revised metric edition

Reflective Insulation and the Control of Thermal Environments.

ACI

Fibreglass Insulation

Foreword to Metric Edition

Architects and engineers are frequently inconvenienced by the gap between the limited technical data supplied by trade catalogues, and the text-books which cover the general theory. Much time may have to be spent to bring together the relevant technical data for the less common design problems. I should like to commend ACI Fibreglass for making available in this booklet the technical information relevant to the use of reflective insulation for opaque building elements and windows, and Mr. David Hassall for his work in writing this concise text.
This metric version of the book should prove to be an extremely valuable aid to designers and students now that we are entering the period of maximum activity for metric conversion in the Australian Construction Industry.
One may hope that other manufacturers will follow this excellent example.

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1. INTRODUCTION TO METRIC EDITION

Following the ready acceptance of this reference book in its original form by specifiers and users of SISALATION products both overseas and in Australia and its extensive use by students of the Science of Building in Universities, Institutes and Colleges it was deemed necessary to produce this metric edition. The metric treatment of the subjects is complete, with no reference to the Imperial System of units except for a useful set of selected conversion factors at the end of the book.

A new section has been added, namely "Insulation for Windows" which outlines methods for assessing solar heat flow through windows both without any treatment and with REFLECTO-SHIELD or any other type of window insulation.

This project has been carried out by our Technical Service Department working under the direction of Mr. D. Hassall, B.E., M.Bdg.Sc.—Product Development Manager—Building.

The purpose of this reference book on the use of reflective insulation is twofold, firstly to bring together in one text the data necessary to calculate the "U" values of structures incorporating reflective insulation and secondly to outline the need and method for measuring thermal environment within building structures. There are included a number of examples covering both the theoretical and practical aspects of heat transfer calculations.

Where possible, references have been given to identify the original source of information so that a particular aspect of the subject can be studied in greater detail if required.

Revised July 1977 by F. R. Richards, C.Eng., F.C.I.B.S., M.I.Mech.E., M.N.Z.I.E., Technical Consultant-Building St. Regis-ACI Pty. Ltd.

2. HEAT AND HUMAN COMFORT

Heat, in all its forms, or the lack of it, has had a profound effect on man and his evolution. He has had to contend with such extremes as the Ice Age and present day tropical living conditions.

At one end of the scale the human body shivers in an attempt to create more body warmth, and at the other sweating commences when the body produces the means for additional "evaporative" cooling of the skin. Both shivering and sweating are stress conditions that can be tolerated only for comparatively short periods. Optimum living conditions produce reactions somewhere in between, and many researchers have conducted subjective surveys involving numbers of people dressed in various types of clothing and performing differing amounts of work. Environments on either side of optimum cause discomfort, inefficiency, distress and finally collapse and death, so that the necessity for attaining optimum conditions has been proved from both economical and humanitarian points of view. The main variables which affect human comfort are:—

Dry Bulb Temperature

Wet Bulb Temperature (i.e. Relative Humidity)

Air Movement

Thermal Radiation from Hot Surfaces. (Also loss of heat from the body by radiation to cold surfaces.)

To a lesser extent certain other climatic factors affect human comfort, such as atmospheric pressure, ion concentration, etc., but they are outside the scope of this publication.

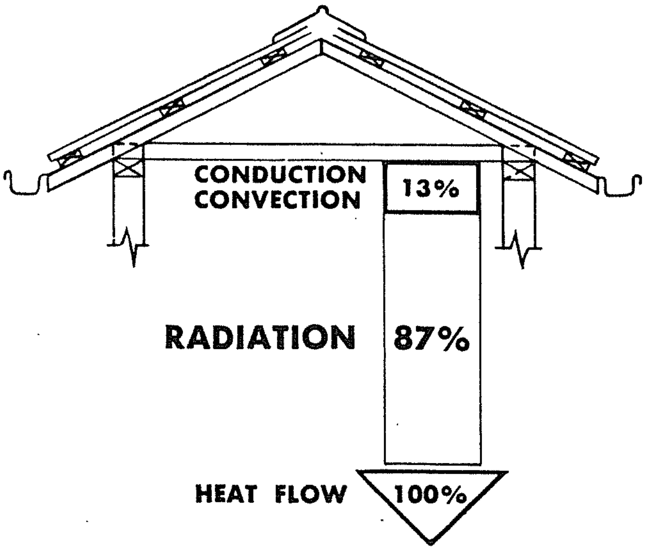
Many attempts have been made in the past to combine the various factors which affect environment into a simple **Comfort Index**[15]. The most important of these are covered in this text.

In **cold climates** man has had considerable experience in creating warmer conditions and it is usually acknowledged that average people feel comfortable within the "dry bulb" temperature range 18°C to 24°C with air speed at about 0·13 m/s.

To combat loss of body heat by radiation to cold surfaces, Rogers[11] has suggested that with an inside ambient of 24°C "superior comfort" can be achieved by ensuring that surfaces are at least 21°C. (See Fig. 6.1.)

In **hot climates** man has largely had to "put up with" the conditions encountered, and only recently has technology enabled him to remove heat and moisture from air so as to produce more comfortable environments. However, there are millions of people living in tropical areas around the circumference of the earth who must be housed but who cannot afford refrigerated air conditioning systems. Therefore examination and control of other factors affecting comfort, such as air movement, and radiation of heat from ceilings and walls are most important.

Excessive thermal radiation from the ceiling or other surfaces can be extremely uncomfortable, particularly in a hot environment. Radiation may increase the globe temperature by only a very few degrees. However, this instrument is an "integrating" device which gives an average, whereas the human body can receive the radiation all from one direction and can feel a "hot spot". Thus, radiation has an undesirable directional property which has so far not been allowed for in comfort indices.



PROPORTIONS OF HEAT FLOWING DOWNWARDS FROM CEILING
FIG. 2.1

As a practical example Fig. 2.1 illustrates that, of the total amount of heat entering a room from a normal domestic ceiling, approximately 87% is by radiation, hence the necessity for keeping the ceiling cooler by use of insulation. An additional measure is to use a ceiling material with a low emissivity (i.e. a low rate of emission of radiant heat).

As regards radiation from hot surfaces it is clear that a limit should be placed on the excess of ceiling temperature above ambient. Very little research has been done into the effects of excessive radiation on everyday living in warm to hot climates. Drysdale[12] found with some exploratory investigations (3 subjects only), that a rise of 2·8K in temperature of a radiant source panel was equivalent to a rise of 0·6K Dry Bulb temperature. For example, the threshold of discomfort of 30°C in a temperate climate, coupled with low air movement, will be reduced to 29·4°C with a ceiling temperature of 32·8°C.

Based on available references[12, 13, 14] it is not unreasonable to conclude that the discomfort effect of hot ceilings is actually greater than this, and that optimum thermal conditions are more likely to be achieved if the ceiling temperature is not more than 32°C with an indoor ambient of 30°C and equal to indoor ambient when the latter reaches the approximate head temperature of 31·7°C quoted by Billington[14]. (Beyond this point, increased air movement or air conditioning becomes essential in order to achieve optimum thermal conditions.)

The American Society of Heating, Refrigerating and Air Conditioning Engineers is sponsoring research into the effect of heated or cooled, walls and ceilings on comfort. The application of their findings will be a step forward in comfort design.

Long neglected in building design, thermal radiation is beginning to receive the attention it warrants.

3. THE NATURE OF HEAT

3.1 The Kinetic Theory

When a bar of iron is placed in a fire it gets "hot" and when removed from the fire and allowed to remain in the room air for a time, it becomes "cool". It is important to understand exactly what the fire does to the bar to give it the property of being "hot".

The question can be answered by an examination of the Kinetic Theory of Matter. Any material, such as the bar of iron, is made up of myriads of particles called molecules; a molecule being the smallest part of a material which can exist independently. These molecules have continuous movement relative to each other and when heat (which is a form of energy) is applied to the bar of iron, the molecules move more rapidly and the bar is said to be **"hotter"**. Alternatively if the bar is placed in a refrigerator and heat is removed from the bar, its molecules will slow down and move less rapidly and it is said to be **"colder"**.

The Kinetic Theory applies also to liquids and gases. It should be noted that most compounds have solid, liquid and gaseous stages depending on the speed of movement of their molecules.

For example, consider a block of ice. This is the **solid** state, with molecules vibrating to a limited degree, but more or less anchored relative to each other so that the block of ice retains its shape.

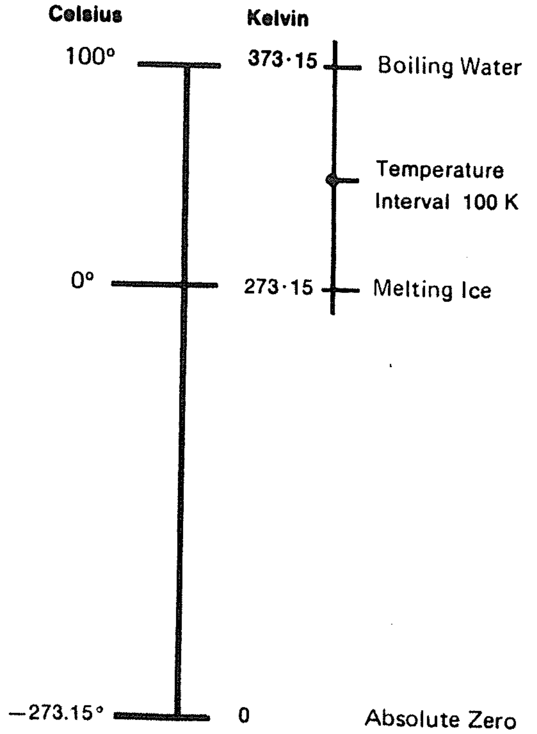
If heat is supplied to the ice the molecules begin to move more rapidly and break free from their anchored position and behave as a **liquid** (water). If still more heat is applied the molecules eventually attain such a speed that they begin to fly off into the atmosphere as a **gas** which is known as "water vapour".

3.2 The Difference Between Heat and Temperature

A bar of iron allowed to remain in a fire will eventually become red hot. In contrast a pin placed in the flame of a match will become red hot in a few seconds. It is not difficult to comprehend that although the two pieces of iron, the large bar and the small pin, are both at the same **temperature,** it has taken different quantities of heat to achieve the result. This example illustrates the concept that the temperature of a body indicates its "level of hotness" above a certain fixed datum.

3.3 Absolute Zero

Absolute zero is the temperature at which, theoretically, the previously mentioned molecular vibration stops altogether. In the **Celsius scale,** where 0°C is the temperature of melting ice and 100°C is the temperature of boiling water, absolute zero is 273·15 degrees below freezing. Unit temperature difference is written 1K.



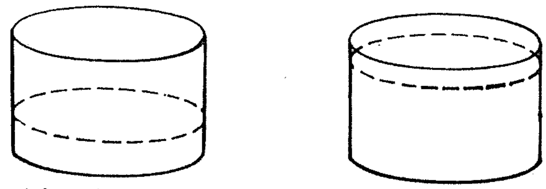
COMPARISON OF CELSIUS AND KELVIN TEMPERATURE SCALES

FIG. 3.1

3.4 Experiments to Demonstrate "Quantity" of Heat

Consider that two identical containers, A and B, are filled with "cold" water in the following proportions:

Container A = 1 kg of water (1 litre)
Container B = 2 kg of water (2 litres)



Container A — 1 kg water Container B — 2 kg water

FIG. 3.2

If these containers, so filled with water, are placed on identical gas jets and brought to the boil, the water in Container B will take approximately twice as long to heat as that in Container A. This is because Container B needs twice as much heat to come to the boil as does Container A.

3.5 Units of Heat and Heat Flow Rate

The unit of heat is the JOULE (J). The heat capacity of water at 15°C is 4185·8 Joules per kilogram for a 1K rise in temperature.

In the example above, suppose that the water placed in the containers had an initial temperature of 10°C and heat was added until the temperature of each was 20°C, i.e., a rise of 10K.

$$\text{Heat gain in Container A} = \text{Number of kg of water} \times \text{Temperature rise, K} \times 4185.8$$

$$= 1 \times 10 \times 4185.8$$

$$= \mathbf{41\,858\ J}$$

$$\text{Heat gain in Container B} = \text{Number of kg of water} \times \text{Temperature rise, K} \times 4185.8$$

$$= 2 \times 10 \times 4185.8$$

$$= \mathbf{83\,716\ J}$$

i.e., the heat gain of B was double the heat gain of A though its temperature increase was the same.

The unit of heat flow rate is the WATT (W). A WATT is the power used when work is done or energy is expended at the rate of one joule per second.

$$\text{i.e., } 1\ \text{W} = 1\ \text{J/s}$$

The size of this unit can be gauged by imagining the amount of heat generated by a 1 kW domestic radiator element.

3.6 Specific Heat Capacity

Originally the basic heat unit was defined in terms of raising a unit mass of water through one degree and specific heats were compared with that of water taken as unity. This was not satisfactory, since heat units were defined for different positions in the temperature scale by different authorities and specific heats vary with temperature. There is now an international agreement defining the joule as the basic heat unit which does not involve the use of water. The term now adopted includes the word capacity to distinguish it from the old term. It is now Specific Heat Capacity instead of as previously Specific Heat.

3.7 The Three Modes of Heat Transfer

A knowledge of the heat transfer processes is required to understand how to control heat flow through building structures.

There are three methods of heat transfer; namely

(a) Conduction
(b) Convection
(c) Radiation

(a) Conduction

If the end of an iron bar is placed in a fire, heat will be transferred along the bar until eventually the other end becomes hot to the touch. This is called **conduction** of heat.

Similarly the end of a piece of wood can be placed in the fire and it can be comfortably held until the end is on fire. It is evident therefore that wood is not as good a "conductor" of heat as is iron. All materials vary in their ability to conduct heat and those that conduct to the least extent are called "insulators". As a matter of interest gases and liquids **at rest** also conduct a certain amount of heat by transfer of Kinetic energy from molecule to molecule.

