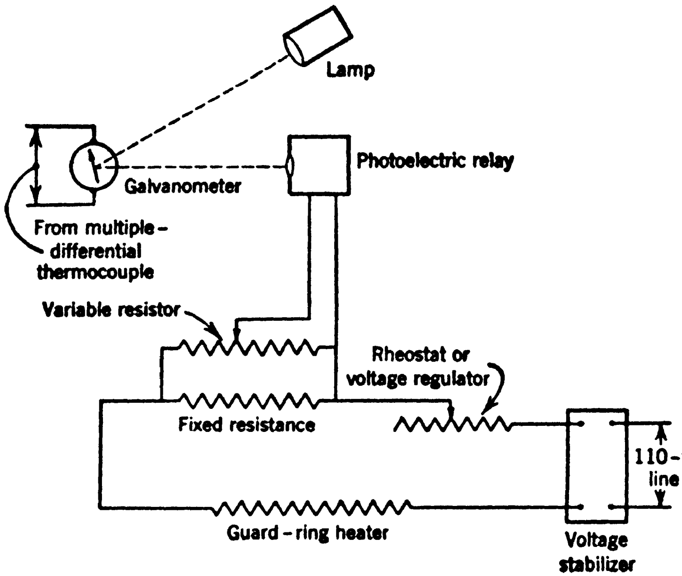


FIG. 3–2. Photoelectric control. Plate test.

or aluminum, which also results in greater resistance to heat transmission from the metal surface to the insulation. Since, in practice, insulation generally faces high-emissivity surfaces, such as wood, plaster, and brick, it is now recognized that determinations with the plate test should be made with high-emissivity surfaces. Many of our early published K values were made with polished aluminum, brass, or copper plates, and the values are undoubtedly somewhat lower than would be obtained for the same materials with high-emissivity plates as required today. Some emphasis is placed on this phase of determining K values because it is quite important in loose, fibrous insulators, powders, etc., especially if the sample is thin (1 in. or less).

Laboratory tests (Berchtold 13) indicate that a sample of glass wool, 3 lb, ft^{-3}, had a K value of 0.33 for a 1-in. thickness with low-emissivity surfaces ($e = 0.05$) and had a K value of 0.37 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$ with surfaces having an emissivity of 0.9. This result represents an increase of 12% in the K value owing to radiation.

Surface temperatures of the insulating material may be measured with fine-wire thermocouples attached to the sample whenever possible and to the asbestos paper facing the sample with powders and loose, fibrous materials. The A.S.T.M. standard test permits the attachment of thermocouples directly to the metal plates with samples having a K value less than 1.0 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$. This method has the advantage of making it easier to change samples and measure their thickness, which is taken as the distance between the plates, but it also has certain disadvantages. Any resistance to heat flow between the plates and the sample is included in the resistance of the sample, tending to give a lower value of K. Also, if multiple or differential thermocouples are employed, the problem of electrical insulation is not easy to solve. The plates must be coated with a high-emissivity paint that may be rubbed off in time and thus give lower emissivities. It has been the practice for many years in the Heat Measurements Laboratory of M.I.T. to place asbestos paper or blotting paper between all metal plates, to attach thermocouples to the sample whenever possible, and otherwise to attach them to the surface of the paper facing the sample.

Plate tests may be mounted either vertically or horizontally. When flexible, fibrous, or powdered materials are tested in the horizontal position, small insulating spacing blocks may be placed in the outer edge of the guard to keep the plates apart; the thickness of these blocks will also be the thickness of the sample under test. The powders may be kept in place by circular or square rings of asbestos paper, cardboard, or thin wood of the same size as the guard heater. Care must also be taken that the sample does not settle, thus making poor thermal contact with the upper surfaces. In placing a powder in such a tester, it is helpful to arrange it so that it is slightly thicker than the test thickness; then, when the plates are lowered, the powder is compressed slightly and makes good thermal contact with the bounding surfaces.

Many thermal insulators are somewhat hydroscopic and become better conductors when exposed to high humidities. To secure uniform test results, all samples should be dried at 215°F until the free moisture is driven off, as indicated by a constant-weight determination. If the sample would be affected by a temperature of 215°F, it should be dried in a desiccator at some lower temperature (120°F to 140°F). Insulating materials that are nonhydroscopic have the same K value in service as that determined by test, as far as the effect of humidity is concerned, whereas hydroscopic materials generally have a higher K value in service than that determined by test.

The A.S.T.M. standard test suggests the following relations between the thickness of the sample and the minimum dimensions of the heater and guard.

Sample Thickness	Minimum Diameter or Edge of Heater	Minimum Width of Guard
1 in.	4 in.	1.5 in.
1.5	8	2.25
2	12	3
4	12	6

It is generally preferable to avoid the smaller sizes because with larger test areas (12-in. edge or diameter) a more representative value of K is obtained, since most heat insulators are not perfectly homogeneous.

For the measurement of the watt input to the heater, which actually is a measure of the rate of heat flow through the specimen, the best type of wattmeter should be obtained and care taken that it is connected in such a fashion that the watts in the voltage coil of the meter are included in the wattmeter reading. Knowing the resistance of the voltage coil, as given on the instrument, the resistance of the heater, and the gross watt input as measured on the wattmeter, one can calculate the correction that must be subtracted from the observed wattmeter reading.

$$\text{Watts used in voltage coil} = \frac{\text{Resistance of heater}}{\text{Resistance of voltage coil}} \times \text{Heater watts}$$

With good wattmeters, this correction, which is always negative, is usually in the neighborhood of 1 or 2%.

A voltage stabilizer is almost essential for the proper operation of a guarded-plate tester in order to insure a constant heat input to the heater regardless of small variations in the line voltage.

The water-cooled plates should be supplied from an overhead tank that is thermostatically controlled so as to supply water at a constant temperature and a constant head.

The thermocouple electromotive force should be measured with a potentiometer having a sensitivity of at least 5 microvolts.

A condition of thermal equilibrium is very important for the precise determination of K values. The A.S.T.M. standard test calls for a 5-hr test after a thermal balance has been obtained, with readings taken at intervals of not more than 1 hr, which give K values that are constant to plus or minus 1%. It would be preferable to make a similar 5-hr test 24 hr later and see whether it checks within 1% of the earlier test. Occasionally, with certain types of materials, the second test has indicated a difference of 2 to 3% from the earlier test. It may take as long as 4 days to complete a test at one mean temperature. Since it is frequently advisable to make tests at three or four mean temperatures, differing by at least 30°F, in order to determine the variation of K with the mean temperature, a plate test requires considerable time. Any steps to hasten this procedure should be adopted only after careful checking to make sure that precision is not sacrificed.

If this method is followed to determine values at mean temperatures lower than 70°F, cold brine may be circulated in place of water but extreme care must be taken that water vapor does not condense within the sample or on the cold plates. Usually condensation can be prevented by enclosing the entire apparatus in a box with a low-temperature coil arranged so that the atmosphere within the box has a dew point lower than any portion of the guarded-plate equipment.

The data required for a determination are

l = thickness of the sample, in.
A = area through which heat flows normally from the heater through the sample, sq ft.

t_2 = average temperature of warmer surface, °F.
t_1 = average temperature of cooler surface, °F.
q = corrected watt input to heater × 3.412, Btu, hr^{-1}.

From Eq. 2–1, the coefficient of thermal conductivity is

$$K = \frac{ql}{A(t_2 - t_1)} \quad \text{Btu, hr}^{-1}\text{, ft}^{-2}\text{, in., °F}^{-1}$$

In reporting this value, the following are usually included:

(1) Identification of material.
(2) Thickness under test, in.
(3) Bulk density of the sample, dry, lb, ft^{-3}.
(4) Percentage of moisture regain during test, if any.
(5) Warmer surface temperature, °F.
(6) Cooler surface temperature, °F.
(7) Mean temperature of sample, °F.
(8) Coefficient of thermal conductivity, K.

Guarded-box method

The guarded-box method is designed primarily for the determination of the overall coefficient of heat transmittance, U, or the coefficient of thermal conductance, C, for built-up wall sections.

An insulated box is constructed of 2-in. corkboard or similar material with one face open (Fig. 3–3). The minimum inside dimensions would be 32 in. by 32 in. for the open face and 10 in. for the depth. An outer insulated box (guard box) surrounds the inner box leaving a minimum of 12-in. clearance on all sides except the open face, and its side walls extend at least 6 in. beyond the open face of the inner box.

The wall section under test is made to fit into the guard box and press against the inner box, closing the open side of both boxes. The edges of the inner box, in contact with the wall, are beveled from the outside, and a narrow gasket is placed between the wall surface and the beveled edge. The inner box is mounted in such a way that it will move readily a small distance toward the wall section, and springs are placed so that they exert a pressure on the inner box toward the wall surface. This arrangement results in good contact between the inner box and the wall and effectively prevents any appreciable flow of air from one box to the other.

The inner box is electrically heated. A baffle is placed between the test wall and the heating unit to prevent the transfer of heat by radiation from the heater to the wall surface. The outer box also is electrically heated, and the air is circulated by fan to give a uniform distribution of temperature all around the inner box. The air within the two boxes is automatically maintained at the same temperature by means of a multiple-differential thermocouple in the same way as in the guarded-plate method (Figs. 3–1 and 3–2).

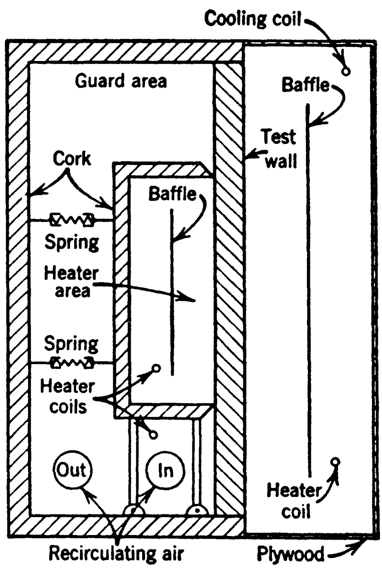


FIG. 3–3. Heat transmittance. Guarded box.

Shielded thermocouples measure the air temperature in the inner box and the air temperature outside the test wall. Thermocouples are also attached to the inner and outer surfaces of the test wall and can be placed at various points within the wall if desired. The watt input to the inner-box heating unit is a measure of the rate of heat flow through the wall for an area equal to the open face of the inner box, after thermal equilibrium has been established.

The overall (air-to-air) coefficient of heat transmittance, U, can be calculated as follows:

$$U = \frac{\text{Watts} \times 3.412}{(t_2 - t_1) \times \text{Area}} \text{ Btu, hr}^{-1}\text{, ft}^{-2}\text{, °F}^{-1} \quad \text{(air-to-air)}$$

The coefficient of thermal conductance, C, can be similarly calculated:

$$C = \frac{\text{Watts} \times 3.412}{(t_{s2} - t_{s1}) \times \text{Area}} \text{ Btu, hr}^{-1}\text{, ft}^{-2}\text{, °F}^{-1} \quad \text{(surface-to-surface)}$$

If there is a possibility of convection in the air spaces or inside the insulation within the wall, the test area must be definitely blocked off from the guard area as shown in Fig. 3-4.

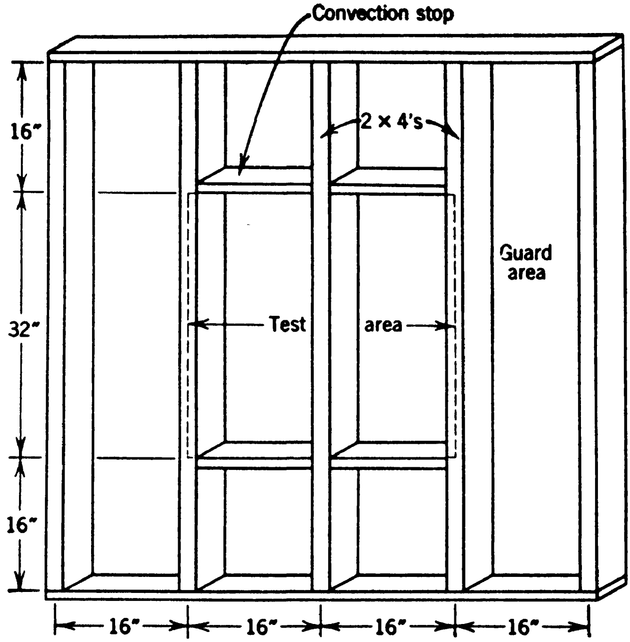


FIG. 3-4. Wall section showing test area. Guarded box.

This procedure applies to nearly all walls except those that are solid from the inner to outer surfaces.

The air on the outside of the test wall must be maintained at a constant temperature. One method is to place the equipment in a constant-temperature room or enclosure. The outer air temperature can be approximately room temperature, 70°F, with the air in the inner box at a higher temperature, say 140°F, thus giving a 70°F drop in temperature from the air on one side of the wall to the other. The objection to these temperatures is that the mean temperature of the wall is considerably higher than one would find ordinarily in practice. The alternative is to place the test equipment in a refrigerated room where the temperature of the air may be kept at 0°F while the temperature of the air in the inner box is maintained at 70°F, thus giving a 70°F temperature drop from one side of the wall to the other and also having a mean temperature of the wall at 35°F, which is common in severe winter weather in many parts of the United States. This latter method involves more expensive equipment, greater operating expense, more skill in maintaining constant temperatures, and more likelihood of serious trouble from condensation of water vapor within the test wall.

As in all heat-transfer determinations under steady-state conditions, it is important that thermal equilibrium is established throughout the test wall before any measurements are made. With walls up to 6 in. thick, it may take as long as 3 days to reach these conditions. See Chapter Seven for data giving the time required to achieve equilibrium under certain conditions.

The question often arises as to the value of transmittance or conductance factors determined in this way with controlled and steady conditions of temperature, when in practice the outdoor temperature varies greatly and, in addition, there are the effects of sunshine, wind, rain, and snow.

During the fall of 1934, four small test houses of frame construction were erected on the flat roof of an unheated building in the suburbs of Boston, Mass. (Wilkes and Bemis 115). One of these houses was uninsulated. The other three were insulated with three common types of house insulation, namely 3-in. loose rock wool, 1-in. Balsam Wool blanket, and two 7/16-in. fiberboards. They were heated electrically and the air temperature inside was automatically maintained at 70°F. These houses had no windows and so the test consisted of determining the rate of heat flow through walls only, after corrections had been made for heat loss through the heavily insulated floors and roofs. Multiple-recording potentiometers kept a continuous record of the air temperature inside and out as well as both surface temperatures. Watthour meters measured the electric input. The tests were conducted throughout the winter months, and the average *U* value compared very favorably with *U* values found with similar walls by means of the guarded-box method. The day-by-day *U* values sometimes varied considerably. On cold, winter days with bright sunshine and little wind, the *U* values were much lower than accepted test values, but on cloudy or rainy windy days they were higher than the average. Snow apparently had little effect on the heat flow through the walls but, of course, it undoubtedly reduced the rate of heat flow through the roofs.

On some clear, cold nights with no wind, the outer surface of the insulated wall was colder by 1°F to 2°F than the outside air. Thus heat was flowing from the outside air to the surface as well as through the wall. This interesting condition was due to heat radiation from the outer wall surface to the sky, which has an effective temperature between −40°F and −50°F (Dines 28) on clear nights in England and probably is about the same in the United States.

The guarded-box method of determining *U* or *C* values is much more dependable than values calculated for the same walls from *K* values based on plate tests and surface and air-space coefficients determined independently. The calculation of the rate of heat transfer through shingles or clapboards would be very complicated and uncertain, the *K* values from the plate test are determined under "bone-dry" conditions, and our knowledge of surface and air-space coefficients is not all that we desire.

Pipe-insulation tester

The rate of heat transmission from pipes radially outward through insulation to the ambient air can be determined with an electrically heated pipe of some standard diameter with electrically heated end guards to insure only radial flow of heat from the test section. Figure 3–5 shows the general setup for such equipment. The temperature of the end guards is automatically maintained at the same temperature as that of the test section in the same manner as employed for the guard ring and heater sections in the guarded-plate method.

The pipe is usually suspended in a horizontal position by slings of canvas or similar material around the outside of the insulation on the end guards. Temperature measurements of the pipe surface, the pipe-cover surface, and the ambient air are made at several points with fine-wire thermocouples. As the outer surface of the insulation varies in temperature from top to bottom, a sufficient number of thermocouples should be attached so that a good average surface temperature can be found.

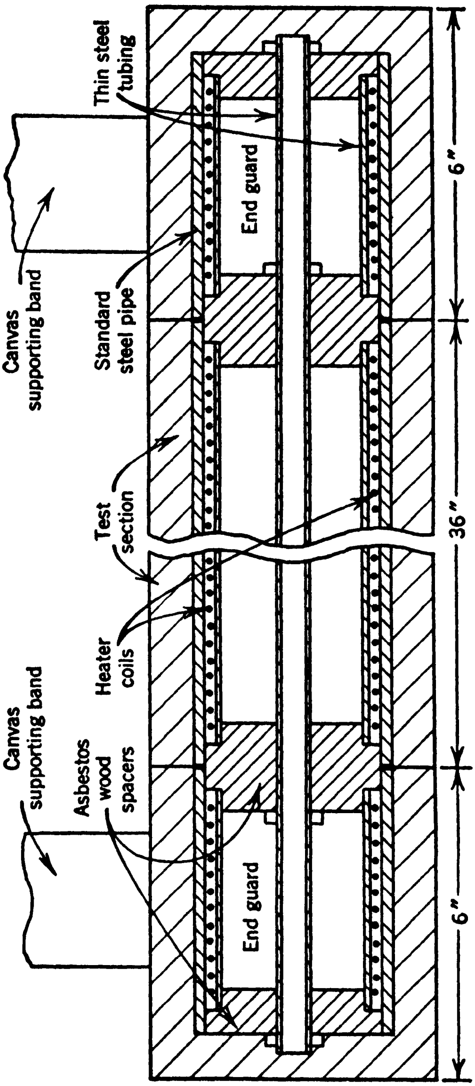


FIG. 3–5. Thermal conductivity. Pipe-cover tester.

The insulating value of the pipe cover can be expressed in various ways, such as:

(1) Btu, hr^{-1}, ft^{-2} (pipe surface), $°F^{-1}$ (pipe surface to air) for specified insulation thickness and pipe diameter.
(2) Btu, hr^{-1}, lin ft^{-1}, $°F^{-1}$ (pipe surface to air) for specified insulation thickness and pipe diameter.
(3) Percentage of efficiency, which is based on the amount of heat saved by the insulation compared to heat loss from a bare pipe under the same temperature conditions.

Both (1) and (2) appear to be satisfactory for expressing the insulating value of a pipe cover, but the percentage of efficiency, although quoted by many manufacturers of pipe covering, is a much less precise method because it is based on the rate of heat loss from a bare pipe, which is affected by air currents and the condition of the metal surface much more than that from an insulated pipe.

None of these methods of expressing the insulating value of pipe insulation can be employed to compute accurately the rate of heat loss for pipe insulation of other thicknesses or for other pipe diameters. For the same conditions of temperature, doubling the thickness of a pipe cover does not reduce the rate of heat transfer by one half, nor is the rate of heat loss per unit area for a 1-in.-thick pipe covering the same on a 5-in.-diameter pipe as on a 1½-in. diameter pipe.

If we take the average outside surface temperature of the pipe covering, the K value for the insulation can be determined from test data by Eq. 2–2.

$$q \text{ per unit length} = \frac{2\pi K(t_3 - t_2)}{\ln (r_2/r_1)}$$

or

$$K = \frac{(\text{Watts per unit length})(3.41)[\ln (r_2/r_1)]}{2\pi(t_3 - t_2)}$$

By making test determinations at three or more pipe temperatures, a plot can be made that indicates how the value of K varies with the mean temperature.

The emission coefficient, E, $(h_r + h_{cv})$ for canvas-covered pipe insulation is shown in Fig. 3–6, the data being taken from the average of values determined by McMillan (64) and Bagley

(9). No change in the value of E is necessary for different pipe diameters, because the outside surface diameter is never very small where the biggest changes in E would occur and, furthermore, the E values play a minor part in the calculation of the heat loss from a well-insulated pipe.

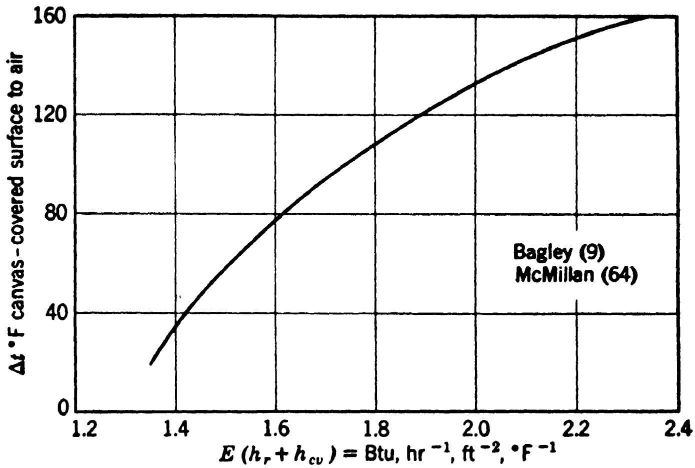


FIG. 3–6. Surface coefficient, E, vs. Δt. Canvas-covered pipes.

The rate of heat transmission from an insulated pipe of some other diameter and (or) thickness of covering from that of the test can now be calculated with Eq. 2–21.

$$q \text{ per unit length} = \frac{2\pi(t_3 - t_1)}{\dfrac{\ln (r_2/r_1)}{K} + \dfrac{1}{r_2 E}}$$

Since K varies with the mean temperature and E varies with the temperature difference between the outer surface temperature and that of the air, for a first approximation an outer surface temperature is assumed in order to choose values for K and E.

For example, if one had an 8-in. nominal-diameter pipe at 400°F with a 2-in.-thick covering and if the ambient air were at 70°F, then

$$r_1 = 4.31 \text{ in.} \quad \text{and} \quad r_2 = 6.31 \text{ in.}$$

because the outside diameter of an 8-in. pipe is 8.62 in.

If the outer surface temperature were 120°F, the mean temperature of the insulation would be $(400 + 120)/2 = 260°F$, and the temperature difference between the surface and the air would be $120 - 70 = 50°F$. If the value of K for a mean temperature of 260°F were 0.45 Btu, hr^{-1}, ft^{-2}, in., °F^{-1} and if E were found by Fig. 3–6 to be 1.47 Btu, hr^{-1}, ft^{-2}, °F^{-1} for a temperature difference of 50°F, the outside surface temperature could be found by Eqs. 2–20 and 2–21.

$$t_2 = \frac{\dfrac{1}{r_2 E}}{\dfrac{\ln (r_2/r_1)}{K} + \dfrac{1}{r_2 E}} \times (t_3 - t_1) + t_1$$

With consistent units, K must be expressed as $0.45/12 = 0.0375$ Btu, hr^{-1}, ft^{-2}, ft, °F^{-1}, $r_2 = 0.525$ ft, and $r_1 = 0.359$ ft.

$$t_2 = \frac{\dfrac{1}{1.47 \times 0.525}}{\dfrac{\ln (0.525/0.359)}{0.0375} + \dfrac{1}{1.47 \times 0.525}} \times (400 - 70) + 70$$

$$= 107°F$$

The assumption of 120°F for the outside surface temperature was obviously in error, and so a second approximation must be made, based on the assumption of a 107°F surface temperature. The mean temperature of the insulation is now 254°F, and the K lowered to 0.037 Btu, hr^{-1}, ft^{-2}, ft, °F^{-1}. The temperature difference between the surface of the cover and the air is now 37°F, and the corresponding E is 1.41 Btu, hr^{-1}, ft^{-2}, °F^{-1}.

The surface temperature of the insulation is now calculated, by means of these new values, and found to be

$$t_2 = \frac{\dfrac{1}{1.41 \times 0.525}}{\dfrac{\ln (0.525/0.359)}{0.037} + \dfrac{1}{1.41 \times 0.525}} \times (400 - 70) + 70$$

$$= 108°F$$

This result is sufficiently close to the assumed surface temperature of 107°F so that the rate of heat loss from the insulated pipe can now be calculated without serious error.

$$q = \frac{2\pi(400 - 70)}{\dfrac{\ln (0.525/0.359)}{0.037} + \dfrac{1}{1.41 \times 0.525}}$$

$$= 178 \text{ Btu, hr}^{-1}, \text{ lin ft}^{-1}, {}^\circ\text{F}^{-1}$$

The terms in the denominator of the above equation represent the relative resistance to heat transfer through the insulation and from the surface to the air. It should be noted that here the resistance from the surface to the air is approximately 12% of the total resistance. Thus the E value could vary appreciably with only a minor effect on the total rate of heat transfer from the pipe.

High-temperature conductivity

High-temperature-conductivity test equipment, developed in the Heat Measurements Laboratory, M.I.T., in 1933 (Wilkes 110), has not been standardized but it is used, with modifications and improvements, in a considerable number of research laboratories.

Figure 3–7 shows the general principles of this apparatus for the determination of the thermal conductivity of refractories, insulating firebrick, powders, etc., at high temperatures. It consists of a globar heated furnace with the top and walls joined as a unit that can be raised by means of a crank and counterweights. The bottom of the furnace consists frequently of three standard bricks laid close together forming a block 13½ in. by 9 in. by 2½ in. thick. The test area is a 3-in. square in the middle brick, the remainder of the three bricks forming the guard. The rate of heat flow is determined by a water calorimeter placed below the test area and surrounded by a water-cooled guard plate, approximately 13½ in. by 9 in., outside dimensions. It has been found advantageous to have an outer guard, which is a water-cooled plate surrounding the primary guard.

A water calorimeter causes trouble frequently by the release of air bubbles as the water is warmed passing through it. This difficulty can be reduced materially by recirculation of the water in the calorimeter and guards. The water, after passing through the calorimeter and guards, drains into a tank on the floor, is cooled to room temperature by a coil containing running water, and then is pumped to an overhead tank from which it flows by gravity back to the calorimeter.

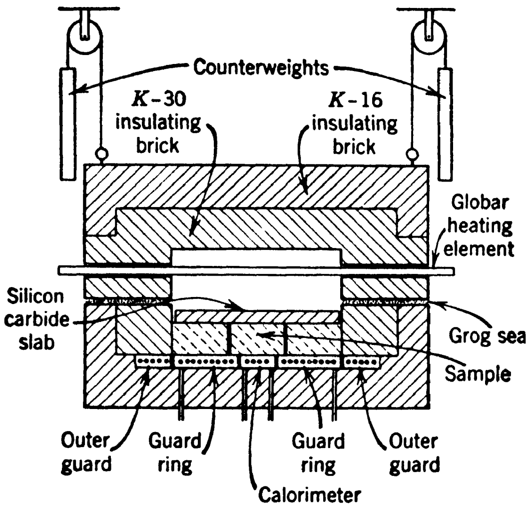


FIG. 3–7. Thermal conductivity. High-temperature test.

Hot surface temperatures of 2500°F can be attained with this equipment, and, if high mean temperatures are desired, a layer of insulating material may be inserted between the sample and the cooling plates, thus raising the temperature of the cooler side of the sample to any desired amount. The temperatures of both sides of the specimen are measured with fine-wire thermocouples employing platinum-platinum, 10% rhodium couples if the temperatures exceed 1800°F.

The furnace temperatures can be controlled by a thermocouple or a radiation pyrometer connected to a potentiometer with a mirror type of galvanometer. With a light source, photoelectric cell, amplifier, and relay, a small change in the current input to the furnace is automatically made in much the same way as described for the guarded plate.

This equipment is suitable for powders if spacing blocks are placed in the outer corners of the guard. Care must be taken that the powder does not settle during the test, or an air space will be left between the silicon carbide plate and the hot surface of the powder. Since, for powders, the thermocouple is attached to the silicon carbide plate, considerable error might be introduced if the powder settled.

For a more complete description of this type of apparatus see Norton (70) and Austin and Pierce (5).

Low-temperature conductivity measurement

One method of determining the coefficient of thermal conductivity at temperatures as low as −300°F is shown in Fig. 3–8 and described by Wilkes (109).

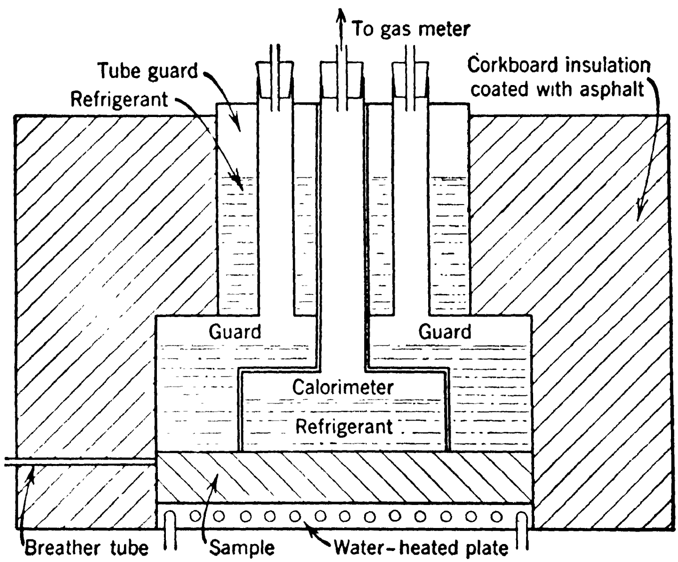


FIG. 3–8. Thermal conductivity. Low temperatures.

The calorimeter, guard, and tube guard (Fig. 3–8) may be filled with liquid oxygen (−297°F). The specimen is placed between the calorimeter and the water-heated plate, thus producing a large temperature difference between the two sides of the specimen. The entire equipment is heavily insulated with corkboard, which is coated with asphalt on the outside to minimize the condensation of water vapor within the sample. After placing a new specimen in the tester, the water-heated plate is put in position and sealed with asphalt to the adjacent corkboard. As an additional precaution against water-vapor penetration, a breather tube is run through the corkboard to the sample. This tube is connected to a drying tube or bottle filled with activated alumina or similar drying agent before opening into the air. When liquid oxygen is poured into the calorimeter, guard, and tube guard, the residual air in the sample is cooled and the pressure is reduced below that of the atmosphere, thus tending to draw moist air from the atmosphere into the specimen. The breather tube permits the pressure to be equalized quickly without the addition of moist air to the specimen.

The rate of heat flow is measured by a wet-gas meter attached to the calorimeter, which measures the volume of gas liberated in a given time. All the heat required to produce this gas by vaporizing the liquid oxygen in the calorimeter must come through the test area of the specimen, since all the calorimeter, including the outlet tube, with the exception of this test area is surrounded with liquid oxygen at the same temperature. Once the latent heat of vaporization of oxygen at the pressure within the calorimeter is known, the rate of heat flow can be calculated from the meter readings after suitable corrections are made for the gas pressure and temperature.

The temperature on each surface of the specimen is measured with fine-wire copper-constantan thermocouples attached to the surface of the specimen. Additional thermocouples may be attached to the specimen in the guard area to prevent lateral flow of heat from the test area.

For other temperatures above −297°F, other liquefied gases could be employed in the same manner. Solid carbon dioxide (Dry Ice) at approximately −110°F is also suitable, but, to insure good heat transfer rates, measured amounts of alcohol or similar liquid must first be placed in the calorimeter, guard, and tube guard. The crushed Dry Ice is then added very slowly at first, and then as the temperature is reduced it can be added at a rapid rate until the containers are filled with a mixture of alcohol and Dry Ice. The calorimeter outlet is connected and sealed to the gas meter. The calorimeter must contain sufficient refrigerant to last 8 to 10 hr, which is ample time for thermal equilibrium to be established. This procedure allows about 5 hr for test readings on a sample not more than 1 in. thick. Refrigerant should be added to the guard and tube guard about once an hour throughout the time of the experiment. Liquid oxygen and Dry Ice are the most readily available refrigerants and have given satisfactory results for several years in the Heat Measurements Laboratory. The equipment is applicable to powders if spacing blocks are placed in the guard area.

Conductivity of metals

Although metals are not generally classed as insulators (reflective insulation utilizes metals to retard heat transfer), in many insulated structures metal pipes, rods, plates, etc., pass through the insulation and frequently become an important factor in calculating the total rate of heat transmission.

Figure 3–9 shows an apparatus devised by Wilkes (112) that can serve to determine the thermal conductivity of metals. The test specimen is a 1-in. diameter rod, 12 in. long. One end of it is threaded into a copper casting, and the other end is turned down and threaded into a brass water calorimeter. The water passage is a square thread superimposed on the V thread. A guard tube is attached to the copper casting at one end, and a square thread is cut on the other end to make a channel for water cooling. Insulation is placed between the test specimen and the guard to prevent any transfer of heat by radiation from the guard to the specimen. A Chromel heating unit is attached to a projection on the copper casting. Three thermocouples are inserted into small holes drilled in the specimen, spaced 3 in. on centers. With the whole equipment heavily insulated, any heat entering the specimen from the copper casting flows lengthwise through the specimen to the water calorimeter, there being essentially no loss or gain of heat from the guard because practically no temperature difference exists to produce radial flow of heat.

Conductivity determinations at a mean temperature of 1000°F can be made with this equipment. It is possible to obtain very high mean temperatures, with some loss in precision, by fitting the specimen and guard into a graphite block, which in turn is placed in a molten tin bath that can be heated to 2500°F. In one set of tests with this modified equipment, one end of a steel specimen was actually melted.

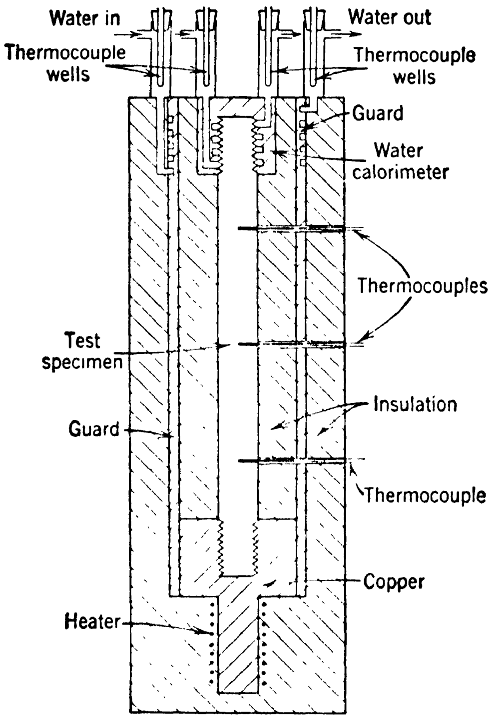


FIG. 3–9. Thermal conductivity. Metals.

Conductance of air spaces

Since the rate of heat transfer across an air space is affected materially by the orientation of the space, equipment for the determination of conductance values must be so designed that the air space can be rotated to various positions. Wilkes and Peterson (115*a*) designed such equipment in 1937, and it has been in almost constant use since that time, with some refinements added.