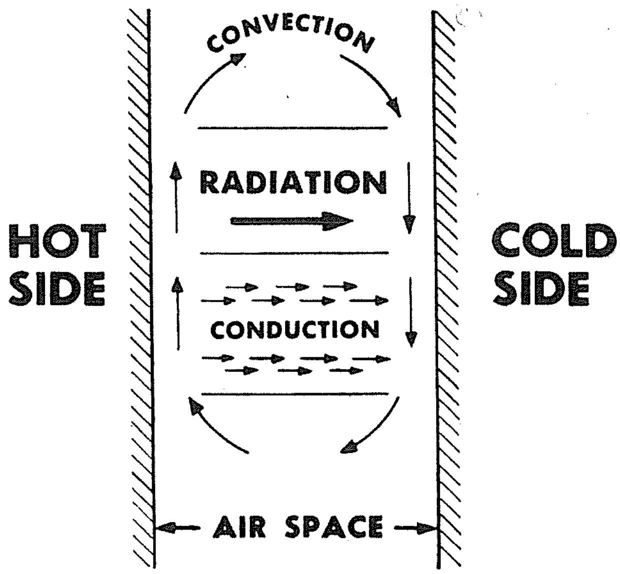


FIG. 3.3

(b) Convection

It is well known that hot air rises. This fact is demonstrated by studying a convector in a closed room. The hot air can be felt rising vertically upwards from the appliance, its place being taken by colder air from near the floor. An observer standing on a chair would be able to feel the accumulation of hot air near the ceiling. A similar thing happens in the case of liquids. Heated gases and liquids usually expand (i.e. become less dense), and rise, being displaced by denser, colder material at the lower level. Anyone who has swum in a still pool of water will realise that if heat is applied at the top of a liquid, in this case from the sun, convection will not take place. Heat is transferred downwards very slowly by conduction only and the deeper water can be quite cold.

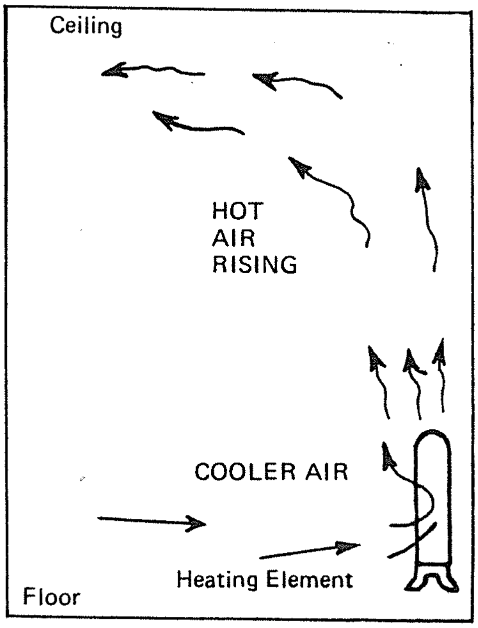


FIG. 3.4

From the foregoing it will be clear that heat is transferred from place to place in gases and liquids by convection.

(c) Radiation

The best example of this phenomenon is the radiation of heat from the sun. The sun emits electromagnetic waves which come to the earth through space (a virtual vacuum) at the speed of light (300 000 km/s). This emission from the sun contains electromagnetic waves of various wave lengths:—

(i) Ultra Violet: 0·01 to 0·39 μm (causes sunburn)
(ii) Visible Light: 0·39 to 0·76 μm (visible)
(iii) Infra-red: 0·76 to 100 μm (felt as warmth)

In a similar way a white hot bar of iron emits visible light plus radiant heat which can be felt from a distance. As the bar cools the colour changes to red —then darker and darker till it is black. But even though visible light is no longer emitted the heat radiated is still considerable and can serve as a warning that the bar has a high temperature. Radiant heat is independent of visible light, and depends on the **absolute temperature** of the emitting body. Heat radiates from one body to another through a vacuum or through air, and travels at the speed of light.

An examination of the curves of "black-body radiations" for varying temperatures shows that in the consideration of the heating and cooling of building interiors, radiations with wave lengths shorter than 5 μm are not significant, refer Fig. 3.5.

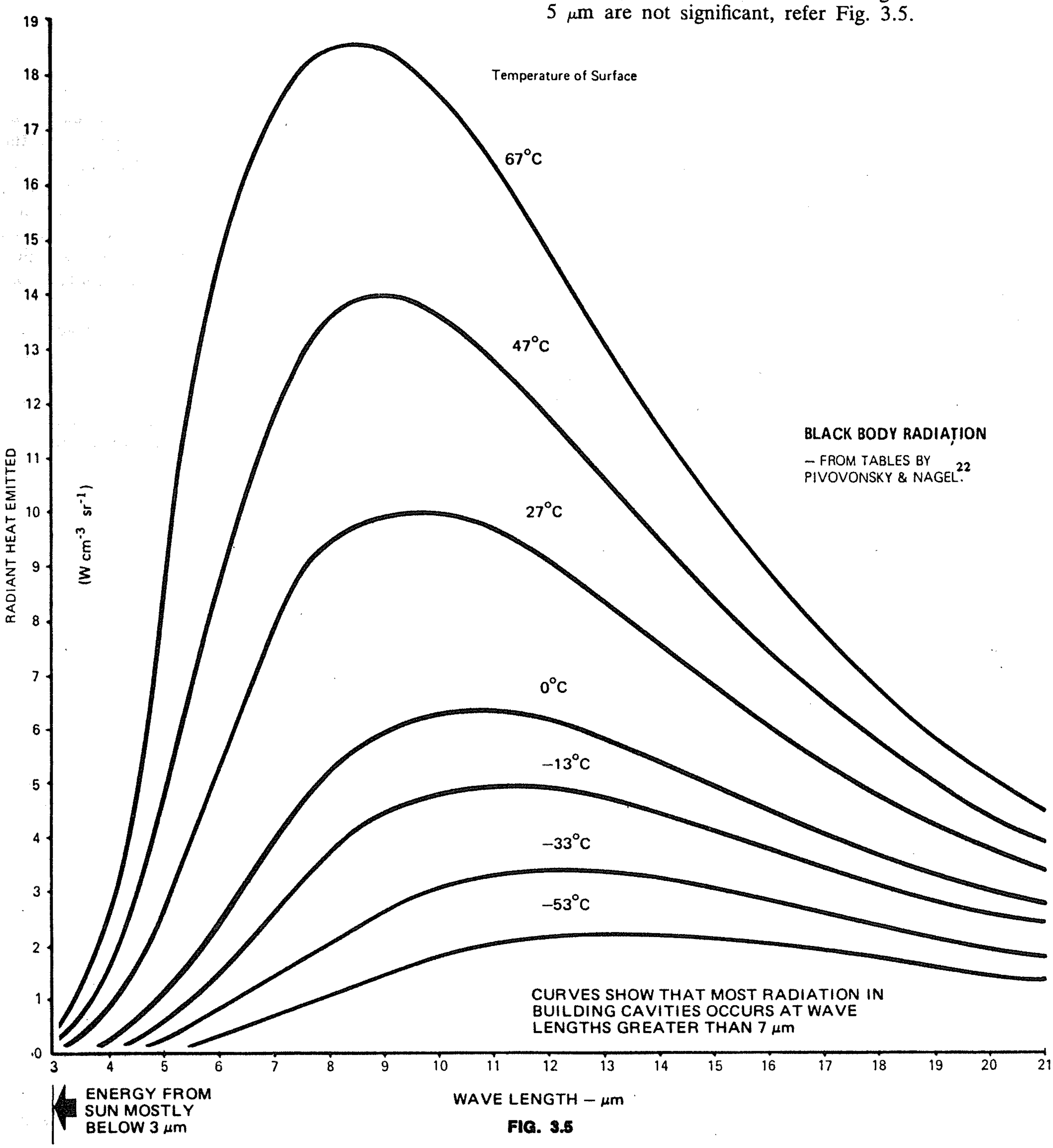


FIG. 3.5

4. INSULATION

It has been seen that some materials conduct heat more rapidly than others. Those that have a high rate of heat transference are called CONDUCTORS, whereas those that have a low rate of heat transference are called INSULATORS.

4.1 Thermal Conductivity (k)

(Refers to unit thickness)

The thermal **conductivity** (k) of a homogeneous material specifies its rate of transference of heat. If a material has a "k" value of 1, it means that a 1 m cube of the material will transmit heat at the rate of 1 watt for every degree of temperature difference between opposite faces. Its conductivity would be written as 1 W/(m.K), (Refer to Fig. 4.1) and this would be unit conductivity.

A conductivity value can not be given for an air space since its effect on heat flow is not directly proportional to its thickness. Variations in direction of heat flow, the position of the air space (i.e. vertical, horizontal, etc.), and such things as mean temperatures have differing effects. Refer to Table 4.1 for conductance and resistance values of air spaces.

4.2 Thermal Conductance (C)

(Refers to any thickness of a material or structural component such as a wall.)

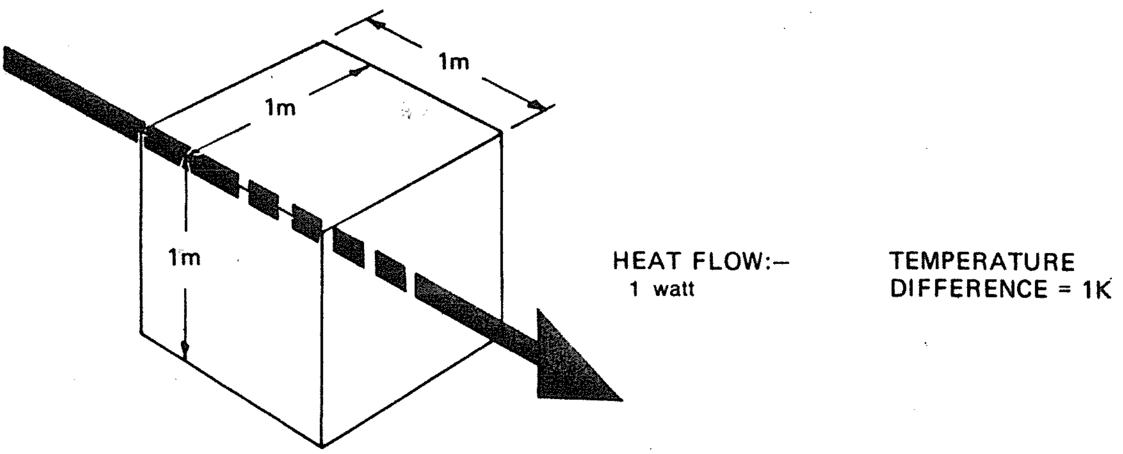
"THERMAL CONDUCTANCE IS THE THERMAL TRANSMISSION THROUGH UNIT AREA OF A STRUCTURAL COMPONENT OR OF A STRUCTURE (E.G. A WALL CONSISTING OF BRICKS, THERMAL INSULATION, CAVITIES, ETC.) PER UNIT TEMPERATURE DIFFERENCE BETWEEN THE HOT AND COLD FACES (UNIT; $W/(m^2.K)$)."

If the thickness in Fig. 4.1 were halved the rate at which heat would be transferred from one face to the other would also be halved. However since time is of unit duration in a conductance value the conductance will now be doubled to $2W/m^2K$. For a thickness other than 1 m or for a non-homogeneous structure the term CONDUCTANCE (C) is used. In such cases the thickness of the component or components making up the structure must be stated.

4.3 Heat Storage Capacity

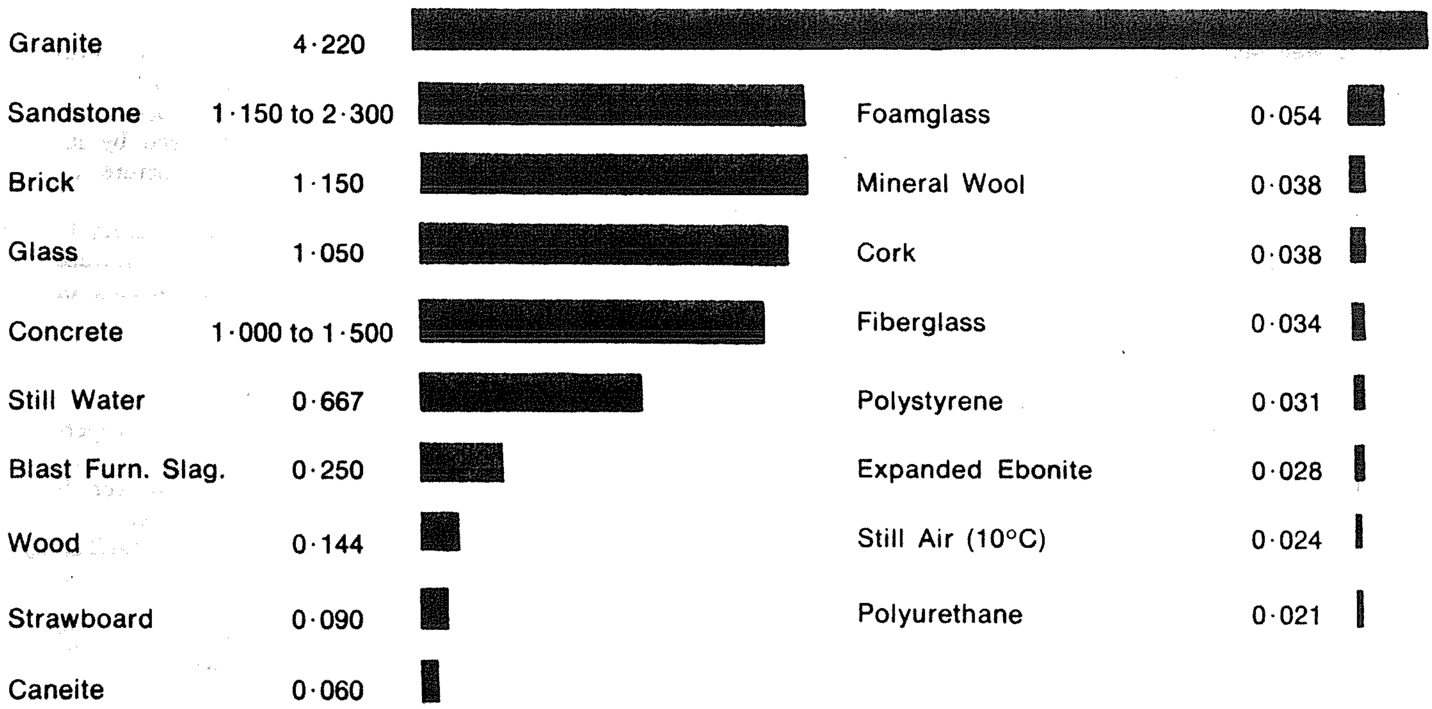
Where two insulating materials have the same "k" value, the one with the lowest heat capacity is the most desirable whenever intermittent heating and/or cooling is necessary. This is because less energy is required for heating or cooling the insulation. Similarly in warm humid climates with relatively small diurnal fluctuations in air temperature, the lighter insulation will cool down more quickly thus taking more rapid advantage of any drops in temperature.

On the other hand, in climates with large daily variations in outdoor air temperature and solar radiation intensity, there is merit in having an insulation which has a high heat capacity or "thermal inertia" thus adding to the flywheel effect of the constructional mass which tends to iron out peak differences in temperature.



ILLUSTRATING UNIT CONDUCTIVITY

FIG. 4.1



Note:—These values can vary depending on the density and moisture content of the material.

TYPICAL CONDUCTIVITIES ("k" VALUES) (W/(m.K))

FIG. 4.2

4.4 Thermal Resistivity — $r = \frac{1}{k}$ (For a 1 m thickness)

Thermal Resistivity is defined as "THE RECIPROCAL OF THERMAL CONDUCTIVITY"[6].

4.5 Thermal Resistance — $R = \frac{1}{C}$ (For any thickness)

The thermal resistance of a structure is "the reciprocal of its thermal conductance". It refers to the thermal resistance (Heat Resistance Units) of any section or assembly of building components and is particularly useful in computing the overall transfer rate of a building section. The method is to calculate the resistance of each individual section of the wall and then to **add** the resistances to get the TOTAL RESISTANCE.

In carrying out such calculations the resistance of the air films on the inside and the outside of the building must be taken into account, because these comparatively motionless layers of air adjacent to the solid wall sections offer appreciable resistance to the flow of heat. The values of these resistances depend on the air velocity and their reciprocals are known as "SURFACE CONDUCTANCE COEFFICIENTS" (f). Refer to Table 4.3 for surface resistances for still and moving air.

4.6 Surface Conductance Coefficients

The C value for the thin layers of air on either side of a building section are designated:—

f_i = inside surface conductance coefficient (still air).

f_o = outside surface conductance coefficient (moving air).

The units of f_i and f_o are the same as for C, i.e., $W/(m^2.K)$.

4.7 The 'U' Value—"Coefficient of Thermal Transmittance"

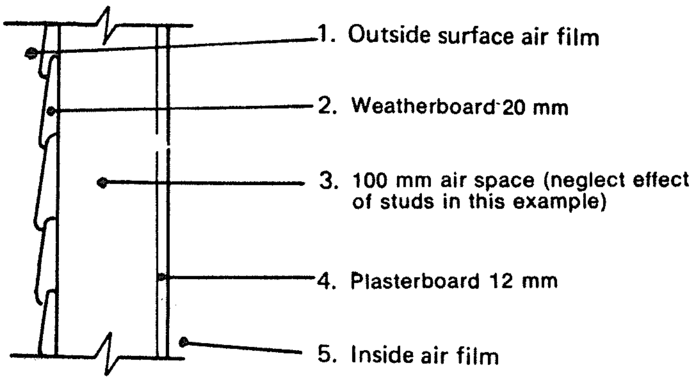
The "U" value is similar to C and has the same units—$W/(m^2.K)$. The term is a measure of the heat transferred through complete building components, such as a wall, INCLUDING the outside and inside air films. The 'U' value is determined by taking the reciprocal of the total resistance of a building section,

i.e., $U = \dfrac{1}{\text{TOTAL R}}$.

In calculating 'U' values, AIR SPACES (including attic spaces which have a significant resistance) must be taken into account. The method of determining the thermal resistance of air spaces is covered in a later section.

Example—the calculation of the 'U' value of a wall section.

a) Uninsulated



AN UNINSULATED STUD-FRAME WEATHERBOARD WALL.
FIG. 4.3

Consider the wall section shown in Fig. 4.3. The 'U' value is determined in the following manner:

Calculation Winter mean temp. 10°C T.D. 17K	k	C	R
1. Outside surface air film	—	—	·044
2. Weatherboard 20 mm	·144	7·25	·138
3. 100 mm Air space (resistance taken from data given later in paper)	—	—	·165
4. Plasterboard 12 mm	·173	14·42	·069
5. Inside air film	—	—	·120
(Heat resistance units) TOTAL			0·536

$$U = \frac{1}{\text{Total R}} = \frac{1}{0{\cdot}536} = 1{\cdot}87\ \text{W/(m}^2\text{.K)}$$

To decrease the 'U' value of the above wall section, the "R" value of the 100 mm air space must be increased by using insulation.

4.8 An Analysis of Heat Flow Across an Air Space

It is obvious that if insulation is used in the wall of Fig. 4.3 it must be placed in the air space. Heat will cross this space by three distinct methods. For a vertical uninsulated air space bounded by normal building materials such as wood and gypsumboard, the approximate proportions of the modes of heat transfer are:

Radiation 70%
Convection 25%
Conduction 5%

This can be demonstrated by an analytical method[7].

It can be seen that radiation poses by far the greatest problem as it is responsible for 70% of the heat transferred through the wall air space. In the case of heat flow through ceilings, the situation is similar, as under summer conditions 87% or more of the downward heat flow is by radiation. If radiation can be controlled the overall heat flow can be reduced considerably. The heat flow can also be reduced by increasing the thermal resistance of the structure by installing mass insulation in the air space.

In order to control radiation it is first necessary to understand how heat is transferred from one surface to another by radiation. This involves a knowledge of the nature of radiation from a surface.

4.9 "Black Body" Radiation

A "black body" is a theoretical body with properties such that it will absorb all radiation falling on its surface, reflecting and transmitting none. However, it does emit radiation depending on its absolute temperature T, in accordance with the STEFAN-BOLTZMANN law—

$$q = \sigma A T^4$$

where q = quantity of heat (J)
σ = Stefan-Boltzmann constant
A = Surface area
T = temperature, °K

4.10 Emissivity

Emissivity (e) is defined as:

"THE RATIO OF THE THERMAL RADIATION FROM UNIT AREA OF A SURFACE TO THE RADIATION FROM UNIT AREA OF A FULL EMITTER (BLACK BODY) AT THE SAME TEMPERATURE"[6].

The emissivity of aluminium foil is 0·05 at temperatures of surfaces in building spaces, e.g., attics, habitable rooms, etc. This means that the material "emits" only 5% of the amount of heat that a black body would emit if it were at the same temperature.

Most building materials such as wood and tiles have an emissivity of 0·90. The surface of mass insulants also has an emissivity of approximately 0·90, i.e., their surface emits 90% of the amount of heat which would be emitted by a black body at the same temperature.

4.11 Reflectivity

Reflectivity (r) is defined as:

"THE RATIO OF THE AMOUNT OF THERMAL RADIATION REFLECTED FROM A SURFACE TO THAT WHICH FALLS ON ITS SURFACE".

Aluminium foil has the high reflectivity of 0·95, i.e., it reflects 95% of the incident thermal radiation, whereas the surface of most building materials and the surfaces of mass insulants have low thermal reflectivities —usually about 0·10.

4.12 Wavelength

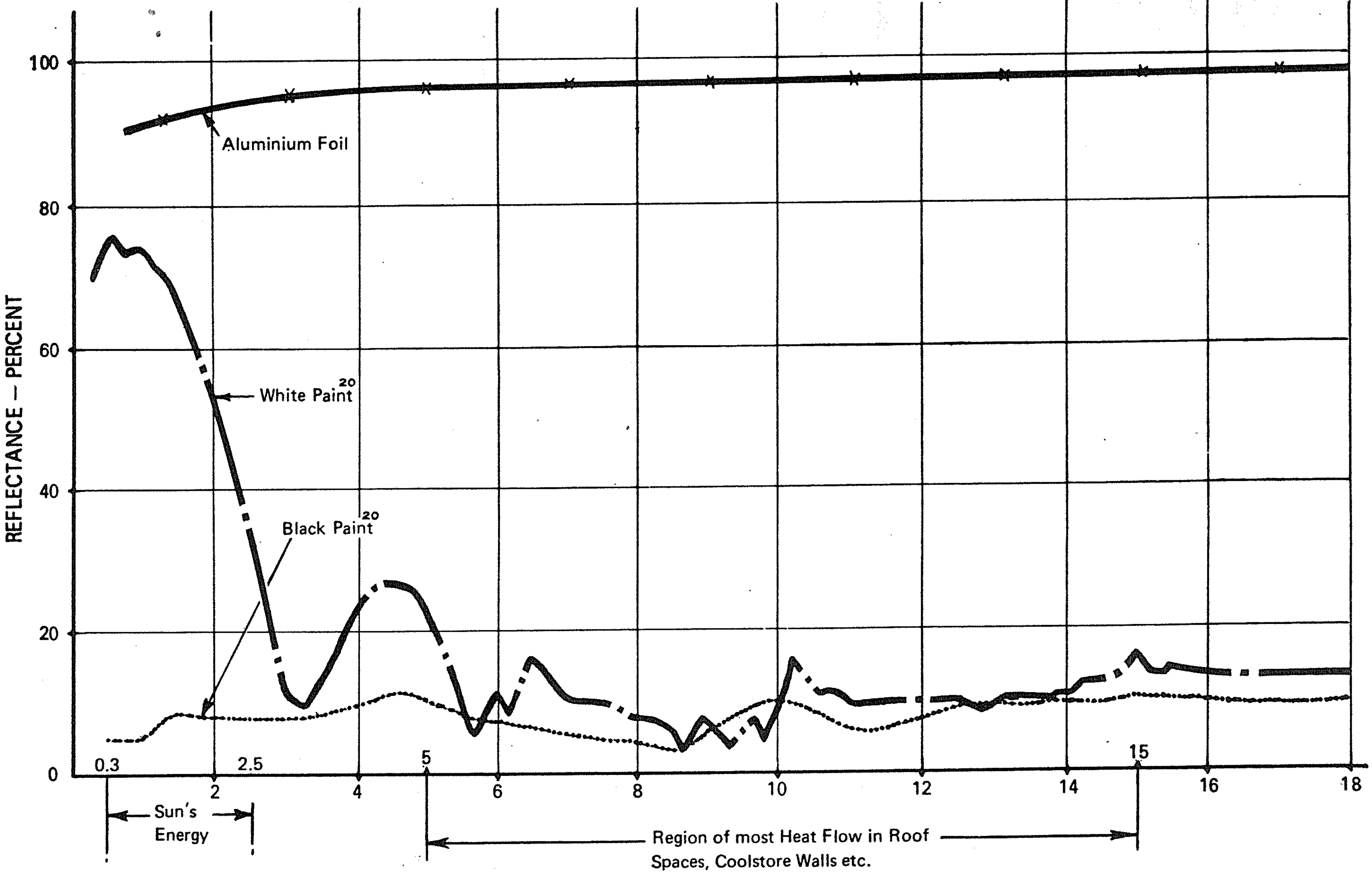
For any given wavelength;

emissivity (e) + reflectivity (r) = 1·00

Reflectivity of surfaces varies with the wavelength of the electromagnetic energy striking it.

REFLECTIVITY OF SURFACES

ALUMINIUM FOIL ——— WHITE PAINT —·—·· BLACK PAINT ·······



WAVELENGTH — MICROMETRES (μm)

REFLECTIVITY OF SURFACES

FIG. 4.4

For example:

Polished Aluminium:—is a good reflector at all wavelengths.

White Paint:— is a good reflector of visible light and solar radiation but a poor reflector of room-temperature radiation.

Black Paint:— is a poor reflector at all wavelengths.

The wavelengths of significance from the point of view of thermal insulation are in the "low temperature" range of from 5 μm upwards, a range in which aluminium foil is an excellent reflector and both white and black paints are very poor reflectors. (Refer to figure 4.4.)

4.13 Roof Surface Temperatures

The temperature of a roof depends on:

(i) its surface reflectivity to the sun's energy,

(ii) the loss of heat by emission from its surface.

An aluminium roof may have the same reflectivity for solar radiation (say $0 \cdot 80$) as a white painted roof and yet because of its low emissivity of $0 \cdot 11$ for low temperature radiation compared with $0 \cdot 90$ for a painted roof, it can be deduced that the aluminium roof would be hotter than the white painted roof. This deduction is endorsed by the experimental results shown graphically in figure 4.5.

4.14 The Use of Aluminium Foil as an Insulator

Because of its high reflectivity and low emissivity, aluminium foil is an excellent insulating material if used in conjunction with an airspace where "low temperature" radiation is the principal mode of heat transfer.

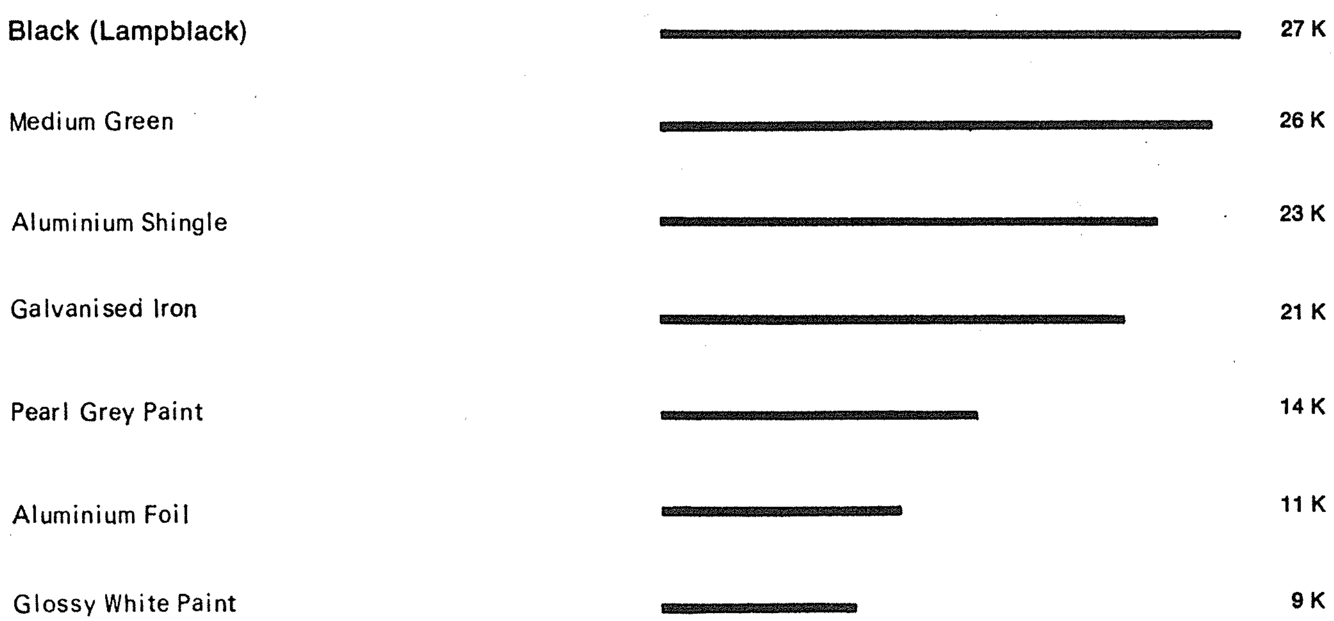
If aluminium foil is placed in the weatherboard wall of figure 4.3, the 'U' value becomes $1 \cdot 2$ W/(m^2.K), which represents a reduction in heat flow of 36%. The new arrangement is shown in figure 4.6.