Figure 3–10 indicates the general principles of this equipment. It consists of an electrically heated plate with guard, one side being insulated with 3-in. corkboard. A water calorimeter, 8 ft by 32 in., is spaced any desired distance from the heated plate, thus forming the test air space. A water-cooled guard surrounds the calorimeter, giving a guard air space. The surface of the

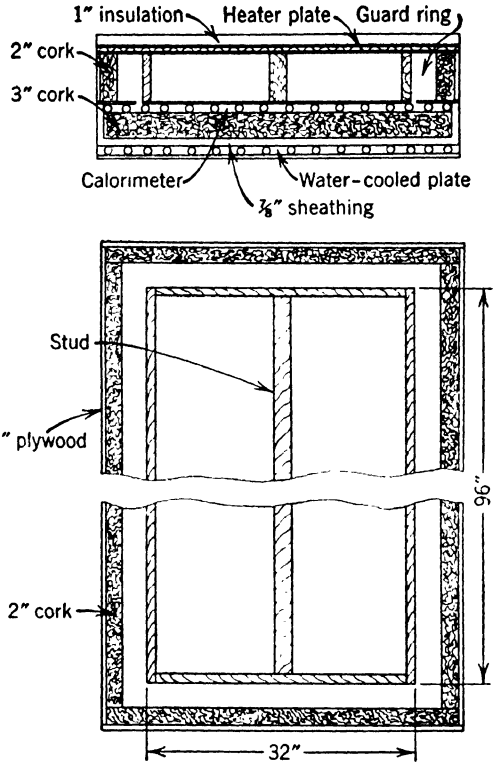


FIG. 3-10. Thermal conductance. Air spaces.

heater and calorimeter may be covered with any material such as aluminum foil, if low-emissivity surfaces are desired, or with nonmetallic paint for high-emissivity surfaces.

The heater, calorimeter, guard heater, and water-cooled guard are placed in a wood box insulated with 3-in. corkboard, and the box is mounted on trunnions so that it can be rotated at least 180°. Temperature measurements of the two surfaces facing the air space are made with eight thermocouples attached to each surface with symmetrical spacing. The quantity of heat transferred per unit time across the air space is measured with the water calorimeter. The thermal conductance can be calculated by the formula

$$C = \frac{q}{A(t_2 - t_1)}$$

The conductance of the stud space in a frame wall or the joist space in a floor or ceiling, with or without insulation, can be readily determined with this apparatus. The test area, being 8 ft high, corresponds closely to that of the average room, and the width, being 32 in., is suitable for studs, joists, or rafters on 16-in. centers. The conductance value so determined includes any heat transfer through the wood, as well as that through the space.

Surface coefficient $(h_r + h_{cv})$

The surface coefficient, E, $(h_r + h_{cv})$, for large plates can be determined with the same equipment as that for air spaces, only the air space must be open on all sides and have a width of 6 in. or more. Wilkes and Peterson (115*b*) describe this operation in considerably more detail.

For cylinders, the same test apparatus as that for pipe covering may be used, with no insulation around the pipe.

Wires can be electrically heated and the temperature determined by their electrical resistance. Knowing the heat input and the temperature difference between the wire and the ambient air, one can readily calculate the surface coefficient.

Convection coefficients

The various types of equipment suitable for surface coefficients can also be employed to determine the coefficient of convection, C', or the value of h_{cv} by covering the surface of the heater with a material of known low emissivity, such as aluminum foil. Then the radiation loss can be calculated and subtracted from the total loss to find the loss of heat by convection.

Low-temperature tester for reflective insulation

Apparatus for the determination of the rate of heat transfer through reflective insulation at low temperatures has recently been constructed in the Heat Measurements Laboratory, M.I.T.

It was designed primarily for a coated steel insulation that comes in sheets 24 in. by 32 in.

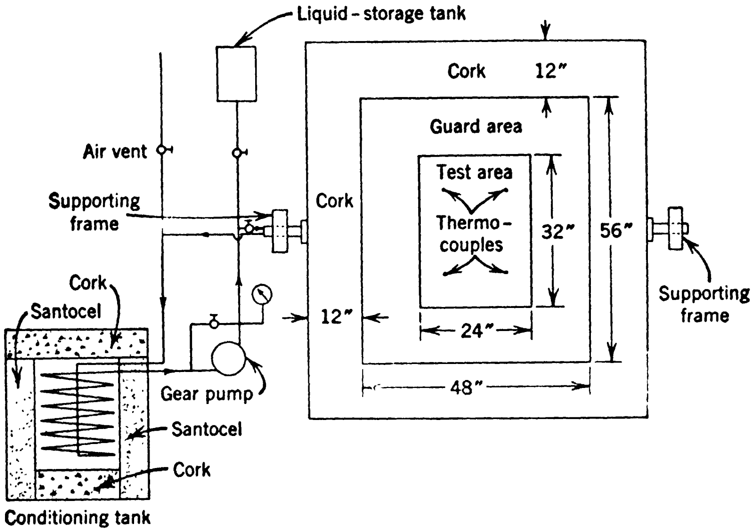


Fig. 3–11a. Conductance of reflective insulation. Low temperatures.

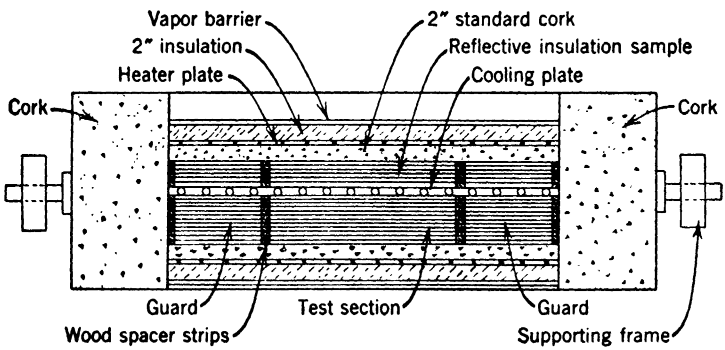


Fig. 3–11b. Conductance of reflective insulation. Cross-section, low temperatures.

Figure 3–11 (*a* and *b*) indicates some of the more important details of this method. A cold plate is placed between two test walls of different thickness, if desired. The test walls have a central test section 24 in. by 32 in., which is surrounded by a 12-in. guard of the same construction as the corresponding test wall. Two-inch corkboard, whose coefficient of thermal conductivity has been previously determined, is placed over each wall, and outside of the cork is an electrically heated plate. The entire assembly is placed in a rectangular wood frame with 12 in. of corkboard insulation between the wood frame and the walls. The wood frame is mounted on trunnions so that the walls may be rotated into different positions. Thermocouples serve to determine the temperature of both sides of the wall, as well as both sides of the 2-in. cork.

Knowing the thermal conductivity of the 2-in. corkboard, and the temperature drop through both the cork and the wall, one can determine the conductance of the wall, since the rate of heat flow is the same through the wall as through the cork.

Since

$$q = CA_{\text{wall}}\,\Delta t_{\text{wall}} = \frac{K_{\text{cork}}A_{\text{cork}}\,\Delta t_{\text{cork}}}{l_{\text{cork}}}$$

and

$$A_{\text{wall}} = A_{\text{cork}}$$

$$C = \frac{K_{\text{cork}}\,\Delta t_{\text{cork}}}{l_{\text{cork}}} \times \frac{1}{\Delta t_{\text{wall}}}$$

With Dry Ice to cool the alcohol circulated through the cold plate, mean temperatures as low as $-60°$F can be maintained. The warmer surface temperature of test wall can be varied by changing the electric input to the heater plate on the warmer side of the calibrated cork. If high mean temperatures are desired, the circulating alcohol can be cooled with water, and mean temperatures as high as 140° F can be secured.

"Northrup" method

Circa 1912, Dr. E. F. Northrup suggested a method of determining the coefficient of thermal conductivity of insulating materials by placing a slab of material whose conductivity was known in contact with one whose conductivity was unknown. The two slabs were then placed between a heated and a water-cooled plate. This principle had previously been applied by Tuchschmidt (100) in 1883 and Lees (55) in 1892 for the determination of the thermal conductivity of single crystals at low temperatures. If the K value of one of the slabs and the thickness of each slab are known and if the temperature drop through each slab is measured, the K value of the unknown sample can be readily calculated.

This method, although not so precise as the guarded-plate, is much cheaper, simpler, and quicker. It is especially useful if a large number of determinations are necessary and precision is not of prime importance, as in the development of new insulating materials by a laboratory.

The general layout of this type of equipment is shown in Fig. 3–12. Greater accuracy is obtained with relatively large

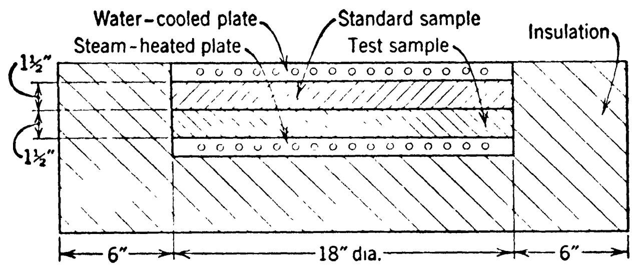


FIG. 3–12. Thermal conductivity. "Northrup" method.

areas and with samples 1 to 2 in. thick. Electrically heated plates can be used, but steam-heated plates are very satisfactory if the pressure and corresponding temperature are constant.

Three thermocouples are employed to measure the temperature drop across the two materials. The thermocouple at the interface measures the temperature of the cooler side of the test sample, as well as the warmer side of the standard material. All these thermocouples should be located as near the center of the materials as possible, and if the sample is not homogeneous it may be helpful to solder the couples to thin, blackened copper disks about 2 in. in diameter. The heat flow at the center is approximately normal to the hot and cold plates; consequently, the rates of heat flow through the center of the test sample and standard sample are essentially the same.

Therefore, for unit area,

$$q = \frac{K_1\,\Delta t_1}{l_1} = \frac{K_2\,\Delta t_2}{l_2}$$

or

$$K_2 = \frac{K_1 \, \Delta t_1}{l_1} \times \frac{l_2}{\Delta t_2}$$

"Northrup" method for crystals

Knapp (51) measured the thermal conductivity of crystals between mean temperatures of 200°F and 900°F with the apparatus shown in Fig. 3–13. In his work the comparison material of known thermal conductivity was stainless steel.

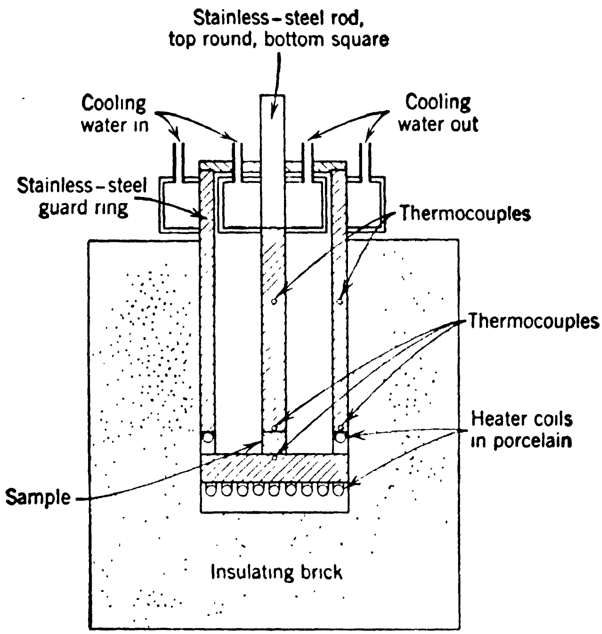


FIG. 3–13. Thermal conductivity. Crystals. Knapp (51).

The test equipment consisted of a 1-cm cube crystal mounted between a stainless-steel disk, electrically heated, and a 1-cm square stainless-steel bar that was water-cooled at the far end. A stainless-steel guard tube surrounded the bar, and the space between the two was filled with crushed insulating brick to prevent cross-radiation and retard conduction between the bar and the guard. The temperature of the bar was measured at two points, one of which was as near the cooler face of the crystal as feasible. The difference in temperature between these two points represents the drop in temperature for a given length of bar. A third thermocouple was placed in the stainless-steel disk as close to the crystal surface as possible. In order to avoid contact resistance between the crystal surfaces and the stainless steel, pure tin foil, 0.001 in. thick, was inserted, which became molten at 450°F and thus made intimate thermal contact.

In Fig. 3–14,

S = crystal cube with thermal conductivity, K_s.
R = stainless-steel bar with conductivity, K_r.
D = stainless-steel disk.
t_3 = temperature of warmer side of crystal.
t_2 = temperature near the cooler end of bar.
t_1 = temperature of bar near cooler side of crystal.
A = distance between t_2 and cooler face of crystal.
B = distance between t_2 and t_1.
l = distance between the two faces of crystal.

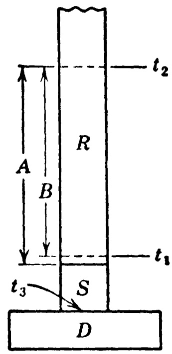


FIG. 3–14. Thermal conductivity. Crystals. Knapp (51).

Then, if there is no transfer of heat between the guard tube and the crystal and bar,

$$q = \frac{K_S[t_3 - (t_1 - t_2)(A/B) - t_2]}{l} = \frac{K_R(t_1 - t_2)}{B}$$

or

$$K_S = \frac{K_R(t_1 - t_2)}{B} \times \frac{l}{t_3 - (t_1 - t_2)(A/B) - t_2}$$

This apparatus, which required the services of a skilled technician to set it up and operate it, gave some excellent results and checked previous observers, although most thermal conductivity determinations on crystals had previously been done at mean temperatures ranging from 200°F down to −300°F. Figure 3–15 shows how the 1911 results of Eucken (29*a*) for crystal quartz at low temperatures compare with the findings of Knapp in 1942 at much higher temperatures.

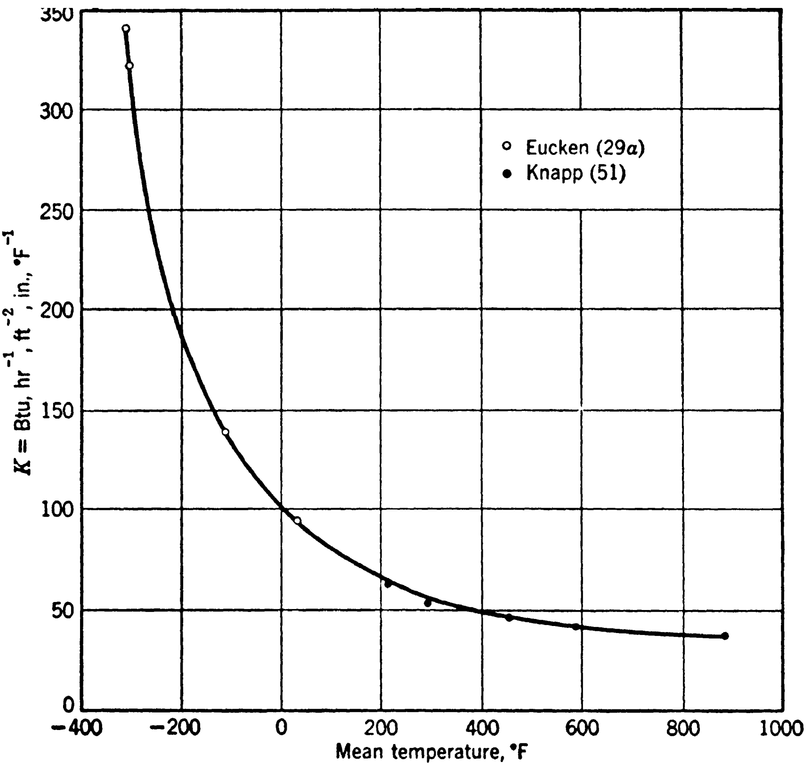


FIG. 3–15. Thermal conductivity of quartz. Parallel to C axis.

"Northrup" method for small samples of high conductivity

Sometimes it is difficult or even impossible to procure sufficiently large samples for the various types of test equipment mentioned previously in this chapter. Adaptations of the "Northrup" method may then prove advantageous if one is willing to sacrifice precision for an inexpensive and simple apparatus.

Figure 3–16 shows one of these adaptations, made for the determination of the thermal conductivity of samples of tungsten carbide, a relatively good conductor, that had a cross-section of ¼ in. by ½ in. and a length of ¾ in. The specimen was placed between two copper bars of the same cross-sectional dimensions as the specimen. In this case the nature of the specimen was such that it could be silver-soldered to the bars.

The outer end of one of these bars was electrically heated, and the outer end of the other bar was water-cooled. Two thermocouples were attached to each bar, one as near the interface as feasible and the other at a measured distance away, say 3 in. The whole assembly was placed in a box and heavily insulated with a powder insulation, such as Santocel, to minimize the heat lost from the copper bars and the specimen to the surroundings.

With the above arrangement, obviously some heat is lost from the bars and sample to the surroundings, and so the rate of heat flow

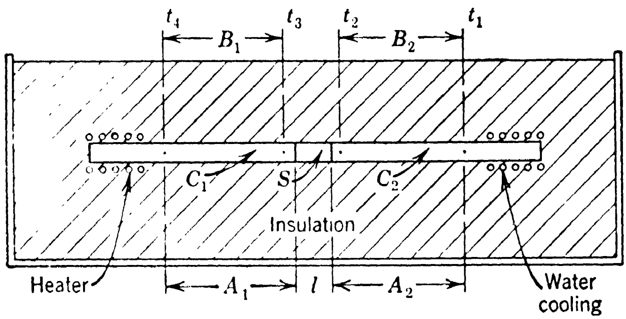


FIG. 3-16. Thermal conductivity. Small samples.

through the assembly is not constant but diminishes slightly with the distance from the warm end. The average rate of heat flow through the warmer bar, C_1, is equal to $q_{C_1} = [K_1(t_4 - t_3)A]/B_1$, and, similarly, the average rate of heat flow through the cooler bar is equal to $q_{C_2} = [K_2(t_2 - t_1)A]/B_2$. The rate of heat flow through the test specimen should be approximately the average of these two, or $(q_{C_1} + q_{C_2})/2$.

The temperature of the warm surface of the specimen may be found approximately by extrapolation of the measured temperature drop in C_1, or $t_h = t_4 - (t_4 - t_3)(A_1/B_1)$, and, in similar fashion, the temperature of the cooler surface of the specimen would be equal to $t_C = t_1 + (t_2 - t_1)(A_2/B_2)$.

The coefficient of thermal conductivity of the sample may now be readily calculated by the usual formula.

$$K = \frac{ql}{A(t_h - t_C)}$$

Conductivity of liquids

Liquids are not ordinarily classed as heat insulation, and so no detailed description of the methods of determining their coefficients of thermal conductivity will be given here. For more information on this subject, with a complete description of the apparatus, see Bates (11).

Emissivity

In calculating the rate of heat transfer by radiation from surfaces to their surroundings, one is generally concerned with the total normal emissivity (all wave lengths) of the surface in question and possibly that of the surroundings as well. Schmidt (88) and McDermott (62) describe excellent methods of determining emissivity of surfaces near room temperature. Figure 3–17 shows the principal details of an apparatus that gives approximate results that check very closely with those of Schmidt. It was designed primarily to determine the emissivity of metals at room temperature but can be used without appreciable error for thin papers that can be cemented to a metal plate.

A sensitive thermopile is enclosed in a steam-heated chamber that has a conical opening at the bottom so that the thermopile "sees" the opening. The opening is first closed by a water-cooled black body, and an electromotive force is developed by the thermopile corresponding to a surface with an emissivity of 1.00. The black body is a water-cooled cone with an angle of about 20° and painted with high-emissivity paint. Any ordinary oil paint is satisfactory and the color has no effect on the results.

The sample forms the cover to a water-cooled plate. It is made watertight by means of a metal frame and gasket that can be screwed firmly to the plate. If this assembly is placed so that it closes the conical opening, the thermopile "sees" the surface and an electromotive force develops correspondingly to the emissivity of the surface. The emissivity of the sample is merely the ratio of the electromotive force due to the surface in question to that due to a black body under the same temperature conditions.

In many methods of measuring emissivity, convection currents are a disturbing factor, but this arrangement, with the heat at the top and the cold sample at the bottom, minimizes them.

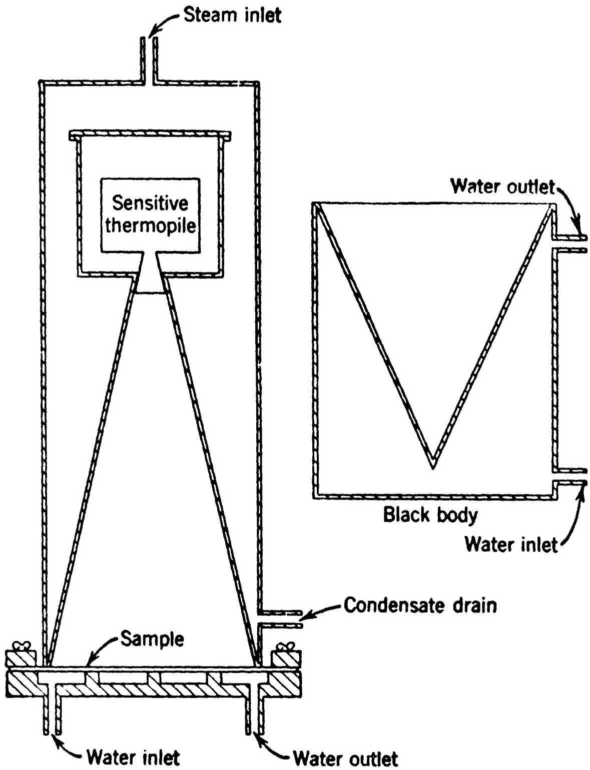


FIG. 3-17. Total normal emissivity apparatus.

Brownlow (14) in 1948 developed an apparatus for determining the total normal emissivity of refractories at high temperatures which shows considerable promise, but sufficient data have not been collected as yet to warrant definite conclusions as to its value.

The apparatus is shown in Fig. 3-18. A rotating cylinder, 2 in. in diameter and 3 in. high, of the sample was enclosed by a Globar furnace except for a small opening about ¾ in. square. A radiation pyrometer was sighted on this opening so that the radiation temperature of the surface could be measured. The true temperature of the surface of the cylinder was measured with a platinum-platinum, 10% rhodium thermocouple inserted in a small coaxial hole. It was found that a negligible temperature gradient occurred radially from the center of the cylinder to the surface at the middle of the cylinder, where all tempera-

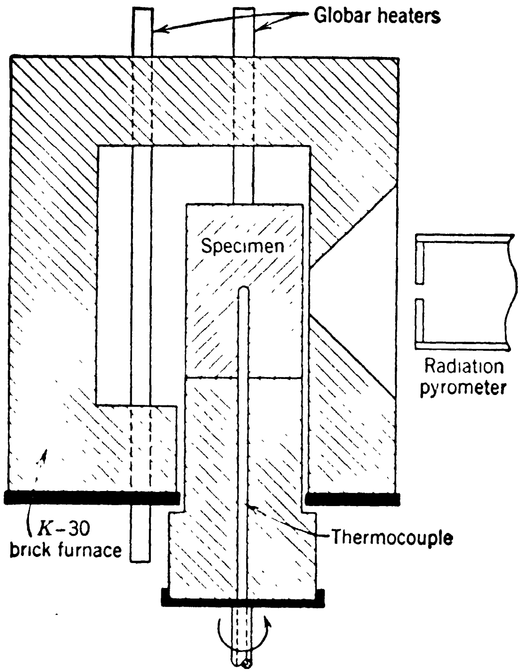


FIG. 3–18. High temperature emissivity. Brownlow (14).

ture measurements are made. The emissivity, after thermal equilibrium has been established, can be found by the following equation:

$$e = \frac{T_R^4}{T^4}$$

where T_R = radiation or apparent temperature (°F absolute) of the surface as determined by a radiation pyrometer calibrated for black-body conditions.

T = true surface temperature (°F absolute) as determined by the thermocouple.

This equipment makes it possible to determine the total normal emissivity from 400°F to 2700°F.

Figure 3–19 shows the results obtained with it for a zirconium oxide cylinder furnished by the Norton Company.

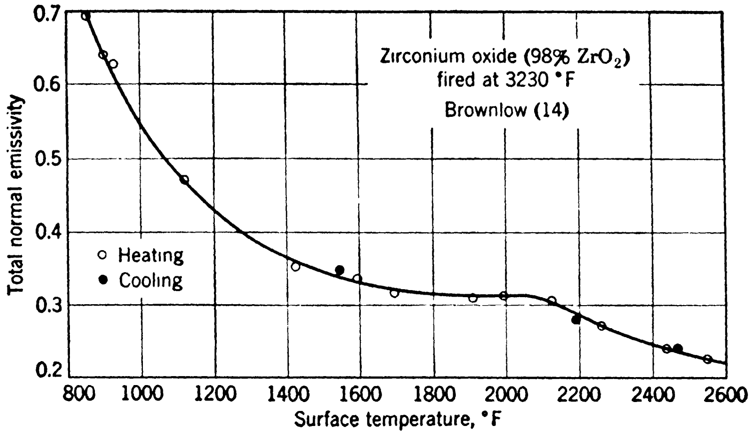


Fig. 3–19. Total normal emissivity of zirconium oxide vs. temperature.

CHAPTER FOUR

Factors Affecting the Coefficient of Thermal Conductivity

Temperature

The formulas derived in Chapter Two for the calculation of the rate of heat transfer by conduction are based on the assumption that the coefficient of thermal conductivity, K, is constant for the material at all temperatures considered in the calculation.

The K value for most insulating materials increases with increasing temperatures. Since, in making any test to determine the coefficient of thermal conductivity, it is necessary to maintain a temperature difference across the sample, the K value varies from the cooler side to the warmer side of the sample. If this temperature drop across the sample is small, say 110°F to 70°F, no appreciable error would be introduced by stating that the K_{90} was equal to the K_m between 110°F and 70°F or at a mean temperature of 90°F. If the relation between K and the temperature is linear, where $K = a + bt$, the K_m between t_2 and t_1 will be equal to K at the mean temperature of the two surfaces, or $(t_2 + t_1)/2$.

For most insulating materials, the assumption that K varies linearly with the temperature is legitimate, and in such instances it is correct to state that the K_m is equal to K at the mean temperature of the specimen. If K should differ considerably from the linear relation to the temperature (Wilkes 110), one can assume that $K = a + bt + ct^2$, and then

$$K_m(t_2 - t_1) = \int_{t_1}^{t_2} K\,dt = \int_{t_1}^{t_2} (a + bt + ct^2)\,dt$$

$$= a(t_2 - t_1) + \frac{b}{2}(t_2^2 - t_1^2) + \frac{c}{3}(t_2^3 - t_1^3) \quad (4\text{–}1)$$

If three determinations of K_m are made at different temperatures, the constants a, b, and c can be calculated and thus K can be evaluated for any temperature within the range of the determinations. Then the K_m between any two temperatures can also be found by Eq. 4–1.

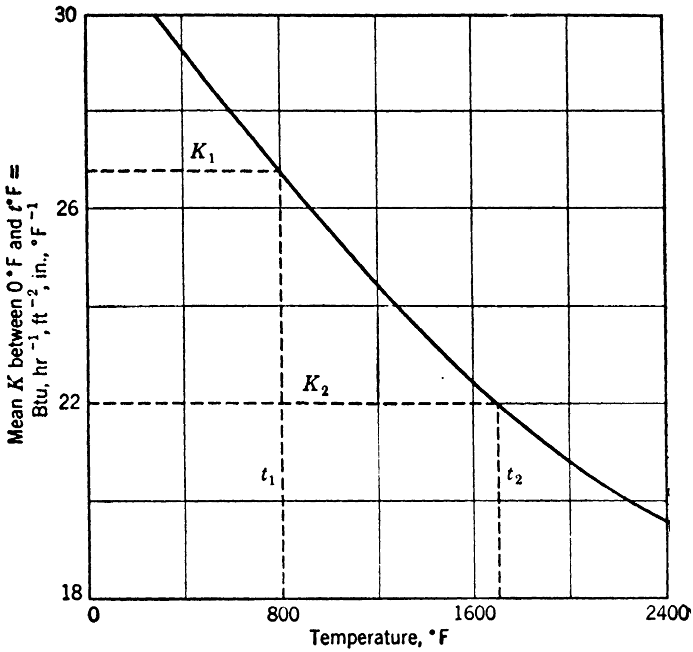


FIG. 4–1. Mean K between 0°F and t°F.

A less laborious method of calculating the K_m between any two temperatures in insulating materials is suggested by Wilkes (110). It is to calculate the K_m between 0°F and four or five values of t°F and then to make a plot of these K_m values against the corresponding temperatures, as shown in Fig. 4–1.

The K_m between any two temperatures t_2 and t_1 can now readily be determined by reading off the plot the K_m values between 0 and t_2 (K_{m2}) and between 0 and t_1 (K_{m1}) and substituting these values in the following equation:

$$K_m \text{ (between } t_2 \text{ and } t_1) = \frac{K_{m2}t_2 - K_{m1}t_1}{t_2 - t_1} \tag{4-2}$$

The rate of heat flow for unit area and unit thickness between t_2 and 0°F is equal to $K_{m2}t_2$, and, similarly, between t_1 and 0 it is equal to $K_{m1}t_1$. The difference between these values represents the rate of heat flow between t_2 and t_1. The K_m between t_2 and t_1 is, consequently, equal to

$$\frac{K_{m2}t_2 - K_{m1}t_1}{t_2 - t_1}$$

Although it is true that the K for most insulating materials increases with increasing temperatures, it does not necessarily increase at the same rate for different materials. The K values of very porous insulators generally have a greater temperature coefficient than materials with fine pores. The K for many loose, fibrous insulators, such as rock wool and glass wool, increases at a higher rate with increasing temperatures than a material such as 85% magnesia. In comparing different insulating materials, the K value loses much of its significance unless the corresponding temperature is also known.

In some materials the K values decrease with an increase of temperature, at least over a fairly wide range of temperature. Eucken (29*a*) found that for a considerable number of single crystals, the K value increased rapidly as the temperature fell from 212°F to −300°F. More recent work by Knapp (51) at temperatures between 212°F and 1000°F indicates that the curve of K vs. temperature flattens as the temperature increases and that, if the temperature is raised sufficiently, the K value increases with further increase of temperature. See Fig. 4–2 for values of crystal quartz by Eucken and Knapp. Sosman (92*a*) states that at liquid-air temperatures (−300°F) crystal quartz is a better conductor of heat than gold. Commercial magnesite brick and recrystallized silicon carbide brick have negative temperature coefficients for K from room temperature to very high temperatures, according to Wilkes (111), but at the higher temperatures the curve of K vs. temperature flattens, showing a tendency to increase with rising temperatures if the temperature is sufficiently high. The curves are similar in shape to those for pure crystals, but the temperature corresponding to a minimum K value is much higher.

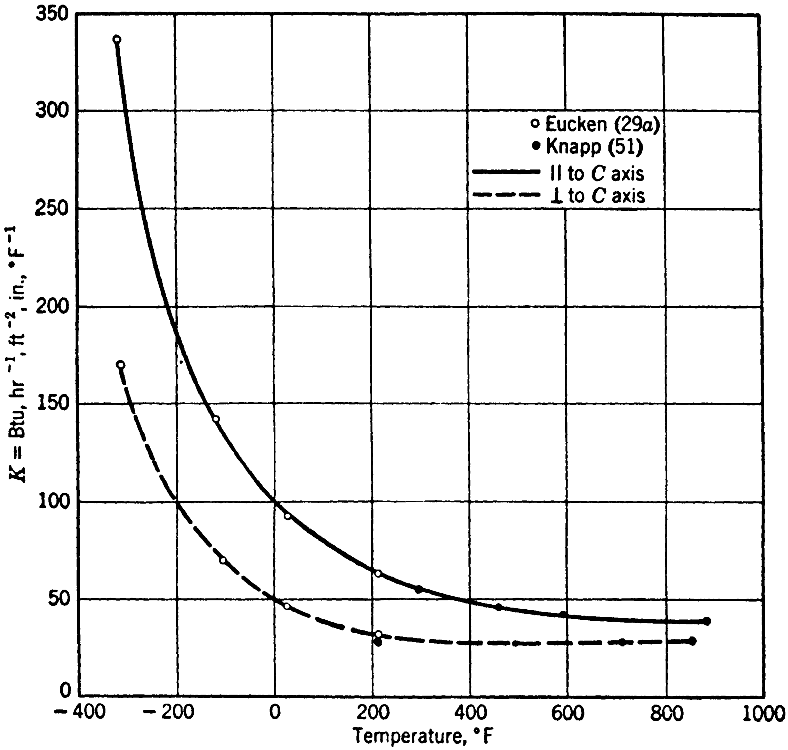


FIG. 4-2. Thermal conductivity. Crystal quartz.

The thermal conductivity of metals, in general, does not vary rapidly with temperature, some showing an increase of K with rising temperatures and others showing a decrease.

Density

The general consensus that the bulk density of a material is a good indication of the relative insulating value is roughly accurate as a whole, but since this assumption has numerous exceptions the indications have limited value. Many fibrous materials and some powders have a minimum K at some particular bulk density, and the K increases from this value if the density is increased or decreased. Figure 4-3 shows conductivity vs. bulk-density curves for two materials with this characteristic.

That bulk density of some pure oxide refractories is no cri-

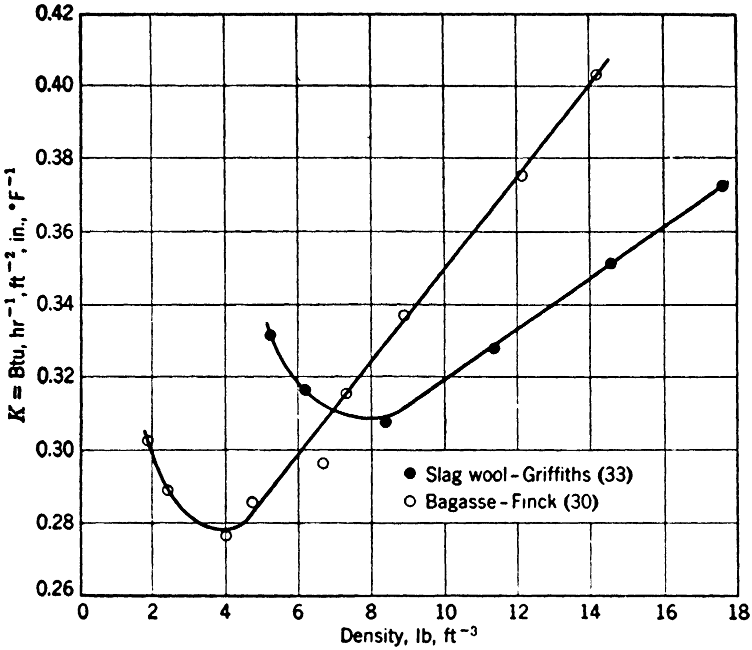


FIG. 4-3. *K* vs. density. Loose, fibrous insulators.

terion of their thermal conductivity is proved by Whittemore (106*a*). The accompanying table clarifies this lack of simple relationship between the bulk density and the thermal conductivity of refractories.

Refractory	Bulk Density	K_{1000}	K_{2000}
Silica brick (111)	95 lb, ft^{-3}	9.6	13.6
Silicon carbide recrystallized (111)	136	151	96
Fireclay brick (111)	143	9.6	11.6
Magnesite brick (110)	159	19.7	13.8
96–97% Magnesium oxide (106*a*)	165	25	16.7
99% Aluminum oxide (106*a*)	195	22	16.9
98–99% Zirconium oxide (106*a*)	250	5.3	6.0

K is expressed in Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$.

Nor has any relationship been found between the thermal conductivity and the density of metals or single crystals.

For any given material, the bulk density varies inversely with the porosity, and the bulk density is a good guide to the thermal conductivity for the particular material in question. It is well known that many of our insulating firebrick consist of the same material as the firebrick but are much more porous with consequent less bulk density and a lower K value. Figure 4–4 shows two examples of the effect of porosity on the K value. For relatively high temperatures, it is important that the pore size be kept relatively small, such as a few hundredths of an inch, or otherwise the transfer of heat by radiation across the pores may become an important factor in the rate of heat transfer,

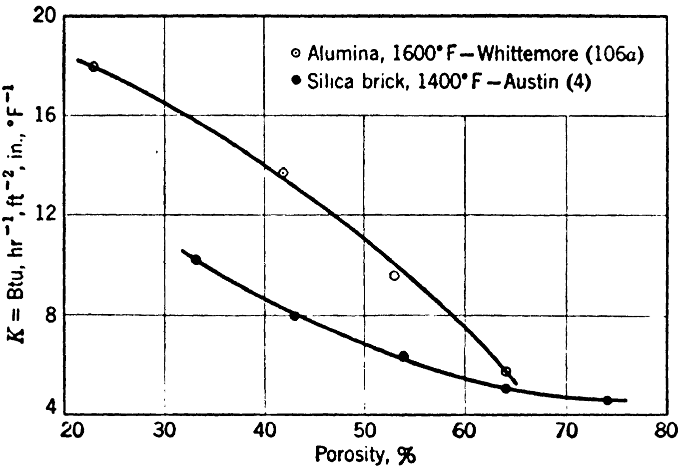


FIG. 4–4. K vs. porosity. Refractories.

with consequent higher K values. Russell (87*a*) emphasizes the importance of small pore size to efficient high-temperature insulation, whereas Austin (4) concludes that pore size is not an important factor in the thermal conductivity of ordinary refractory materials. Austin evidently refers to the small pores usually found in insulating refractories, because if the pores were large, ¼ in. or more in diameter, there is no doubt but that the conductivity would seriously increase at high temperatures.

MacLean (58) made thermal conductivity tests on "bone-dry" samples of eighteen varieties of our common woods with average specific gravities (dry) ranging from 0.16 for balsa wood to 0.76 for rock elm. The relationship between the thermal conductivity at a mean temperature of 85°F and the specific gravity is shown in Fig. 4–4*a*. It appears to be a straight-line relationship.

MacLean recommends the equation $K = 1.39S + 0.165$, where K is the thermal conductivity expressed in Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, and where S is the specific gravity of the dry wood.

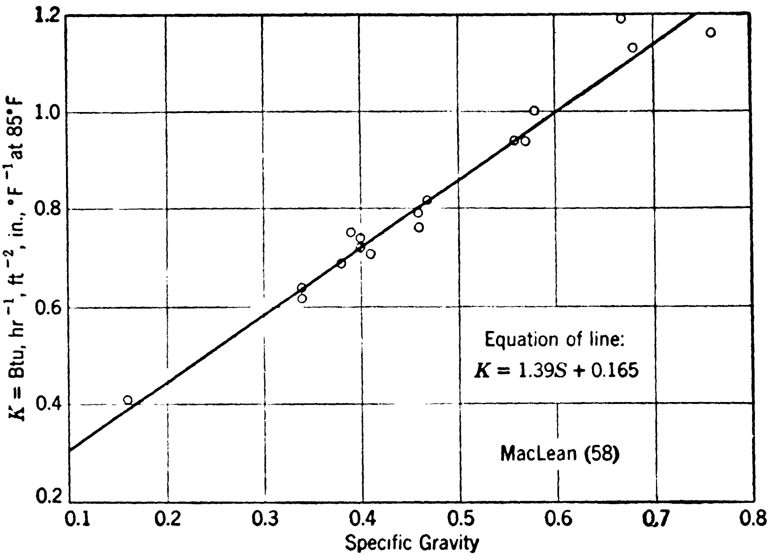


FIG. 4–4*a*. K vs. density. Woods.

Specific heat

Austin (4) states that it was pointed out some years before that in any given system, such as a brick, the rate of flow of thermal energy is proportional to the gradient of internal energy, dE/dx. If this is so, it is proportional to $(dE/dT)(dT/dx)$. For unit dimensions and unit temperature gradient $[(dT/dx) = 1]$, $(dE/dT) = C_v$, the specific heat at constant volume, and the rate of heat flow equals K by definition under these conditions. Since the rate of heat flow is proportional to C_v and at the same time equals K, a correlation should exist between C_v and K so that if C_v were doubled by a temperature change then K should also be doubled. Since C_v and C_p, the specific heat at constant pressure, are nearly equal, C_p should have the same relationship to K as has C_v. There is some evidence that the ratio of C_p/K is roughly constant for amorphouslike materials, such as silica, fireclay, and porous alumina brick, over a wide range of temperatures.

Obviously, this relationship does not exist in materials that have a negative temperature coefficient of thermal conductivity, as have many crystals and some pure oxide refractories. Nor does it exist in very porous or fibrous insulators, nor in powders, which, in addition to conduction of the solid, have radiation across the pores plus conduction and/or convection by a gas.

Apparently there is no relationship between the specific heat of commercial insulators and their *K* value.

Direction of heat flow

The direction of heat flow through most heat insulators has little effect on the thermal conductivity, but there are some important exceptions to this statement. Griffiths (33) measured the thermal conductivity of some lightweight woods perpendicularly and in parallel to the grain and found the *K* values 60 to 80% greater when the heat flow was parallel instead of perpendicular to the grain. (See accompanying table.)

Light Woods	*K*	Mean Temperature
Musanga Smithii (Nigeria)		
parallel to grain	0.84	73°F
perpendicular to grain	0.52	
Herminiera Elaphroxylon (Nigeria)		
parallel to grain	0.61	73
perpendicular to grain	0.38	
Janganda wood (Brazil). 9.5 lb/cu ft		
parallel to grain	0.83	79
perpendicular to grain	0.46	

Austin (4) states that measurements on the thermal conductivity of diatomaceous earth bricks indicate a *K* value 25 to 50% greater when the heat flow is parallel to the layers than when perpendicular to them.

Finck (30) made thermal conductivity determinations on various fibrous materials and showed the effect produced on the *K* value by different arrangement of the fibers. Some of his results are given below, as well as the percentage of change in the *K* value when the fiber arrangement differs from approximate perpendicularity to the direction of heat flow. (See accompanying table.)

	Density	K_{90}	Percentage of Change in K
Flax fibers, perpendicular to heat flow	4.9 lb, ft^{-3}	0.238	...
parallel to heat flow	4.9	0.533	124
random arrangement	4.9	0.278	17
Flax fibers, perpendicular to heat flow	9.7	0.266	...
parallel to heat flow	9.7	0.834	213
Glass fibers, perpendicular to heat flow	10.0	0.260	...
parallel to heat flow	10.0	0.555	113
Fiberboard fibers, perpendicular to heat flow	....	0.34	...
parallel to heat flow	....	0.72	112

In the process of manufacturing fiberboards, the fibers usually are approximately perpendicular to the direction of heat flow. The fiberboard in the tests mentioned was tested first in the form in which it was received from the manufacturer, and then it was sawed into 1-in. strips, which were placed on edge in the plate apparatus for the test, indicating that the fibers were parallel to the direction of the heat flow. In other words, these tests indicate that the edgewise conductivity of fiberboards may be more than double the conductivity when the heat flow is perpendicular to the flat side of the board.

If we assume that there is no convection in fibrous insulating materials, heat is transferred by conduction of the fibers, by conduction of the air, and by radiation across the air spaces. All known fibers have a much higher conductivity than air. Consequently, when the fibers are perpendicular to the heat flow, they play a minor part in the rate of heat transfer, but, when they are parallel to the heat flow, conduction by continuous fibers from the warm side to the cool side becomes an important factor. It is also interesting to note the effect of higher density in flax fibers. The percentage of increase of K for fibers parallel to the heat flow is over 200% for the flax fibers of 9.7 lb, ft^{-3}, density.

If the structure of an insulating material is sufficiently open to permit convection currents, the rate of heat flow is affected by orientation.

Air spaces of sufficient width for convection to occur (stud space in frame walls) transfer heat at different rates, depending upon whether the direction of heat flow is horizontal, vertically upward, or vertically downward. This transference is much more pronounced in air spaces formed by reflective insulation (Chapter Six).

Figure 4–2 shows the K values for a single quartz crystal with the heat flow perpendicular to the C axis and then parallel to the C axis, as determined by Knapp (51) and Eucken (29*a*). Isotropic (cubically symmetrical) crystals do not show this directional effect on thermal conductivity, but crystals with hexagonal, tetragonal, or trigonal symmetry have a maximum or minimum thermal conductivity when the heat flow is parallel to the principal axis and, conversely in all directions, perpendicular to this axis.

In crystals belonging to the rhombic, monoclinic, or triclinic systems, the thermal conductivity is different for the three axes. This directional effect in single crystals has no bearing on the thermal conductivity of bricks made largely of crystals, on account of the random distribution of the crystals in the brick.

Convection in insulators

For many years, there has been considerable debate among heat insulation experts as to whether convection really occurs in some types of insulation to a sufficient extent so that the rate of heat transfer is appreciably affected. Enough evidence is now available to warrant making the statement, with assurance, that with some insulators under suitable conditions convection is present to an extent that very materially increases the rate of heat transfer through the material.

A space filled with loose, fibrous materials, for example rock wool, glass wool, shredded redwood bark, consists largely of air. The weight of solid limestone or glass is roughly 165 lb, ft^{-3}, but if rock wool, which is usually made from limestone, is packed in a space to a density of 8 lb, ft^{-3}, the space must consist of approximately 95% air by volume. Glass wool packed to a density of 3 lb, ft^{-3}, would occupy only about 2% of the volume of a space, the remainder being air. Since the air passages in this type of material are continuous, they offer little resistance to the flow of air, when loosely packed. In a typical frame wall with the stud space filled with a loose, fibrous insulating material, the relatively warm air on one side of the insulation would tend to rise, whereas the cooler air on the other side of the insulation would tend to fall, on account of the temperature-caused difference in density of the air on each side. This tendency for the air to rise on one side and fall on the other would set up convection currents, provided that it could overcome the resistance to air flow due to the fibrous insulating material and the spatial dimensions.

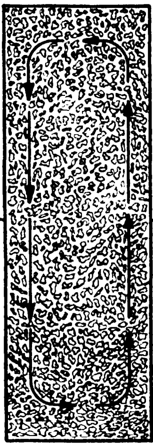
0°F  70°F

Fig. 4–5. Convection in rock wool.

In Fig. 4–5, the air within the insulation near the cold side is approximately 15% heavier than the air near the warm side, thus producing a strong tendency toward the production of convection currents.

Griffiths (33) in 1929 found, when using a vertical guarded plate for the determination of K values, that the temperature of the bottom of the guard was colder than that of the upper part of the guard in certain types of materials. Consequently, he constructed the guard so that it had four independently controlled heating circuits corresponding to sections *A*, *B*, *C*, and *D* in Fig. 4–6.

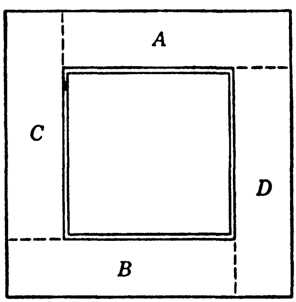


Fig. 4–6. Convection in guard area of vertical plate tester. Griffiths (33).

With materials such as corkboard, the power input to each of these four sections was essentially the same in order to produce a uniform temperature over the entire guard area, but with coarsely granulated cork, he found that the following watt input to the various sections was necessary to equalize the guard temperature.

Section A, top panel	2.3 watts
Section B, bottom panel	4.8 watts
Section C, left-hand panel	2.7 watts
Section D, right-hand panel	3.1 watts

Griffiths states that "the only reasonable explanation of these effects is that convection currents are possible in granular material."

Further investigations with other apparatus indicated that convection occurred in granular cork and granular charcoal when placed in vertical testers. Griffiths came to the conclusion also that granular charcoal prevents convection when the plate tester is in a horizontal position with heat flowing upward. He explains that this effect is perhaps due to the rather close proximity of the cold plate to the hot plate, thus causing a greater restriction to air flow than when the tester is in a vertical position.

To bring further light on this matter of convection in insulation, Griffiths made a comparison of the thermal conductivity of slag wool packed to various densities in vertical plate testers of three sizes. His three square plate testers had the following dimensions:

Designation	Heater Area	Width of Guard	Sample Thickness	Range of Temperature
12 in.	8 in. × 8 in.	2 in.	1.13 in.	50–97°F
20	12 × 12	4	1.5	36–54
60	36 × 36	12	6	46–58

The results of these tests, which are plotted in Fig. 4–7, give very interesting information concerning convection in loose, fibrous insulation. It will be noted that the K values, as determined on all three testers, agree when the density is between 12 and 20 lb, ft^{-3}. The 60-in. tester indicates a minimum K value of 0.33 at a density of 12 lb, ft^{-3}, and, as the density is further decreased, the K value increases sharply, reaching 0.43 at a density of 6.5 lb, ft^{-3}.

The K values from the 20-in. plate decrease with decreasing density, reaching a minimum value of K equal to 0.31 at a density of 8.5 lb, ft^{-3}. As the density is further decreased, the K value increases, reaching a K value of 0.33 at a density of 5 lb, ft^{-3}.

With the 12-in. plate, the K values decrease with decreasing density until a minimum value of 0.290 is found at a density of 7 lb, ft^{-3}. A K value of 0.293 was found at a density of about 6 lb, ft^{-3}, but, as this was the only point taken below the density corresponding to the minimum K value, it does not prove that the minimum has been reached.

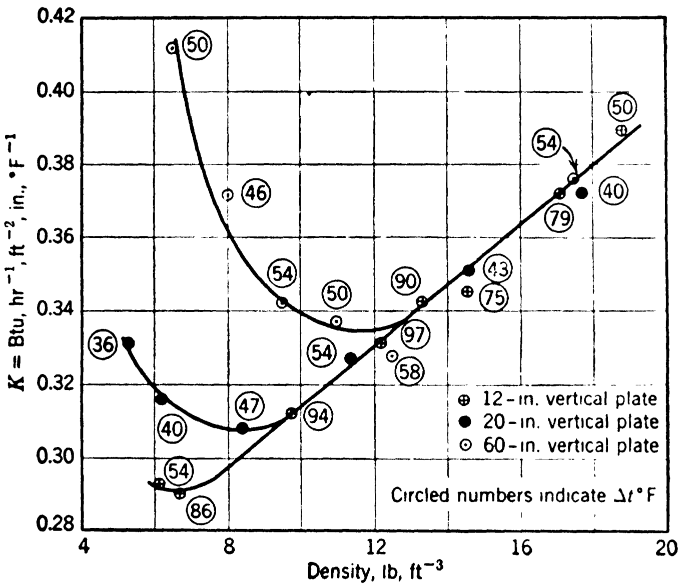


FIG. 4–7. K vs. density. Slag wool, various sizes of vertical plate testers. Griffiths (33).

If convection currents are present within an insulating material, they would be expected to increase with decreasing density, and this trend is shown definitely by the portion of the curves in Fig. 4–7 to the left of the density corresponding to the minimum K values. One would also expect that convection currents within the insulation would increase with greater width of space, and this expectation is also justified by the experimental data. At densities greater than those corresponding to minimum K values, all three plates give approximately the same K for any given density, indicating that at these densities the resistance to air flow through the insulation is so high that convection does not occur to any appreciable extent for the given temperature conditions and dimensions of plates. Over this range the relation between density and K is approximately a straight line. Although the temperature drop between the hot and the cold plates is not the same in all the tests, it is of the same order of magnitude throughout. The circled figures in Fig. 4–7 show the temperature difference between plates for each test point.

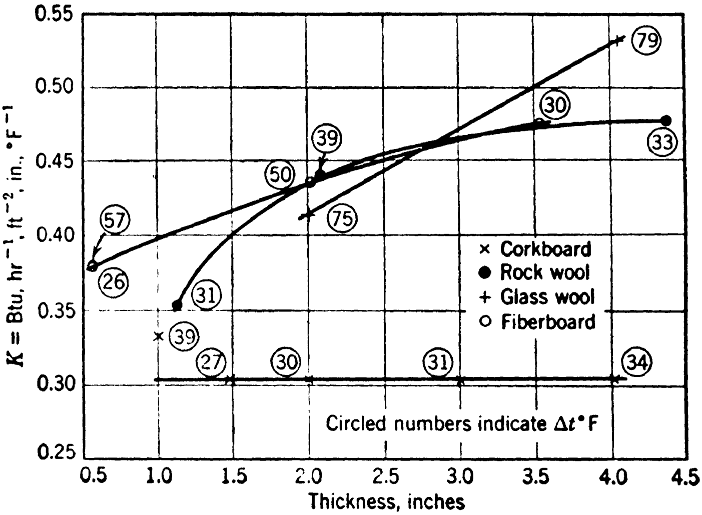


FIG. 4–8. K vs. thickness. Allcut and Ewens (3).

Allcutt and Ewens (3) in 1937 investigated the question of convection in certain insulators. Using a square vertical plate apparatus with a heater 16 in. square and a guard 4 in. wide surrounding the heater, they found, with thicknesses varying from 1 to 4 in., that in some instances the K value varied with the thickness although the mean temperature was maintained essentially constant. Corkboard showed no variation of this kind but rock wool, 10 lb, ft^{-3}, glass wool, 1.5 lb, ft^{-3}, and fiberboard, 15.5 lb, ft^{-3}, showed a marked variation, as indicated in Fig. 4–8. They concluded that the variation of K with thickness was due in part to contact resistance between the plates and the sample and in part to convection currents within the sample.

These investigators made further tests on a 4-in.-thick rock wool sample packed to a density of 10 lb, ft^{-3}. The temperature drop across the sample was varied, and a considerable variation of K was noted. The mean temperature was not constant during the tests, but the variation was much greater than could be accounted for by mean temperature alone. Horizontal cardboard separators were then placed in the sample so that it left rock wool sections only 2 in. high. The K values were very much reduced, and it is logical to conclude that the convection

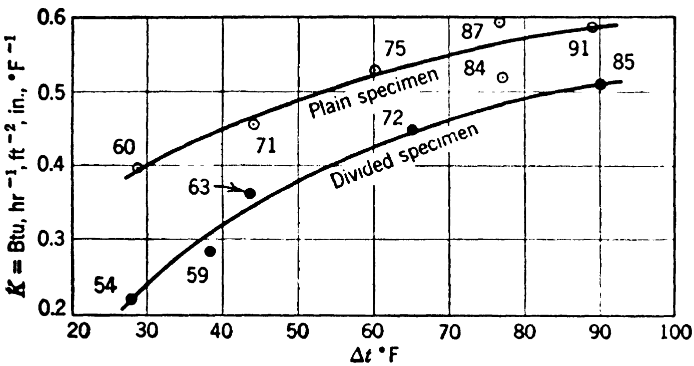


FIG. 4–9. Variation of K with height of space and Δt. Allcut and Ewens (3).

currents were also reduced in this latter experiment. The contact resistance between plate and sample must have been essentially the same in both sets of tests. The results are shown graphically in Fig. 4–9, and the circled figures indicate the mean temperature for each test point.

Wilkes and Vianey (116) in 1943 made tests to determine the rate of heat loss through ceilings with heat flowing upward through various loose insulators of different thicknesses. For comparison purposes in one test the ceiling was insulated with 3-in. corkboard.

The method consisted of a typical guarded-box apparatus oriented to make the test section in a horizontal position with the heat flow upward. The test area was completely separated from the guard area by wood strips to eliminate the possibility of convection currents carrying heat from one area to the other. It was found by experience that it required from 3 to 4 days to establish thermal equilibrium with ceilings 3 to 6 in. thick.

The conductance of an uninsulated ceiling in this equipment checked reasonably closely to the accepted values. When the ceiling was insulated with 3-in. corkboard, the K value of the corkboard as determined by this type of test proved to be almost exactly the value determined by the guarded-plate method. However, when low-density loose, fibrous materials comprised the insulation, the conductance values were considerably higher than would result from calculation employing accepted K values for the insulating material. With rock wool of various thicknesses and a density of 5 to 6 lb, ft^{-3}, the apparent K value was approximately double the accepted value for this material. Expanded mica with a density of 7.2 lb, ft^{-3}, showed a similar result. With batt-type insulation the apparent K values were from 7 to 61% greater than the usual values. These batts came from different manufacturers. Two types of blanket insulation were tried but there was no appreciable change in the apparent K value from the accepted values for these materials.

The only conclusion that the authors could reasonably make for the difference between the apparent K as determined in these tests and the K value from plate tests was that convection currents were present which aided the transfer of heat through the material.

Rowley and Lund (87) in 1943 mounted a guarded box onto trunnions so that a wall or ceiling could be tested either in the vertical position or in the horizontal position with heat flowing upward. They came to the conclusion after a large number of tests that there was no appreciable difference between the thermal conductivity of the insulation in the vertical position and that in the horizontal position with heat flow upward. These tests were made very rapidly, and, in one period of 9 days, eight walls were tested, one of which had 6-in. rock wool insulation. It is difficult to understand how thermal equilibrium could have been definitely established in these tests. Furthermore, the test area was not separated from the guard area in the insulation, and, if convection currents were present, the effect could not be accurately measured. It is unfortunate that some tests were not made in the horizontal position with heat flow downward, which would have eliminated any convection currents.

Finck (30) attempted to show the effect of orientation on the

K value of Kapoc with a density of 0.2 lb, ft^{-3}. The sample was 1 in. thick and 8 in. square, including the guard area. Tests were made only in the vertical position and in the horizontal with heat flow downward. Convection was apparently of minor importance in these tests.

Lander (53*a*) made tests with 12-in.- and 24-in.-square vertical plate testers to determine the effect of air circulation in insulation. He concluded that convection effects were slight for loose, fibrous materials.

In the experiments of both Lander and Finck, the plate equipment was relatively small and apparently no attempt was made to prevent convection passing from the guard area to the test area. These two facts alone might make it impossible to detect any appreciable convection in the test area, because in a vertical plate, if convection were present in the insulation, the heat transfer by convection would take place primarily at the top and the bottom of the guard area. The air currents would naturally tend to rise near the heated plate, cross to the cooler side at the top of the guard area, then fall along the cool side and cross to the heated plate near the bottom of the guard area. Furthermore, neither of the investigators attempted to measure convection in the most favorable position, horizontal with heat flow upward.

Although there is much more to be learned concerning the effect of convection in insulation, present evidence appears sufficient to warrant the statement that convection occurs in low-density fibrous or granular insulators if conditions such as orientation, temperature difference, and dimensions of the insulated space are favorable.

Moisture

The thermal conductivity of water is 4.8 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$, which is more than ten times the K value of most of the insulating materials for temperatures below 212°F. If moisture is present in insulation, one would expect the K value to be greater than that of the dry material. Moisture may be added by absorption of water vapor in the insulating material, itself, due to exposure to air containing water vapor. Then the amount of water present depends upon the relative humidity and the temperature of the ambient air. Materials such as wood, corkboard, and fiberboards gain or lose moisture in accordance with the humidity of the surrounding atmosphere. Some insulating materials, such as rock wool, glass wool, and metal foils, are not appreciably affected by the relative humidity of the air if the temperature throughout the insulation is above the dew point. The A.S.T.M. standard guarded-plate method to determine K values calls for the test to be made on a "bone-dry" sample. This requirement is essential in order to secure consistent results from different laboratories and different atmospheric conditions, but, when K values are being taken from various tables, consideration should be given to the fact that these values are minimum values and that in service the majority of insulating materials that absorb moisture from the air have a higher K value than that determined by test.

The effect of moisture content on the K value is very difficult to determine accurately, because in making a thermal conductivity test a temperature difference between the two sides of the sample is maintained and moisture always tends to migrate from the warmer to the colder side. This difficulty leads to uncertain results but sufficient experimental work has been done to show that an increase in moisture content always means an increase in the K value of insulators.

Miller (66) found the following changes in the K values as they were determined first in the dry condition and then after exposure to an atmosphere of 50% relative humidity at 70°F.

	K—Dry	K—50% Relative Humidity	Percentage of Increase in K
Wood fiberboard	0.290	0.316	9.0
Cane fiberboard	0.353	0.364	3.1
Flax fiberboard	0.326	0.337	3.4
Dry Zero	0.230	0.266	15.6
Corkboard	0.323	0.357	10.5

The moisture content of a sample of insulating material is usually reported as percentage by weight. This method has certain objections. If one were comparing two materials, A and B, with densities of 10 lb, ft^{-3}, and 20 lb, ft^{-3}, respectively, and if the percentage of moisture by weight was found to be

10% for A and 5% for B, the conclusion might be drawn that sample A contained twice as much moisture as B. Actually, the weight of water per unit volume would be the same for both samples. If we calculated on a percentage by volume basis, we would have

$$A = \frac{10 \times 10}{62.5} = 1.6\% \text{ by volume}$$

$$B = \frac{5 \times 20}{62.5} = 1.6\% \text{ by volume}$$

Wilkes (113) in 1934 took four different makes of insulating boards and stored them in an unheated warehouse for approximately one week. These samples were wrapped in moisture-proof paper, and the K value was determined. The samples were then dried to determine the moisture content and were allowed to remain in a heated laboratory for a week or more, and the K value was again determined. Figure 4–10 shows the results of these determinations.

The Cork Insulation Manufacturers Association (25) reports the variation in the K of corkboard with moisture content, these findings also being shown in Fig. 4–10.

Rowley and Algren (85*a*) in 1937 determined the thermal conductivity of a large number of American woods with different moisture content. Three typical examples of their results are shown in Fig. 4–11.

The experiments cited above deal mainly with the variation of K in some types of insulating materials when exposed to air at various relative humidities. Examination would reveal no visible moisture in any of them, although the moisture content might be as high as 20% by volume. A much more serious effect from moisture is found when the temperature of part of the insulation is below the dew point of the air on the warmer side. If water vapor from this air penetrates the insulation it will condense as visible water when it arrives at that part of the insulation where the dew point temperature is reached. Nearly all insulating materials transmit water vapor readily, but there are a few exceptions, such as Foamglas and metal reflective insulation, and even they have joints that are vulnerable.

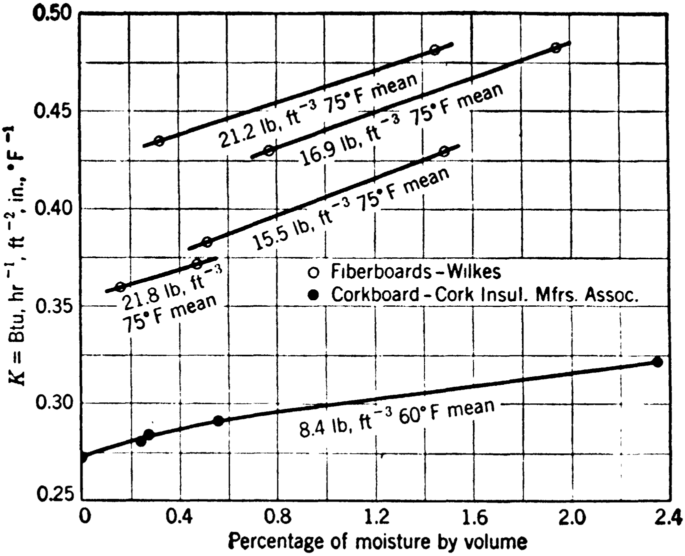


FIG. 4-10. K vs. percentage of moisture by volume. Insulating boards.

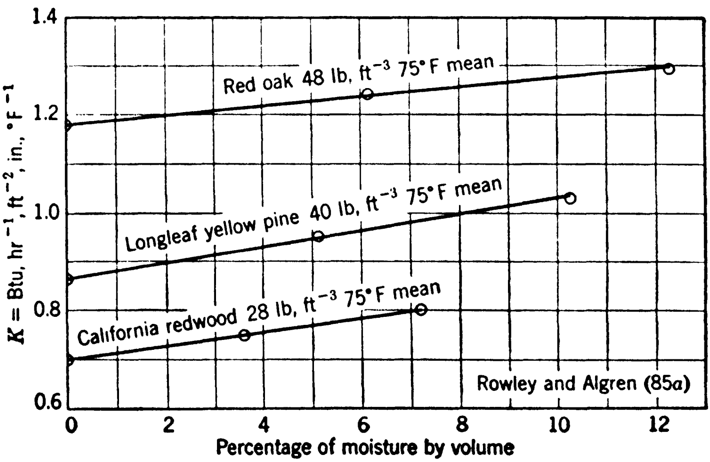


FIG. 4-11. K vs. percentage of moisture by volume. Woods.

Nearly all attempts to solve this problem which have been more or less successful have followed the same general idea. If the warm side of the insulation could be protected with a perfect vapor seal, water vapor could not get into the insulation and there would be no condensation within the wall. Some manufacturers enclose the insulation between sheets of asphalted paper, thus making what is known as blanket insulation. This method helps considerably in preventing water-vapor transmission, but joints and cut edges permit a limited amount of penetration. Others install so-called vapor seals, consisting usually of asphalted paper, but none of them are 100% perfect. Fortunately for house insulation, the conditions are not very severe and either of the above methods prove reasonably satisfactory. For severe conditions, such as found in cold-storage warehouses, freezers, etc., asphalt mastic, moppings of hot asphalt, or Foamglas blocks are usually employed with considerable success.

Willey (118) made an investigation of the effect of moisture on the U value of uninsulated frame walls. He found an increase of 12.4% in the U value for a wall exposed to rain over that for a dry wall.

Radiation

The majority of insulators contain air either in sealed pores, as in Foamglas, or in a continuous pore space, as found in the loose, fibrous materials such as rock wool. The transfer of heat across these air spaces must occur by radiation from one side of the space to the other, conduction, and (or) convection. Some of the lightweight insulating materials actually transmit heat by radiation directly from the warmer to the cooler side.

Finck (30) studied the effect of dusting fibrous insulating materials with aluminum powder in an attempt to reduce the radiation within the sample. With low-density materials he found that the addition of aluminum powder reduced the K value appreciably.

	Density	K_{90}	Percentage of Reduction
Kapoc	0.19 lb, ft^{-3}	0.374	....
dusted with aluminum powder	0.19	0.316	15.5
Asbestos	2.2	0.380	....
dusted with aluminum powder	2.2	0.312	18.0

With the same materials packed to higher densities, the effect of the aluminum powder was negligible.

Berchtold (13) found that with glass wool 1 in. thick he could reduce the K value by means of aluminum foil on each side of the test specimen.

	Density	Percentage of Reduction in K due to Aluminum Foil
Glass wool	1 lb, ft^{-3}	20
	3	9
	6	7

Johnson and Ruehr (47) determined the K value of ½-in. fiberboard by the plate method, taking plate surfaces of different

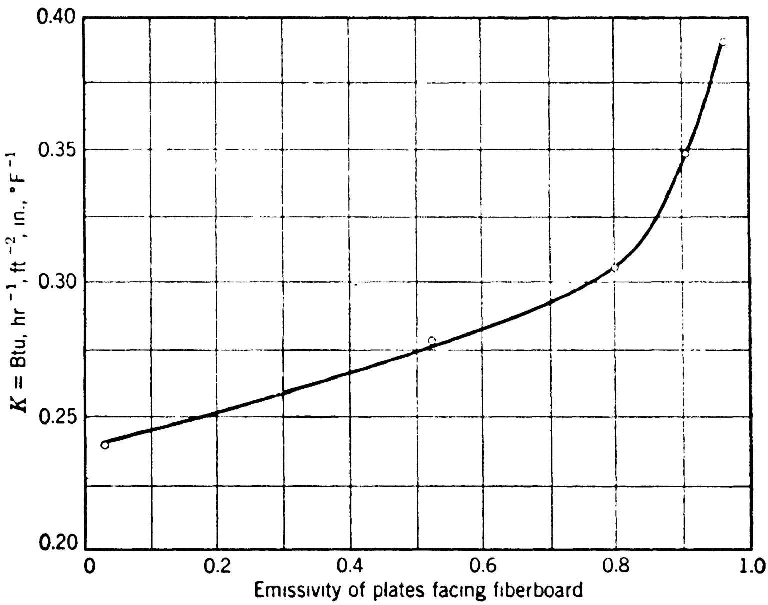


FIG. 4-12. Effect of emissivity of plates on K of fiberboard. Johnson and Ruehr (47).

emissivities. Figure 4-12 shows their findings. Reduction in K values up to 38% of the highest value were observed.

Silica aerogel (Santocel) is somewhat transparent to infrared radiation, and White (105) shows the effect on the K value when powdered metals are added to reduce radiation. Figure 4–13 shows the results graphically. He further experimented with different amounts of powdered silicon and found, for the particular type of aerogel used, that 15% by weight was the optimum amount for the lowest K value.

The American Society of Testing Materials recognized the effect of low-emissivity surfaces facing the insulating material

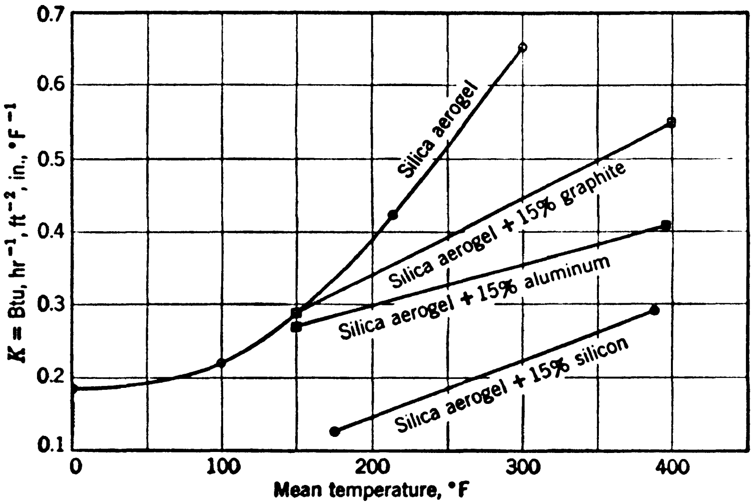


FIG. 4–13. K vs. mean temperature. Silica aerogel with opacifiers. White (105).

in the guarded-plate method of determining coefficients of thermal conductivity, and now the standard test code requires that the surface of the plates facing the sample must have a high emissivity. Many plate tests before 1935 were made with bright metal plates in contact with the test sample, thus giving lower values for K than would ordinarily be found in practice.

It is evident from the foregoing that radiation does play an important part in the rate of heat transfer through some types of heat insulation.

Air pressure

Insulating materials are seldom used under greatly reduced pressure, but data concerning the variation of K as the pressure is reduced to a high vacuum gives valuable information about the mechanism of heat transfer in such materials.

Smoluchowski (92), Aberdeen and Laby (1) and Kistler and Caldwell (50) have performed experiments along this line. Kistler and Caldwell found that the K_{95} value for commercial corkboard dropped from 0.29 at atmospheric pressure to 0.087 under a high vacuum. This result indicates that nearly 75% of the heat transfer through corkboard at atmospheric pressure is due to the confined air, leaving only 25% for conduction by the cork and radiation across the pores. It explains why one can make

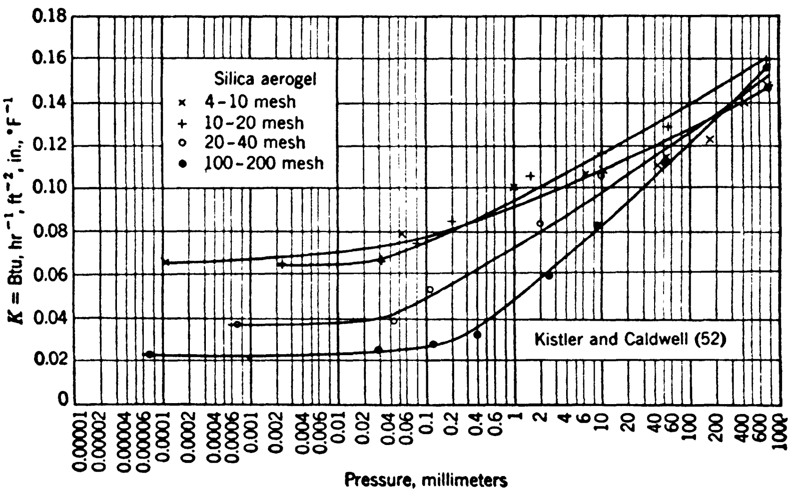


FIG. 4–14. K vs. pressure. Silica aerogel.

excellent insulating material from fine fibers of relatively good conducting material, such as glass or rock.