The change in heat flow occurs because 95% of the radiant heat which passes across the air space from the outside to the inside is reflected. This has the effect of greatly increasing the resistance of the wall to heat flow.

If the heat flow were reversed, i.e., from inside to outside, the result would be the same as in this case the low 'emissivity' of the aluminium ($0 \cdot 5$) would have the same effect in reducing the heat flow across the air space. The radiant energy emitted, because of the foil surface, is reduced by 95%.

4.15 Effect of Multiple Air Spaces

By installing double sided reflective insulation at the centre of the air space in Fig. 4.6 at the point marked 'X', **two** air spaces are created each with a polished aluminium surface, one being reflective and the other having a low emissivity. With this arrangement, the 'U' value is further reduced to $0 \cdot 74$ W/(m^2.K). This is a 60% reduction in heat flow compared with the uninsulated wall. (It should be noted that the thickness of the air spaces is important. This aspect of reflective insulation is discussed in a subsequent section.)



DAILY MEAN RISE IN TEMPERATURE (K) OF TEST PANELS EXPOSED TO SUN[21] (30°, AUGUST, U.S.A.)

FIG. 4.5

Example—the calculation of the 'U' value of an insulated wall.

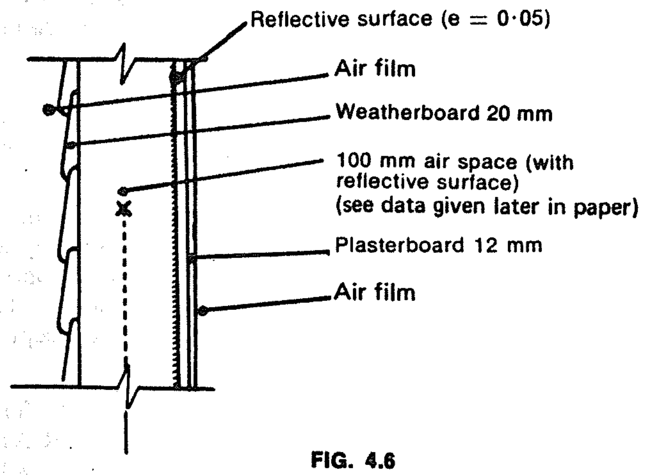


FIG. 4.6
Reflective insulation placed on the face of studs or at centre of air space.

The 'U' value of the wall section shown in Fig. 4.6 is determined as follows:

Calculation Winter mean temp. 10°C T.D. 17K (total)	k	C	R
1. Air film (12 km/h)			·044
2. Weatherboard 20 mm	·144	7·25	·138
3. 100 mm Air space (with reflective surface) (see data given later in paper)			·461
4. Plasterboard 12 mm	·173	14·42	·069
5. Air film (still air)			·120
			TOTAL R — 0·832

$\therefore U = \frac{1}{0\cdot832} = 1\cdot2\ W/(m^2.K)$ This is a 36% reduction in heat flow.

4.16 Factors Affecting the Thermal Conductance of an Air Space

A study of the behaviour of both plane air spaces (as in walls) and wedge shaped spaces (as in "attic" or roof spaces) is obviously essential in view of the importance of the thermal conductance of air spaces in building structures.

From the previous discussion it will be clear that when an air space is bounded by a polished foil surface the insulating effect is improved considerably, due to the fact that the reflectivity is comparatively high at 0·95, the emissivity is low being 0·05. Furthermore, because of thermodynamic considerations, the side of the air space on which the foil is placed is not important. However, a **polished foil surface must face the air space.** The factors affecting the thermal conductance of an air space are as follows:

a. Air space thickness and shape.
b. Orientation (vertical, sloping or horizontal).
c. Direction of heat flow, i.e., up or down.
d. Emissivity of surfaces on either side (e_1 and e_2).
e. Temperature difference across air space (T_1 and T_2).
f. Mean temperature (T_m).
g. Convection between adjacent air spaces.
h. Ventilation of air spaces (if any).

4.17 The Thermal Insulating Properties of Air Spaces

The theory of heat flow across air spaces has been well treated by two of the major researchers in this field, Robinson and Powlitch[7]. For theoretical purposes assuming the air spaces to be parallel and infinite the heat flow is expressed by the following formula.

$$\text{Total heat flow across air space} = \underbrace{\text{Heat flow by Radiation}} + \underbrace{\text{Heat flow by Convection} + \text{Heat flow by Conduction}}$$

$$\text{i.e.}\quad C = Eh_r + h_c$$

where (i) **"E"** is the **"Effective emissivity"** of the air space and is equal to—

$$\frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$$

(this can be proved mathematically), and e_1 and e_2 are the emissivities of the two surfaces facing the air space.

- If e_1 and e_2 are both 0·05 (Bright foil on both sides):
$$E = 0\cdot03.$$
- If one of these is 0·90 (ordinary building material) and the other 0·05 (foil):
$$E = 0\cdot05.$$

Therefore it makes little difference to the total heat flow if the foil is on one or both sides of the air space!

- If there is a non-reflective surface on both sides of the air space then $e_1 = e_2 = 0\cdot90$ and $E = 0\cdot82$.

Refer to Table 4.2 for commonly used values of r, e and E.

(ii) $\mathbf{h_r}$ is the heat flow by radiation ("Radiation Conductance")—calculated using the basic physical concept embodied in the Stefan-Boltzmann law. h_r varies with the **mean** temperature of the air space but for a mean temperature of 10°C is 5·16 W/(m².K). We will use this figure in the following curves which are all based on a mean air space temperature of 10°C.

$$\text{Thus } \mathbf{Eh_r} = 0\cdot05 \times 5\cdot16 = 0\cdot258\ W/(m^2.K).$$

(iii) $\mathbf{h_c}$ is the sum of the conduction and convection components which has been found experimentally. The **conduction only** component can be calculated using the known conductivity of still air, and by subtracting this from h_c, the convection component can be found.

Thus the three separate items can be illustrated graphically, the sum of the three being the total heat flow across the air space.

4.18 Curves Illustrating Conductances of "Reflective" Air Spaces

In order to illustrate the change in the thermal conductance for reflective air spaces with various orientations, various directions of heat flow, and thickness, the following graphs have been included.

Fig. 4.7	—Vertical Air Space	—heat flow horizontal
Fig. 4.8	—Horizontal Air Space	—heat flow down
Fig. 4.9	,, ,, ,,	—heat flow up
Fig. 4.10	—45° Air Space	—heat flow down
Fig. 4.11	,, ,, ,,	—heat flow up

4.19 Thermal Resistance Tables of Air Spaces

The ASHRAE Handbook of Fundamentals[5] has tabulated the results obtained by Robinson and Powlitch[7] and their figures are reproduced in metric units in Table 4.1. Figures pertaining to roof attic spaces are given in Table 4.4.

It should be remembered that conductance varies according to many conditions, and corrections may have to be made in unusual cases.

4.20 Effect of Convection and Ventilation

It is assumed that the air spaces referred to are sealed by the aluminium foil and the structural members to keep inter air space convection currents to a minimum. Convection between tandem air spaces has a limited effect in the case of horizontal air spaces, but with sloping and vertical spaces it can be appreciable, resulting in an increase in thermal conductance[8].

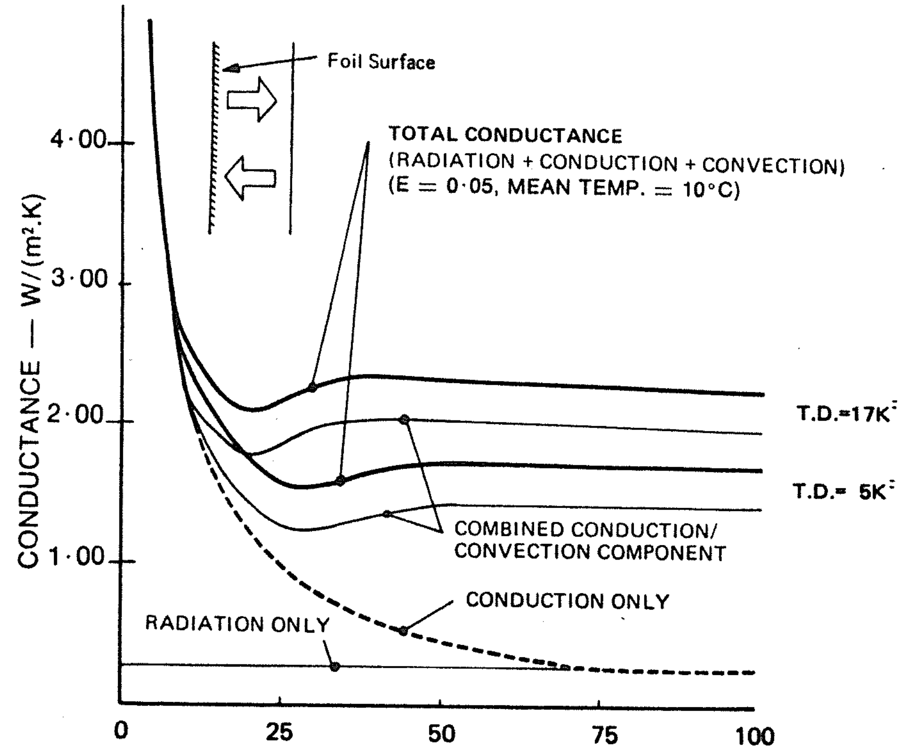
Under winter conditions, with heat flow up, heat loss is increased by ventilation but under summer conditions where the movement of heat is downward, mainly by radiation, **ventilation has been shown to have little effect in reducing heat gain**[16].

4.21 "Sol-Air Temperature"

In calculating the summer heat load on a building the direct and diffused rays of the sun on a building will increase the heat load above that which will apply if the outside air temperature is considered alone. This problem is dealt with by the use of the concept ot "Sol-air Temperature".

> "**THE SOL-AIR TEMPERATURE** IS THE TEMPERATURE OF THE OUTDOOR AIR, WHICH IN THE ABSENCE OF ALL RADIATION EXCHANGES, WOULD GIVE THE SAME RATE OF HEAT ENTRY INTO THE SURFACE AS WOULD EXIST WITH THE ACTUAL COMBINATION OF INCIDENT SOLAR RADIATION, RADIANT ENERGY EXCHANGE WITH THE SKY AND OTHER OUTDOOR SURROUNDINGS, AND CONVECTIVE HEAT EXCHANGE WITH THE OUTDOOR AIR."[5]

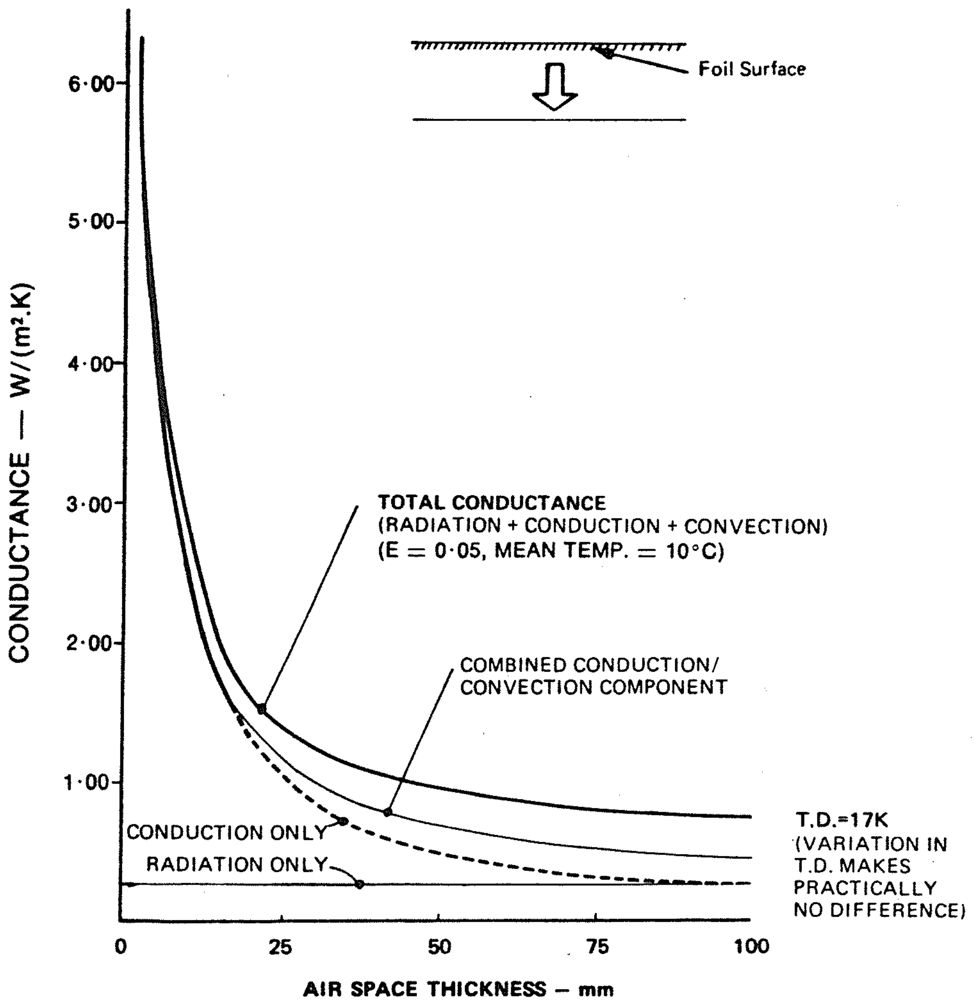
The Sol-air temperature concept applies to **roofs** (flat or sloping) and also to vertical **walls** having various aspects. For engineering calculations the Sol-air temperature can be estimated using figures published in the ASHRAE Handbook of Fundamentals.



Thermal Conductances for a Reflective Air Space having $e_1 = 0{\cdot}90$ (ordinary building material); $e_2 = 0{\cdot}05$ (foil); i.e. $E = 0{\cdot}05$

AIR SPACE VERTICAL—HEAT FLOW HORIZONTAL

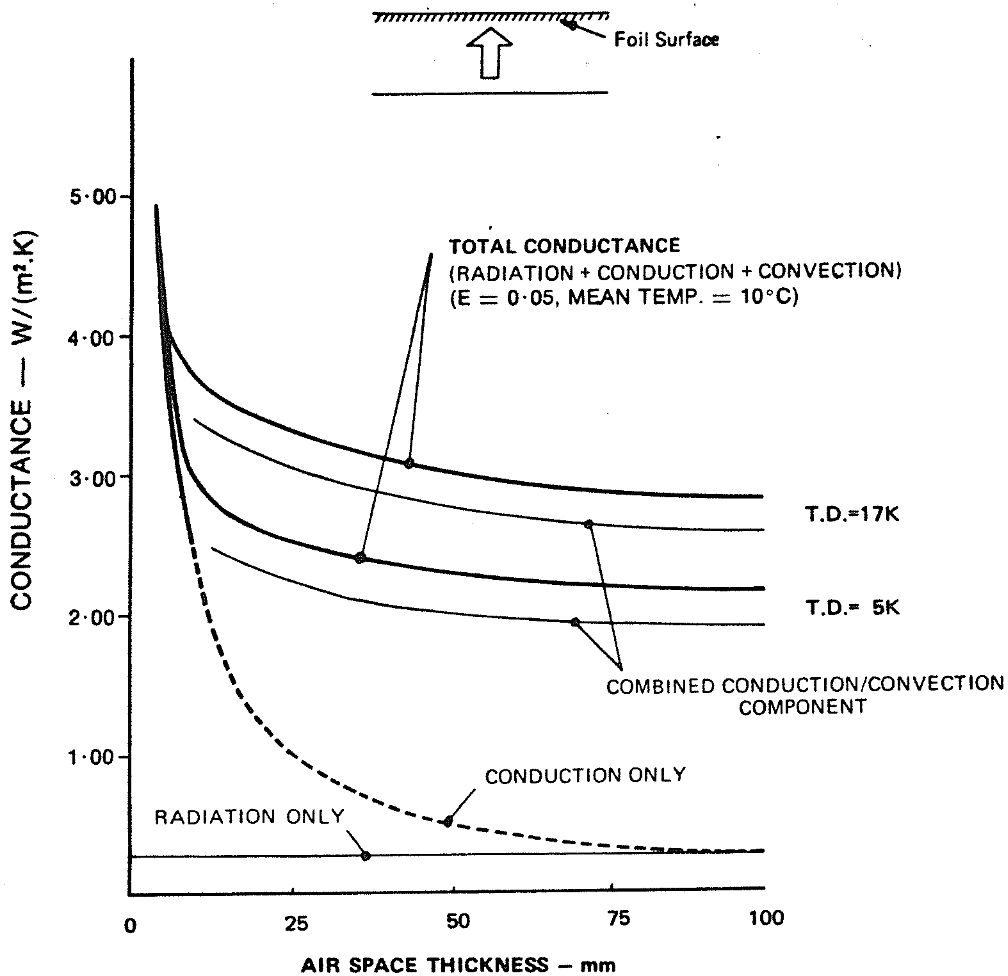
FIG. 4.7



Thermal Conductances for a Reflective Air Space having $e_1 = 0{\cdot}90$ (ordinary building material), $e_2 = 0{\cdot}05$ (foil); i.e. $E = 0{\cdot}05$

AIR SPACE HORIZONTAL – HEAT FLOW DOWNWARD

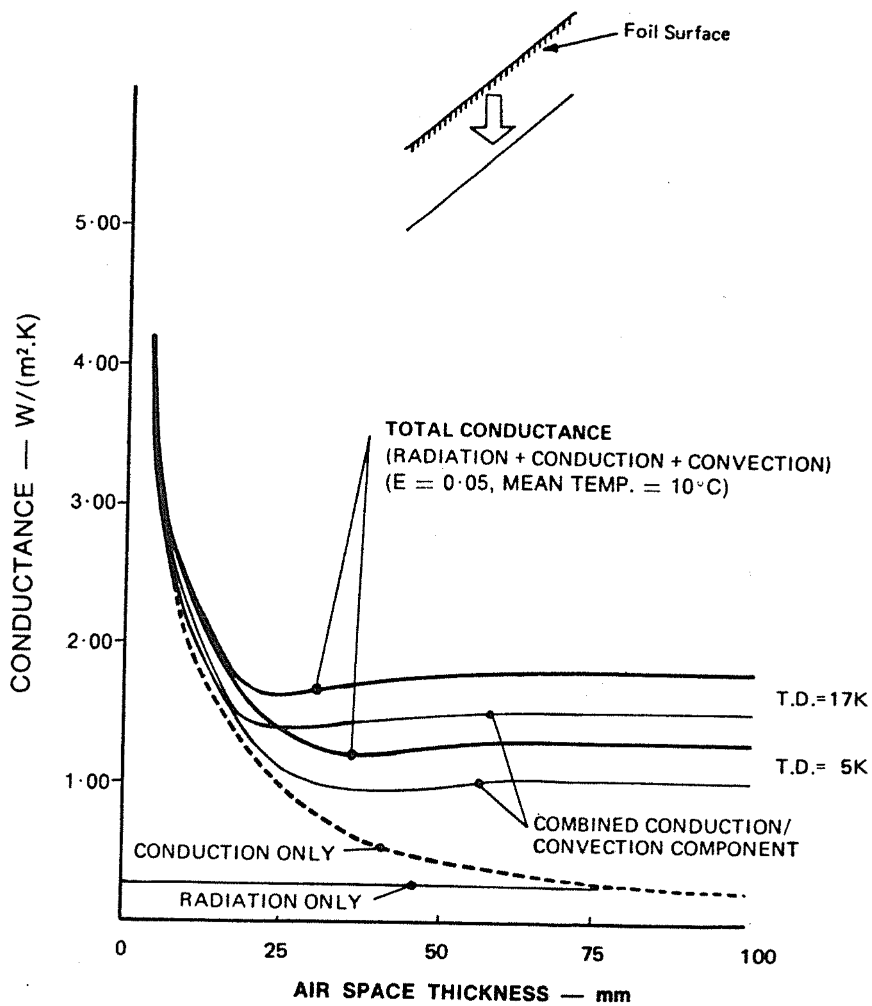
FIG. 4.8



Thermal Conductances for a Reflective Air Space having $e_1 = 0{\cdot}90$ (ordinary building material); $e_2 = 0{\cdot}05$ (foil); i.e. $E = 0{\cdot}05$

AIR SPACE HORIZONTAL – HEAT FLOW UPWARD

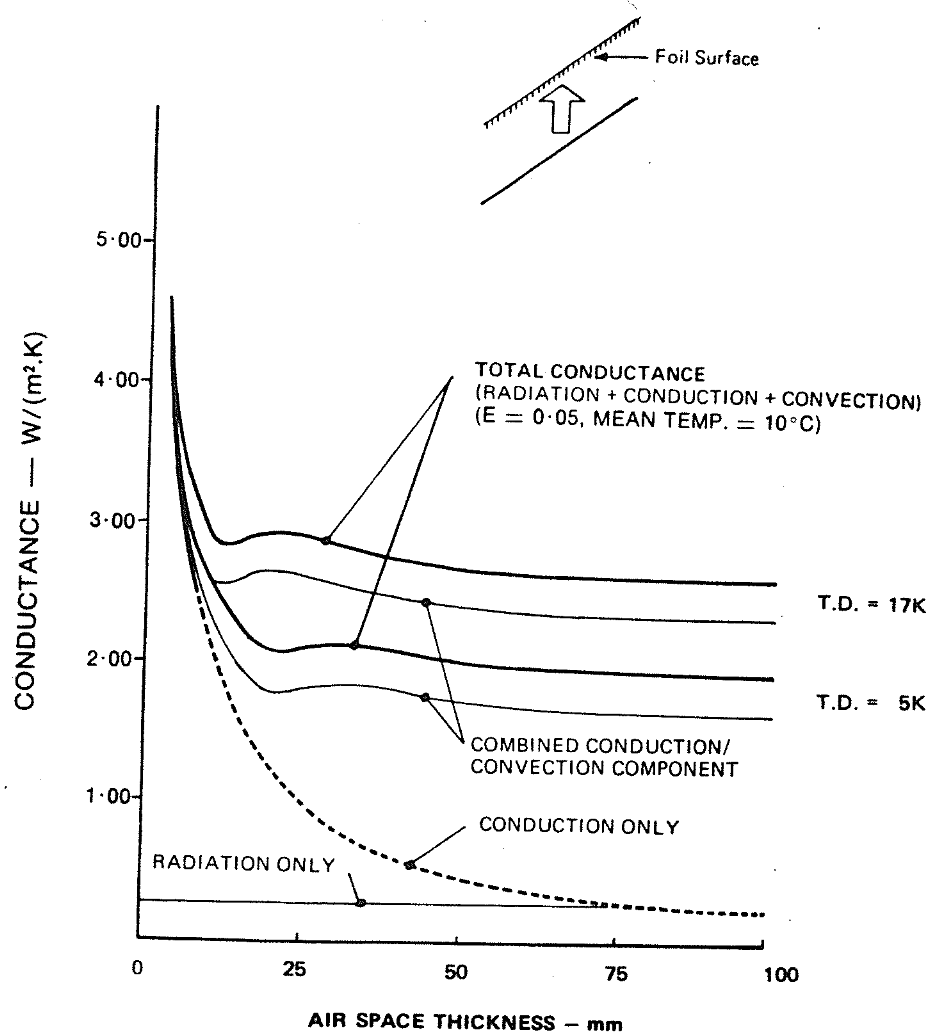
FIG. 4.9



Thermal Conductances for a Reflective Air Space having $e_1 = 0{\cdot}90$ (ordinary building material); $e_2 = 0{\cdot}05$ (foil); i.e. $E = 0{\cdot}05$

AIR SPACE 45° – HEAT FLOW DOWNWARD

FIG. 4.10



Thermal Conductances for a Reflective Air Space having $e_1 = 0{\cdot}90$ (ordinary building material); $e_2 = 0{\cdot}05$ (foil); i.e. $E = 0{\cdot}05$

AIR SPACE 45° – HEAT FLOW UPWARD

FIG. 4.11

TABLE 4.1*

THERMAL RESISTANCES OF AIR SPACES[a,e]

Units: $(m^2.K)/W$ or: H.R.U.

Position of Air Space	Direction of Heat Flow	Air Space[c] Mean Temp. °C	Temp Diff. K	20mm Thickness[b] 2 Foil Surfaces[c] E = 0·03	1 Foil Surface E=0·05	1 Partly Reflective Surface E=0·20	No Reflective Surface E=0·82	40mm Thickness[b] 2 Foil Surfaces E=0·03	1 Foil Surface E=0·05	1 Partly Reflective Surface E=0·20	No Reflective Surface E=0·82	100mm Thickness[b] 2 Foil Surfaces E=0·03	1 Foil Surface E=0·05	1 Partly Reflective Surface E=0·20	No Reflective Surface E=0·82
Horizontal	Down	30	see note f	·627	·572	·366	·148	1·050	·923	·486	·164	1·747	1·423	·595	·174
		10		·669	·625	·421	·180	1·129	1·011	·565	·201	1·884	1·574	·708	·217
45°	Down	30	5	·618	·570	·368	·148					·852	·768	·440	·158
		10	17	·611	·576	·400	·178					·636	·597	·410	·180
		10	5	·673	·629	·427	·180					·843	·777	·484	·190
Vertical	Horizontal	30	5	·623	·578	·370	·148					·657	·606	·380	·160
		10	17	·519	·493	·359	·169					·482	·461	·342	·165
		10	5	·655	·613	·416	·178					·650	·607	·412	·178
45°	Up	30	5	·530	·495	·335	·143					·567	·528	·349	·144
		10	17	·356	·343	·271	·146					·408	·391	·301	·155
		10	5	·518	·489	·356	·166					·560	·528	·375	·169
Horizontal	Up	30	5	·421	·398	·287	·134					·518	·484	·329	·141
		10	17	·303	·294	·241	·137					·377	·363	·285	·150
		10	5	·410	·393	·301	·153					·507	·481	·350	·165

MAXIMUM INTERPOLATION ERROR

Position / Direction	2 Foil Surfaces E=0·03	1 Foil Surface E=0·05	1 Partly Reflective Surface E=0·20	No Reflective Surface E=0·82
45° Down, 30 °C, 5 K	−26%	−25%	−15%	−6%
45° Down, 10 °C, 17 K	see notes b and c			
45° Down, 10 °C, 5 K	−21%	−20%	−12%	−6%
Vertical Horizontal, 30 °C, 5 K	−11%	−10%	−6%	
Vertical Horizontal, 10 °C, 17 K	+12%	+ 9%	+7%	
Vertical Horizontal, 10 °C, 5 K to Horizontal Up	see notes b and c			

a Spaces of uniform thickness bounded by moderately smooth surfaces.

b Interpolation between 20 mm and 100 mm thick air spaces gives values that are within ±5% of original data, except for those values indicated by the table of maximum error – inset above. For more precise values, the reference should be consulted.

c Interpolation, and moderate extrapolation of resistance values are permissible for other values of mean temperature, temperature difference and effective emissivity E.

d Effective emissivity of air space, E, is given by $\frac{1}{E} = \frac{1}{e_1} + \frac{1}{e_2} - 1$, where e_1 and e_2 are the emissivities of the two surfaces facing the air space.

e The resistance values in this table were obtained from the original published source[5] using the relationship 1 HRU (Imperial) = 0·176 110 29 HRU (Metric).

f The resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference

NOTE:
An air space has a resistance to heat flow which depends on its orientation etc., and also on the emissivity of its bounding surfaces.
A REFLECTIVE AIR SPACE is bounded on one or both sides by a bright Aluminium Foil surface.