Kistler and Caldwell also made thermal conductivity determinations on silica aerogel at various pressures and for various screen sizes, as shown in Fig. 4–14. The percentage of heat transfer due to the air was found to vary between 56% for the coarse mesh and 85% for the finest mesh. In order to show the variation of K with the pressure, the pressure is plotted on a logarithmic scale, which gives one the impression that the K value changes more rapidly with change in pressure near atmospheric pressure than at a high vacuum. Actually, the change in K value for the 4–10 mesh sample is approximately 0.035 for a 1-mm drop in pressure at 1-mm pressure and only 0.000023 for a 1-mm drop in pressure at 760-mm pressure. If it were possible to plot the K value against pressure on a uniform scale, the slope of the curve at low pressures would be much greater than near atmospheric pressure. Figure 4–15 shows K plotted against pressure in this way, and the abrupt drop in K as the pressure is reduced below 1-mm pressure is shown but it is impossible to read a plot of this kind at pressures below 1-mm pressure.

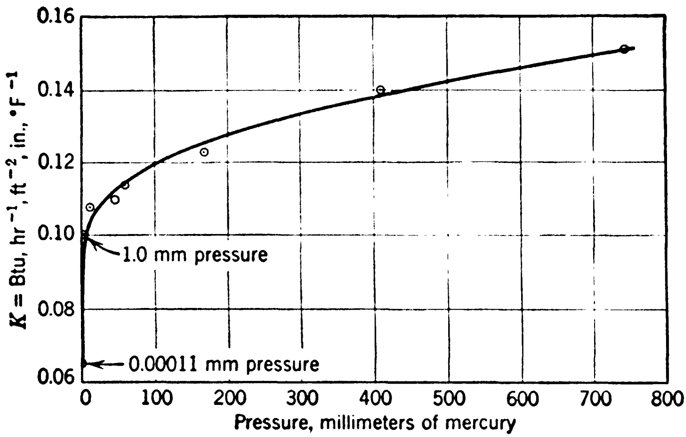


FIG. 4–15. K vs. pressure. Silica aerogel. Kistler and Caldwell (50).

The conduction of heat through air (or any other gas) is caused primarily by molecular motion. The molecules in the air move with high velocity but cannot travel far before a collision takes place with other molecules. The average distance that the molecules travel between collisions has been determined and is called the *mean free path* for the temperature and pressure involved. The mean free path for the molecules in air at 32°F for various pressures is given below.

Pressure	Mean Free Path
760 mm of mercury	3.8×10^{-6} in.
10	2.89×10^{-4}
1	2.89×10^{-3}
0.1	0.0289
0.01	0.289
0.001	2.89
0.0001	28.9

Except at fairly high vacuum, the mean free path is very small, and the distance across most air spaces in insulating materials at atmospheric pressure is greater than that of the mean free path. If the pressure is reduced from 760 mm to 76 mm, temperature remaining constant, there will be $\frac{1}{10}$ as many molecules per unit volume but because of this reduction in pressure the mean free path of these molecules will be increased tenfold. In other words, at the lower pressure one will have $\frac{1}{10}$ as many molecules, but they will, on the average, travel ten times as far between collisions; thus the conductivity would be the same for both pressures. It has been experimentally determined that the conductivity of gases does not change with the pressure as long as the width of the gas space exceeds the mean free path of the molecules.

If the distance across an air space becomes less than that of the mean free path, the conductivity of the air is directly proportional to the pressure, reaching zero at zero pressure. The commercial vacuum bottle is evacuated to approximately 0.001-mm pressure and the mean free path is 2.89 in. at 32°F. The width of the space between the inner and outer walls might be approximately $\frac{3}{8}$ of an inch, which is much less than the mean free path, and the conductivity of the remaining air is only a small fraction of the conductivity at higher pressures. If the width of the interwall space in a vacuum bottle were greater than that of the mean free path, lowering the pressure to 0.001 mm would be of little value in reducing the conduction across the space.

Referring to Fig. 4-14, we observe that the 100–200 mesh silica aerogel is a better insulator under high vacuum than the coarser mesh material. This superiority is undoubtedly due to the smaller air spaces in the finer mesh material: the mean-free-path distance is reached at higher pressures, and, consequently, the reduction of conductivity with pressure begins at higher pressures than in the coarser samples.

Before Kistler and Caldwell's paper, it was generally believed that no insulator could have a lower K value at normal pressure than that of air, since all known solids have a higher K than air and since heat insulators consist of solids and air spaces. The thermal conductivity of silica aerogel is shown to be less than that of air at the same temperature, and this fact has been confirmed by many others. Kistler and Caldwell estimated that the air spaces in silica aerogel were probably well under 4×10^{-6} in. and thus of the same order of magnitude as the mean free path of the molecules in air at atmospheric pressure and temperature. If the structure of the aerogel is such that the distance of molecular travel is less than the mean free path, then one would expect to find an insulator whose K value might be less than that of air under normal conditions. Kistler and Caldwell's results show that the air in aerogels is responsible for a large part of the conduction, and, since the mean free path is greater than the width of the spaces, we have a heat insulator that is a better insulator than air. A form of carbon black (Smith and Wilkes 91) also has been found to have a lower K value than air, presumably for the same reasons.

If the interwall space of vacuum bottles were filled with a fine powder and then evacuated, approximately the same insulating effect could be attained at somewhat higher pressures, possibly 0.1 mm, than 0.001 mm.

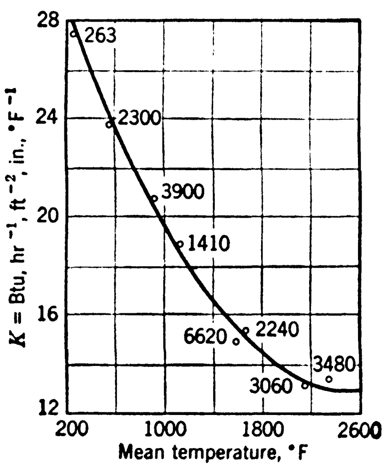


FIG. 4-16. K vs. mean temperature. Magnesite brick. Variation with rate of heat flow. Wilkes (110).

Rate of heat flow

Frequently the question arises whether K varies with the rate of heat flow. Wilkes (110), in determining the conductivity of magnesite brick, investigated this matter to a limited extent. Figure 4-16 shows a curve of K vs. mean temperature, and the rate of heat flow for each point on the curve is given in Btu, hr^{-1}, ft^{-2}. The rate of heat flow varies from 263 to 6620 Btu, hr^{-1}, ft^{-2}, but there is no evidence that the K value is influenced by this factor.

CHAPTER FIVE

Types of Insulating Materials

Insulating materials are frequently classified in accordance with the temperatures at which they are used, such as low-temperature (<212°F), moderate- or steam-temperature (212°F to 1000°F), and high-temperature (>1000°F). This arrangement is not entirely satisfactory because many insulating materials can be properly employed in two or more of these temperature ranges. It seems better to classify these insulators according to physical structure, such as loose-fill, rigid-board, and insulating-brick. At the end of this chapter a list of the commonest types of insulation for various purposes, for example, refrigeration, house, and furnace, will be given. In the appendix, a table will be found giving the K value, bulk density, and mean temperature for many well-known insulating materials. Reflective insulation will be discussed in Chapter Six, since the mechanism of heat transfer and calculation of the rate of heat transfer are entirely different from those encountered with most insulating materials.

Loose-fill insulation

This type of material is generally made in powdered, granulated, cellular, or fibrous form. The advantage of loose-fill insulators is that they can generally be applied, particularly to existing structures, by pouring or blowing. One disadvantage in many cases is that settling must be guarded against, and another is the fact that it is difficult to be sure that the material does not "bridge" over, leaving empty pockets in the walls.

Mineral Wool. The term mineral wool applies to rock wool, glass wool, slag wool, and kaolin wool. Rock wool is prepared by melting rock, usually an argillaceous limestone. As the molten rock issues in a small stream from the furnace, it encounters a blast of steam converting the rock into fine fibers that freeze almost instantly. After the "shot" or drops of the rock that are not converted to fibrous form have been eliminated, the remainder is collected and bagged for hand-packing insulation. It can readily be run through a machine that changes this fibrous material to granulated or nodulated rock wool, which pneumatic machines can blow into house walls, etc. Glass and slag wools are made in much the same way except that glass and slag are substituted for limestone. Kaolin wool (Wilkes 113) is a new product made from molten aluminum silicate in much the same way as the other mineral wools, but it will withstand much higher temperatures because of the refractory nature of kaolin.

Diatomaceous Earth. Diatomaceous earth consists of the tiny silica remains of "diatoms," or microscopic plants, deposited thousands of years ago in layers hundreds of feet thick. When ground to a fine powder, it can serve as a loose-fill insulation.

Vermiculite. Vermiculite is a form of mica (aluminum-magnesium silicate) that expands to many times its original volume when heated. After crushing and grading for size, the material is bagged and ready for pouring into place.

Gypsum. Gypsum when ground is sometimes used for a loose-fill insulation.

Silica Aerogel. White (105) states that when a silica aquagel is dried by heating at normal pressures, it shrinks to about one fifth the original volume and the product is similar to the well-known silica gel. If the water in the aquagel is replaced with alcohol and the resultant product heated in an autoclave to the critical temperature of the alcohol with a pressure in excess of the critical pressure, shrinkage is eliminated and the product is left with a bulk density of about 6.5 lb, ft^{-3}. The material consists of a system of extremely fine pores of submicroscopic size, and it has been calculated that the pore volume is approximately 94% of the total. This material has some extremely interesting properties. It has a lower K value than air and has the lowest reported K value of any known material. The insulating value can be further reduced by the addition of powdered silicon, which retards the transmission of infrared radiation through the silica aerogel. At present, this material is available only in the powdered form.

Crushed insulating brick, asbestos fibers, granulated cork, and many other materials sometimes serve as loose-fill insulation.

Shredded redwood bark is used as a hand-packed insulator for house insulation, particularly on the Pacific coast of the United States.

Batts or pads

Many manufacturers make batts or pads of their fibrous insulating material of such a size that they can be handled readily and will fit into standard stud spaces. (See Fig. 5–1.) The materials for this purpose are usually rock wool, glass wool, lead slag wool, or cotton. Generally they are fastened by an adhesive on one side to a sheet of asphalted paper, which acts as a vapor seal. Some manufacturers, however, place the insulating material between two sheets of paper, one of which may be a vapor seal. The vapor seal should be installed on the warm side of the insulation. These batts are supplied in various thicknesses ranging from 1 in. to 4 in.

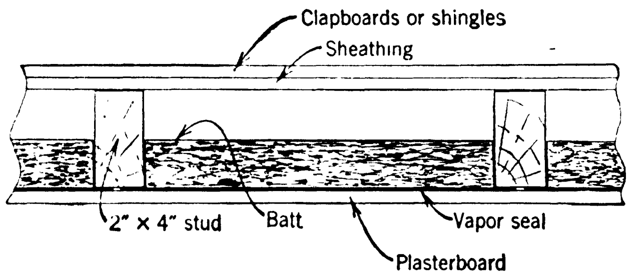


Fig. 5–1. Batt insulation in frame wall.

Blanket or flexible insulation

This type of insulation is made by enclosing the fibrous material between sheets of paper, cloth, or wire mesh. The blankets are usually supplied in rolls of various widths and thicknesses. They are frequently applied for house insulation in frame construction as shown in Fig. 5–2. It is better from an insulation standpoint to have an air space on each side of the blanket although the labor cost may be somewhat greater. For solid-brick, stone, or cement-block walls, furring strips are required if blanket insulation is to be applied.

For house insulation, blanket types of the following materials are available.

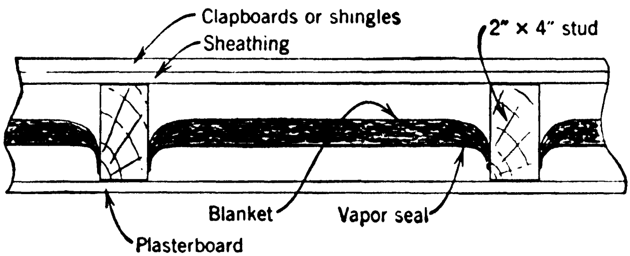


FIG. 5–2. Blanket insulation in frame wall.

Mineral wools including rock wool, glass wool, and slag wool.
Wood wool between heavily asphalted papers. (Balsam Wool.)
Cotton between sheets of paper, sometimes reflective.
Eelgrass stitched between paper or cloth. (Cabot's Quilt.)
Macerated paper between paper sheets. (Jiffy Blanket.)
Thin crepe paper stitched together. (Kimsul.) This material comes in compressed form and when applied can be stretched to several times its compressed length with no appreciable change in thickness.
Hair felt between paper or cloth.
Aluminum foil in blanket form. (See Chapter Six.)

For steam temperatures, blanket insulation of the various mineral wools and asbestos fibers is available. For the insulation of irregular surfaces, such as turbines, blanket insulation is very adaptable.

It is very important from a fire-prevention standpoint that combustible materials in blanket insulation are treated so that they are permanently fire-retardant or slow-burning.

Rigid-board or slab insulation

Rigid-board insulation for dwellings consists usually of wood or sugar-cane fibers, processed with an adhesive making sheets

4 ft by 8 ft and approximately ½ in. thick. These sheets have some structural strength. Plasterboard coated on one side with aluminum foil gives about the same insulating value in a wall, provided that an air space of ¾ in. or more is adjacent to the foil side. If ½-in.-thick fiberboard replaces 1 in. of wood sheathing there would be little gain in insulating value of a wall.

Slab insulation consisting of wood shavings or excelsior cemented so as to produce a rigid board is available for insulation. This material has sufficient structural strength to be employed as wall or roof slabs between suitable supports.

For refrigeration and other low-temperature applications, a number of slab or block forms of insulation are available.

Corkboard slabs are made from the bark of the cork oak by crushing the bark to ¼-in. to ¾-in. mesh, placing it in molds, compressing it to suitable density, and then baking it in an oven. The natural gums in the cork bark serve as the binding material and so no additional binder is necessary. Attempts have been made to grow cork oaks in the United States but the great majority of commercial cork bark comes from Spain, Portugal, and Morocco, where the bark is stripped from the trees at intervals and generally shipped elsewhere for manufacture into corkboard.

Rock Cork (a trade name) consists of rock wool, wood pulp, and an asphaltic binder. It contains no cork. It is marketed usually in the same type of slabs as corkboard.

Balsa wood is a very lightweight wood of variable density that is found in the tropical sections of South America. The low-density varieties are sawed into boards and can be utilized as insulating material, particularly where more structural strength is needed than usual. Blocks of this wood were coated with asphalt and used as mine floats in the North Sea during World War I. When these floats were removed after several years immersion in salt water, they appeared to be in excellent condition.

Rubatex is the trade name for a board insulation for low-temperature application, consisting of synthetic rubber with a large number of sealed pores.

Foamglas is the trade name for a porous glass block insulation with sealed pores. It is suitable for the exterior wall and the floor of low-temperature rooms on account of its structural strength and the fact that the block, itself, is impervious to moisture. Figure 5–3 illustrates a combination of Foamglas and corkboard which frequently serves to insulate a low-temperature room.

Glass blocks are hollow translucent blocks of glass. In some types the hollow space is sealed and a partial vacuum is produced by the cooling of the air as the block cools from a high temperature during the process of manufacture. This partial vacuum has no effect on the insulating value of the block because the vacuum is not sufficiently high to change the conductivity of the enclosed air. (See Chapter Four.)

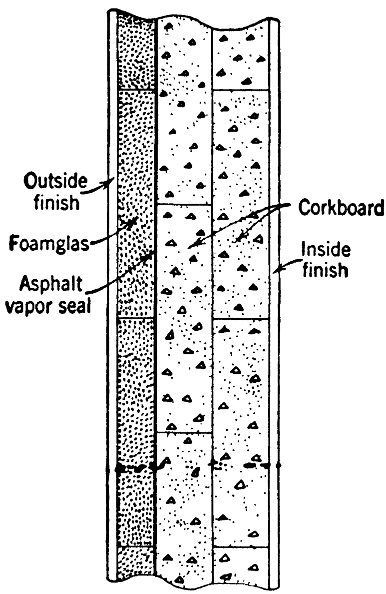


Fig. 5–3. Wall of refrigerated room.

For steam temperatures, block insulation is supplied from various types of materials.

Eighty-five per cent magnesia is made from the light carbonate of magnesia [expressed variously as $5MgCO_3Mg(OH)_23H_2O$; $5MgCO_32Mg(OH)_27H_2O$; $4MgCO_3Mg(OH)_2$] with approximately 15% asbestos fiber added to give strength. This material after being molded and baked can be employed at temperatures up to 500°F.

Corrugated asbestos paper, assembled so that there are many small air spaces, is made into block form for insulating hot water and low-pressure steam vessels.

For temperatures up to 1000°F and in some instances up to 2000°F, block insulation is available. One type consists of diatomaceous earth and asbestos with a binding material, which is combined, molded into block form (frequently 6 in. by 18 in. by 2 in.), and then dried and baked. Blocks of glass wool, rock wool, and slag wool, with suitable binding material, are also frequently employed.

Pipe covering

Sectional pipe covering for various standard-size pipes is made from the following materials.

Cork
Wool felt
Corrugated asbestos paper
85% Magnesia
Diatomaceous earth
Glass wool
Foamglas
Hair felt
Foam rubber—for small pipes or tubing in domestic refrigeration

It is important to remember that a small-diameter pipe and a large-diameter pipe with the same thickness of insulation do not have the same rate of heat loss per square foot of pipe surface, all other conditions remaining the same. (See Eq. 2–4.)

Insulating firebrick

Norton (71) states that insulating firebrick are generally made by mixing an organic material with the refractory before molding, and the organic material is subsequently burned during firing, leaving voids. The organic material is usually ground wood or cork although many other materials are also suitable. The molded brick is dried, fired, and then cut to a standard size of 2½ in. by 4½ in. by 9 in.

Some insulating brick are made by producing bubbles in the refractory mix and adding some stabilizing device, such as gypsum, to set the bubbles. This process, at present, is employed for insulating brick having a low-temperature limit, such as 2000°F.

Insulating firebrick for higher temperatures are made from fireclay, kaolin, silica, etc. Some of these materials will withstand temperatures of 3000°F on the hot side.

In general, the insulating value decreases with the ability to withstand higher temperatures. The following data, illustrating this point, are taken from literature supplied by The Babcock & Wilcox Company.

Maximum Temperature	Bulk Density	K_{500}	K_{1000}	K_{1500}	K_{2000}
2000°F	30 lb, ft^{-3}	1.47	1.91	2.36	
2200	34	1.61	2.01	2.43	
2600	34	1.66	2.33	3.05	3.84
3000	40	1.89	2.37	2.93	3.52

In furnaces, insulating firebrick are applied outside of heavy refractories. Where service conditions are not too severe, however, such as found in many heat treating furnaces, insulating firebrick are often applied without the lining of heavy refractory, thus reducing the heat capacity of the furnace walls, with consequent gain in the time required to bring the furnace up to the specified temperature.

Classification of insulating materials

The accompanying list includes common insulating materials and their various classifications. Important data, such as K value, are given in the appendix.

Refrigeration

Loose fill	Shredded redwood bark Silica aerogel Granulated cork Kapoc
Blanket	Hair felt Wool felt
Rigid board	Corkboard Rock Cork Glass-wool board Sugar-cane fiberboard Synthetic rubber board (Rubatex) Foamglas
Reflective	*See Chapter Six*

House Insulation

Loose fill	Rock wool Slag wool Glass wool Gypsum plaster crushed Vermiculite Shredded redwood bark
Blanket	Rock wool Glass wool Slag wool Cotton Wood fiber (Balsam Wool) Eelgrass (Cabot's Quilt) Kimsul
Batts	Rock wool Glass wool Slag wool
Rigid board	Wood fiberboard Sugar-cane fiberboard Corkboard
Reflective	*See Chapter Six*
Special	Ground paper and silicate of soda (Spray-o-flake)

Moderate Temperatures (212°F–1000°F)

Loose fill	Diatomaceous earth Asbestos fiber Silica aerogel (Santocel)
Blanket	Rock wool Glass wool Asbestos fiber Slag wool Kaolin wool
Block	85% Magnesia Diatomaceous earth and asbestos Corrugated asbestos paper Mineral-wool blocks Sponge felt
Reflective	*See Chapter Six*

High Temperature (Above 1000°F)

Diatomaceous earth
Mineral wool
Insulating firebrick
Crushed insulating firebrick
Kaolin wool

CHAPTER SIX

Reflective Insulation

The effectiveness of a bright metallic surface for retarding heat transfer was well known to physicists early in the nineteenth century. The calorimetric experiments of Rumford, Joule, and other early investigators were usually conducted in polished metal containers so that the rate of heat transfer could be diminished between the calorimeter and its surroundings. Peclet (75) in 1878 recognized the excellent insulating value of multiple sheets of tin separated by air spaces. Dewar in 1892 invented the Dewar flask, which utilized the silvered surfaces of the glass that made the double-walled vessel in order to minimize the transfer of heat by radiation between the inner and the outer wall. Conduction and convection were practically eliminated by maintaining a high vacuum in the space. This invention was of tremendous importance in connection with the liquefaction of gases, and it is, of course, the basis of our common Thermos bottle.

Before 1925 there was practically no commercial utilization of reflective insulation except that of the Thermos bottle, but in that year Schmidt and Dykerhoff filed patents in Germany for the use of aluminum foil, less than 0.0005 in. thick, in crumpled form for commercial insulation. These two men are primarily responsible for an important advance in the art of insulation. Since then, a number of other investigators have discovered many applications of the principle of reflective insulation, and during the period between 1925 and 1950 millions of square feet of reflective insulation have been applied to cold storage walls, ovens, furnaces, houses, ships, etc.

Since the commercial application of reflective insulation is relatively new and the theory connected with its functioning is based on principles much more complex than those of the usual type of insulation, it seems necessary to describe it in considerable detail.

A large amount of experimental work with reflective insulation is still required to make our knowledge of its performance reasonably complete, but the following points have been fairly well established and the reader should become thoroughly familiar with them before attempting to understand reflective insulation.

1. *If the rate of heat flow is to be retarded, reflective insulation always involves the utilization of an air space adjoining the reflective surface.* If there is no air space, heat will be transmitted by conduction from the reflective surface to the adjoining wall, and the benefit of the reflective surface will be nil. If a sheet of paper, coated with aluminum foil on one side, is cemented to a wall with the foil facing the wall, no appreciable gain in insulating value will result except possibly that due to less infiltration of air and water vapor.

2. *If convection occurs in an air space faced with a reflective surface, the insulating value should be reported as thermal conductance,* C, *rather than a coefficient of thermal conductivity,* K. The rate of heat transfer is inversely proportional to the thickness for pure conduction. Thus, a 4-in. thickness would have one half the rate of heat flow that would occur in a 2-in. thickness of the same material, if all other conditions remained the same. However, in a vertical air space faced with reflective surfaces, it is very difficult to detect any difference in the rate of heat flow between a 2-in.- and a 4-in.-wide space. Much confusion has been caused by assigning K values to reflective insulation, although in the absence of convection and under certain conditions an apparent K value may be used.

3. *If an air space is faced on one side with a reflective surface, it makes no appreciable difference on which side the reflective surface is placed as far as the rate of heat transmission is concerned.* It is not surprising that the layman often thinks that the reflective surface must face the warmer side of an air space in order to reflect the radiant heat. Reflective insulation might just as well have been called low-emissivity insulation, because, as stated in Chapter Two, a good reflector of radiation must also be a poor emitter of radiation. If a surface reflects 95%

of the incident radiation (a black body would reflect none), it will emit only 5% as much radiation as a black body under the same conditions. The rate of heat flow by radiation across an air space faced on one side with a reflective surface is directly proportional to the effective emissivity of the two surfaces bounding the space, and the effective emissivity will have the same value regardless of the direction of heat flow. (See Eqs. 2–14 and 2–15.) It should be kept in mind that, although direction of heat flow does not affect radiation, it may change the rate of heat flow by convection considerably.

4. *The width of an air space often is important with reference to the conductance of reflective insulation.* If the temperature drop across an air space is sufficient to produce convection currents, the conductance value for a vertical space is essentially the same for widths varying from ¾ in. to 16 in. As the width of the air space is reduced to less than ¾ in., the conductance increases rather rapidly, becoming infinite at zero width. The above statements apply to temperature drops up to 50°F. Convection in an air space can be retarded and eventually stopped by making the width or the temperature drop sufficiently small. When convection becomes a negligible factor, heat is transferred across the space by radiation, which is independent of the width, and by conduction of the air, which increases as the width diminishes. If reflective surfaces bound the air space, radiation becomes small, the heat is transferred across the space primarily by conduction, and a K value can be assigned, provided that there is no convection.

Figure 6–1 illustrates how the conductance value, C, of a vertical air space varies with the width of space. All the values shown were experimentally determined. It will be noted that the conductance under the conditions stated changes little as the width varies from ¾ in. to 16 in. but the value for the ½-in. space is nearly twice that for the widths greater than ¾ in. If the temperature drop across the space were reduced so that convection essentially ceased, the values for the widths greater than ¾ in. would be considerably reduced.

5. *The orientation of an air space usually affects the conductance significantly.* Wilkes and Peterson (115*a*) made many tests on a 4-in. air space, 8 ft high, corresponding to the air

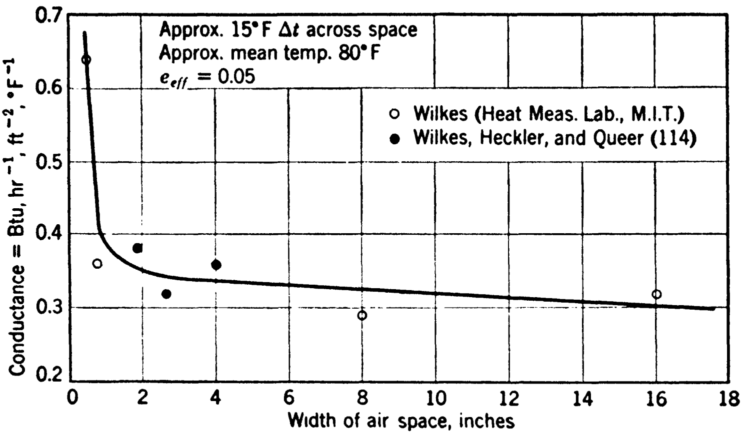


FIG. 6–1. Conductance of vertical air spaces vs. width. Reflective insulation.

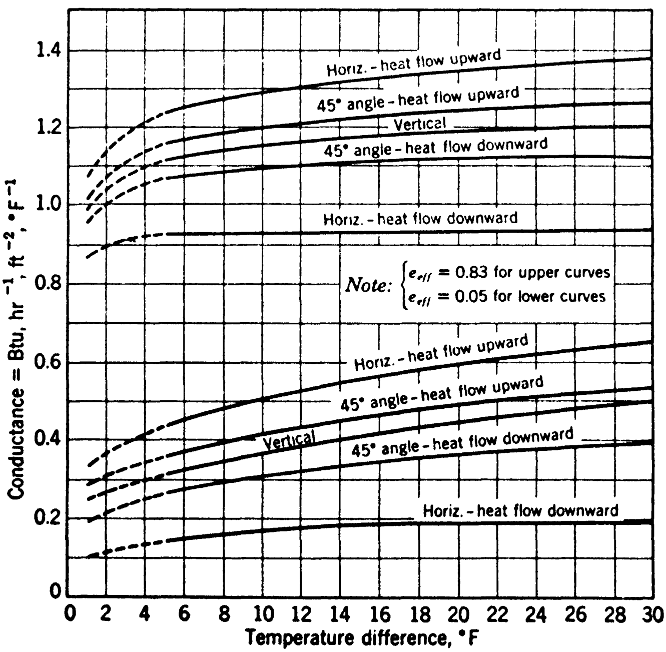


FIG. 6–2. Conductance of air spaces vs. Δt. Wilkes and Peterson (115a).

space between the studs in frame construction. Figure 6–2 shows the effect of orientation and direction of heat flow on the conductance of such a space. With the effective emissivity equal to 0.05 (aluminum foil on one side) and a temperature drop of 30°F, the rate of heat transmission across an air space in a horizontal position with heat flowing upward is more than three times what it would be with the heat flow downward under

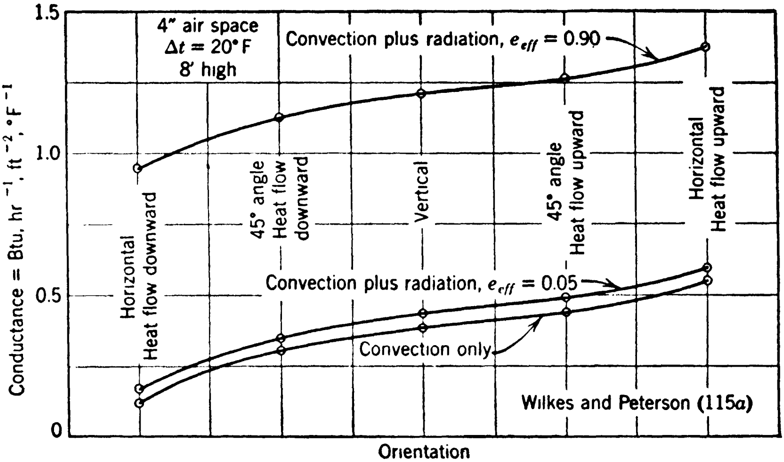


FIG. 6–3. Conductance of air spaces vs. orientation.

similar conditions. With ordinary surfaces, such as wood, plaster, paper, having an emissivity of 0.90, the effect of orientation is not so great but that it still should be taken into account in calculating the rate of heat transfer through ceilings, floors, and walls. Figure 6–3 shows the variation of the conductance of an air space with orientation, as well as the portion played by convection and radiation for high- and low-emissivity surfaces.

6. *The height of a vertical air space affects the conductance if convection is present.* Although the information on this subject is meager, some experimental evidence is available that indicates the influence of height on the conductance. Griffiths and Davis (35) found little effect by changing the height from 4 ft 1.6 in. to 2 ft 0.8 in. with air spaces ½ in. and 1 in. wide. Figure 6–4, based on data from Wilkes and Peterson (115*a*), Griffiths

and Davis (35), and Steenkamp (93), indicates the change of conductance due to height with air spaces 2 to 4 in. wide and height varying from 6 in. to 8 ft. It would appear from these data that reducing the height of an air space actually increases the rate of heat transmission. This result is probably due to the decreased resistance to convection currents as the height

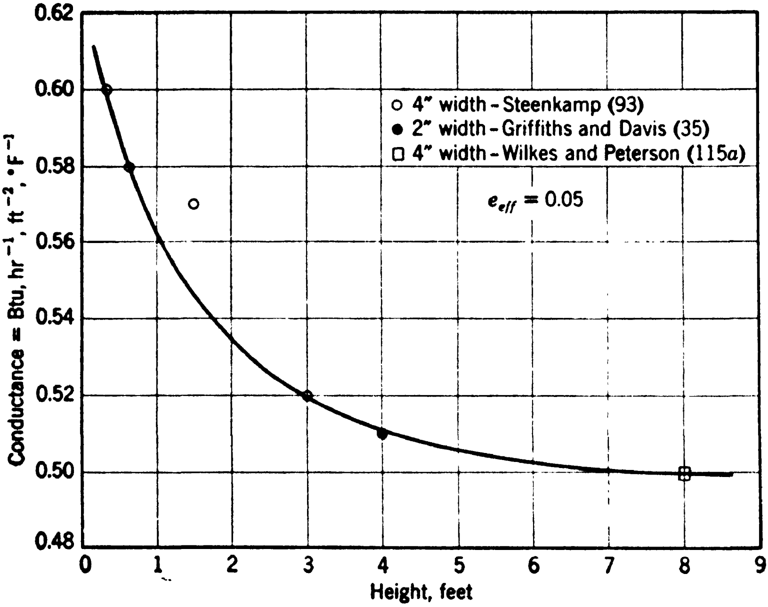


FIG. 6–4. Conductance of air spaces vs. height. Reflective insulation.

becomes less. In an 8-ft-high space the air probably travels up nearly the entire distance and then returns on the cooler side. It is probably subject to more interference with the upward and downward currents than in a 4-ft height. Figure 6–4 indicates that conductance is 20% greater for a 4-in. height than for an 8-ft height. In other words, subdividing an 8-ft-high air space with horizontal strips, thus making many 4-in.-high spaces, would actually increase the rate of heat transfer across the total area.

7. *The emissivity of the surfaces bounding an air space affects the heat transfer by radiation but it does not affect the transfer by conduction and (or) convection.* Figure 6–5 indicates the

variation of the conductance across a vertical air space with change of effective emissivity. It will be noted that a shift in the effective emissivity from 0.05 to 0.20 increases the conductance value by about 32%. The transmittance of a frame wall, insulated with a single sheet of reflective material in the center of the stud space, would be approximately 16% greater for this

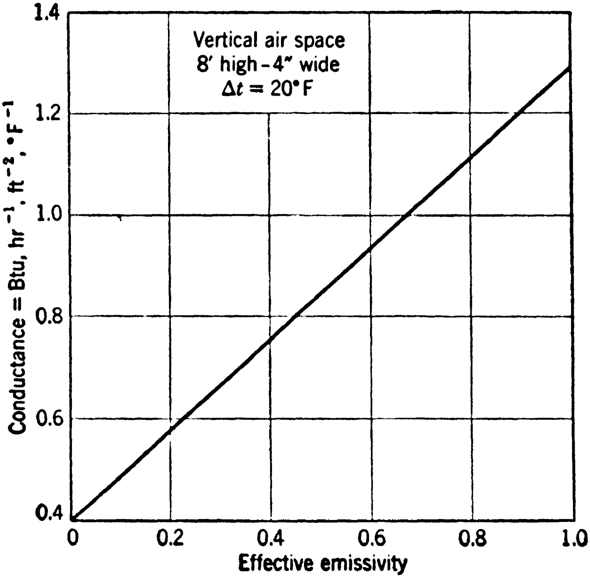


FIG. 6-5. Conductance of air spaces vs. effective emissivity.

same change in effective emissivity. In an air space in the horizontal position with heat flow downward, an increase in emissivity would make even a greater change in the conductance because radiation plays a more important part in the transfer of heat in this position than in the vertical one.

8. *Temperature difference and (or) mean temperature affect the conductance values of air spaces faced with reflective insulation.* If the temperature difference is sufficient to produce convection in an air space faced with a reflective surface, the conductance will vary roughly with the $\frac{5}{4}$ power of the temperature difference and it will be essentially independent of variation in mean temperature. If the convection is not appreciable because of orientation, slight temperature difference, or

narrow space, the conductance will vary with the mean temperature and be essentially independent of the temperature difference. Wilkes (109) in Fig. 6–6 shows this effect for lead-tin alloy-coated steel sheets spaced ½ in. apart. For small Δt's, convection is apparently negligible and the conductance varies with the mean temperature primarily, regardless of whether the heat flow is upward or downward with the air space in a hori-

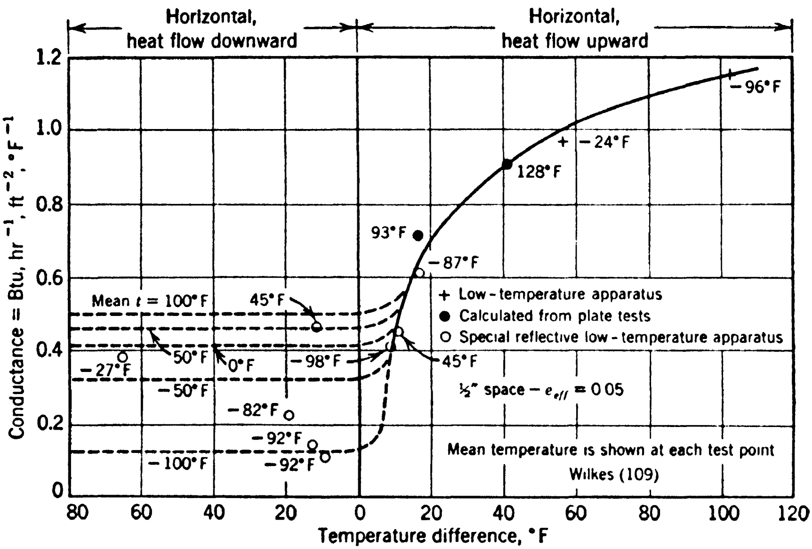


FIG. 6–6. Conductance of ½ in. air spaces vs. Δt. Reflective insulation. Heat flow upward and heat flow downward.

zontal position. When the Δt is raised above 7.5°F and the heat flow is upward, the conductance begins to increase rapidly with an increase in Δt, and it then becomes independent of the mean temperature.

Figure 6–7 shows the relation between the conductance of a ½-in. air space faced with reflective insulation and the mean temperature. For a horizontal position with heat flow downward this curve applies for any Δt, but for other positions the Δt across the space must be less than 7.5°F. If the Δt is increased much above 7.5°F, convection will probably begin and the conductance value will increase rapidly with the Δt and be independent of the mean temperature.

Figure 6–8 gives the conductance values for various numbers of air spaces, each faced with reflective material giving an effec-

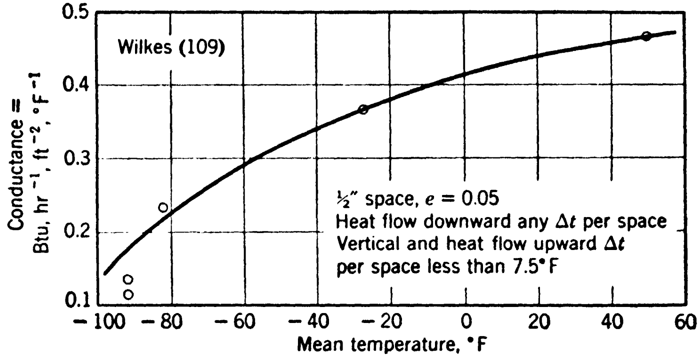


FIG. 6–7. Conductance of ½ in. air space vs. mean temperature. Reflective insulation.

tive emissivity of about 0.05. These curves apply only to the horizontal position with heat flow downward.

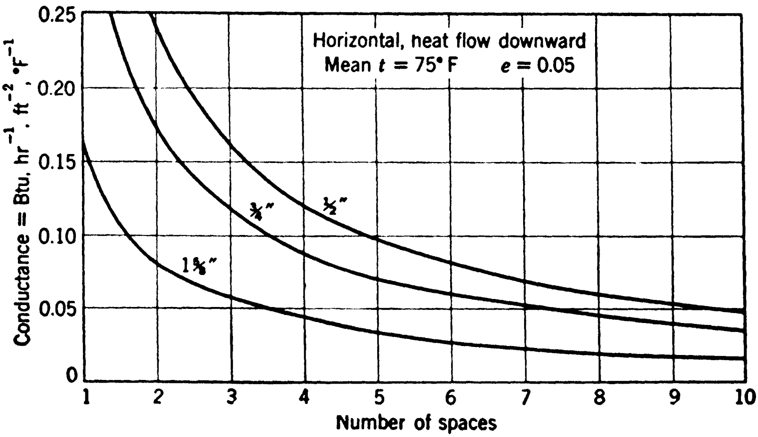


FIG. 6–8. Conductance vs. number of air spaces. Reflective insulation.

The conductance values for various numbers of air spaces in the vertical position are shown in Fig. 6–9 and, similarly, in the horizontal position with heat flow upward in Fig. 6–10. The values given in Figs. 6–8, 6–9, and 6–10 were determined in the Heat Measurements Laboratory, M.I.T., and are presented through the courtesy of C. T. Hogan & Company, New York City.

It should be noted that when the air space is in the horizontal position with heat flow upward, the width of the space has little effect on the conductance value with a Δt across the space of

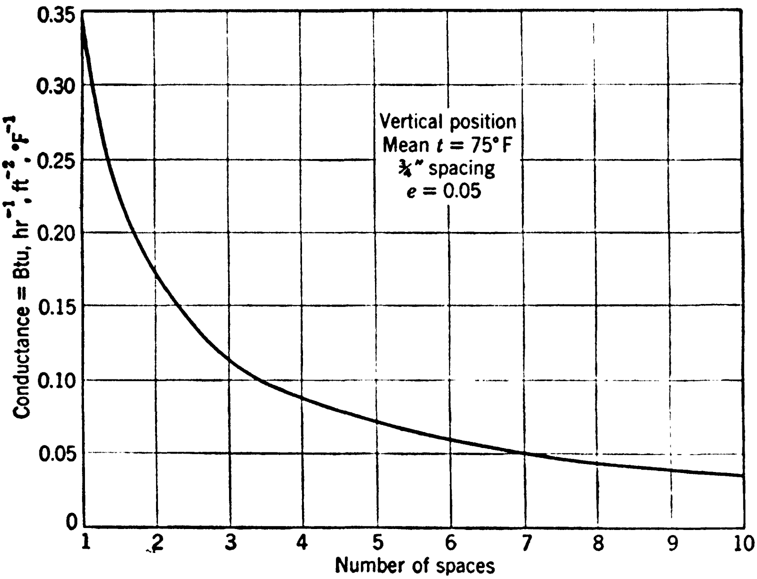


FIG. 6–9. Conductance vs. number of spaces. Reflective insulation.

7.5°F. If the Δt across the space were decreased, a point would soon be reached where convection would diminish and then the conductance value would undoubtedly decrease with an increase of the width of the space.

In Fig. 6–9, the ¾-in. and 1⅝-in. spaces have nearly identical conductance values in the vertical position, and, furthermore, the width of the space could be increased to 4 in. or more with little change in the conductance. If the space were made less than ¾ in., the conductance value would gradually increase.

For the horizontal position, with heat flow downward (see Fig. 6–11), the width of space becomes an important factor in the conductance value because convection is minimized under these circumstances. Up to a width of 8 in., the conductance decreases with width of space, the decrease being very rapid between ½-in. and 1-in. widths. Thus it is often possible to

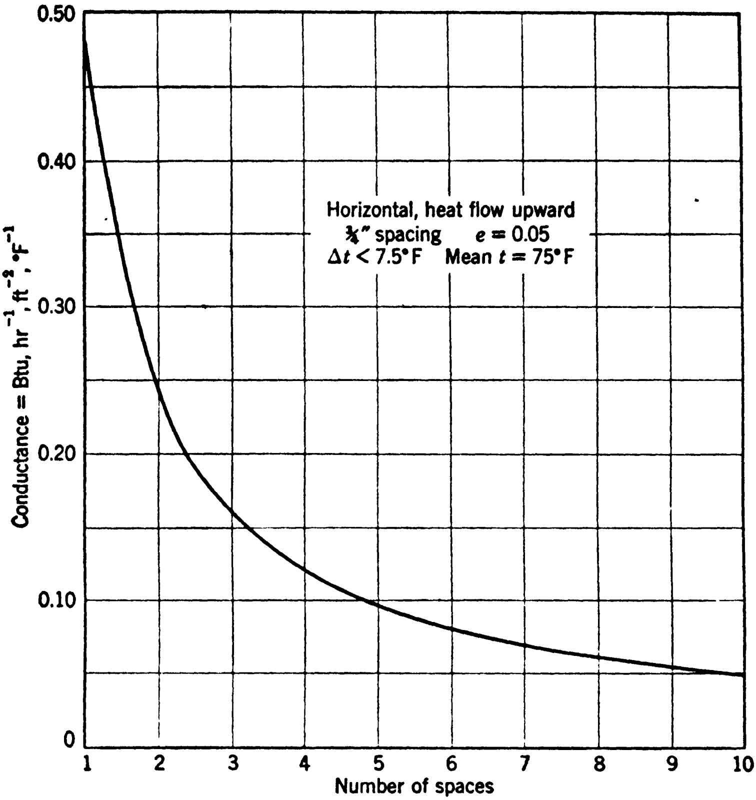


FIG. 6–10. Conductance vs. number of spaces. Reflective insulation.

improve insulation by applying fewer sheets of reflective material when wide air spaces are available. This statement applies only when the reflective surfaces are horizontal and heat flow is downward. Figure 6–12 shows the approximate conductance for various numbers of spaces with widths ranging from ½ in. to 2 in. Since data are meager on this point, the exact values must be employed with caution, but the effect of changing the width of space should be approximately that shown in the plot.

If the desired conductance for the insulation in a cold storage roof is 0.05 Btu, hr^{-1}, ft^{-2}, $°F^{-1}$, one can readily determine

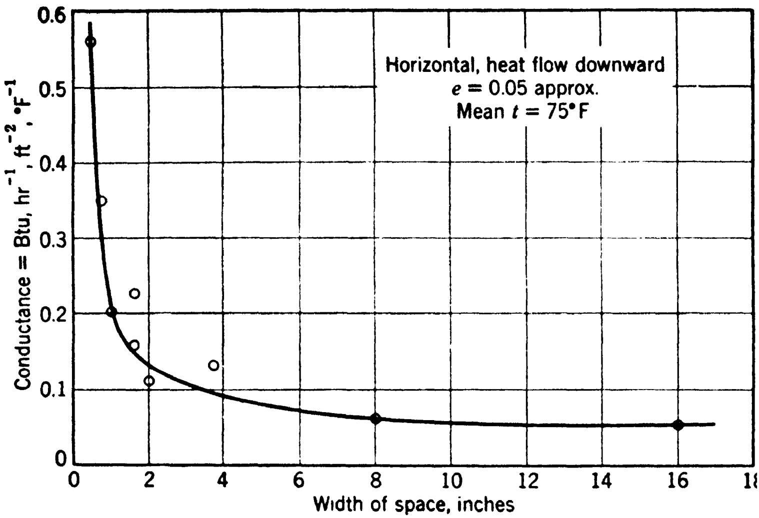


FIG. 6–11. Conductance vs. width of space. Reflective insulation.

from Fig. 6–12 the number of air spaces needed to obtain this value for various widths as shown in the accompanying table.

No. of Spaces	Width of Space	No. of Sheets	Total Thickness
3	2 in.	2	6 in.
4	1	3	4
7	¾	6	5¼
11	½	10	5½

In other words, for this particular condition, two sheets of reflective insulation, installed so as to give three 2-in. air spaces, would give essentially the same insulating value as ten sheets providing eleven ½-in. spaces. Furthermore, the total wall thickness would be nearly the same.

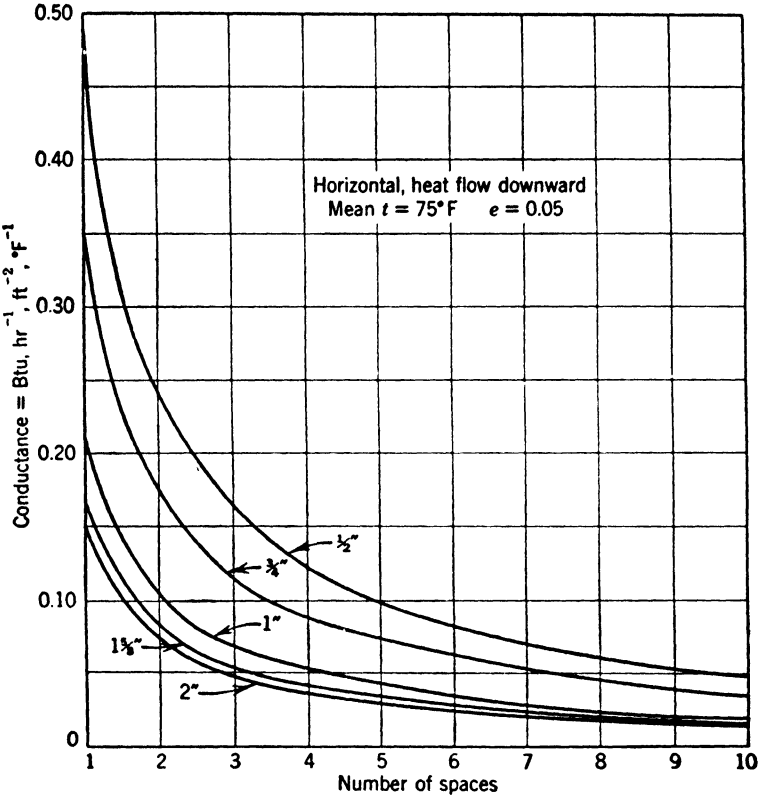


FIG. 6–12. Conductance vs. number of spaces. Reflective insulation. Various widths.

TYPES OF REFLECTIVE INSULATION

One of the simplest ways to convert an ordinary air space into effective insulation is to subdivide the original space into two or more air spaces with one or more sheets of material with reflective surfaces. In general, the spaces should be at least 0.5 in. wide, and it would be better to have them 0.75 in. or greater. The following list, although not complete, will illustrate some of the more common materials and methods employed by the reflective-insulation manufacturers.

Aluminum foil (Alfol), Reflectal Corporation

Alfol is supplied in rolls of various widths up to 24 in. and of thicknesses ranging from 0.0003 in. to 0.0005 in. Before application, this type of foil is usually crumpled by hand, to some extent, for two reasons. Crumpling eliminates any rattle and it also allows for considerable "come and go" of the supporting members. If the foil were stretched tightly across studs it might be torn apart by slight movement of the studs. The crumpled sheet is smoothed out and is then ready to be attached to the edges of an air space with cardboard strips or battens with the aid of a stapling hammer. The foil is very frail and it should be applied by an experienced person for best results. Once properly applied and protected, it is sufficiently strong for building purposes, as has been demonstrated.

Aluminum foil on paper (Metallation), Reynolds Metals Company

This material consists of very thin foil (0.0003 in. thick) cemented to kraft paper, either on one side or both, as desired. It can be applied to the side walls of an air space in much the same way as the plain aluminum foil, but since it has much more mechanical strength the battens may be omitted. Any carpenter or skilled workman can attach it to the studs in a house wall by means of a stapling hammer. From the viewpoint of insulation value, the two sheets of foil plus the paper give no better insulation than a single sheet of foil.

Figure 6–13 shows some ways of applying this material in the air space of a frame wall. Part *a* shows Metallation stapled to the 2-in. face of the studs allowing sufficient material between the studs so that it will bow, making two air spaces with a reflective surface facing each one. This is a relatively cheap method of application but it is not so effective as the method shown in *b*, where the Metallation is stapled to the 4-in. side of the studs. This arrangement divides the original space into two spaces of equal width, if the work is carefully done. The labor cost is somewhat greater but it is warranted by the better result. Part *c* shows a way of applying two sheets of Metallation in an air space, thus making three air spaces. One sheet is coated with foil on both sides and one with foil on one side. The small

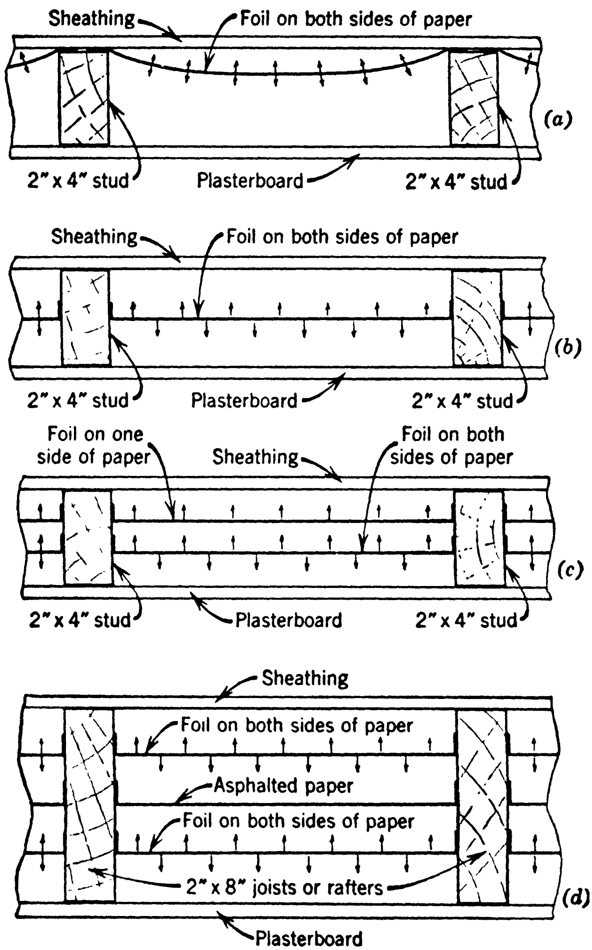


Fig. 6–13. Methods of application of spaced reflective insulation.

arrows indicate a reflective surface in Fig. 6–13. Part *d* illustrates a means of achieving greater insulation at relatively little additional cost. A piece of ordinary asphalted paper is placed between two sheets of Metallation coated on both sides, thus making four air spaces with a reflective surface facing each one. This arrangement gives very effective insulation.

Papers coated with reflective surfaces

Paper may be coated with aluminum powder and, if installed in the way shown in Fig. 6–13, may serve as insulation. Its emissivity is considerably higher than that of aluminum foil, ranging from 0.15 to 0.35. As shown in Fig. 6–5, the conductance value is somewhat higher because of the increased emissivity. The coated papers are usually reinforced with a mesh of string or thread, giving them considerable strength. Sisalation, manufactured by the American Reinforced Paper Company, and Silvercote, manufactured by the Silvercote Products, Inc., are excellent examples of this type of insulation. In general, the cost is much less than for aluminum foil.

Special blanket types of reflective insulation

Figure 6–14*b, c* (arrows indicate reflective surfaces) shows two types of reflective blankets, known as Alfol, manufactured by the Reflectal Corporation. They are constructed in such a manner that, when opened, they can be stapled directly to the studs, making either three or four air spaces faced at least on one side with aluminum foil. The labor cost of installation is much less than for placing separate sheets of foil but the results are not quite so effective.

Infra Insulation, made by the Infra Insulation, Inc., is similar to the Alfol blankets, but it comes in long, narrow strips. These strips can be cut to the proper height for the air space and then opened to stretch across the air space. After attaching them to the studs, one has practically three air spaces, each faced at least on one side by aluminum foil. This type of blanket is sometimes referred to as "accordion" insulation on account of its construction, shown in Fig. 6–14*a*.

Another type of reflective blanket consists of a layer of cotton or another insulating material, approximately 1 in. thick, with aluminum foil on each side. It is attached between the studs so as to make two air spaces faced on one side with reflective insulation plus the cotton blanket.

Plasterboards coated with aluminum foil

Plasterboards are not very effective insulators, having a conductance for the ½-in.-thick boards of 2.60 Btu. hr^{-1}, ft^{-2}.

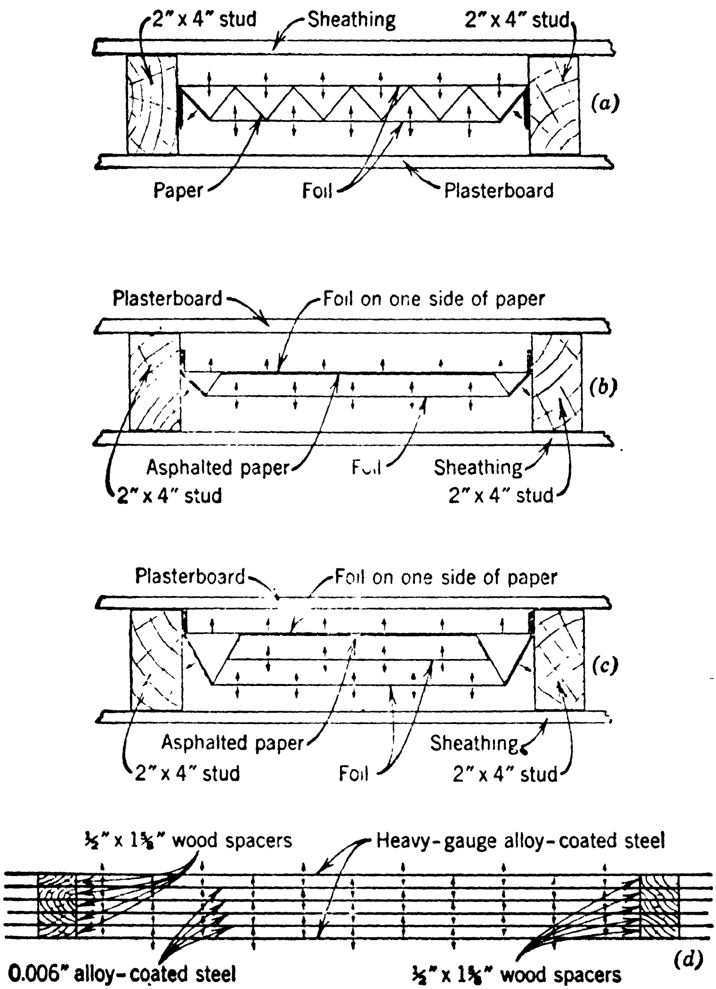


FIG. 6–14. Types of spaced reflective insulation.

$°F^{-1}$. Their effectiveness as insulators can be greatly enhanced by coating with aluminum foil the side that faces the air space between the studs. Plasterboard treated in this way, in conjunction with the air space, offers more resistance to heat flow than ½ in. of insulating board in conjunction with an air space faced with ordinary surfaces, as shown in the accompanying table.

	Resistance		Resistance		Resistance
½-in. plasterboard	0.38	½-in. plasterboard	0.38	½-in. insulating board	1.50
Ordinary air space	0.83	Reflective air space	2.17	Ordinary air space	0.83
Total resistance	1.21		2.55		2.33
Conductance	0.83		0.39		0.43

The conductance in this table is expressed in Btu, hr^{-1}, ft^{-2}, $°F^{-1}$. Incidentally, the aluminum foil also serves as an excellent vapor seal.

There are many irregular places in frame walls where it is either difficult or impossible to apply reflective insulation. It is common practice to fill them with loose, fibrous insulation. Considerable ingenuity is often required to apply reflective insulation properly.

Low-temperature reflective insulation

For refrigerated rooms and cold-storage warehouses, the reflective sheets are usually of greater thickness than those for house insulation.

Lead-tin Alloy-coated Steel Sheets (Ferro-Therm), American Flange and Manufacturing Company. Ferro-Therm is usually supplied 24 in. by 32 in. by 0.006 in. thick. For the outside and inside wall surfaces a thicker-gauge sheet can be utilized. The sheets are attached to ½-in.-thick furring strips by stapling. Walls of any desired thickness may be constructed by adjusting the number of furring strips and sheets. Figure 6–14*d* shows typical construction of this kind.

Aluminum Sheets (Alumiseal), C. T. Hogan and Company, Inc. Alumiseal comes in rolls or coils of 0.006-in. thickness and various widths. It is applied in much the same way as Ferro-Therm except that thicker furring strips are employed, giving greater width of air space. For walls and floors of refrigerated rooms, the furring strips are nominally 1 in. by 2 in., making an air space actually ¾ in. wide. For the roof, the spacing is usually 1⅝ in., thus saving material and labor at no loss in insulating value because the air space is horizontal with heat flow downward (see Fig. 6–8). Since this material comes in coils of considerable linear footage, the number of joints are

FIG. 6–15. Installation of reflective insulation (Alumiseal) on cold-storage roof. Courtesy of C. T. Hogan and Company, Inc.

reduced to a minimum. A special lead tape with adhesive on one side seals all the joints on the warm side of the insulation so as to prevent the infiltration of water vapor with consequent condensation within the wall. Heavier aluminum sheets are used for exposed wall surfaces. Figure 6–15 is a photograph of the installation of this type of insulation on a cold-storage roof, and Fig. 6–16 shows the measured temperature gradient through a cold-storage wall.

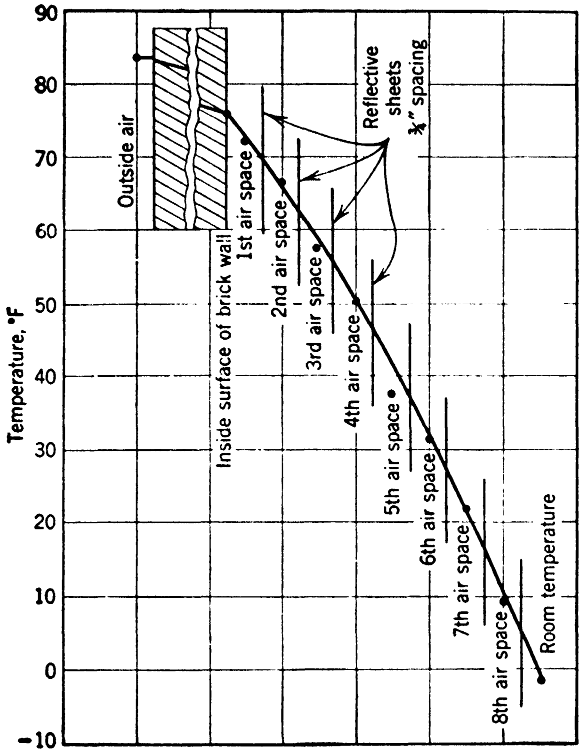


FIG. 6-16. Temperature gradient in wall of cold-storage plant. Reflective insulation. Courtesy of C. T. Hogan and Company, Inc.

Crumpled aluminum foil (Alfol), Reflectal Corporation

This type of insulation is sold in rolls of various widths and 0.0003-in. thickness, which, when unrolled at the job, are crumpled by hand and then roughly straightened before application. Layers of this crumpled foil are placed one over the other and slightly compressed so that there are about 2½ layers per inch of thickness. An allowance of about 15% should be included in calculating the amount of foil for any given area on account of the crumpling. This type of insulation is suitable primarily for industrial application, such as ovens, furnaces, prefabricated panels, storage tanks, ships, and railway cars. The dome of the new 200-in. telescope at Palomar, Cal., is insulated with crumpled aluminum foil in panel form. Crumpled aluminum foil has been employed at temperatures up to 900°F without serious loss of emissivity.

Reflective insulation in high vacuum

In laboratory work where high vacuum (1 micron or less) is maintained, extremely effective insulation can be made with one or more sheets of reflective material. It gives the effect produced by a Dewar flask or Thermos bottle multiplied by the number of spaces provided. A vacuum bottle is not very satisfactory at temperatures above 300°F, because there is only one space between the inner and outer walls and there is sufficient radiation at high temperatures to make the heat transfer appreciable. By placing multiple reflective sheets in the vacuum chamber, one can get very efficient insulation at temperatures as high as 3000°F. The reflective material must be able to withstand any temperature to which it is exposed. The limiting temperatures for some foils are given in the accompanying list.

Platinum foil	3000°F
Nickel foil	2500
Gold foil	1700
Aluminum foil	1000

In a high vacuum, there should be no serious trouble from oxidation of the above metals.

Reflective insulation to prevent condensation on cold pipes

A layer or two of loose, crumpled foil wrapped around cold-water pipes will usually prevent condensation of water vapor on the pipe and also the subsequent dripping of water. Cold-water pipes in cellars where condensation is troublesome can be readily protected in this way at very little expense and labor. It is important that the foil be wrapped loosely in order to achieve the best results.

Reflective insulation behind radiators

Aluminum foil is available for placing behind radiators in homes to increase the rate of heat transfer from the radiators to the air in the room. The aluminum surface reflects the radiant heat that would otherwise be absorbed by the wall.

A reflective surface behind a stove will aid greatly in preventing the woodwork from becoming overheated with consequent risk of fire.

PERMANENCE OF REFLECTIVE INSULATION

Before the installing of reflective insulation, the question of permanence naturally arises. Aluminum foil has been in use since 1930 and one can be assured that when it is employed under proper conditions, protected from the outside atmosphere, it will last as long as the average house without any serious change in emissivity. If dust should collect in sufficient quantity to obscure the aluminum surface, the emissivity of that surface would be much increased. This could not happen in a vertical or inclined wall, but the top layer of foil in an attic ceiling might be affected in this way. The lower surface of the top layer would still be effective, as well as the succeeding layers below. Taylor and Edwards (96) state:

> The eye may be a good judge in appraising the reflectivity of a surface for visible light but can not evaluate the infrared radiation characteristics. A piece of aluminum may have very high reflectivity for visible light and low reflectivity for infrared radiation, or it may have only fair reflectivity for light and be an excellent reflector of infrared radiation, depending on the presence or absence of surface films.

A mirror coated on the back side is an excellent example of a surface that reflects light remarkably well but is a very poor reflector of long-wave-length radiation such as is emitted from surfaces at room temperature. It would be no better as a reflector of low-temperature radiation than a black-painted surface. Glass is opaque to long infrared radiation and has a high emissivity with consequent low reflectivity for these wave lengths. However, it is quite transparent to the short-wave-length radiation of visible light, and so this radiation travels through the glass and is reflected by the silver coating on the back.

Extremely thin, transparent lacquer coats can be applied to aluminum foil without impairing the emissivity to any great extent, but heavier coats destroy the low emissivity of the surface for room-temperature radiation, although the surface will still appear very reflective for light.

Alkalies attack aluminum readily, and aluminum foil should always be protected from wet plaster.

The author has exposed aluminum foil for 30 days in an electric furnace with an oxidizing atmosphere at a temperature of 1050°F. The surface changed to a dull gray during this exposure but the emissivity increased only from 0.05 to 0.075. This change would not affect the conductance of an air space appreciably.

One would expect that papers coated with aluminum powder would also be long-lived when installed under suitable conditions.

CHAPTER SEVEN

Specific Heat of Insulating Materials

The scope of this book is not intended for unsteady-heat-flow calculations, and the reader is referred to Fishenden and Saunders (31), McAdams (60), Carslaw (19), Ingersoll and Zobel (44*a*), and others for this type of heat calculation. In order to make unsteady-heat-flow calculations, the specific heat and density of the material must be known. This chapter will describe laboratory methods of determining the specific heat of insulating materials over a wide range of temperature and show some important applications of its effect upon the time required to reach thermal equilibrium.

The flow of heat in an insulator is said to be steady when the temperatures at all points remain constant with time. Under such a condition, the specific heat and the density of a material have no effect on the rate of heat flow. When an insulated furnace or oven is being raised to some desired temperature, the temperature of points within the insulation change with time. Under these conditions of unsteady heat flow, the specific heat and density of the insulation are important factors in determining the rate of rise in temperature of points within the insulation, as well as the furnace temperature itself.

For a number of years insulating firebrick with low heat capacity as a replacement for standard firebrick in intermittently heated kilns and furnaces have shown decisively that large savings in fuel and time can be obtained. McCullough (61) gives the results of tests on two furnaces, identical except for the type of lining, which were heated empty in order to compare the fuel requirements and heating time. The furnace with an insulating firebrick lining was heated to 2000°F with only one third the quantity of gas required to heat the furnace lined with regular firebrick to the same temperature. Also, it was noted that the furnace lined with insulating firebrick reached 2000°F in approximately one third the time required for the other furnace.

Example. Let us look at another example comparing the behavior of three insulating materials with very different heat capacities but with essentially the same K value. Suppose that three large, identical ovens are insulated with (1) 4 in. of 85% magnesia block, (2) 4 in. of glass wool, and (3) 4 in. of crumpled aluminum foil. These ovens, if maintained at a constant temperature of 500°F, will lose approximately the same amount of heat per unit time, since their respective K values are equal.

If the period during which their temperatures are being raised is considered, the heat quantities involved in the three insulating materials will be substantially different. The following specific-heat and bulk-density values will be taken for the three types of insulation.

	Mean Specific Heat at 290°F	Bulk Density
85% Magnesia block	0.284 Btu, lb^{-1}, $°F^{-1}$	15 lb, ft^{-3}
Glass wool	0.234	3
Crumpled aluminum foil	0.24	0.19

The heat required per cubic foot of insulating material will be approximately as follows:

85% Magnesia block $\frac{500 - 80}{2} \times 0.284 \times 15 = 1235$ Btu, ft^{-3}

Glass wool $\frac{500 - 80}{2} \times 0.234 \times 3 = 204$ Btu, ft^{-3}

Crumpled aluminum foil $\frac{500 - 80}{2} \times 0.24 \times 0.19 = 13$ Btu, ft^{-3}

The heat capacity of the 85% magnesia is ninety-three times and glass wool fifteen times that of the aluminum foil. The heat capacity of the framework and casings would reduce these ratios of heat capacity considerably but a large difference would still remain in the amount of heat required to raise the temperature of the ovens from 80°F to 500°F.

In order to estimate the time required to heat an empty oven to some required temperature, the thermal conductivity of the insulating material must be considered in addition to the heat capacity. The thermal conductivity divided by the product of the specific heat and the density, $K/c\rho$, gives a term commonly called thermal diffusivity. The reader should be cautioned about the use of consistent units in the determination of this value. For materials of the same thickness, the heating time is inversely proportional to the diffusivity. For materials of the same diffusivity, the heating time is directly proportional to the square of the thickness. Consequently, for any material of any thickness, the controlling factor is $1/(K/c\rho) \times L^2 = c\rho L^2/K$, where L is the thickness. Since the three samples have the same thickness, the thermal diffusivity, alone, can be used for a comparison of the relative heating times. If the thermal conductivity of all three samples is taken as 0.50 Btu, hr^{-1}, ft^{-2}, in., $°\text{F}^{-1}$ (for consistent units use $K = 0.042$ Btu, hr^{-1}, ft^{-2}, ft, $°\text{F}^{-1}$), the thermal diffusivity will be:

$$\text{85\% Magnesia} \quad \frac{0.042}{0.284 \times 15} = 0.0099 \text{ ft}^2, \text{hr}^{-1}$$

$$\text{Glass wool} \quad \frac{0.042}{0.234 \times 3} = 0.060 \text{ ft}^2, \text{hr}^{-1}$$

$$\text{Aluminum foil} \quad \frac{0.042}{0.24 \times 0.19} = 0.92 \text{ ft}^2, \text{hr}^{-1}$$

The time required to heat the ovens is inversely proportional to the diffusivity, and so the 85% magnesia will take ninety-three times and the glass wool will take fifteen times as long as the crumpled foil. This calculation applies to the insulation only; the relative times would be somewhat reduced if the framework and casings were included.

Laboratory methods of determining specific heat

The determination of the specific heat of insulating materials is rather complicated because the materials are light in weight and transfer heat very slowly, thus making it difficult to secure equilibrium of temperature throughout the sample. Two methods of making such determinations will be described.