* Derived from data presented in ASHRAE Handbook of Fundamentals, 1967.

Table 4.2

REFLECTIVITY AND EMISSIVITY VALUES OF VARIOUS SURFACES AND EFFECTIVE EMISSIVITIES OF AIR SPACES

(For Low Temperature Radiation as found within Building Cavities)

MATERIAL	Reflectivity r (percent)	Average Emissivity e	Effective Emissivity of Air Space—E	
			With one surface having emissivity e and other 0·90	With both surfaces of emissivity e
Aluminium foil, bright	95%	0·05	0·05	0·03
Aluminium sheet, oxidised	89%[2,3]	0·11[2,3]	0·108	0·06
Steel, Galvanised, bright	77%[2,3]	0·23[2,3]	0·22	0·13
weathered	72%[2,3]	0·28[2,3]	0·27	0·20
Aluminium Paint	50%	0·50	0·47	0·35
Building Materials: wood, asbestos cement, tiles, paper, glass, masonry, white paint, black paint....	10%	0·90	0·82	0·82

Table 4.3[5]

SURFACE RESISTANCES—STILL & MOVING AIR

POSITION OF SURFACE	Direction of Heat Flow	Surface Emissivity**				
		e=0·90 Non Reflective	e= 0·28	e= 0·20	e= 0·11	e=0·05 Reflective
Still Air						
Horizontal	Down	·162	·4	·475	·57	·801
Sloping 22½°*	Down	·148	·34	·384	·48	·595
Sloping 45°	Down	·134	—	·294	—	·391
Vertical	Horiz.	·120	—	·238	—	·299
Sloping 45°	Up	·109	—	·201	—	·241
Sloping 22½°*	Up	·109	·16	·197	·22	·236
Horizontal	Up	·107	·16	·194	·22	·232
Moving Air						
Any Orientation: 3·35 m/s	Any	·044				
6·70 m/s	Any	·030				

**Values for 22½° slope found by interpolation.*
***Emissivity is explained in the text.*

Table 4.4

THERMAL RESISTANCE OF ATTIC SPACES

	Thermal Resistance—R (Heat Resistance Units)	
	Heat flow down	Heat flow up
	Summer	Winter
Non-Reflective Attic Space:—		
—No Ventilation	·282	·176
—Natural vent. of approx. 0·005m³/s for each m² of ceiling area	·458	·11[2]
Reflective Attic Space:—		
—No Ventilation	1·092	·564
—Natural vent. of approx. 0·005m³/s for each m² of ceiling area	1·356	·335

NOTE:—A "reflective attic space" is defined as one in which either the upward facing surface or the downward facing surfaces (or both) comprises bright aluminium foil, having an emissivity of 0·05.

Table 4.5

TYPICAL CONDUCTANCES AND RESISTANCES OF MATERIALS NOT HAVING UNIT THICKNESS

MATERIAL	Thick-ness mm	Conduc-tivity (k) W/(m.K)	Conduc-tance (C) $W/(m^2.K)$	Resis-tance (R) H.R.U.
Aluminium Sheet	0·5	200	400 000	·000
Galvanised Steel Sheet	0·5	50	100 000	·000
Asbestos Cement Sheet	5	·650	130	·008
Caneite	12	·060	5·00	·200
Plasterboard (Gypsumboard)	12	·173	14·42	·069
Hardboard	6	·200	33·33	·030
Roof Tiles	12	·836	70·00	·014
Weatherboard	20	·144	7·20	·139

Table 4.6

TYPICAL PROPERTIES OF SOME COMMON MATERIALS*

MATERIAL	Density kg/m^3 (p)	Specific Heat Capacity (C_f) J/kg K $\times 10^3$	Volumetric Heat Capacity $J/(m^3.K)$ h $= (C_f \times p) \times 10^3$	Conductivity W/(m.K) k	Thermal Resistivity H.R.U. $r (= \frac{1}{k})$	Thermal Diffusivity m^2/s $(\frac{k}{p \times C_f})$ or $(\frac{k}{h})$ or $(\frac{1}{hr})$ $\times 10^{-6}$
Air (Dry 30°C)	1·2058	1·009	1·2166	·026	38·5	21·4
Aluminium	2560	0·900	2304	200	·005	87·0
Steel	7850	0·502	3940	50	·02	12·7
Water at 20°C	998·2	4·183	4175·47	0·60	1·67	0·14
Building Materials						
Asbestos Cement	1600	0·837	1339	·650	1·5	0·5
Brickwork	1600	0·837	1339	1·150	·9	0·9
Concrete, aerated	320	0·879	281	·086	11·6	0·3
Concrete, dense	2320	0·879	2039	1·500	·7	0·7
Fibreboard (Caneite)	270	1·507	407	·060	16·7	0·14
Glass	2480	0·67	1661	1·050	1·0	0·6
Granite	2640	0·816	2154	4·220	·2	1·96
Plasterboard	990	1·047	1037	·173	5·8	0·17
Hardboard	1120	—	—	·201	5·0	—
Sandstone	2000	—	—	1·3	·77	—
Tiles, Clay	1920	0·921	1768	·836	1·2	0·47
Wood	480	1·884	904	·144	6·9	0·16
Insulating Materials						
Corkboard	160	1·8	288	·038	26·3	0·13
Mineral Wool	52	0·879	42·2	·040	25·0	0·9
Polystyrene (exp. board)	15	1·21	18·2	·037	27·0	2·0
Polyurethane (foamed)	32	1·59	50·9	·021	47·6	0·4
Vermiculite: Exfoliated, Loose	80	0·879	70·3	·065	15·4	0·09
Fibreglass	11	0·837	9·2	·042	23·8	4·6

NOTE:— *These figures will be found very useful for approximate calculations. For very exact calculations care should be taken to allow for variations in temperature, composition of material, etc.*

4.22 The Thermal Resistance of Roof Attic Spaces

Roof spaces, otherwise known as attic spaces, are different from plane air spaces in two respects—

1. they are often ventilated.
2. they are often not bounded by parallel planes, but may, for example, be triangular or wedge shaped.

Practical experimentation has been carried out to determine the thermal resistance of attic spaces by Professor F. A. Joy, *et. al.*[17, 18].

Following a study of this work and other information on the subject[2, 5], the figures quoted in Table 4.4 are considered adequately accurate for practical purposes.

4.23 Effect of Dust

Upward facing foil surfaces will have their reflectivity impaired by dust which settles over a period of time. If presence of dust is expected then all upward facing foil surfaces should be considered to be non-reflective when performing calculations. Air spaces, wherever possible, should be bounded on the **upper** side by foil by incorporating spacing battens, etc., at the design stage.

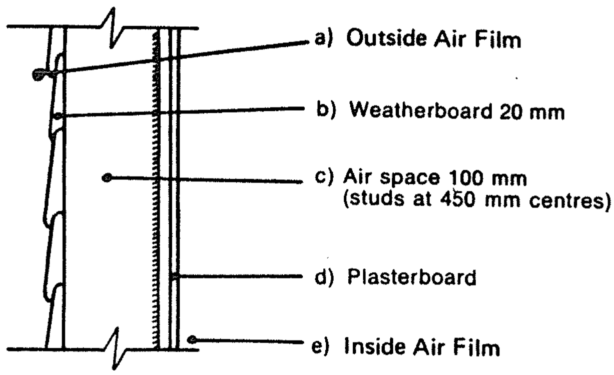
Vertical and downwards facing foil surfaces are not affected by dust.

4.24 'U' Values of Typical Building Structures

With the information provided herein, it is possible to calculate the U value of various building sections. The examples below are included to illustrate the use of the data provided. It should be noted that unless otherwise stated all upward facing foil surfaces are assumed to be completely covered with dust. Thus for some time after installation, reflective insulation will actually give better performance than stated.

Example 1—WEATHERBOARD WALL

"A weatherboard wall with lining of gypsumboard and reflective insulation between the studs and the gypsumboard."



WALL SECTION — Example 1.
FIG. 4.12

The calculation shows the "U" values for the wall under summer and winter conditions.

Correction for Studs

The resistance to heat flow of timber studs may be greater than or less than the resistance of the air space between and if a highly accurate U value is required then a correction should be made, using the resistance figures for wood (Table 4.6) for 1/9th of the wall, (i.e. assuming 50 mm studs spaced at 450 mm centres). In practice however, this refinement is usually not necessary.

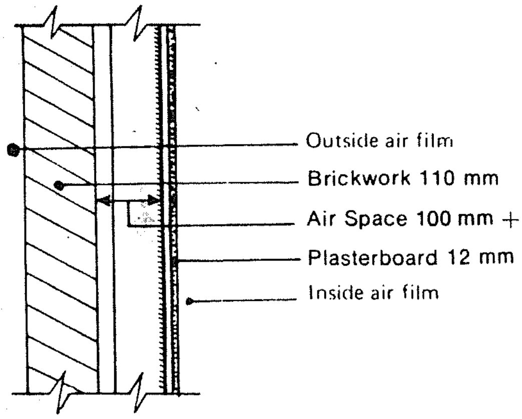
Calculation—Example 1

ITEM	SOURCE	HEAT RESISTANCE UNITS: NO INSULATION (E = 0·82)		REFLECTIVE INSULATION (E = 0·05)	
		Summer M.T. 30°C T.D. 5K	Winter M.T.10°C.17K T.D. (Total)	Summer	Winter
Outside air film	Table 4.3	·044	·044	·044	·044
Weatherboard 20 mm	Table 4.5	·138	·138	·138	·138
Air Space 100 mm	Table 4.1	·160	·165	·606	·461
Plasterboard 12 mm	Table 4.5	·069	·069	·069	·069
Inside Air Film	Table 4.3	·120	·120	·120	·120
	TOTAL R	0·531	0·536	0·977	0·832
	$U(=\frac{1}{R})$	1·88	1·87	1·02	1·2
	REDUCTION IN HEAT FLOW			46%	36%

Example 2—BRICK VENEER WALL

"A brick veneer wall with reflective insulation on **inside** of studs underneath gypsumboard sheets."

There is practically no difference in U value for summer and winter conditions with a wall section, i.e., with heat flow horizontal, if a mean temperature of 10°C and a temperature difference of 5K is taken for the winter condition, so for this case only two sets of figures, namely insulated and uninsulated, need be calculated.



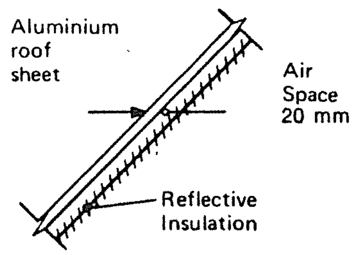
Section of brick-veneer wall with reflective insulation — Example 2.
FIG. 4.13

Calculation— Example 2.	SOURCE	RESISTANCE (H.R.U.) NO INSULATION Summer	Winter	INSULATED Summer	Winter
Outside air film	Table 4.3	·044	·044	·044	·044
Brickwork — 110 mm	Table 4.6	·099	·099	·099	·099
Air space — 140 mm	Table 4.1	·160	·165	·606	·607
Gypsum board	Table 4.5	·069	·069	·069	·069
Inside air film	Table 4.3	·120	·120	·120	·120
TOTAL RESISTANCE		0·492	0·497	0·938	0·939
"U" Value $= \frac{1}{R} =$ W/(m².K)		2·01	2·03	1·07	1·07
"U" in round figures		2·02		1·07	
REDUCTION IN HEAT FLOW		47%			

Example 3—FACTORY ROOF

"A saw-tooth factory roof, 22½°, with new aluminium roof sheeting."

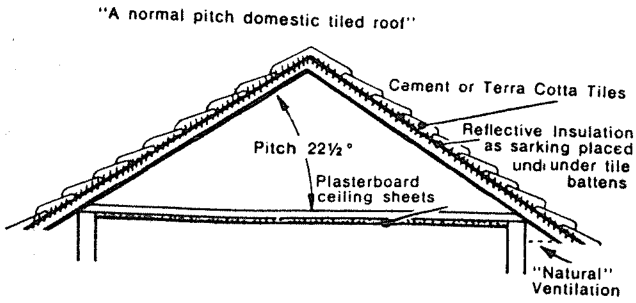
The "e" value for mill finish aluminium roof sheet is 0·11. The average air space thickness is assumed to be 20 mm and the emissivity of the upper foil surface is taken as being 0·90 (assuming dust covered).