Griffiths (33) found that the ordinary method of mixtures was "unworkable due to practical difficulties and to the fact that the moisture content of the material was altered by heating to about 100°C, before dropping it into the calorimeter." Figure 7-1 is a sketch of Griffiths' specific-heat apparatus. The calorimeter is a large cylindrical vessel (4300 cc) of thin sheet aluminum, carried on a horizontal axis rotating at a uniform speed. The sample, in a state of fine subdivision, occupies about one third to one half the volume of the calorimeter. Helical fins, attached to the inside wall of the calorimeter, move the contents around, thus giving thorough mixing with consequent uni-

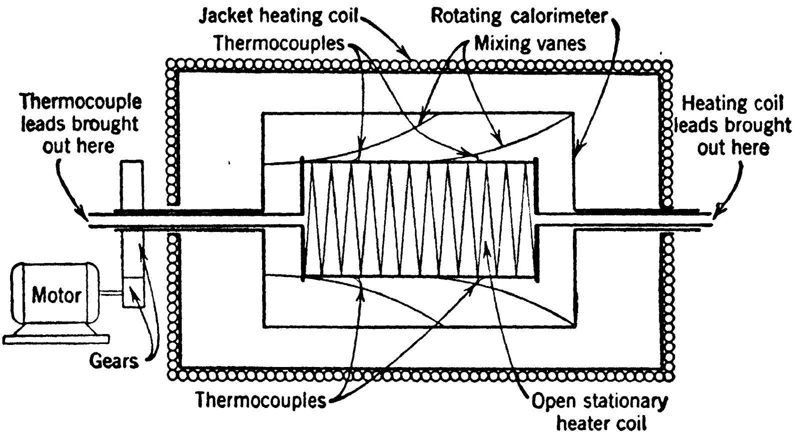


FIG. 7-1. Specific heat equipment. Griffiths (33).

formity of temperature. An electric heating coil projects into the calorimeter and is held stationary. The watt input to the coil can be measured during the time of heating. The temperature of the sample was measured at six different points but the differences in temperature between these points were negligibly small. To prevent heat loss to the surroundings the calorimeter was enclosed by an electrically heated copper drum, which could be heated at the same rate as the calorimeter, thus avoiding any correction for radiation and convection losses. The actual time that heat was supplied approximated 10 min, with a corresponding rise in temperature of the calorimeter and jacket of about 15°C.

Wilkes and Wood (117) describe a method of determining the specific heat of insulating materials at temperatures ranging from 1350°F to room temperature. The calorimeter was a cylindrical block of brass or aluminum through which a hole had been drilled, parallel to the axis as shown in Fig. 7–2. A hinged cover was held open by a rod attached to a false bottom. When the sample was dropped into the calorimeter, the false bottom dropped and the cover closed. The calorimeter was mounted inside a jacket consisting of two metal walls, electrically insulated from each other and spaced about ¾ in. apart. This space was filled with city water and the two walls connected to a 110-volt circuit with a switch so that the jacket could be heated at the same rate as the calorimeter, thus minimizing any heat transfer between the calorimeter and the jacket. Thermocouples were inserted into holes drilled in the calorimeter in order to measure the temperature rise. A differential couple with one junction attached to the calorimeter and the other in the jacket water was connected to a spotlight galvanometer so that any difference in temperature could be detected quickly and corrected immediately.

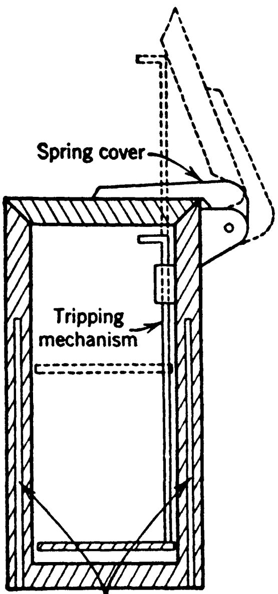


FIG. 7–2. Specific heat calorimeter. Wilkes and Wood (117).

To eliminate the difficulties connected with a lightweight, relatively nonconducting sample, the specimens were prepared by pulverizing and then compressing in a hydraulic press under a pressure of roughly 100,000 lb, in.$^{-2}$, so as to form a solid pellet. Next the pellet was cut down to the desired weight (from 2 to 15 grams), depending upon the material and the temperature to which it was to be heated, correlated with a calorimeter temperature rise of from 4°F to 7°F. The sample was then dried and wrapped in blackened aluminum or platinum foil, weighings being made so that the exact weights of the sample and the foil were known. The sample was placed in a stoppered test tube of Pyrex or fused silica and then put in a steam bath or electrically heated furnace and allowed to remain there until equilibrium was established.

Next the test tube with the sample was quickly removed and the sample "poured" into the calorimeter. In this way very little heat was lost from the sample during transfer as most of the heat loss would have been from the test tube, which did not enter into the calculation. The rise in temperature of the calorimeter was measured accurately with two thermocouples.

The calorimeter was calibrated by inserting a small electric heater into the calorimeter and measuring the watt input for a given rise in temperature corresponding to the test conditions. The heat capacity of the calorimeter in Btu, $°F^{-1}$ could then be readily determined.

The mean specific heat of the sample was calculated from the experimental data as follows:

$$\frac{W_1}{454}(t_3 - t_2)c_1 + \frac{W_2}{454}(t_3 - t_2)c_2 = W_3c_3(t_2 - t_1)$$

where W_1 = weight of sample, grams.
W_2 = weight of foil, grams.
t_1 = initial temperature of calorimeter, °F.
t_2 = final temperature of calorimeter, °F.
t_3 = temperature of sample in furnace, °F.
c_1 = mean specific heat of sample between t_3 and t_2.
c_2 = mean specific heat of foil between t_3 and t_2.
W_3c_3 = heat capacity of calorimeter, Btu, $°F^{-1}$.

c_2 may be taken from tables, as it is a minor correction. All the other terms are found experimentally except c_1, and this value can be found by solving the equation above.

If determinations at three different temperatures are made, the true specific heat, c_t, at any given temperature within the range of the experiments can be calculated, if we assume that

$$c_t = A + Bt + Ct^2$$

then

$$c_1(t_3 - t_2) = \int_{t_2}^{t_3} (A + Bt + Ct^2)\,dt$$

Appropriate values of c_1, t_3, and t_2 for the three tests can be substituted in the above equation. Then, by solving the three simultaneous equations, the constants A, B, and C can be evaluated.

Wilkes (109) extended these specific-heat measurements down to -300°F by using essentially the same technique except that he placed the calorimeter in a Thermos bottle and chilled the samples by immersing the test tubes in carbon dioxide snow or liquid oxygen.

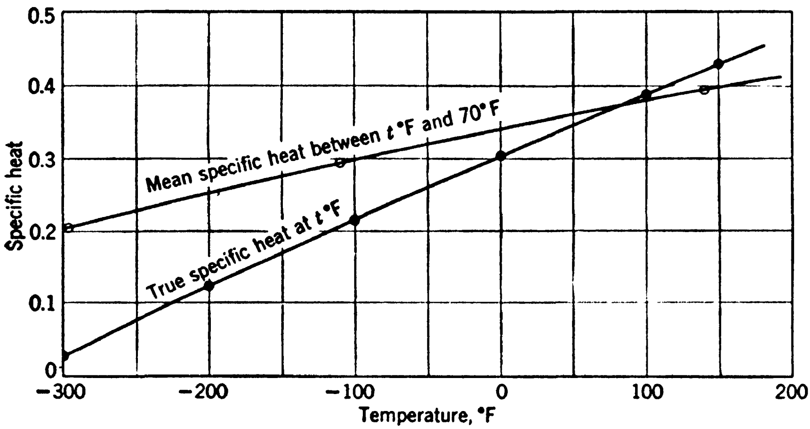


FIG. 7-3. True and mean specific heat between t and 70°F vs. Δt. Wilkes and Wood (117), Wilkes (109).

Figure 7-3 shows the mean specific heat of corkboard as experimentally determined in these tests as well as the true specific heat calculated from the data. Note the large difference between the mean specific heat from -300°F to 70°F and the true specific heat at -300°F.

The specific heat of refractory oxides can be determined by a method described by Wilkes (108) and shown in Fig. 7-4. The method of mixtures was followed, but, to avoid the vaporization of any of the water in the calorimeter when the hot sample was immersed, a special metal container similar to that shown in Fig. 7-2 but of much lighter weight was employed. Before the sample was transferred to the calorimeter, this container was supported in the calorimeter so that no water entered. When the sample was dropped from the carbon tube furnace into this container, the cover was immediately closed and the container sank so that it was totally immersed in the calorimeter water.

This method gave excellent results on aluminum and magnesium oxides up to temperatures of 1800°C (3272°F).

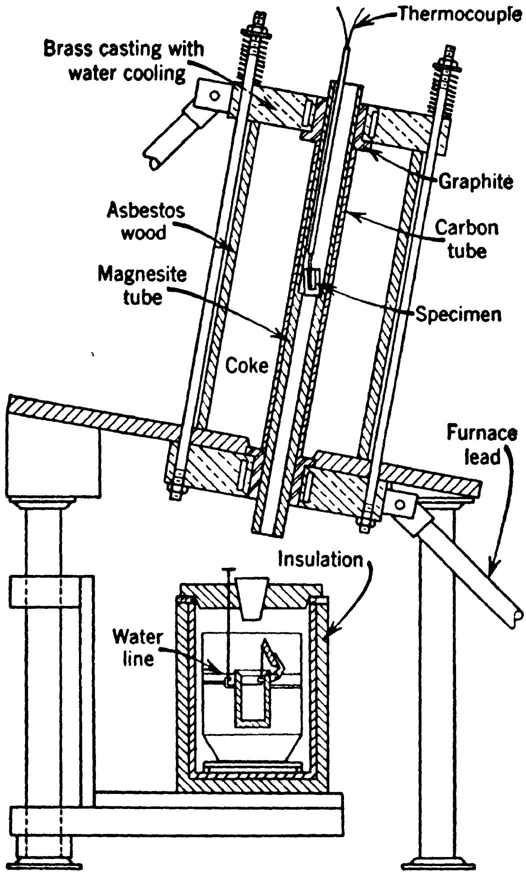


FIG. 7-4. Specific heat equipment for refractory oxides. Wilkes (108).

Applications of heat capacity and thermal diffusivity

The heat capacity, $c\rho$, and thermal diffusivity, $K/c\rho$, are important factors in the choice of insulation wherever a condition of unsteady heat flow exists. The following are a few examples in which one could achieve much better results by considering these two properties.

(*a*) *Intermittently Heated Furnaces, Ovens, Etc.* If it is desirable to have quick heating and quick cooling, an insulating material with high thermal diffusivity is essential. In certain situations, such as in some annealing operations, one wants slow cooling, and then insulation with low thermal diffusivity is desirable.

(*b*) *Intermittently Cooled Refrigerator Trucks, Etc.* Use insulation with high thermal diffusivity for quick cooling.

(*c*) *Test Chambers with Rapid Changes of Temperature.* Aircraft are tested sometimes in chambers where the temperature varies from −100°F to 100°F in a relatively short time. In such chambers, insulation with high thermal diffusivity is almost imperative.

(*d*) *Roof Insulation to Retard Solar Heat.* Roofs with insulation are heated by the sun during the daytime and give off this heat during the night. For reasons of comfort, the roof should be of such construction that it will cool rapidly at night. Low heat capacity and high thermal diffusivity of the insulation help in this direction.

(*e*) *Storage of Vegetables to Prevent Freezing.* Vegetables are sometimes stored in heavily insulated structures to prevent freezing during the winter months. Govan (32*a*) states that they have been stored in this fashion, with no external heat, in a climate such as that of southern Canada. Such storage requires slow cooling and consequently insulation with high heat capacity and low thermal diffusivity.

(*f*) *Prevention of Freezing of Water Pipes.* Water pipes exposed to freezing temperatures for relatively short periods of time may be kept from freezing by means of insulation with high heat capacity and low thermal diffusivity. Over extended periods of exposure to freezing temperatures, it is obviously impossible to prevent freezing of still water, insulation merely retarding the action. If water is continuously flowing through the pipes, sufficient insulation of high heat capacity and low thermal diffusivity may prevent freezing indefinitely.

Equilibrium time

In order to determine the time required to reach thermal equilibrium with both a constant electrical input and a nearly instantaneous heating of one surface of a slab, the following experiments were made on various pieces of equipment in the

Heat Measurements Laboratory, M.I.T. (Wood and Lemberger 121). The information gained applies only to the particular setups, but it should prove useful in approximating the time required for equilibrium in similar apparatus.

Uninsulated Pipes. Six cylinders of various diameters were suspended in a horizontal position exposed to the natural convection and radiation losses to the ambient air and walls of the room. Five of the six cylinders were standard iron pipes with electrical heating applied internally. The sixth cylinder was a brass tube, with a ½-in. outside diameter, chrome-plated so as to minimize the radiation loss. The other pipes were covered with aluminum foil for the same purpose. The temperatures of these cylinders were measured with thermocouples attached to the outer surfaces. Temperature readings were taken when the cylinders were in equilibrium with a room temperature of 76°F. The constant watt input necessary to maintain each cylinder at approximately 163°F, having been previously determined, was applied to the pipes, and temperature readings on each pipe were taken at intervals until equilibrium was established. The results follow.

Cylinder number	1 *	2 *	3	4	5	6
Nominal pipe diameter	8 in.	6 in.	4 in.	3 in.	1.5 in.	. . . . .
Actual outside diameter	8.63 in.	6.63 in.	4.50 in.	3.50 in.	1.9 in.	0.5 in.
Length	34 in.	34 in.	34 in.	34 in.	34 in.	36 in.
Time for equilibrium	4.75 hr	4.75 hr	6.0 hr	4.75 hr	1.8 hr	0.6 hr

* These cylinders had heating elements of lighter construction.

Heavily Insulated Copper Block. A solid copper cylinder, 4 in. in diameter and 14 in. long, was wound with resistance wire and placed in the middle of a cylindrical shell 12 in. in diameter and 22 in. high. The block was supported by 4-in.-thick magnesia block insulation on the bottom, and the remainder of the space between the copper and the shell was insulated with Santocel powder, to a thickness of 4 in. The temperature of the copper was measured with thermocouples inserted in small holes that were cast into the middle of the block. A constant watt input was applied and temperature readings of the copper taken at intervals until equilibrium had been attained. The time to reach equilibrium was considerable. At the end of 100 hr, the temperature had risen from room temperature to 339°F. Real equilibrium was not reached until 300 hr had elapsed but the rise in temperature was only 4°F in the last 200 hr. For most practical purposes, equilibrium was established at the end of 100 hr, or approximately 4 days.

Cold-storage Room. A 10-ft-cube low-temperature room, insulated with 8 in. of corkboard with the exception of the door, which was 4 in. thick, was investigated to find the time required for thermal equilibrium with a constant watt input. Three 100-watt lamps were placed on the floor of the room about 1 ft away from the wall. The temperature of the room was measured with shielded thermocouples and also a resistance thermometer.

After initial temperature readings were taken, the 300-watt input was applied and temperatures were read twice a day until equilibrium was established.

This room, although well insulated, had the usual ammonia pipes and various wires for heating and temperature measurements passing through the walls. Because of the construction of this room it was not considered wise to heat to very high temperatures. In order to heat this room from 79°F to 108°F, 260 hr, or nearly 11 days, were required. Continued heating at the same rate showed no increase in temperature for 140 additional hours. Previous calculations as to the required time were 8 days.

Large Insulated Pipe. A nominal 10-in.-diameter iron pipe, 6 ft long, was insulated with three layers of pipe covering as tabulated.

	Inner Cover	Intermediate Cover	Outer Cover
Insulation	High temperature	High temperature	85% magnesia
Thickness	2.0 in.	1.56 in.	1.50 in.
Outside diameter	14.8	17.9	20.9

After taking temperature readings, with the pipe and coverings at room temperature, it was electrically heated at a constant watt input until equilibrium was attained. Equilibrium time was 115 hr. The temperature differences between the room and various portions of the insulation are shown in Fig. 7–5.

Various Thicknesses of Corkboard with Heat Applied Suddenly on One Side. The time required to reach equilibrium for insulating materials in the slab form and of various thicknesses should theoretically be proportional to the square of the thickness when heated on one side instantaneously. The theory is based upon the use of an infinitely thick slab, an instantaneous

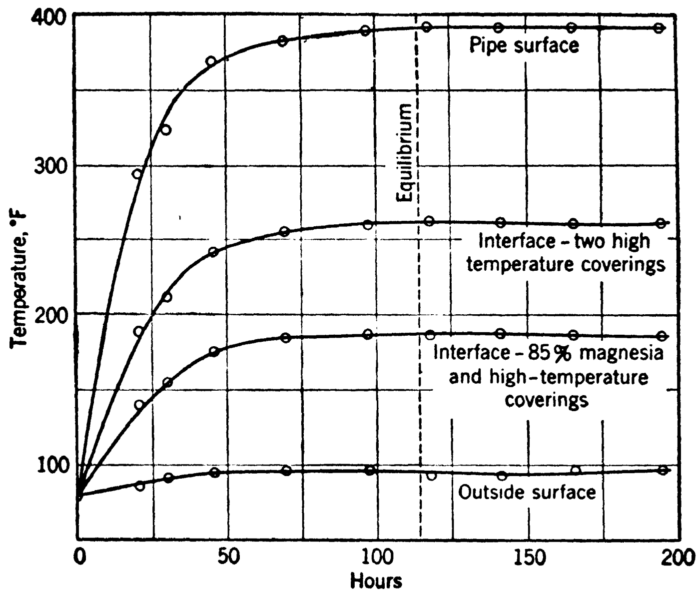


FIG. 7–5. Equilibrium time, compound pipe cover. Wood and Lemberger (121).

rise in temperature on one side, and no change in temperature on the cooler side. It is usually assumed that this theory can be applied without appreciable error (Fishenden and Saunders 31) to slabs of finite thickness until the temperature of the cooler surface becomes appreciably affected. The problem then becomes more complicated, as the heat transfer from the cooler surface to the surroundings must be taken into account.

Wood and Lemberger (121) attempted to check this relation between equilibrium time and thickness of cork slabs, for situations where heat was suddenly applied to one side of the slab. Figure 7–6 is a sketch of the method followed in this investigation. Heat was applied rapidly to one surface by suddenly permitting steam at about 5 lb, in.$^{-2}$, pressure to flow through metal tubing attached to an iron plate in contact with the cork surface. The upper or cooler surface of the slab was maintained at nearly constant temperature by means of a water-cooled plate. Four thicknesses of corkboard were used, 1 in., 2 in., 4 in., and 6 in.

The results are shown on Fig. 7–7 (solid curve), which gives very different values from those calculated on the assumption

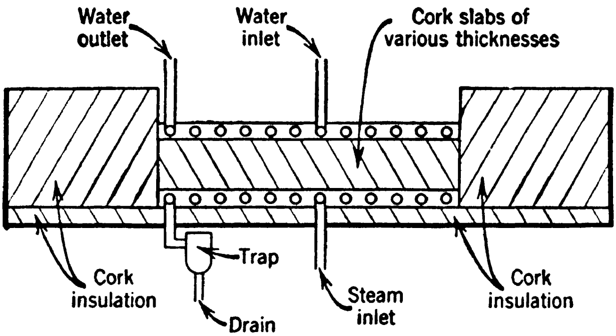


FIG. 7–6. Equipment for determination of equilibrium time. Corkboard. Wood and Lemberger (121).

that the equilibrium time varies with the square of the thickness. The dotted curve shows the calculated equilibrium time based on the 6-in. corkboard's requirement of 780 min to reach equilibrium as experimentally determined. The calculated values for the 1-in. and 2-in. corkboard are less than one half those found by experiment. This large discrepancy may be due to the difference between the conditions of the test and those assumed for the theoretical calculation. The surface was not instantaneously heated but it was rapidly heated to the equilibrium temperature. The final temperature of the hot surface was about 212°F. The time required for the hot surface to reach 200°F for the various thicknesses is given below.

Thickness	Time to Reach 200°F
1 in.	11 min
2	10
4	10
6	8

Heat was also transferred from the cooler side of the corkboard to the water-cooled plate, particularly for the 1-in. and

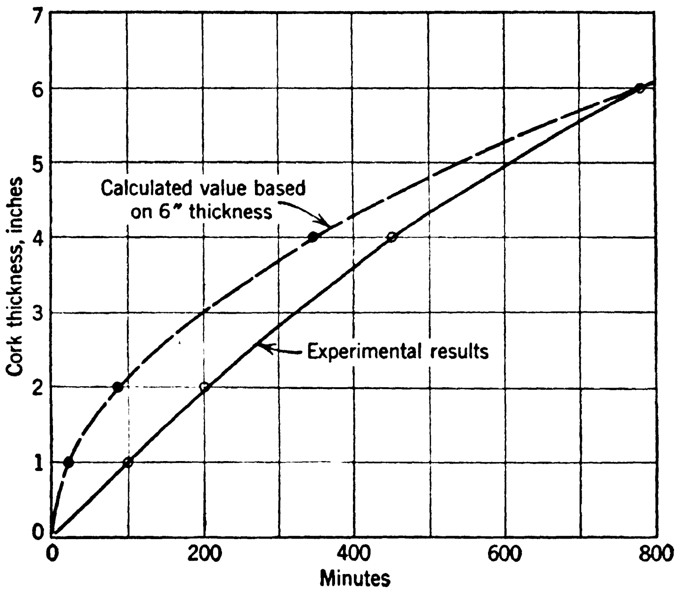


FIG. 7-7. Equilibrium time vs. thickness. Corkboard. Wood and Lemberger (121).

2-in. thicknesses. The temperature of the cool side increased during the test as shown below.

Thickness	Initial Temperature	Equilibrium Temperature
1 in.	79°F	103°F
2	82	95
4	81	91
6	82	88

In dealing with the time required for equilibrium when one side of an insulated wall is rapidly heated to some higher temperature while the other remains nearly constant, the experimental values are probably nearer the correct answer than the values based on the assumption that the time varies directly with the square of the thickness of the wall. This is primarily due to the fact that the conditions assumed for the theoretical calculation are not generally met in practice with insulated walls. Much more work is required to substantiate these conclusions but, in the meantime, calculated values should be employed with caution for conditions similar to those of the tests here described.

CHAPTER EIGHT

Moisture in Insulation

Chapter Four included a discussion of the effect of moisture content on the *K* value, but, since moisture is such an important factor in the effectiveness of low-temperature insulation, more detailed information will be given here concerning the effects of moisture and preventative methods.

Condensation of water vapor on surfaces

Everyone is familiar with the condensation of water vapor on the outside of a cold glass in warm, humid weather or on the inner surface of a window pane during cold weather when the conditions are suitable. Atmospheric air always contains some water vapor in addition to the other constituents. Although the percentage of nitrogen, oxygen, carbon dioxide, and the rarer gases remain nearly constant, the amount of water vapor varies considerably. There is a definite limit to the amount of water vapor that can exist in a given volume of air for any given pressure and temperature. If one attempts to add more water vapor under these conditions, an equal amount of water vapor will condense to drops of water and the actual amount of water vapor in the space will remain the same. This is what is known as a saturated condition or 100% relative humidity. If the temperature of this air is lowered, there will be further condensation because the maximum amount of water vapor that the space can hold decreases as the temperature is reduced. The relative humidity will still be 100% but the actual weight of water vapor will be less. If air at 100% relative humidity is heated, the water-vapor content will remain the same but the relative humidity will become less than 100% because then water vapor can be added without immediate condensation. Relative humidity merely means the ratio of the quantity of water vapor in a given volume of air to the maximum amount the volume could hold at the same pressure and temperature.

If air containing water vapor is gradually cooled, the pressure remaining constant, a temperature will be reached at which the air is saturated (100% relative humidity). Any further lowering of the temperature will produce condensation. This tem-

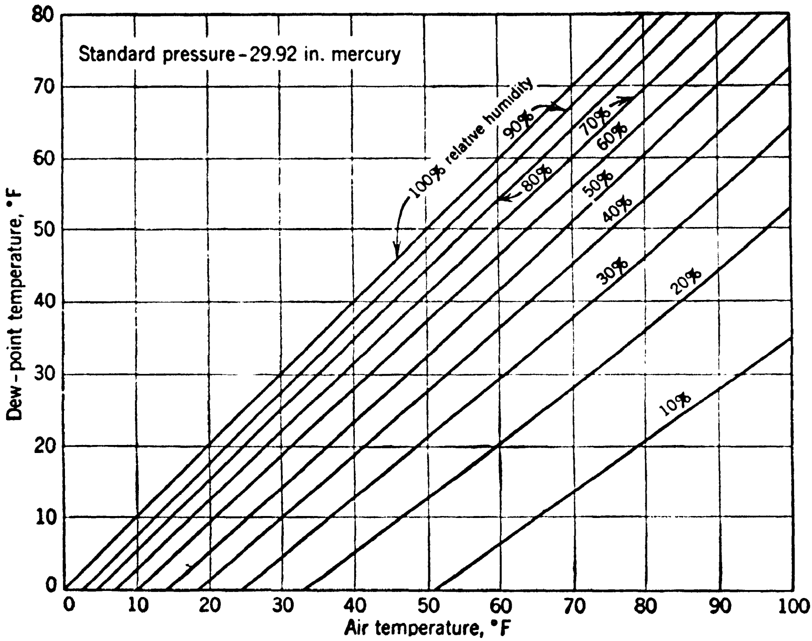


FIG. 8-1. Dew point vs. air temperatures at various relative humidities.

perature at which saturation is reached is called the dew-point temperature or commonly just the dew point. Figure 8-1 can serve to determine the dew point for air under standard pressure for various temperatures and relative humidities.

Cold-water Pipes. Frequently, during the summer months, people are troubled by condensation of water vapor on cold-water pipes, particularly in the cellar. If the air is 80°F with 70% relative humidity, one can readily determine from Fig. 8-1 that the dew point is 69°F. If the cold-water-pipe surface temperature is 65°F, condensation will take place on the pipe surface and continue to do so as long as the surface is below the dew-point temperature. The consequent dripping of water from the pipe may cause considerable damage. Condensation can be prevented by insulating the pipe so that the outer surface of the insulation will have a surface temperature above the dew point. A very simple and inexpensive method of minimizing this trouble when it occurs in cellars is to wrap *loosely* a layer or two of thin aluminum foil around the pipe and fittings.

Window Panes. During cold weather, condensation of water vapor on the warm side of single window panes often takes place, causing damage to the window sills and wallpaper from the water. The dew point for air at 70°F and 30% relative humidity is about 37.5°F. On a cold day the surface of the window pane is often below 32°F, and condensation with freezing can be expected. It can be avoided for the most part by means of storm windows, double glass, fan circulation of air near the pane, or reduction of the relative humidity of the air in the room.

Toilet Tanks. In many localities, the cold-water supply, particularly from wells, is sufficiently cold to cause condensation on toilet tanks during warm, humid weather. Insulation usually is not feasible in such circumstances. Sometimes drip pans are installed but they are far from pleasing in appearance. A tempering tank in the basement helps to raise the water temperature and reduces the condensation. For serious cases, some of the domestic hot water can be mixed with the cold water by means of a modulating valve set to raise the temperature of the incoming water above the dew point.

Roofs of Manufacturing Plants. In many buildings, the relative humidity is high on account of the nature of the industry (paper mills, laundries, etc.), or it is artificially maintained at a high level on account of the material being processed (textile mills, etc.) In cold weather, the temperature of the lower surface of a concrete roof may well be lower than the dew point. Condensation with consequent dripping of water from the ceiling may cause serious damage to the machinery or the product. It can be prevented by insulating the roof, thus bringing the ceiling temperature above the dew point.

Assume that we have a roof slab (Fig. 8–2*A*) consisting of 6-in. concrete and a built-up roofing which will be exposed to

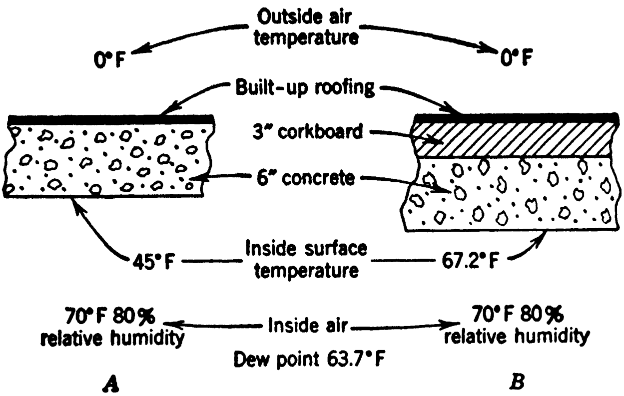


FIG. 8–2. Prevention of condensation on ceilings by insulation.

outside air at 0°F, inside air at 70°F, and 90% relative humidity. First determine whether condensation will occur on the lower side of the slab under these conditions. By means of Eq. 2–20, the temperature, t_s, can be determined.

$$t_s = t_i - \frac{1/f_i}{(1/f_i) + (l_1/K_1) + (1/C_2) + (1/f_o)}(t_i - t_o)$$

where t_o = outside air temperature, 0°F.
t_i = inside air temperature, 70°F.
f_i = inside surface coefficient, 1.96 Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
f_o = outside surface coefficient, 5.9 Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.
l_1 = thickness of concrete, 6 in.
K_1 = coefficient of thermal conductivity of concrete, 12 Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$.
C_2 = (K_2/l_2) = conductance of built-up roofing, 3.6 Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.

$$t_s = 70 - \frac{1/1.96}{(1/1.96) + (6/12) + (1/3.6) + (1/5.9)}(70 - 0)$$

$$= 45.6°F$$

From Fig. 8–1, the dew point for 70°F and 90% relative humidity is found to be 67.1°F. Condensation would occur under these conditions on the lower side of the concrete roof slab. The thickness of insulation necessary to bring the surface temperature up to the dew point of 67.1°F can be found by again employing Eq. 2–20, with $t_s = 67.1$°F. If the added insulation is corkboard with a K value of 0.27 Btu, hr^{-1}, ft^{-2}, in., $°\text{F}^{-1}$ and with a thickness of x in., we have

$$t_s = t_i - \frac{1/f_i}{(1/f_i) + (l_1/K_1) + (x/K) + (1/C_2) + (1/f_o)} (t_i - t_o)$$

x is the only unknown in this equation and is found to be 2.93 in. In this problem, 3 in. of corkboard would eliminate condensation on the underside of the slab under the stated conditions (Fig. 8–2B).

The above expression can be simplified by letting U = the coefficient of thermal transmittance of the uninsulated roof =

$$\frac{1}{(1/f_i) + (l_1/K_1) + (1/C_2) + (1/f_o)} = \frac{1}{R_{\text{wall}}}$$

then

$$t_s = t_i - \frac{1/f_i}{R_{\text{wall}} + R_{\text{cork}}} (t_i - t_o)$$

By substituting $1/U$ for R_{wall} and x/K for R_{cork}, and transposing, we have

$$x = K \left[\frac{t_i - t_o}{f_i(t_i - t_s)} - \frac{1}{U} \right]$$

Cellar Walls. Condensation on the inside of cellar walls often happens because the temperature of wall surface is below that of the dew point on warm, humid days. It can be prevented by the application of insulation or by the rapid circulation of air over the surface so that the temperature of the wall surface will rise above the dew point. Drying agents such as silica gel are also used for this purpose.

Condensation in walls

Whenever the temperature at any point within a wall becomes lower than the dew point of the air on the warmer side, condensation tends to occur at that point, provided that water vapor can be transmitted through the wall from the warmer side. This is a very serious problem in connection with low-temperature insulation, and it also causes trouble occasionally in building walls. Moisture can be transmitted into a wall by hygroscopic action, as well. Some materials such as fiberboards, wood, and cork, will absorb a certain amount of moisture when exposed to air containing water vapor. This is not visible moisture, such as occurs when the dew point is within the wall, but is moisture absorbed by the fibers of the insulating material because of the relative humidity and temperature of the ambient air, which may be well above the dew point. When such materials are placed in the standard plate test (Chapter Three) after being exposed to air with some water vapor present, the moisture in the sample will tend to migrate from the warm side to the cooler side of the sample. MacLean (58) proved this statement with a variety of woods by measuring the average moisture content in each sample before testing, and then immediately after completion of the test he removed the sample and cut thin slices from the warm and cool sides. In every instance, the cool side increased in moisture content, whereas the warmer side decreased in moisture content during the test. The amount of moisture migration varied with the type of wood, the original moisture content, the temperature difference, and probably the time. The K values tended to be high during this moisture migration until moisture equilibrium was reached.

A few of MacLean's values for the distribution of moisture content in Douglas fir under these conditions are in the accompanying table as percentage of dry weight. The duration of the tests was one day.

Average Moisture Content before Test	Moisture Content near Warm Plate after Test	Moisture Content near Cool Plate after Test	Thickness	Δt
10.2%	8.6%	10.8%	0.660 in.	52°F
14.0	8.6	15.6	0.628	49
20.4	10.8	33.0	0.707	46
27.7	12.6	47.0	0.713	47

The usual ways to prevent the condensation of water vapor in walls when some point within the wall is below the dew point are to place a vapor barrier on the warm side, reduce the relative humidity on the warm side, and vent the cool side. Venting the cool side helps if the outside is always cooler, but in some localities during the summer months the air may be warmer on the outside and have high relative humidity. Under such conditions, venting the outside of a wall just invites condensation within the wall.

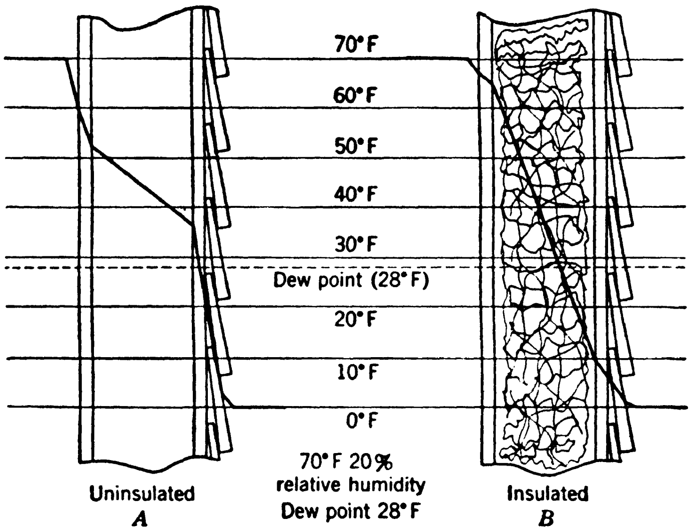


FIG. 8-3. Location of dew point in walls.

Building Walls. The insulation of homes, although saving a considerable amount of fuel and making more comfortable living quarters, is occasionally the cause of condensation of water vapor in the walls. Figure 8-3 shows the temperature gradient through an uninsulated (A) and an insulated (B) frame wall. With an inside air temperature of 70°F and with 20% relative humidity, the dew point is 28.4°F. In the uninsulated wall, the dew point would be in the sheathing. This condition would probably not be very serious unless these temperature conditions held for a long period of time. With fluctuating temperatures in conjunction with the air circulation in the stud space, condensation in the sheathing should not cause much trouble. In the insulated wall, the dew point would be located near the middle of the insulation. This portion would become wet and be very slow to dry out under different temperature conditions. Also, when condensation occurs, the wet insulation becomes a better conductor of heat, and the dew point would tend to move toward the warm side of the insulation. In time, this condition might become the source of serious damage. With new construction, a vapor seal could be placed on the warmer side of the insulation, which would reduce materially the danger of condensation.

The rate of water-vapor transmission through a vapor seal is generally assumed to be proportional to the water-vapor pressure differential that exists between the two sides of the seal. The water-vapor pressure at 70°F and 30% relative humidity is equal to 13.6 lb, ft^{-2}, whereas at 0°F and 70% relative humidity, the vapor pressure is only 1.9 lb, ft^{-2}. This leaves a pressure differential of 11.7 lb, ft^{-2}, to transmit water vapor from the inside to the outside of a wall.