Factory Roof Section — Example 3.
FIG. 4.14

Calculation—Example 3	Source	THERMAL RESISTANCES (H.R.U.) SUMMER M.T. 30°C T.D. 5K: No Insulation	SUMMER: With Reflective Insulation	WINTER M.T. 10°C T.D. 17K (Total): No Insulation	WINTER: With Reflective Insulation
Outside air film (Moving air)	Table 4.3	·044	·044	·044	·044
Aluminium roof sheet (Negligible thickness)	Table 4.5	·000	·000	·000	·000
Reflective Air Space (e=0·108, Table 4.2)	Table 4.1	—	·52	—	·30
Reflective Insulation (Negligible thickness)	—	—	—	—	—
Inside Air Film (e = 0·11, Table 4.2) ...	Table 4.3	0·48	·595	·22	·236
TOTAL RESISTANCE		0·524	1·159	0·264	0·570
"U" Value $= \frac{1}{R}$ W/(m².K)		1·91	0·86	3·79	1·75
REDUCTION IN HEAT FLOW			55%		54%

Example 4—PITCHED ROOF



PITCHED ROOF — Example 4.
FIG. 4.15

Comments on Technique for Calculations

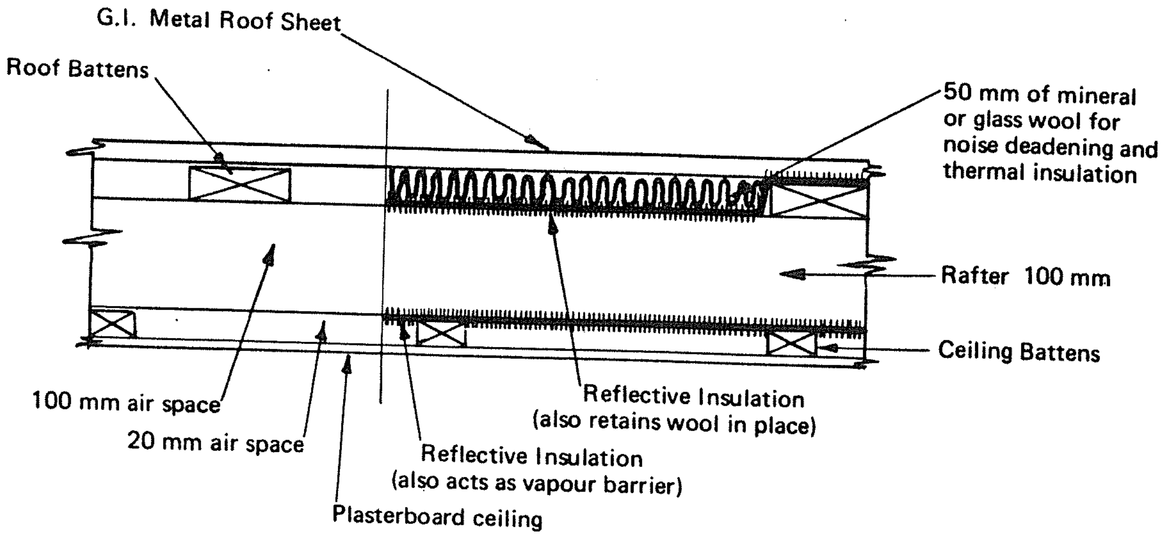
In summer the resistance value of a ventilated attic space is high but in winter it is much lower. It has been argued in the past that a resistance value for the tiles should be taken only for the summer condition, assuming that in the winter the upper surface of the ceiling sheet is the outside surface with a surface resistance equivalent to still air conditions.

This does not take into account that there is always an exchange of radiant heat between surfaces of differing temperatures, and that this applies to both winter and summer conditions (i.e. heat flow up or down). The tiles and enclosed attic space must thus perform as a resistance to heat flow in both cases.

This is the method adopted by learned bodies overseas.

Calculation—Example 4	SOURCE	THERMAL RESISTANCES (H.R.U.)			
		SUMMER M.T. 30°C T.D. 5K		WINTER M.T. 10°C T.D. 17K (Total).	
		No Insulation	With Reflective Insulation	No Insulation	With Reflective Insulation
Outside Air Film (see comments above)	Table 4.3	·044	·044	·044	·044
Tiles	Table 4.5	·014	·014	·014	·014
20 mm Air Space (approx.)	Table 4.1	—	—	—	—
Attic Space	Table 4.4	·458	1·356	·11	·335
12 mm Gypsumboard	Table 4.5	·069	·069	·069	·069
Inside Air Film	Table 4.3	·162	·162	·107	·107
TOTAL R =		0·747	1·645	0·344	0·569
$U = \frac{1}{R}$ W/(m².K)		1·34	0·61	2·91	1·76
REDUCTION IN HEAT FLOW			54%		40%

Example 5—FLAT METAL ROOF



Calculation — Example 5	Source of Info.	H.R.U.			
		SUMMER M.T. 30°C T.D. 5K		WINTER M.T. 10°C T.D. 17K (Total)	
		No Insulation	With Insulation	No Insulation	With Insulation
Outside Air Film	Table 4.3	·044	·044	·044	·044
Metal Roof Sheet	Table 4.5	Nil	Nil	Nil	Nil
Mineral or Glass Wool 50 mm	Table 4.6	—	1·350	—	1·35
Air Space 100 mm +	Table 4.1	·50 (E = 0·27)	1·423	·25	·481
Air Space 20 mm	Table 4.1	—	·572	—	·393
Plasterboard Ceiling	Table 4.5	·069	·069	·069	·069
Inside Air Film	Table 4.3	·162	·162	·107	·107
TOTAL R		0·775	3·62	0·470	2·444
$U = \frac{1}{R}$ W(m².K)		1·29	0·28	2·13	0·41
REDUCTION IN HEAT FLOW		78%		80%	

4.25 Endorsement of Reflective Insulation

The benefit of reflective insulation has been acknowledged by both the Commonwealth Experimental Building Station, Sydney, and the C.S.I.R.O. Division of Building Research, Melbourne, who have acknowledged that in resisting the downward flow of heat in summer, a 100 mm air space with a reflective surface on one side, is equivalent to 50 mm of mineral wool[10].

Reflective insulation has wide application in industrial and domestic buildings where experience has confirmed that the product is effective, practical and economic. In most cases no additional structural or supporting material is required and reflective insulating air spaces can be created within the confines of the basic structure.

5. CONDENSATION

Possible problems from condensation should be considered irrespective of the type of insulation used.

5.1 Water Vapour—A Gas

Air is a mixture of many gases and includes a very small amount of gaseous WATER VAPOUR, a TRANSPARENT, TASTELESS, ODOURLESS gas, **not** to be confused with VISIBLE condensed water droplets which make up clouds, fog, and the visible so called "steam" which is emitted from a kettle.

Each gas in the air exerts a "partial pressure" according to Boyle's gas law and the total sum of all the pressures adds up to atmospheric pressure which is measured by the height of a column of mercury supported by the air.

Normal atmospheric pressure is 760 mm of mercury and to give an idea of how small is the partial pressure of the water vapour (named vapour pressure), the water vapour pressure in a sample of air at 20°C containing the maximum amount of water vapour without "fogging", is 17·535 mm of mercury.

5.2 Relative Humidity

When the previously mentioned air sample at 20°C contains the maximum amount of water vapour, it has a Relative Humidity of 100% and is said to be "SATURATED". In other words 17·535 mm of mercury is the "SATURATED" VAPOUR PRESSURE for air at 20°C.

The weight of water vapour present in moist air is often expressed in gm per gm of dry air, this being the humidity ratio.

A glance at the Psychrometric chart (Fig. 5.1) will show that in the above mentioned example, the moisture content of air at saturation (i.e. 100% R.H.) is approximately 0·0145 gm/gm of dry air.

If the Relative Humidity were 50%, the moisture content would be 50% of 0·0145 gm, i.e. approximately 0·007 25 gm of moisture per gm of dry air.

If the same sample of air at 20°C and 100% Relative Humidity were heated to 25°C (without adding or subtracting any moisture), it would then be capable of holding more moisture and its Relative Humidity would drop. (This can be followed graphically using the Psychrometric chart.)

Relative Humidity is defined as:

> "THE RATIO, EXPRESSED AS A PERCENTAGE, OF THE VAPOUR PRESSURE IN THE AIR AT A GIVEN TEMPERATURE AND TOTAL PRESSURE, TO THE VAPOUR PRESSURE OF SATURATED AIR AT THE SAME TEMPERATURE AND TOTAL PRESSURE."

It should be noted that by quoting a particular Relative Humidity the moisture content is not specified unless the temperature is also given. **However,** the moisture content can be specified exactly by quoting the **DEW POINT.**

5.3 Condensation and Dew Point

If the sample of moist air used in the previous example were cooled again to 20°C the Relative Humidity would again be 100% and if it were cooled to 19°C there would be a slight **FOG.** That is, some of the moisture would "condense" to form water droplets visible as a FOG. The temperature of 20°C is defined as the "DEW POINT" of this sample.

Condensation can also be caused by bringing the air into contact with a surface which is below its dew point, e.g. a surface having a temperature of 15°C would collect condensed moisture from the previously mentioned sample of air.

5.4 Psychrometric Chart

The **DEW POINT** of any particular sample of air can be determined using the "PSYCHROMETRIC CHART" which correlates most of the physical properties of air such as DRY BULB TEMPERATURE, DEW POINT, RELATIVE HUMIDITY, and other parameters which have not been considered, such as WET BULB TEMPERATURE (not to be confused with DEW POINT), and ENTHALPY.

The Psychrometric Chart for normal atmospheric pressure at sea level (760 mm of mercury) is given in Fig 5.1.

5.5 A Practical Problem

If the air underneath a sheet metal roof has a dry bulb temperature of 12°C and a relative humidity of 70%, what is the temperature below which the roof sheeting must not fall if condensation on the under surface is to be prevented? The answer of 7°C is determined from the Psychrometric Chart. (Refer point P.)

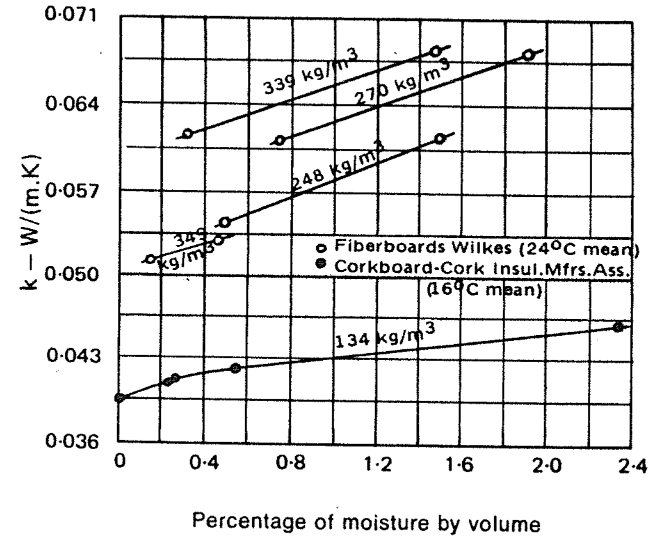
For altitudes which are significantly above sea level where the atmospheric pressure is well below the standard, a special psychrometric chart should be used[5]. However, the assessed dew points will be practically the same using either chart, even though other psychrometric values may differ.

5.6 Condensation in Roofs

During winter in many parts of the world, conditions at times encourage condensation of water vapour on the under side of roofing materials. Condensation and resulting drips cause damage by staining ceiling sheets and by causing rotting of timbers and corrosion of metals. Condensation can also cause wetting of mass insulation thus increasing its conductivity markedly (see Fig. 5.2). Care should be taken at the design stage to guard against condensation.

Even in the warm humid tropics, condensation can occur, particularly when air conditioning is used.

In hot arid climates, the depression of roof temperatures at nights due to the radiation loss to the night sky has been known to induce condensation under sheet metal roofs.



k vs. percentage of moisture by volume. Insulating boards.

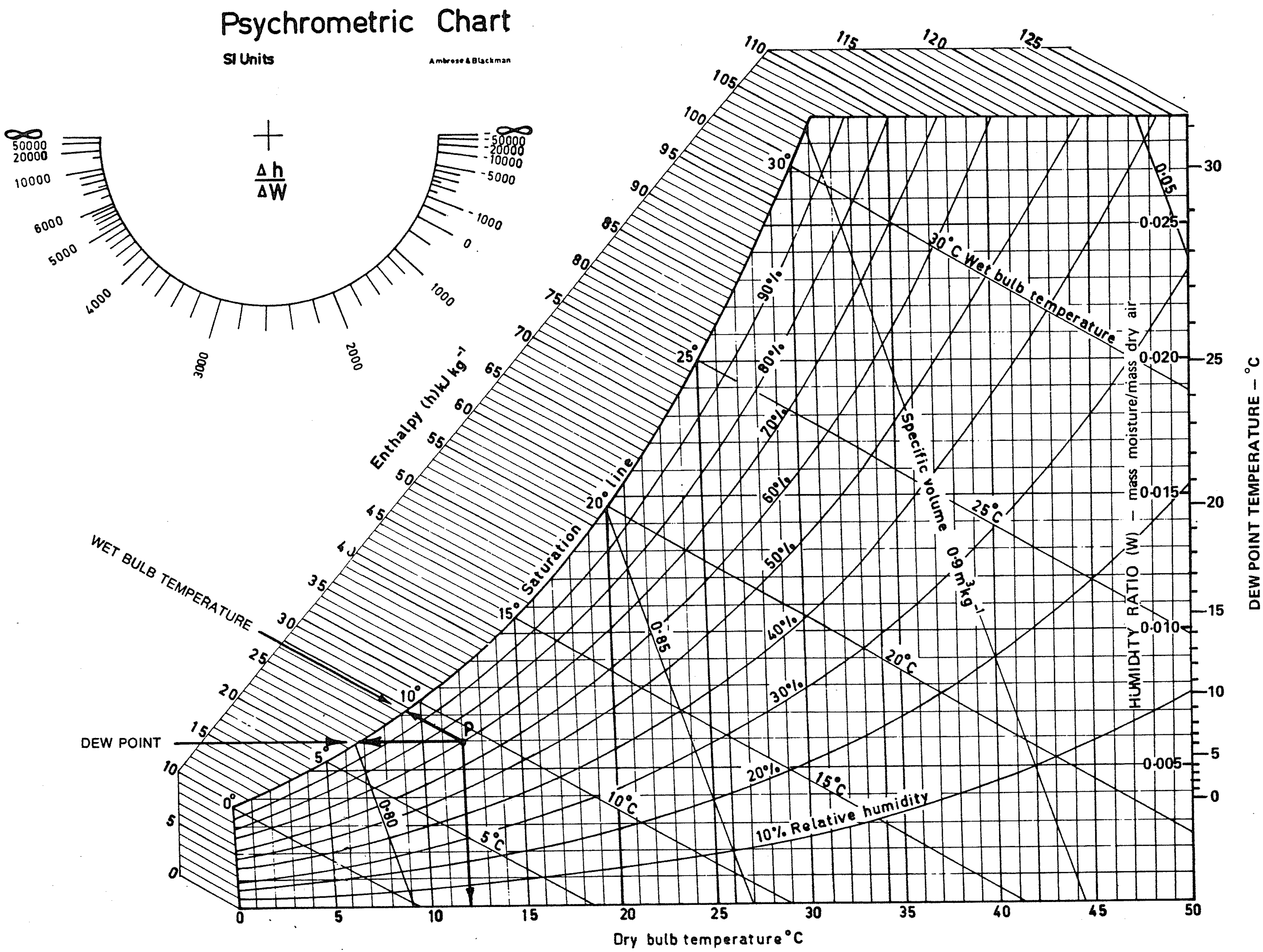
EFFECT OF MOISTURE ON INSULATIONS

FIG. 5.2

5.7 Vapour Barriers

One solution to the condensation problem is to include a vapour barrier to prevent water vapour reaching any surface which is cold enough to cause condensation. The vapour barrier should be placed on the warm side of the structure—in order to keep its temperature above the 'Dew Point'.