Most of the tests on the permeability of vapor seals are made with essentially leakproof joints at the edge of the vapor seal. This condition is not usually found in actual practice. One may have a vapor seal, with very low permeability, that when applied to a wall will permit much more water vapor to be transmitted than indicated by the permeability tests, because of the leakage around the joints.

The results of tests on the permeability of various materials are given in Table 5 in the appendix.

A glance at the table on water-vapor permeability shows that there is considerable variance in the results of the different investigations, which is probably due to the nonstandardization of method as well as nonuniformity of samples. The numerical results should be employed with caution but relative values by the same investigator should be very helpful in choosing a vapor seal.

Moisture in Walls of Low-temperature Rooms. When refrigerated spaces are maintained at 40°F or less, the dew point is usually located somewhere in the insulation for long periods of time; and at very low temperatures, as in deep freeze units, the dew point is permanently in the insulation. These are much more severe conditions than usually found in building walls and are a great problem to refrigerating engineers.

The troubles connected with the condensation and frequently the freezing of water vapor in the walls of low-temperature rooms can be reduced by the following precautions.

(*a*) Apply the best possible water-vapor seal on the warm side of the insulation. Materials are available that are essentially impervious to water vapor, but there are nearly always joints that also must be made vaportight and at the same time must withstand some "come and go" due to thermal expansion. Foamglas, Rubatex, and metal foils or sheets are examples of this type of vapor seal. Foamglas is usually supplied in blocks 12 in. by 18 in., Ferro-Therm in sheets 24 in. by 32 in., Rubatex in sheets 3 ft by 4 ft, and Alumiseal in coils 32 in. wide and many feet long. A lead-foil adhesive tape is supplied with Alumiseal which, if carefully applied, should make the joints essentially vaportight. Asphalt has been used for many years to seal the warm side of cold-storage rooms. It must be an excellent grade and, if applied over an open mesh fabric, better results should be obtained, as the fabric will aid in preventing cracks. Domestic refrigerators are usually made with metal walls that can be sealed fairly effectively.

(*b*) Do not place a vapor seal on the cold side of refrigerator walls if the room is to be continuously maintained at a low temperature. This seal would prevent any moisture, that might come into the insulation from the warm side, from slowly migrating to the refrigerator coils in the room and thus aid in keeping the insulation dry.

(*c*) Since it is nearly impossible with a commercial installation to have a perfectly vaporproof seal, sooner or later some moisture will get into the insulation. It is, therefore, good practice to choose an insulating material that is least affected by moisture penetration, provided that its other properties are suitable. The Cork Insulation Manufacturers Association (26) presents data on the increased power consumption and moisture absorption of boxes constructed of several low-temperature insulating materials. The temperature of the air inside these boxes was maintained at 90°F and a very high relative humidity. The outside air was maintained at 20°F. When no vapor seal was on the warmer wall, the power input to maintain a constant temperature difference increased considerably in a 10-day period and there was a marked difference in this increase for different materials. The results of some of these tests after 18 weeks are given in the following table. The insulating materials tried were corkboard, mineral fiberboard, vegetable fiberboard, wood fiberboard, animal hair board, and loose shredded redwood bark.

	Cork	Non-cork 1	Non-cork 2	Non-cork 3	Non-cork 4
Power at start (dry)(watts)	1673	1727	2023	1869	2599
Power after 18 weeks (watts)	2300	4891	5410	4575	5121
Percentage of increase in power	38	183	167	145	97
Pounds of water condensed	54.9	203.5	266.0	236.3	267.6
Percentage of water by volume	7.8	28.8	37.6	33.4	37.8

With a vapor seal on the warm side, much longer tests were necessary because the rate of water-vapor transmission into the insulation was much reduced.

The following quotation from the report is enlightening.

> Surface coverings on the hot side retarded the absorption of moisture and decreased the rate of increase of power demand for all materials. Asphalt emulsion improved the moisture resistance of all materials, even those with a factory-applied surface covering. However, so far as the results of these tests show, neither asphalt emulsion trowelled on nor factory-applied coatings give permanent protection against moisture. All coverings of this kind may fail in service and permit moisture to get into the insulation; moisture reduces the insulating value and may ultimately destroy the insulation itself. The development of better methods for securing and maintaining permanently waterproof construction is of great economic importance and should receive encouragement and support.

The results of tests similar to those mentioned above are shown in an advertised report of the Engineering Experiment Station, Pennsylvania State College, to the Rubatex Division of the Great American Industries, Inc., in 1942. The average temperature of the air inside the boxes was 100°F, and that of the air outside was 5°F. No vapor seal was used on the warm side in order to accelerate the tests. The walls were 4 in. thick, and in this experiment all six sides of the box were constructed of the insulating material, whereas in the previously mentioned tests only four walls and the top consisted of the insulating material under test. Some of the results are tabulated below.

	Cork	Rubatex Chemical Blown	Rubatex Gas Blown
Watts at start (dry)	1711	1383	1230
Watts after 18 weeks	2449	1572	1471
Percentage of increase in power	43	14	20
Pounds of water absorbed	58.2	7.0	7.0
Percentage of water by volume	6.5	0.8	0.8

Since Rubatex, itself, is essentially impervious to water vapor, all the moisture absorbed during these tests probably came through the joints between the blocks.

CHAPTER NINE

Economics of Insulation

If the actual yearly cost of insulation were not less than the yearly saving in fuel and equipment, it is obvious, from a purely economic view, that one would not use insulation. However, it should be stressed that in most cases other factors must be considered which often far outweigh the economic factor. See Chapter One for a discussion of some of these other reasons for insulation. It is very difficult to evaluate these other factors in terms of dollars.

House insulation

An example showing the relative saving by insulating different parts of a house follows.

The cost of the insulation must include material, labor, and incidental expenses. If the insulation replaces other material such as wood sheathing, the cost of the wood sheathing plus labor must be subtracted from that of the insulation. If a smaller and less expensive heating plant can be used, due to the insulation, this saving must also be subtracted from the cost of the insulation. The average yearly cost of insulation is usually found by applying some factor that includes interest on the investment, depreciation, repairs, insurance, etc. For a dwelling, a figure of 10% of the cost of the insulation is frequently satisfactory for the yearly cost.

The yearly cost of 1,000,000 Btu available for heating purposes may be determined as follows:

Fuel	oil
Assumed Btu per gallon	144,000
Assumed heater efficiency	70%
Cost of oil	$0.12 per gallon

$$\text{Cost of 1,000,000 Btu} = \frac{1{,}000{,}000 \times 0.12}{144{,}000 \times 0.70} = \$1.19$$

The saving in Btu due to the insulation must be calculated. For dwellings use the *American Society of Heating and Ventilating Engineers Guide* or the Federal Housing Authority *Technical Bulletin* 7. The F.H.A. bulletin takes into account the variation of surface coefficients and air-space conductances due to orientation, which sometimes make considerable difference. For low-temperature work the *American Society of Refrigerating Engineers' Data Book* is convenient. U values can also be calculated from the tables given in the appendix.

For a 30 ft by 30 ft, two-and-one-half-story frame house, the following assumptions will be made.

	U Uninsulated	U Insulated
Windows, 191 sq ft	1.13	0.45
Doors, 39 sq ft	0.51	0.35
Walls, 1810 sq ft	0.26	0.10
Second-floor ceiling, 900 sq ft	0.33	0.10
Degree days for season, 7400		

The number of degree days are found by subtracting the average daily outdoor temperature from 65 and adding these results for the entire heating season. The average degree days for each month of the heating season for the principal cities of the United States can be found in the *A.S.H.V.E. Guide*.

The Btu loss from the uninsulated windows, for example, can be found for the entire heating season as follows:

$$7400 \times 24 \times 1.13 \times 191 = 38{,}000{,}000 \text{ Btu per season}$$

The fuel saving that might be expected under perfect conditions by insulating various parts of this house can be found in the following table.

Case 1. Storm sash and storm doors only.
Case 2. Wall insulation only.
Case 3. Ceiling insulation only.
Case 4. Combine Cases 1 and 3.
Case 5. Combine Cases 1, 2, and 3.

LOSS IN MILLION BTU PER YEAR

	Uninsulated	1	2	3	4	5
Single windows	38	......	38	38	......	......
Storm sash	......	15	......	......	15	15
Doors, single	3.5	......	3.5	3.5	......	......
Storm doors	......	2.4	......	......	2.4	2.4
Walls	83	83	......	83	83	......
Insulated	......	......	32	......	......	32
Ceiling	53	53	53	......	......	......
Insulated	......	......	......	16	16	16
Infiltration (estimated)	50	25	50	50	25	25
Total	228	178	177	191	141	90
Fuel cost @ $1.19 per million Btu	$271	$212	$211	$227	$168	$107
Fuel saving per year	......	$59	$60	$44	$103	$164
Percentage of saving in fuel	......	22	22	16	38	60

The following costs of insulation will be assumed. The actual prices vary considerably, depending upon the types of materials used and the locality.

			10% of Cost
Case 1.	Storm sash and doors @ $10 each	$190	$19.00
Case 2.	Wall insulation @ $0.15 per sq ft	272	27.20
Case 3.	Ceiling insulation @ $0.15 per sq ft	135	13.50
Case 4.	Case 1 plus Case 3	325	32.50
Case 5.	Case 1 plus Case 2 plus Case 3	597	59.70

The return on the investment will be:

Case 1. $$\frac{59 - 19}{190} \times 100 = 21\%$$

Case 2. $$\frac{60 - 27.2}{272} \times 100 = 12\%$$

Case 3. $$\frac{44 - 13.5}{135} \times 100 = 23\%$$

Case 4. $$\frac{103 - 32.5}{325} \times 100 = 22\%$$

Case 5. $$\frac{164 - 59.7}{597} \times 100 = 18\%$$

These values are merely calculated values based on certain assumptions that may vary considerably in actual situations. For example, any of the following conditions will affect the fuel consumption.

Storm sash not fitting tightly.
Doors and windows open excessive amount.
Dampers in fireplaces left open.
Moisture in the insulation.
Improper application of insulation.
Heater in poor condition.
Excessive infiltration of air due to poor construction.

The calculations do indicate that the house owner should get the best return on his investment by using storm windows and ceiling insulation. In a climate such as that of Boston, Mass., insulating a new house is well worth while both from an economic and comfort viewpoint. Storm windows and doors, although a nuisance to put up and take down each year, probably add more to the comfort of an uninsulated house than any one thing that the house owner can do. Double glazing (two panes of glass separated by a small air space and mounted in the same sash) is very convenient but less effective in reducing heat transfer than storm sash properly applied with an air space of 3 to 4 in. between windows (see Fig. 6–1). The transmission of heat through the nonglass portion of the sash is much greater, especially with metal sash, for double glazed sash than for separate storm windows. Unless good weather stripping is used, the infiltration of air is also greater with double glazing than with separate storm sash.

With an uninsulated house, already built, it is certainly advisable to install storm windows and insulate the ceiling or roof if the house is located in the northern parts of the United States. The insulation of the walls of a house after it has been built is much more expensive than during construction, and blowing insulation into the walls is a questionable procedure in many instances.

The National Bureau of Standards, Technical Information on Building Materials 3, March 4, 1936, indicates estimated fuel savings up to 40% due to the application of simple heat-loss preventives to walls and roof of an uninsulated house, and with suitable weather stripping and storm sash and doors the total savings were boosted to 60%. These values are essentially the same as given in the previous sample calculation.

A well-insulated residence should have U values for the walls and ceiling or roof which approximate 0.10 and also should have storm sash and doors.

Commercial insulation

The question concerning insulation for low-temperature rooms, ovens, furnaces, etc., is not the advisability of insulating but

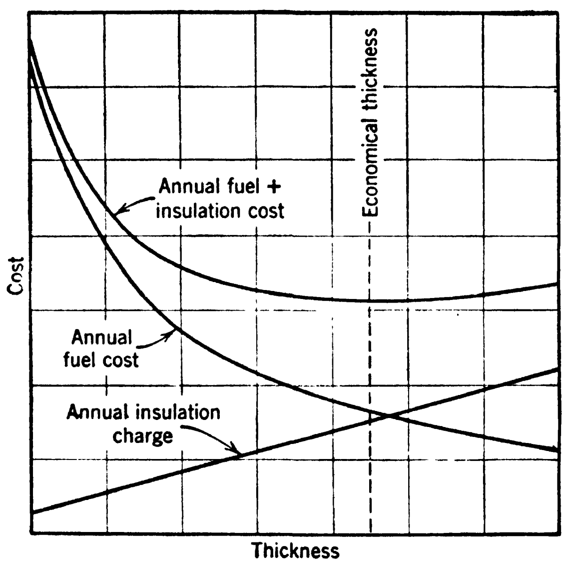


Fig. 9-1. Economical thickness of insulation.

the proper thickness of insulation. The problem is somewhat simpler than that of house insulation because conditions here are usually more controlled. The economical thickness can be calculated as shown by Patton (74) and many others. In general these calculations are based on the assumption that the cost of insulation varies proportionally with the thickness, which is not usually the situation in practice. The decrease in the cost of fuel or power with the increase in the thickness of insulation is gradually offset by the increase in the cost of insulation, and so a point is soon reached where it is not economical to provide greater thickness of insulation. Figure 9–1 shows the general relationship between the insulation thickness and the annual fuel costs, the annual insulation costs, and the sum of these two costs. The economical thickness for insulation is sometimes calculated but very seldom employed in practice. In general, the thickness can vary considerably from the economical one with not much change in the total cost. It is seldom that one encounters a commercial structure that is overinsulated from the economic viewpoint.

Appendix

TABLE 1

Coefficients of Thermal Conductivity

A. Powders

K is expressed in Btu, hr^{-1}, ft^{-2}, in., $°F^{-1}$. For conversion factors to other units see Table 2–3, p. 10.

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Alumina, compressed powder	115	117	4.7	45
powdered	4.5	100	0.26	113
Alundum, grain	69	1000	2.04	113
		1200	2.15	
		1500	2.40	
		1800	3.10	
Ash, volcanic	51	300	1.48	113
		500	1.57	
		700	1.68	
		900	1.82	
Ashes, soft wood	12.5	68	0.22	45
Carbon black	12.0	133	0.148	91
	16.2	133	0.141	
	20.2	118	0.182	
	24.8	125	0.346	
Coal dust	46	100	0.78	45
		200	0.87	
Coke dust	63	68	1.04	45
Charcoal	11.5	63	0.35	33
Cork, granulated, coarse	5.4	23	0.32–0.35	33
	7.3	26	0.35–0.38	

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

A. Powders (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Cork, granulated, fine	6.5	32	0.30	33
	7.3	50	0.32	
Diatomaceous earth	12.5	50	0.36	45
		200	0.43	
		400	0.50	
		600	0.59	
		800	0.67	
	18.7	50	0.42	45
		200	0.49	
		400	0.59	
		600	0.69	
		800	0.79	
	25	50	0.52	45
		200	0.60	
		400	0.71	
		600	0.83	
		800	0.95	
Flotofoam (U. S. Rubber Co.)	1.6	92	0.20	113
Fuller's earth	33	86	0.70	45
Graphite, powdered				
100-mesh	30	100	1.26	45
40-mesh	26	100	2.7	45
20-mesh on 40-mesh	44	100	8.2	45
Gravel	116	68	2.6	45
Pearlite (Arizona)	9.1	112	0.42	113
spherical shells of silicious material	13.1	100	0.44	113
		200	0.47	
		300	0.53	
Peat, dry	11.9	86	0.36	45
Plaster of Paris	......	68	7.5	45

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

A. Powders (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Pumice, powdered	49	300	1.32	113
		500	1.42	
		700	1.55	
		900	1.78	
Sand, fine river, dry	95	50	2.2	45
normal (moisture 6.9% weight)	102	68	7.8	45
Sawdust, various kinds	12.5	86	0.41	45
Silica, finely ground	50	600	0.54	113
		900	0.57	
		1200	0.62	
		1500	0.72	
Silica aerogel, Santocel	5.3	−100	0.117	113
		0	0.138	
		100	0.160	
		200	0.184	
		300	0.212	
Silica gel	32.5	131	0.59	113
Slag, blast furnace				
2–5 mm grain	22.5	68	0.72	45
3 cm grain	22.5	68	1.04	45
Snow	7–31	32	0.34–1.3	45
Soil, dry	......	68	0.9	45
normal including stones	127	32	3.5	45
		68	3.6	
		158	4.0	
wet	......	68	4.6	45
Titanium oxide, finely ground	52	1000	0.49	113

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

A. Powders (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Zirconia grain	113	600	1.35	113
		900	1.50	
		1200	1.70	
		1500	1.95	
		1800	2.30	

B. Loose, Fibrous Materials

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Cotton, tightly packed	5.0	−200	0.28	45
		0	0.37	
		100	0.42	
		200	0.47	
Eelgrass	15.6	86	0.32	45
Glass wool	1.4	114	0.286	113
	2.1	100	0.28	113
		200	0.37	
		300	0.55	
	4.8	100	0.27	113
		200	0.36	
		300	0.47	
	3.0	200	0.31	71
		400	0.36	
		800	0.57	
Kaolin wool	10.6	800	0.71	113
Kapoc, loosely packed	0.9	68	0.24	45
Rock wool	7.0	117	0.29	113
	8–12	200	0.38	71
		400	0.52	
		800	0.78	
Shavings, various	8.7	86	0.41	45
Silk, scrap from spinning mill	6.3	50	0.32	45
		100	0.35	
		200	0.41	

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

B. Loose, Fibrous Materials (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Slag wool	9.4	86	0.29	45
	12.5	86	0.31	
	15.6	86	0.33	
	18.7	86	0.36	
Steel wool	4.8	131	0.62	45
	6.3	131	0.60	
	9.5	131	0.56	
Wood fibers, bark of eucalyptus tree	3.4	32	0.32–0.38	33
shredded redwood bark, Palco Bark	4.0	−300	0.14	109
		−200	0.14	
		−100	0.17	
		0	0.22	
		100	0.30	
Wood wool	2.6	68	0.28	33
Wool, pure	2.5	86	0.29	45
	5.6	86	0.25	

C. Boards or Blocks

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Asbestos paper, corrugated	11	100	0.54	113
	16	100	0.37	113
		200	0.43	
		300	0.50	
laminated	22	100	0.45	113
		200	0.43	
		300	0.50	
		400	0.61	113
Cellulose sponge	2.2	86	0.40	113
	3.4	82	0.40	

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

C. Boards or Blocks (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Corkboard	6.9	−300	0.12	109
		−200	0.17	
		−100	0.22	
		0	0.24	
		100	0.26	
dense	15.7	121	0.31	113
	20.7	121	0.35	
Fiberboards				
glass, Fiberglas PF	2.0	50	0.248	113
		100	0.295	
		150	0.349	
	2.5	50	0.240	
		100	0.278	
		150	0.323	
	3.25	50	0.230	
		100	0.262	
		150	0.305	
	4.25	50	0.220	
		100	0.249	
		150	0.281	
	6.0	50	0.220	
		100	0.245	
		150	0.275	
Fiberglas with asphalt coating	11	−300	0.08	109
		−200	0.13	
		−100	0.17	
		0	0.22	
		100	0.27	
Fiberglas XB-PF	0.5	50	0.245	113
		100	0.280	
		150	0.325	

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

C. Boards or Blocks (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Fiberboards (*Continued*)				
Fiberglas XB-PF	1.0	50	0.214	113
		100	0.240	
		150	0.270	
mineral wool with asphaltic binder	14.3	−300	0.09	109
		−200	0.15	
		−100	0.19	
		0	0.25	
		100	0.29	
sugar cane, Celotex with asphaltic coat	14.4	−300	0.16	109
		−200	0.21	
		−100	0.25	
		0	0.29	
		100	0.33	
wood, various makes	12–19	100	0.32–0.37	113
Glass blocks, expanded, Foamglas	10.6	−300	0.30	109
		−200	0.35	
		−100	0.39	
		0	0.42	
		100	0.43	
Gypsum board	51	99	0.74	113
	65	105	1.06	
High-temperature blocks	14–24	200	0.43–0.62	113
		400	0.50–0.68	
		600	0.59–0.77	
		800	0.70–0.85	
85% Magnesia blocks	11–13	100	0.38–0.42	113
		200	0.41–0.44	
		300	0.43–0.45	
Rubber board, expanded, Rubatex	4.9	−300	0.05	109
		−200	0.13	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

C. Boards or Blocks (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Rubber board, expanded, Rubatex (*Continued*)		−100	0.18	
		0	0.21	
		100	0.22	
Vermiculite blocks	21	200	0.91	113
	19	200	1.50	113
		400	1.62	
		600	1.75	
		800	1.87	
Woods	K is proportional to the density. See Fig. 4–4*A*			
Woolfelt, crepe	14.5	100	0.30	113
solid	24.2	100	0.37	113

D. Insulating Refractories

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Insulating brick				
calcined diatomaceous earth base	43	800	1.4	71
		1200	1.6	
		1600	2.0	
raw diatomaceous earth base	33	200	0.7	71
		400	0.8	
		800	0.9	
		1200	1.0	
Insulating firebrick				
fireclay base	48	400	2.5	71
		800	2.8	
		1200	3.2	
		1600	3.7	
kaolin base	55	800	2.9	71
		1200	3.3	
		1600	3.9	
		2000	4.5	
		2400	5.2	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

D. Insulating Refractories (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Insulating firebrick (*Continued*)				
kaolin base	44	800	2.1	
		1200	2.8	
		1600	3.3	
		2000	3.8	
		2400	4.4	
	26	400	0.8	
		800	1.0	
		1200	1.3	
		1600	1.5	

E. Heavy Refractories

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Alumina, fused, 99% aluminum oxide				
low density	90	800	7	106*a*
		1200	6	
		1600	6	
		2000	6	
	120	800	13	106*a*
		1200	11	
		1600	10	
		2000	9	
	145	800	19	106*a*
		1200	16	
		1600	14	
		2000	12	
high density	195	800	25	106*a*
		1200	20	
		1600	18	
		2000	17	
Chrome brick	...	400	10.0	72
		800	10.8	
		1200	11.3	
		1600	11.6	
		2000	11.8	
		2400	12.0	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

E. Heavy Refractories (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Fireclay brick, average, considerable variation with different brands	...	400	6.6	Various
		800	7.1	
		1200	7.6	
		1600	8.1	
		2000	8.6	
Magnesia, fused, 96–97% magnesium oxide	165	800	28	106*a*
		1200	23	
		1600	20	
		2000	17	
Magnesite brick	159	400	25.8	110
		800	22.0	
		1200	18.2	
		1600	15.7	
		2000	13.7	
		2400	12.6	
Magnesite brick (Canadian)	173	400	15.8	113
		800	14.3	
		1200	13.2	
		1600	12.6	
		2000	12.4	
Silica brick	95	400	7.2	111
		800	9.1	
		1200	10.9	
		1600	12.4	
		2000	14.1	
		2400	15.5	
Silicon carbide brick, clay bonded	139	400	26	111
		800	26	
		1200	29	
		1600	34	
		2000	38	
		2400	50	

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

E. Heavy Refractories (Continued)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Silicon carbide brick, clay bonded (*Continued*)				
recrystallized	136	400	210	111
		800	175	
		1200	150	
		1600	115	
		2000	95	
		2400	81	
recrystallized	...	800	143	71
		1200	126	
		1600	105	
		2000	93	
		2400	80	
Soapstone	175	1040	11.3	113
Spinel brick				
$MgOAl_2O_3$	...	400	10.4	71
		800	11.3	
		1200	12.2	
		1600	13.0	
		2000	13.7	
		2400	14.3	
Zirconia, fused, 98–99% zirconium oxide plus calcium oxide	250	800	5	106*a*
		1200	5	
		1600	6	
		2000	6	

F. Crystals

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Beryl [$Be_3Al_2(SiO_3)_6$]		200	27.5	51
⊥ to C axis		400	28.5	
single crystal, Brazil		600	31.0	
		800	34.5	
		900	37.0	
∥ to C axis		200	29.0	51
single crystal, Brazil		400	31.0	
		600	33.5	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

F. Crystals (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Beryl [$Be_3Al_2(SiO_3)_6$] (*Continued*)				
‖ to C axis		800	36.0	
single crystal, Brazil		900	37.0	
Corundum (Al_2O_3)		200	28	51
⊥ to C axis		400	28	
single crystal (African)		600	30	
		800	35	
		900	40	
‖ to C axis		200	40	51
single crystal		400	43.5	
		600	46	
		800	48.5	
		900	49.5	
Lithium fluoride		200	17.0	51
isotropic		400	23.5	
single crystal, synthetic		600	30.0	
		800	36.0	
		900	39.5	
Periclase, MgO		200	35.0	51
isotropic		400	46.0	
single crystal, synthetic		600	55.0	
		800	61.5	
		900	64.5	
Quartz (SiO_2)		−300	150	29*a*
⊥ to C axis		−150	77	
single crystal		0	52	
		150	36	51
		300	28	
		450	26	
		600	27	
		750	29	
		900	32	
‖ to C axis		−300	300	29*a*
single crystal				

TABLE 1 (*Continued*)

COEFFICIENTS OF THERMAL CONDUCTIVITY

F. Crystals (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Quartz (SiO_2) (*Continued*)				
‖ to C axis		−150	150	
single crystal		0	100	
		150	71	51
		300	50	
		450	42	
		600	39	
		750	36	
		900	36	
Sapphire (Al_2O_3)		200	22.0	51
⊥ to C axis		400	25.5	
single crystal		600	28.5	
synthetic		800	34.5	
		900	39.5	
Topaz, $Al(FOH)_2SiO_4$		200	34.5	51
‖ to C axis		400	35.5	
single crystal		600	36.0	
		800	40.0	
		900	48.0	
Tourmaline		200	21.5	51
⊥ to C axis		400	24.0	
single crystal, Brazil		600	26.0	
		800	28.0	
		900	29.0	
‖ to C axis		200	18.5	51
single crystal, Brazil		400	20.0	
		600	21.5	
		800	23.0	
		900	24.0	
Zircon, ZrO_2SiO_2		200	18.5	51
⊥ to C axis		400	26.0	
single crystal, domestic		600	31.5	
		800	35.5	
		900	36.0	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

G. Pure Metals

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Aluminum	169	32	1400	45
Antimony	417	32	128	45
Bismuth	610	32	58	45
Cadmium	540	32	644	45
Copper	558	32	2680	45
Gold	1210	32	2040	45
Iron	492	32	428	45
Lead	710	32	243	45
Magnesium	109	32	1070	45
Mercury	887	32	58	45
Nickel	557	32	405	45
Platinum	1340	32	480	45
Silver	660	32	2890	45
Steel, 1% carbon	....	32	335	45
Tin, white	457	32	454	45
Tungsten	1210	32	1100	45
Zinc	446	32	781	45

Relatively small changes in K with temperature. In many instances K decreases with rising temperature. See International Critical Tables (45) and current literature for alloys.

H. Miscellaneous

Earth and water				
frozen				
loosely packed, 42% water	108	0	7.4	113
tightly packed, 24% water	116	0	10.0	113
Glass, lead	...	60	4.2	45
Pyrex	...	200	7.1	51
		400	11.2	
		600	15.1	
		800	19.1	
		1000	23.0	
Schott, borosilicate crown	...	−300	3.6	45
		−150	6.6	

TABLE 1 (*Continued*)

Coefficients of Thermal Conductivity

H. Miscellaneous (*Continued*)

Material	Bulk Density, lb, ft^{-3}	Mean Temperature, °F	K	Reference
Glass (*Continued*)				
Schott, borosilicate crown		0	8.0	
		200	9.3	
soda	...	68	5.0	45
		212	5.3	
soda lime	...	200	7.1	51
		400	10.4	
		600	13.5	
		800	16.7	
Silica, fused	...	200	10.0	51
		600	16.5	
		1000	27.0	
		1400	41.5	
		1600	50.0	

TABLE 2

Specific Heat and Thermal Diffusivity

Insulating Materials

Material	Specific Heat, Btu, lb^{-1}, °F^{-1}	Mean Temperature, °F	Diffusivity, ft^2, hr^{-1}	Reference
Aluminum foil, crumpled	0.24	290	0.92	45
Asbestos fibers	0.25	...	......	
Asbestos paper, corrugated	0.245	147	0.0089	117
Cellulose fibers	0.32	...	......	45
Charcoal	0.25	77	......	33
Cork, baked	0.43	77	......	33
granulated	0.44	77	......	33
natural	0.419	124	......	49

TABLE 2 (*Continued*)

SPECIFIC HEAT AND THERMAL DIFFUSIVITY

Insulating Materials (*Continued*)

Material	Specific Heat, Btu, lb^{-1}, $°F^{-1}$	Mean Temperature, °F	Diffusivity, ft^2, hr^{-1}	Reference
Corkboard	0.433	91	......	49
	0.417	150	0.0068	117
	0.204	−115	0.0121	109
	0.292	−19	0.0099	
	0.392	109	0.0083	
Diatomaceous earth	0.21	77	......	33
Fiberglas board	0.129	−117	0.0097	109
	0.192	−22	0.0084	
	0.236	111	0.0097	
Glass block, expanded, Foamglas	0.132	−117	0.0227	109
	0.157	−20	0.0205	
	0.179	112	0.0190	
Glass wool	0.157	...	......	45
	0.210	150	0.036	117
	0.233	285	......	
	0.275	569	......	
	0.279	624	......	
with binder	0.196	150	0.011	117
	0.231	288	......	
Hairfelt	0.334	148	0.0061	117
High-temperature block insulation	0.203	149	0.011	117
	0.227	290	......	
	0.263	569	......	
	0.269	710	......	
Kapoc fiber	0.320	65	......	49
Lead slag wool	0.178	150	0.017	117
	0.198	286	......	
	0.235	565	......	
	0.235	718	......	
85% Magnesia	0.276	150	0.0086	117
	0.283	279	......	

TABLE 2 (*Continued*)

SPECIFIC HEAT AND THERMAL DIFFUSIVITY

Insulating Materials (Continued)

Material	Specific Heat, Btu, lb^{-1}, $°F^{-1}$	Mean Temperature, °F	Diffusivity, ft^2, hr^{-1}	Reference
Mineral-wool board with binder, Rock Cork	0.247	150	0.0066	117
	0.095	−120	0.0116	109
	0.156	−23	0.0085	
	0.190	111	0.0092	
Paper, expanding blanket, Kimsul	0.349	148	0.046	117
Redwood bark, shredded, Palco Bark	0.172	−127	0.0206	109
	0.215	−19	0.0203	
	0.246	109	0.0268	
Rock wool	0.201	149	0.019	117
	0.212	271	......	
	0.250	575	......	
	0.250	652	......	
Rubber board, expanded, Rubatex	0.152	−125	0.0194	109
	0.198	−19	0.0177	
	0.273	111	0.0139	
Silica aerogel, Santocel	0.205	147	0.020	117
	0.222	271	......	
	0.267	571	......	
	0.274	630	......	
Silk	0.33	...	......	45
Slag wool	0.17	77	......	33
Vegetable fiberboard, Celotex	0.171	−116	0.0084	109
	0.217	−21	0.0076	
	0.279	109	0.0070	
Vermiculite, expanded mica, Zonolite	0.205	149	0.025	117
	0.236	290	......	
Wood fiber blanket, Balsam Wool	0.330	150	0.023	117

TABLE 2 (*Continued*)

SPECIFIC HEAT AND THERMAL DIFFUSIVITY

Insulating Materials (Continued)

Material	Specific Heat, Btu, lb^{-1}, $°F^{-1}$	Mean Temperature, °F	Diffusivity, ft^2, hr^{-1}	Reference
Wood fiberboard	0.341	148	0.0055	117
Wool	0.325	...	......	45

TABLE 3

EMISSIVITY AND REFLECTIVITY OF VARIOUS SUBSTANCES

As stated in Chapter Two, the emissivity of a surface at a given temperature is equal to the absorptivity of black-body radiation from a source at the same temperature and is equal to one minus the reflectivity of the surface when exposed to black-body radiation from a source at the same temperature. If a surface is irradiated by a source at a considerably higher temperature than that of the surface, the formula $e = a = 1 - r$ would not necessarily apply because the wave lengths of the radiation from this source would be much shorter than those of the radiation from the surface at a lower temperature. This fact is often neglected in assembling emissivity tables.

In the following table, the temperature t°F indicates the sample temperature if the emissivity is given, but it represents the temperature of the radiating source if values for reflectivity are shown. In this latter situation the sample is usually about room temperature.

In many instances the emissivity values vary considerably with the condition of the surface. A film of oxide on a bright metal surface may increase the emissivity greatly. Direct determinations of emissivity at high temperatures are meager. Many of the values given should be considered only approximate and should be used with discretion.

TABLE 3 (*Continued*)

EMISSIVITY AND REFLECTIVITY OF VARIOUS SUBSTANCES

A. Metals

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Aluminum, polished	100		0.96	23
	500		0.95	
	1000		0.92	
	2000		0.83	
	5000		0.74	
	100	0.03–0.05		88, 62, and 113
	500	0.04		
	1000	0.06		
Aluminum foil, embossed	100	0.05		113
Aluminum oxidized at 1110°F	100	0.11		80
	500	0.12		
	1000	0.18		
Aluminum plate, clean	100	0.07		88, 113
gray and dull	100	0.086		113
Bismuth	176	0.34		88
Chromium, polished	100		0.92	23
	500		0.83	
	1000		0.74	
	2000		0.64	
	5000		0.57	
	Solar		0.51	
	100	0.06		113
	300	0.06		88
Copper, black oxidized	100	0.78		88
etched or scratched	100	0.09		88
mat	100	0.22		98
polished	100		0.96	23
	500		0.95	
	1000		0.92	
	2000		0.83	
	5000		0.74	

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

A. Metals (*Continued*)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Gold, polished	100	0.02		16
	500	0.02		
	1000	0.03		
	2000	0.03		
	1000	0.03		89
unpolished	500	0.02		89
Graphite	100		0.59	23
	500		0.51	
	1000		0.46	
	2000		0.36	
	5000		0.27	
	Solar		0.22	
Iron, cast	100	0.21		80
freshly turned	100	0.44		88
oxidized	100	0.63		80
	500	0.66		
	1000	0.76		
electrolytic deposit	100	0.05		89
	500	0.07		
freshly rubbed with emery	100	0.24		88
liquid	2000	0.015		99
	2000	0.045		104
polished	100		0.94	23
	500		0.92	
	1000		0.87	
	2000		0.78	
	5000		0.65	
	Solar		0.55	
rough oxide layer	100	0.81		88
sheet, red with rust	68	0.69		88

TABLE 3 (*Continued*)

EMISSIVITY AND REFLECTIVITY OF VARIOUS SUBSTANCES

A. Metals (*Continued*)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Lead, gray oxidized	100	0.28		88
oxidized	100	0.43		80
polished	100	0.06		89
	500	0.08		
rough	100	0.43		98
Magnesium, polished	100		0.93	23
	500		0.87	
	1000		0.82	
	2000		0.77	
	5000		0.74	
	Solar		0.70	
Molybdenum, filament	1000	0.08		123
	2000	0.14		
	5000	0.29		
polished	100		0.94	23
	500		0.92	
	1000		0.89	
	2000		0.82	
	5000		0.57	
	Solar		0.45	
Nickel, electrolytic	100		0.96	36
	500		0.94	
	1000		0.90	
	2000		0.84	
	5000		0.72	
	Solar		0.60	
mat	100	0.11		88
oxidized	100	0.31		80
	500	0.46		
polished	100	0.06		88
	500	0.07		89
	1000	0.10		

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

A. *Metals* (*Continued*)

Substance	$t°F$	Total Normal Emissivity	Reflectivity	Reference
Platinum, polished	100		0.95	36
	Solar		0.50	
	500	0.06		89
	1000	0.10		
	500	0.07		80
	1000	0.10		
	2000	0.18		
Platinum black	100		0.07	23
	500		0.04	
	1000		0.03	
	2000		0.03	
	5000		0.03	
	Solar		0.03	
Silver, polished	100		0.99	23
	500		0.98	
	1000		0.97	
	2000		0.97	
	5000		0.96	
	Solar		0.89	
	500	0.02		89
	1000	0.03		
	100	0.02		80
	500	0.02		
	1000	0.03		
Steel, oxidized	100	0.79		80
	500	0.79		
	1000	0.79		
polished	100		0.93	23
	500		0.90	
	1000		0.86	
	2000		0.77	
	5000		0.63	
	Solar		0.55	
	100	0.09		31

TABLE 3 (*Continued*)

EMISSIVITY AND REFLECTIVITY OF VARIOUS SUBSTANCES

A. Metals (*Continued*)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Tin, polished	200	0.05		62
Tungsten, filament	100	0.025		123
	500	0.045		
	1000	0.075		
	2000	0.186		
	5000	0.335		
Zinc, galvanized iron	200	0.07		62
dull	200	0.24		113
oxidized	500	0.11		80
plate, dull	200	0.05		113
polished	100		0.98	23
	500		0.97	
	1000		0.96	
	2000		0.94	
	5000		0.50	
	Solar		0.54	
	500	0.05		89
polished oxidized	100	0.28		88

B. Alloys

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Brass, freshly rubbed with coarse emery	100	0.20		88
oxidized at 1110°F	100	0.61		80
polished	100	0.05		88
rough rolled	100	0.07		88
Duralumin	Solar		0.47	23
Everdur, dull	200	0.11		113
Monel metal, polished	1000		0.90	23
	2000		0.84	

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

B. Alloys (*Continued*)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Monel metal, polished (*Continued*)	5000		0.71	
	Solar		0.57	
smooth, not polished	200	0.16		113
Nickelin, gray oxidized	100	0.26		88
Speculum metal, polished	100		0.92	36
	500		0.89	
	1000		0.87	
	2000		0.81	
	5000		0.70	
	Solar		0.61	
Stainless steel	200	0.22		113
Zin-o-lyte, dull	200	0.05		113

C. Oxides and Refractories

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Aluminum oxide refractory, R.A.98 Alundum	800	0.45		14
	1200	0.37		
	1600	0.31		
	2000	0.28		
	2400	0.34		
	2700	0.37		
Copper oxide, black	100		0.13	23
	500		0.17	
	1000		0.23	
Iron oxide	1000	0.85		16
	2000	0.88		
red	100		0.04	23
	1000		0.33	
	5000		0.41	
	Solar		0.26	
Kaolin insulating brick	800	0.80		14
	1200	0.61		

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

C. Oxides and Refractories (*Continued*)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Kaolin insulating brick (*Continued*)	1600	0.49		
	2000	0.48		
	2400	0.50		
	2550	0.50		
⅛-in.-dense Kaolin coating on Kaolin insulating brick	800	0.44		14
	1200	0.32		
	1600	0.25		
	2000	0.24		
	2400	0.24		
	2550	0.26		
Magnesium oxide refractory, 98% MgO	800	0.56		14
	1200	0.38		
	1600	0.33		
	2000	0.32		
	2400	0.35		
	2550	0.37		
Nickel oxide	500	0.39		80
	1000	0.45		
	1000	0.50		15
	2000	0.81		
Silica, translucent, 3/16 in. thick over Kaolin base	800	0.98		14
	1200	0.80		
	1600	0.70		
	2000	0.68		
	2400	0.67		
	2550	0.65		
Silicon carbide refractory, Crystolon	1000	0.95		14
	1200	0.93		
	1600	0.92		
	2000	0.90		
	2400	0.88		
	2550	0.85		

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

C. Oxides and Refractories (Continued)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Zirconia refractory, 98% Zirconia	800	0.74		14
	1200	0.44		
	1600	0.33		
	2000	0.31		
	2400	0.25		
	2550	0.22		

D. Miscellaneous Materials

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Aluminum paint	100	0.27–0.69		62, 88, and others
Aluminum powder with binder on paper	100	0.10–0.60		113
Asbestos cloth	200	0.90		62
Asbestos paper	200	0.93		62
Asphalt pavement, dust free	Solar		0.07	23
Concrete pavement	Solar		0.17	23
Enamel, white	100	0.92		88
Glass, polished	100	0.90		98
	100	0.95		88
Ice	32	0.96		88
wet	32	0.98		113
Lampblack	100	0.95		38
	750	0.95		
Lacquer, black	100	0.80		38
	750	0.95		
flat black	100	0.96		38
	750	0.98		
Marble, polished	100	0.94		88
Mica	200	0.84		113

TABLE 3 (*Continued*)

EMISSIVITY AND REFLECTIVITY OF VARIOUS SUBSTANCES

D. Miscellaneous Materials (Continued)

Substance	t°F	Total Normal Emissivity	Reflectivity	Reference
Oil on polished steel				
very thin	100	0.06		88
0.001 in. thick	100	0.22		
0.002 in. thick	100	0.45		
0.004 in. thick	100	0.65		
0.008 in. thick	100	0.81		
very thick	100	0.83		
Paper, white	100	0.94		88
Plaster	100	0.93		88
Plaster of Paris	100	0.92		88
Porcelain, glazed	100	0.94		88
Quartz, fused	100	0.94		88
Varnish, dark glossy	100	0.89		88
spirit	100	0.83		88
Water	100	0.96		88
Woods				
oak	100	0.92		88
spruce, sanded	200	0.82		62
walnut, sanded	200	0.83		62

TABLE 3 (*Continued*)

Emissivity and Reflectivity of Various Substances

E. Reflectivity of Pigments—Effect of Color

All values taken from Coblentz (23). Wave length of maximum energy of radiating source and corresponding temperature of black-body radiation are given.

Substance	8.8μ, 125°F	4.4μ, 750°F	0.95μ, 5000°F	0.60μ, Solar
Lampblack paint	0.04	0.03	0.03	0.03
Camphor soot	0.02	0.01	....	0.01
Acetylene soot	0.01	0.01	0.01	0.01
Platinum black	0.09	0.05	0.03	0.02
Blue (CO_2O_3)	0.13	0.14	0.03	0.03
Black (CuO)	....	0.15	0.24	....
Red (Fe_2O_3)	0.04	0.30	0.41	0.26
Green (Cu_2O_3)	0.05	0.33	0.45	0.27
Yellow (PbO)	0.26	0.51	....	0.52
($PbCrO_4$)	0.05	0.41	....	0.70
White (Al_2O_3)	0.02	0.21	0.88	0.84
(Y_2O_3)	0.11	0.34	....	0.74
(ZnO)	0.03	0.09	0.86	0.82
(CaO)	0.04	0.22	....	0.85
($MgCO_3$)	0.04	0.11	0.89	0.85
(ZrO_2)	0.05	0.23	0.84	0.86
(ThO_2)	0.07	0.47	....	0.86
(MgO)	0.03	0.16	....	0.86
($PbCO_3$)	0.11	0.29	0.92	0.88

TABLE 4

Coefficients of Heat Transmittance for Building Walls. Calculated *U* Values

The values given in this table are taken mainly from *Technical Circular* 7, Federal Housing Administration (1947), and the A.S.H.V.E. Guide (1948), with some minor changes and additions. It should be clearly understood that these values are merely calculated values and should serve only as a guide and with the realization that the conductance values are based on "bone-dry" conditions, which are never met in practice. The conductance of some materials is affected much more by convection, humidity and condensation of water vapor than others (see Chapter Eight).

A guarded-box test on these walls would give more dependable values, and occasionally in these tables an actual test value will be given for comparison.

In calculating the transmittance values for building walls, it is simpler to add the resistance ($1/C$ or $1/f$) to heat flow of the various parts of the wall and then take the reciprocal of this sum as the coefficient of heat transmittance or U value.

The following example will illustrate the method for a building wall consisting of 8-in. brick, ¾-in. furring strips, aluminum foil, ¾-in. furring strips, and ½-in. plasterboard.

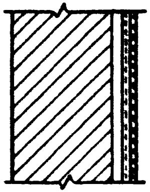
	Resistance
Outside surface coefficient, 15 mph wind, $f_o = 6.0$	$1/f_o = 0.17$
8-in. brick, $K = 5.0$, $C = 5/8 = 0.625$	$1/C = 1.60$
Two ¾-in. vertical air spaces, one side reflective, $C = 0.46/2 = 0.23$	$1/C = 4.34$
½-in. plasterboard, $K = 1.85$, $C = 1.85 \div 1/2 = 3.7$	$1/C = 0.27$
Inside surface coefficient, $f_i = 1.52$	$1/f_i = 0.66$
Total resistance	$= 7.04$

$U = 1/7.04 = 0.14$ Btu, hr^{-1}, ft^{-2}, $°F^{-1}$.

TABLE 4 (*Continued*)

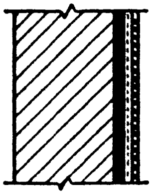
Coefficients of Heat Transmittance for Building Walls. Calculated U Values

A. Walls. 8-in. Brick



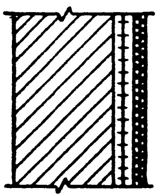
Outside surface, $1/f_o$	0.17
8″ common brick, 8 × 0.20	1.60
¾″ air space	0.83
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	3.68

$$U = \frac{1}{3.68} = 0.27$$



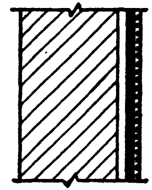
Outside surface, $1/f_o$	0.17
8″ common brick, 8 × 0.20	1.60
¾″ furred space, flexible insulation, 0.75 × 3.33	2.50
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	5.35

$$U = \frac{1}{5.35} = 0.19$$



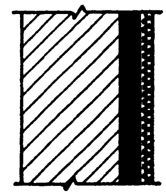
Outside surface, $1/f_o$	0.17
8″ common brick, 8 × 0.20	1.60
2—¾″ air spaces faced one side with reflective insulation, $e = 0.05$, 2 × 2.17	4.34
½″ plaster on ⅜″ gyplath	0.42
Inside surface	0.66
	7.19

$$U = \frac{1}{7.19} = 0.14$$



Outside surface, $1/f_o$	0.17
8″ common brick, 8 × 0.20	1.60
¾″ air space	0.83
½″ insulating board lath, 0.5 × 3.00	1.50
½″ plaster, 0.5 × 0.30	0.15
Inside surface, $1/f_i$	0.66
	4.91

$$U = \frac{1}{4.91} = 0.20$$



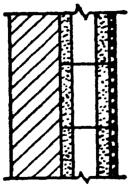
Outside surface, $1/f_o$	0.17
8″ common brick, 8 × 0.20	1.60
1⅝″ furred space, flexible insulation, 1.625 × 3.33	5.41
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	8.26

$$U = \frac{1}{8.26} = 0.12$$

TABLE 4 (*Continued*)

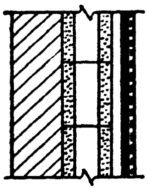
COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

B. Walls. 4-in. Brick, 4-in. Cinder Block



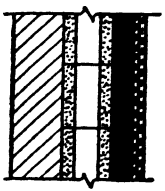
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
½″ plaster, 0.5 × 0.3	0.15
Inside surface, $1/f_i$	0.66
	2.78

$$U = \frac{1}{2.78} = 0.36$$



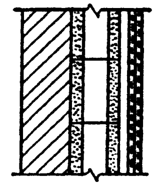
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
¾″ air space	0.83
½″ insulating board, 0.5 × 3.0	1.50
½″ plaster, 0.5 × 0.3	0.15
Inside surface, $1/f_i$	0.66
	5.11

$$U = \frac{1}{5.11} = 0.20$$



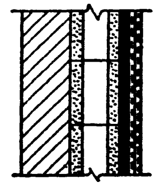
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
1⅝″ furred space, flexible insulation, 1.625 × 3.33	5.41
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	8.46

$$U = \frac{1}{8.46} = 0.12$$



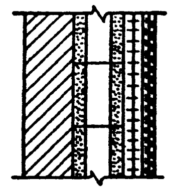
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
¾″ air space	0.83
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	3.88

$$U = \frac{1}{3.88} = 0.26$$



Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
¾″ furred space, flexible insulation, 0.75 × 3.33	2.50
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	5.55

$$U = \frac{1}{5.55} = 0.18$$



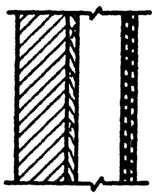
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
4″ cinder block	1.00
2—¾″ air spaces, faced one side with reflective insulation, e = 0.05, 2 × 2.17	4.34
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	7.39

$$U = \frac{1}{7.39} = 0.14$$

TABLE 4 (*Continued*)

Coefficients of Heat Transmittance for Building Walls. Calculated U Values

C. Walls. Brick Veneer, Wood Frame

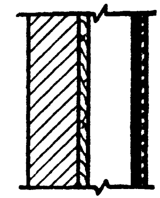


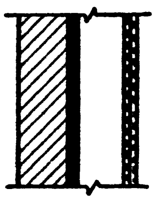
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
25/32″ fir sheathing with building paper	1.16
3 5/8″ air space	0.83
1/2″ plaster on 3/8″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	4.04

$$U = \frac{1}{4.04} = 0.25$$

Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
25/32″ fir sheathing with building paper	1.16
3 5/8″ air space	0.83
1/2″ insulation board, 0.5 × 3.00	1.50
1/2″ plaster, 0.5 × 0.30	0.15
Inside surface, $1/f_i$	0.66
	5.27

$$U = \frac{1}{5.27} = 0.19$$





Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
Building paper	0.18
3/4″ insulation board, 0.75 × 3.00	2.25
3 5/8″ air space	0.83
1/2″ plaster on 3/8″ gyplath	0.42
Inside surface	0.66
	5.31

$$U = \frac{1}{5.31} = 0.19$$

Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
25/32″ fir sheathing with building paper	1.16
1 5/8″ air space	0.83
2″ flexible insulation, 2 × 3.33	6.66
1/2″ plaster on 3/8″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	10.70

$$U = \frac{1}{10.70} = 0.09$$

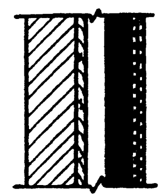
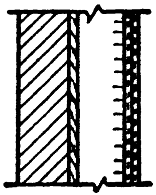


TABLE 4 (*Continued*)

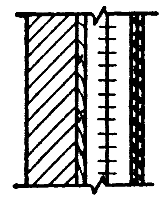
Coefficients of Heat Transmittance for Building Walls. Calculated U Values

C. Walls. Brick Veneer, Wood Frame (*Continued*)



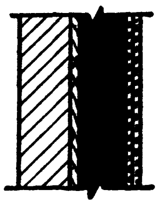
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
$^{25}/_{32}$″ fir sheathing with building paper	1.16
$3\frac{5}{8}$″ air space faced one side with reflective insulation, e = 0.05	2.17
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	5.38

$$U = \frac{1}{5.38} = 0.19$$



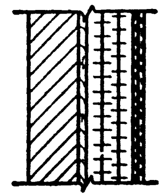
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
$^{25}/_{32}$″ fir sheathing with building paper	1.16
2—1.81″ air spaces one side reflective, e = 0.05, 2 × 2.17	4.34
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	7.55

$$U = \frac{1}{7.55} = 0.13$$



Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
$^{25}/_{32}$″ fir sheathing with building paper	1.16
$3\frac{5}{8}$″ flexible insulation, 3.625 × 3.33	12.07
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	15.28

$$U = \frac{1}{15.28} = 0.07$$



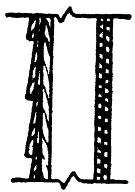
Outside surface, $1/f_o$	0.17
4″ common brick, 4 × 0.20	0.80
$^{25}/_{32}$″ fir sheathing with building paper	1.16
3—1.21″ air spaces faced one side with reflective insulation, e = 0.05, 3 × 2.17	6.51
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	9.72

$$U = \frac{1}{9.72} = 0.10$$

TABLE 4 (*Continued*)

COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

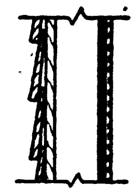
D. Walls. Wood Siding, Wood Frame



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" fir sheathing with building paper	1.16
$3^5/_8$" air space	0.83
½" plaster on ⅜" gyplath	0.42
Inside surface, $1/f_i$	0.66
	4.02

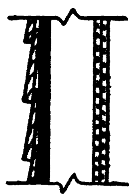
$$U = \frac{1}{4.02} = 0.25$$

Test value $U = 0.26$



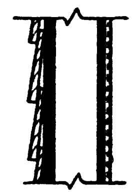
Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" fir sheathing with building paper	1.16
$3^5/_8$" air space	0.83
½" insulating board, 0.5 × 3.0	1.50
½" plaster, 0.5 × 0.3	0.15
Inside surface, $1/f_i$	0.66
	5.25

$$U = \frac{1}{5.25} = 0.19$$



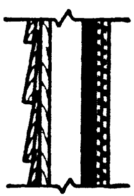
Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" rigid insulation, 0.78 × 3.0	2.34
$3^5/_8$" air space	0.83
½" plaster on ⅜" gyplath	0.42
Inside surface, $1/f_i$	0.66
	5.20

$$U = \frac{1}{5.20} = 0.19$$



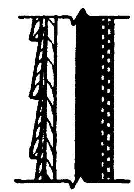
Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" rigid insulation, 0.78 × 3.0	2.34
$3^5/_8$" air space	0.83
½" insulating board, 0.5 × 3.0	1.50
½" plaster, 0.5 × 0.3	0.15
Inside surface, $1/f_i$	0.66
	6.43

$$U = \frac{1}{6.43} = 0.16$$



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" fir sheathing with building paper	1.16
$2^5/_8$" air space	0.83
1" flexible insulation, 1 × 3.33	3.33
½" plaster on ⅜" gyplath	0.42
Inside surface, $1/f_i$	0.66
	7.35

$$U = \frac{1}{7.35} = 0.14$$



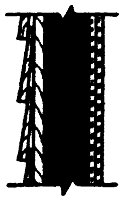
Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$" fir sheathing with building paper	1.16
$1^5/_8$" air space	0.83
2" flexible insulation, 2 × 3.33	6.66
½" plaster on ⅜" gyplath	0.42
Inside surface, $1/f_i$	0.66
	10.68

$$U = \frac{1}{10.68} = 0.09$$

TABLE 4 (*Continued*)

COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

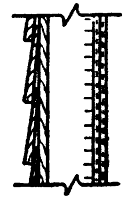
D. Walls. Wood Siding, Wood Frame (*Continued*)



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$″ fir sheathing with building paper	1.16
$3\frac{5}{8}$″ flexible insulation, 3.625 × 3.33	12.05
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	15.24

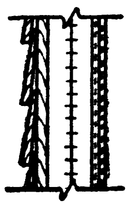
$$U = \frac{1}{15.24} = 0.07$$

Test value $U = 0.08$



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$″ fir sheathing with building paper	1.16
$3\frac{5}{8}$″ air space, one side reflective, $e = 0.05$	2.17
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	5.36

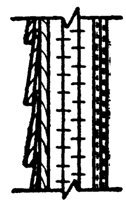
$$U = \frac{1}{5.36} = 0.19$$



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$″ fir sheathing with building paper	1.16
2—1.81″ air spaces, one side reflective, $e = 0.05$, 2 × 2.17	4.34
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	7.53

$$U = \frac{1}{7.53} = 0.13$$

Test value $U = 0.12$



Outside surface, $1/f_o$	0.17
Y.P. lap siding	0.78
$^{25}/_{32}$″ fir sheathing with building paper	1.16
3—1.21″ air spaces, one side reflective, $e = 0.05$, 3 × 2.17	6.51
½″ plaster on ⅜″ gyplath	0.42
Inside surface, $1/f_i$	0.66
	9.70

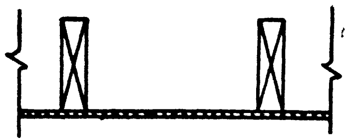
$$U = \frac{1}{9.70} = 0.10$$

Test value $U = 0.08$

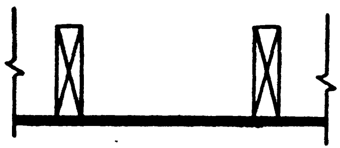
TABLE 4 (*Continued*)

COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

E. Ceilings. Ventilated Space between Ceiling and Roof



	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
3/8" gypsum board, 0.375 × 0.71	0.27	0.27
Inside surface, $1/f_i$	0.51	0.83
	1.29	1.93
	$U = \frac{1}{1.29} = 0.78$	$\frac{1}{1.93} = 0.52$
	Test value $U = 0.77$	



	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
1/2" insulating board, 0.5 × 3.0	1.50	1.50
Inside surface, $1/f_i$	0.51	0.83
	2.52	3.16
	$U = \frac{1}{2.52} = 0.40$	$\frac{1}{3.16} = 0.32$

	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
2" flexible insulation, 2 × 3.33	6.66	6.66
1/2" plaster on 3/8" gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	8.10	8.74
	$U = \frac{1}{8.10} = 0.12$	$\frac{1}{8.74} = 0.11$
	Test value $U = 0.14$	

TABLE 4 (*Continued*)

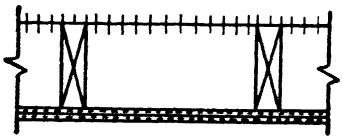
COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

E. Ceilings. Ventilated Space between Ceiling and Roof (Continued)

	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
$3\frac{5}{8}''$ flexible insulation, 3.625 × 3.33	12.05	12.05
$\frac{1}{2}''$ plaster on $\frac{3}{8}''$ gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	13.49	14.13

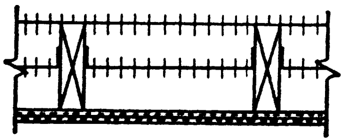
$$U = \frac{1}{13.49} = 0.07 \qquad \frac{1}{14.13} = 0.07$$

Test value U = 0.11



	Heat Flow Up	Heat Flow Down
Attic surface, reflective, $e = 0.05$, $1/f_o$	0.86	2.28
$7\frac{1}{2}''$ air space, one side reflective, $e = 0.05$	1.85	5.00
$\frac{1}{2}''$ plaster on $\frac{3}{8}''$ gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	3.64	8.53

$$U = \frac{1}{3.64} = 0.27 \qquad \frac{1}{8.53} = 0.12$$



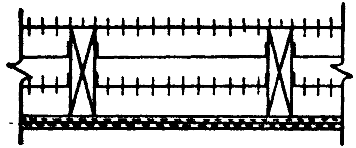
	Heat Flow Up	Heat Flow Down
Attic surface, reflective, $e = 0.05$, $1/f_o$	0.86	2.28
2—$3\frac{5}{8}''$ air spaces, one side reflective, $e = 0.05$	3.70	10.00
$\frac{1}{2}''$ plaster on $\frac{3}{8}''$ gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	5.49	13.53

$$U = \frac{1}{5.49} = 0.18 \qquad \frac{1}{13.53} = 0.07$$

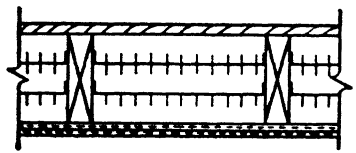
TABLE 4 (*Continued*)

Coefficients of Heat Transmittance for Building Walls. Calculated U Values

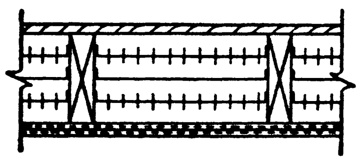
E. Ceilings. Ventilated Space between Ceiling and Roof (Continued)



	Heat Flow Up	Heat Flow Down
Attic surface, reflective, $e = 0.05$, $1/f_o$	0.86	2.28
3—2½" air spaces, one side reflective, $e = 0.05$	5.55	15.00
½" plaster on ⅜" gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	7.34	18.53
	$U = \frac{1}{7.34} = 0.14$	$\frac{1}{18.53} = 0.05$



	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
25/32" Y.P. flooring, 0.78 × 1.25	0.98	0.98
3—2½" air spaces, one side reflective, $e = 0.05$	5.55	15.00
½" plaster on ⅜" gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	7.97	18.06
	$U = \frac{1}{7.97} = 0.13$	$\frac{1}{18.06} = 0.06$

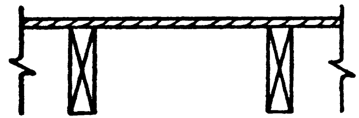


	Heat Flow Up	Heat Flow Down
Attic surface, $1/f_o$	0.51	0.83
25/32" Y.P. flooring, 0.78 × 1.25	0.98	0.98
4—1.9" air spaces, one side reflective, $e = 0.05$	7.40	20.00
½" plaster on ⅜" gyplath	0.42	0.42
Inside surface, $1/f_i$	0.51	0.83
	9.82	23.06
	$U = \frac{1}{9.82} = 0.10$	$\frac{1}{23.06} = 0.04$

TABLE 4 (*Continued*)

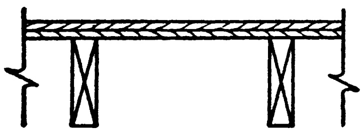
Coefficients of Heat Transmittance for Building Walls. Calculated U Values

F. Floors. Ventilated Space under Floor



	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813 × 0.87	0.71	0.71
Building paper	0.18	0.18
Under floor surface, $1/f_o$	0.51	0.83
	1.91	2.55

$$U = \frac{1}{1.91} = 0.52 \qquad \frac{1}{2.55} = 0.39$$



	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813 × 0.87	0.71	0.71
Building paper	0.18	0.18
$\frac{25}{32}''$ Y.P. subfloor, 0.78 × 1.25	0.98	0.98
Underfloor surface, $1/f_o$	0.51	0.83
	2.89	3.53

$$U = \frac{1}{2.89} = 0.35 \qquad \frac{1}{3.53} = 0.28$$

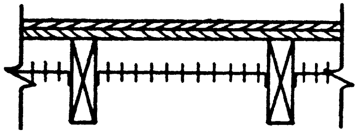
	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813 × 0.87	0.71	0.71
Building paper	0.18	0.18
$\frac{25}{32}''$ Y.P. subfloor, 0.78 × 1.25	0.98	0.98
2″ air space	0.75	1.06
2″ flexible insulation, 2 × 3.33	6.66	6.66
Underfloor surface, $1/f_o$	0.51	0.83
	10.30	11.25

$$U = \frac{1}{10.30} = 0.10 \qquad \frac{1}{11.25} = 0.09$$

TABLE 4 (*Continued*)

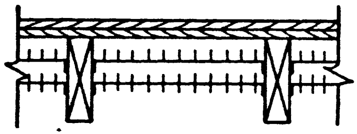
COEFFICIENTS OF HEAT TRANSMITTANCE FOR BUILDING WALLS. CALCULATED U VALUES

F. Floors. Ventilated Space under Floor (*Continued*)



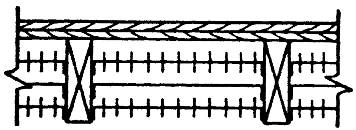
	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813×0.87	0.71	0.71
Building paper	0.18	0.18
$2\frac{5}{32}''$ Y.P. subfloor, 0.78×1.25	0.98	0.98
1—3″ air space, one side reflective, $e = 0.05$	1.85	5.00
Underfloor surface, reflective, $e = 0.05$	0.86	2.28
	5.09	9.98

$$U = \frac{1}{5.09} = 0.20 \qquad \frac{1}{9.98} = 0.10$$



	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813×0.87	0.71	0.71
Building paper	0.18	0.18
$2\frac{5}{32}''$ Y.P. subfloor, 0.78×1.25	0.98	0.98
2—2″ air spaces, one side reflective, $e = 0.05$	3.70	10.00
Underfloor surface, reflective, $e = 0.05$	0.86	2.28
	6.94	14.98

$$U = \frac{1}{6.94} = 0.14 \qquad \frac{1}{14.98} = 0.07$$



	Heat Flow Up	Heat Flow Down
Inside surface, $1/f_i$	0.51	0.83
$1\frac{3}{16}''$ hardwood floor, 0.813×0.87	0.71	0.71
Building paper	0.18	0.18
$2\frac{5}{32}''$ Y.P. subfloor, 0.78×1.25	0.98	0.98
3—2″ air spaces, one side reflective, $e = 0.05$	5.55	15.00
Underfloor surface, reflective, $e = 0.05$	0.86	2.28
	8.79	19.98

$$U = \frac{1}{8.79} = 0.11 \qquad \frac{1}{19.98} = 0.05$$

TABLE 5

Water-Vapor Permeability of Various Materials

It will be noted upon scanning the following table that there is considerable variance in the results of the different investigators. This is probably due to the nonstandardization of the method as well as the nonuniformity of samples. The numerical results should be used with caution, but relative results of the same investigator should be helpful in the choice of a vapor seal.

Permeability of Various Materials to Water Vapor. From Woolley, Nat. Bur. Standards (122)

Babbitt (6, 7, and 8)

Material	Thickness	Permeability, Grains, hr^{-1}, ft^{-2}, lb^{-1}, $in.^2$
Fiberboard	0.492 in.	60.6
	1.06	37
	0.803	43.4
	0.599	56.4
	0.405	74.5
	0.201	133.3
1 surface asphalt, rolled	0.492	8.0
1 surface asphalt, dipped	0.63	17.3
laminated, 2 samples cemented with asphalt	0.985	2.74
laminated, 6 layers with 5 layers of asphalt	0.527	0.23
Wood, spruce	0.563	3.48
	0.480	4.03
	0.405	3.94
	0.323	4.93
	0.232	7.24
	0.161	10.35
pine	0.80	1.88
	0.645	2.52
	0.496	3.45
	0.315	5.55
	0.169	9.65
pine A	0.508	6.47
1 coat aluminum paint		3.42
2 coats aluminum paint		0.92
3 coats aluminum paint		0.71
pine B	0.508	6.68
1 coat aluminum paint		3.85
2 coats aluminum paint		1.95
3 coats aluminum paint		1.53

TABLE 5 (*Continued*)

WATER-VAPOR PERMEABILITY OF VARIOUS MATERIALS

Material	Thickness	Permeability, Grains, hr^{-1}, ft^{-2}, lb^{-1}, $in.^2$
Kraft paper, 1 sheet	0.00394	168
2 sheets		107
3 sheets		80
4 sheets		63.6
5 sheets		53.5
5 sheets		65.3
5 sheets		61.6
7 sheets		45.5
7 sheets		38.3
8 sheets		38.3
8 sheets		33.1
Black vulcanized rubber, hardness 40	0.0791	0.185
Plasticized rubber hydrochloride	0.00581	0.382
30-30-30 Paper A	0.0071	1.83
30-30-30 Paper B	0.0071	1.79
Duplex Scutan 6-6, asphalt between 2 sheets of kraft	0.0071	0.946
Scutan 0-14, kraft infused with asphalt on 1 surface, A	0.0071	8.6
B	0.0071	15.97
Scutan 14, kraft infused with asphalt, A	0.0071	13.9
B	0.0071	15.8
Black building paper, shiny, infused with asphalt	0.0173	0.376
Asphalt felt, 15-lb felt building paper with soft, dull appearance	0.0319	13.5
Pressed corkboard, A	0.905	4.75
B	0.985	5.42
Plaster	1.34	27.1
Plasterboard, plaster between sheets of heavy paper	0.37	70.2
Masonite Presdwood	0.13	21.7
tempered	0.13	9.76
5 layers		6.25
7 layers		4.9
Teesdale (97)		
Foil-surfaced reflective insulation, double-faced		0.172-0.236
Roll roofing, smooth, 40 to 65 lb per roll of 108 sq ft		0.263-0.348

TABLE 5 (*Continued*)

WATER-VAPOR PERMEABILITY OF VARIOUS MATERIALS

Material	Thickness	Permeability, Grains, hr^{-1}, ft^{-2}, lb^{-1}, $in.^2$
Asphalt-impregnated and surface-coated sheathing paper		
50 lb per 500 sq ft		0.433–1.57
35 lb per 500 sq ft		0.348–4.19
Duplex or laminated papers, 30–30–30		2.80 –5.24
30–60–30		1.05 –1.75
Duplex papers, reinforced		1.40 –4.19
coated with metal oxides		1.05 –2.63
Insulation backup paper, treated		1.75 –6.97
Gypsum lath with aluminum foil backing		0.173–0.785
Plaster, wood lath		22.4
3 coats lead and oil		7.5 –7.84
3 coats flat wall paint		8.72
2 coats aluminum paint		2.35
fiberboard or gypsum lath		40.2 –41.9
Slater's felt		10.5 –52.4
Plywood, Douglas fir, soybean glue, plain	0.25	8.72–13.1
2 coats asphalt paint		0.87
2 coats aluminum paint		2.63
5-ply Douglas fir	0.50	5.43– 5.59
3-ply Douglas fir, artificial resin glue	0.25	8.72–13.1
5-ply Douglas fir, artificial resin glue	0.50	5.59– 6.85
Insulating lath and sheathing, board type		52.3 –69.8
Insulating sheathing, surface-coated		6.17– 8.88
Compressed fiberboard	0.19	10.3
Insulating cork block	1.00	12.6
Blanket insulation between coated papers	0.5–1.00	3.90– 4.07
Mineral wool, unprotected	4.0	59.2
Forest Products Laboratory (101)		
Kraft paper		112
Plaster wall, no paint, plasterboard lath		41.7
wood lath		22.2
Slater's felt, best type		10.1
Duplex paper		2.78
Plaster wall, 2 coats aluminum paint, wood lath		2.43
Asphalt-coated paper, 35 lb per 500-sq ft roll		2.08
50 lb per 500-sq ft roll		1.04
Metal-coated paper		0.174

TABLE 5 (*Continued*)

Water-Vapor Permeability of Various Materials

Material	Thickness	Permeability, Grains, hr^{-1}, ft^{-2}, lb^{-1}, in. [2]
International Critical Tables (45)		
Still air	3.63	70.9
	1.00	257
Wray and Van Vorst (124)		
Wood, western yellow pine	0.25	1.8
Wood, 1 coat aluminum paint, Bakelite resin varnish		1.22
glycerol phthalate varnish		1.22
ester gum varnish		2.26
2 coats ordinary paint, linseed oil		3.48
Taylor, Hermann, and Kemp (95)		
Hydrocarbon wax	1.0	0.000052
Thiokol	1.0	0.00014
Gutta percha	1.0	0.00035
Hard rubber	1.0	0.00035
Para gutta	1.0	0.00042
Polystryene	1.0	0.00087
Asphalt sealing compound	1.0	0.00087
Phenol fiber	1.0	0.00148
Soft vulcanized rubber	1.0	0.00157
Benzyl cellulose	1.0	0.00226
Bakelite	1.0	0.0035
Waterproof cellulose film	1.0	0.062
Cellulose acetate	1.0	0.12
Miller (65)		
Plaster base and plaster	0.75	30
Vapor barrier (Kimberly Clark Corp. data)		1.65
Fir sheathing	0.75	6
Waterproof paper		100
Pine lap siding		10
Paint film		7
Celotex	0.75	25.5
Brick masonry	4	2.2
Martley (59)		
Wood, Scot pine	1.0	21.4
Engineering Experimental Station, Pennsylvania State College		
Rubatex	1.0	0.0

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