A vapour barrier is defined as any membrane which will sufficiently restrict the migration of water vapour from the warm moist interior of a building to the wall or roof cavity where it may contact a cold surface. Vapour barriers may be formed by such differing materials as a well preserved film of paint on the ceiling sheet, a polyethylene film or an impervious metallic layer such as aluminium foil. Aluminium foil, laminated to paper and reinforced with fibreglass and/or sisal fibres, is one of the most effective and widely used vapour barriers, for besides incorporating a film which is most resistant to vapour transmission, it can


Psychrometric Chart
SI Units
Ambrose & Blackman
Δh / ΔW
Enthalpy (h) kJ kg⁻¹
WET BULB TEMPERATURE
DEW POINT
P
Saturation 20° line
30°C Wet bulb temperature
Specific volume 0·9 m³kg⁻¹
0·80
0·85
10% Relative humidity
20%
30%
40%
50%
60%
70%
80%
90%
HUMIDITY RATIO (W) — mass moisture/mass dry air
DEW POINT TEMPERATURE — °C
Dry bulb temperature °C

also act as thermal insulation, either in the roof, wall or floor of a building.

Figure 5.3 shows the design of a "flat" sheet metal roof which incorporates a vapour barrier/insulation sheet which acts as (i) a vapour barrier, (ii) thermal insulation, and (iii) sarking (to catch drips of water from accidental leaks). This type of design has been acepted widely by many statutory and lending authorities.

5.8 Ventilation

Ventilation is an important aspect to be considered in relationship to condensation. In controlling condensation the "FLOW THROUGH" or "FAIL SAFE" principle must be observed, e.g., in the design ot roofs it must be easier for water vapour to escape from the roof cavity, than it is for it to enter. Ventilation can greatly assist in some cases.

5.9 The Method for Estimating the Incidence of Condensation on Surfaces

The technique is best understood by considering practical examples. In the wall structure detailed in Fig. 5.4 the thermal resistances of the components of the uninsulated wall illustrated are:

Outside air film	·044
Weatherboard	·138
Air Space	·165
Plasterboard	·069
Inside air film	·120
TOTAL	0·536 H.R.U.

If the outside temperature is 2°C, and the inside ambient is 20°C, the temperature difference across the wall is 18K. Under steady conditions the temperature of the inside surface of the gypsumboard is **below** the inside ambient of 20°C, by a **proportion** of temperature drop which is equal to the proportion of resistance drop.
i.e.

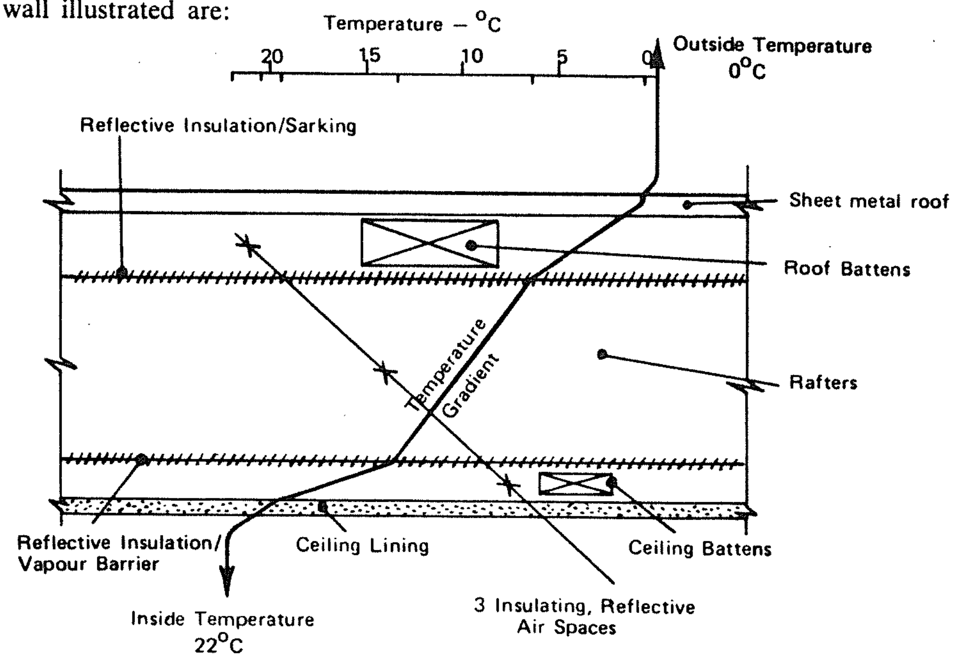
$$\frac{\text{Temp. drop across inner air film}}{\text{Total temp. drop}} = \frac{\text{Thermal resist. of inner air film}}{\text{Total thermal resist. of wall}}$$

Applying figures to this formula:

$$\text{Temperature drop across inner air film} = \frac{0\cdot120}{0\cdot536} \times 18$$
$$= 4\text{K}$$

Therefore the temperature of the inner surface is

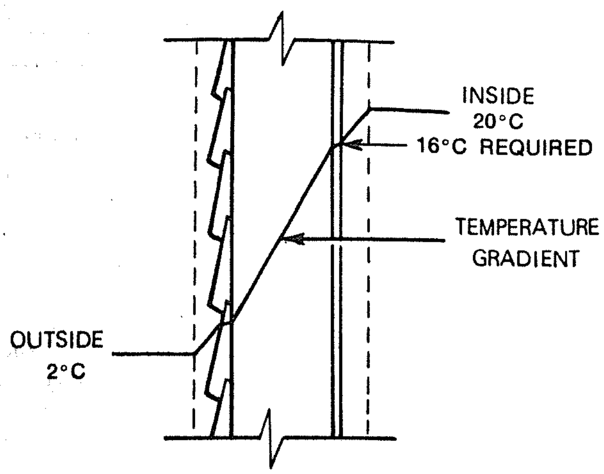
$$20 - 4$$
$$= 16°\text{C}$$



Roof Design accepteu by the major lending authorities of N.S.W. (In the humid tropics the upper sheet of insulation would need to be the vapour barrier). The diagram shows also a typical winter temperature gradient.

FLAT ROOF WITH INSULATION/SARKING & VAPOUR BARRIER

FIG. 5.3



TEMPERATURE GRADIENT ACROSS WALL

FIG. 5.4

Relating this temperature to DEW POINT it will be clear that condensation will not occur on this surface as long as the air with which it is in contact has a dew point below 16°C.

An examination of the Psychrometric chart shows that the air (which is at a temperature of 20°C) must not have a RELATIVE HUMIDITY greater than 77% if condensation is to be prevented.

If a layer of reflective foil insulation were placed between the gypsumboard and the studs the thermal resistance of the air space would be increased from 0·164 to 0·462, the resistances of the various components then being:—

Outside air film	·044
Weatherboard	·138
Air space (Reflective)	·461
Plasterboard	·069
Inside air film	·120
TOTAL	0·832 H.R.U.

New Temperature of Gypsumboard Surface

$$= 20 - \frac{0 \cdot 120}{0 \cdot 832} \times 18$$
$$= 17 \cdot 4°C$$

That is, the inside Relative Humidity can rise from 77% to about 86% before inner surface condensation takes place. A similar effect could be achieved using approximately 20 mm of mineral wool insulation.

This simple concept can be applied to all kinds of roof and wall structures to study the possibility of condensation and the use of insulation and vapour barriers.

The temperature gradient of an insulated flat roof is indicated in Figure 5.3. From this, the effectiveness of the vapour barrier which is placed on the warm or lower side of the roof structure can be evaluated.

5.10 Summary

The rules for preventing condensation in building structures can be summarised as follows:

a. Provide a vapour barrier on the warm side of the structure.

b. Provide enough thermal insulation on the cold side of the vapour barrier to ensure that the vapour barrier does not drop below the expected "dew point" of the air coming in contact with it.

c. Provide ventilation on the cold side of the vapour barrier and insulation, to assist in removal of any water vapour which does happen to penetrate the vapour barrier (flow through principle).

6. ENVIRONMENTAL CONDITIONS AND THEIR MEASUREMENT

Although the Comfort Indices used for describing combinations of conditions which make up man's environment are little known or used at present, in the future it will be essential for those involved with buildings, particularly in hot climates, to be completely familiar with them.

6.1 Physical Parameters Used

Dry Bulb Temperature:—The dry bulb temperature can be measured with an ordinary mercury in glass thermometer taking care to avoid errors due to radiation which could lead to a higher temperature indication. (For very accurate measurements several corrections should be made.[5])

Wet Bulb Temperature:—If a wetted wick is placed over the bulb of an ordinary thermometer the temperature reading is depressed below the dry bulb reading. Wet and dry bulb readings can be used together to determine the relative humidity of air when using this method. Precautions are necessary if accurate results are to be obtained, such as keeping the wick absolutely clean, using distilled water, and keeping air velocity within certain prescribed limits. Details of the procedure are given in reference 5.

Globe Temperature:—This temperature is measured with a globe thermometer which consists of a hollow sphere (usually 150 mm diam. and made of copper), coated with a matt black paint, and containing a thermometer with its bulb at the centre of the sphere. The temperature of the instrument depends on the environment in which it is placed. If the walls and other surfaces which surround the globe are warmer than the air, the temperature recorded by the thermometer inside the globe will be above air temperature because of radiation, and conversely with the surroundings cooler than the air, the globe thermometer temperature will be below air temperature. The globe temperature is also influenced by the velocity of air movement and globe diameter.

Mean Radiant Temperature:—The mean radiant temperature, which does not take into account relative humidity, represents the mean temperature of the surroundings. It can be found by using Figure 7 from Bedford's booklet[19], provided the following information is available:—

(a) Globe thermometer temperature.
(b) Air temperature (dry bulb).
(c) Air velocity.

6.2 Effective Temperature and Other Comfort Indices

(a) **Effective Temperature**—The effective temperature takes into account Humidity as well as dry bulb temperature and air movement but does **not** allow for **radiant heat.** The effective temperature is found by using a special chart, see Fig. 6.2.

(b) **Corrected Effective Temperature.** By using the **globe temperature** in place of the dry bulb temperature when reading the chart to determine effective temperature, a temperature which takes into account radiation is obtained. This temperature is known as the "corrected effective temperature".
The chart, entitled "**Normal** Scale of Corrected Effective (or Effective) Temperature" (Fig 6.2) applies for people in normal light clothing. (There is another scale, called the "**Basic** Scale" which has been prepared for subjects stripped to the waist.)

(c) **The Equatorial Comfort Index.** This index is based on a comfort vote survey embracing representatives of several races living in Singapore. It takes into account temperature, humidity and air speed, the index being read from a nomogram in a manner similar to that used for determining effective temperature. Although not further considered in this paper, the approach, in future, could prove useful in equatorial regions because it is based on assessments by subjects habitually exposed to the climate concerned.

(d) There are several other indices which are sometimes used. For example the **Equivalent Temperature**[14] which makes no allowance for humidity.

Use of Scale to Find "Corrected Effective Temperature"

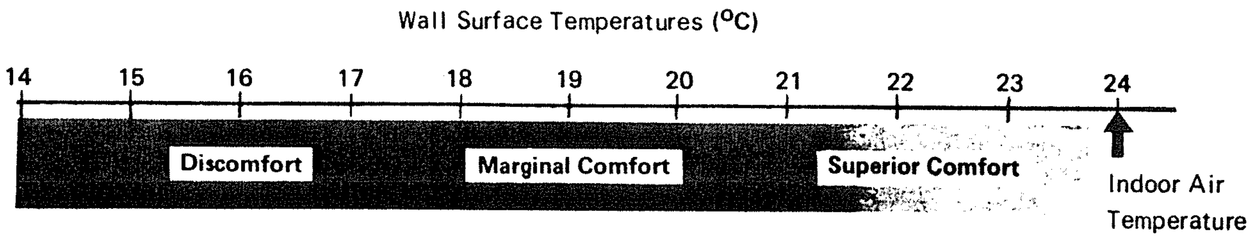
A straight line is drawn through the points on the vertical scales on Fig. 6.2 which represent the observed globe thermometer temperature and the wet bulb temperature, and the point at which the line cuts the line on the grid corresponding with the observed air velocity indicates the corrected effective temperature. Thus, when the globe thermometer is 32·2°C, the wet bulb temperature 24·4°C, and the air velocity 1 m/s, the corrected effective temperature is 26·7°C. That means that the combination of air temperature, humidity, air movement and radiant heat which yields a globe thermometer temperature of 32·2°C and a wet bulb temperature of 24·4°C with air moving at the rate of 1 m/s, is equivalent to an environment in which the air is calm and saturated with moisture at a temperature of 26·7°C and in which the mean temperature of the cold surroundings is also 26·7°C. If, with the same globe thermometer and wet bulb temperatures, the air had a velocity of 3 m/s, the corrected effective temperature would be 25·3°C.

6.3 Directional Radiation

In Section 2 on "Heat and Human Comfort" it was mentioned that the directional property of radiation from hot ceilings and other building components, has not, so far, been allowed for in any comfort indices.

A mathematical simplification of the problem is to assume that the overhead source of radiation is a fairly large horizontal plane, and that the receiving surface, i.e., the top of a person's head, is a very small horizontal plane.

(The head, if considered to be a small sphere, gives a much greater variation in "configuration factor" resulting in a greater variation of radiated heat with changes of dimensions. Therefore the former simplification only is quoted.)



Indoor comfort is partly dependent on the radiation of body heat to nearby cold surfaces—floor, walls and ceiling—or, in summer, the gain of body heat from warmer surroundings. This comfort yardstick shows the approximate relationship of winter comfort to surface temperatures.[11]

COMFORT YARDSTICK

FIG. 6.1

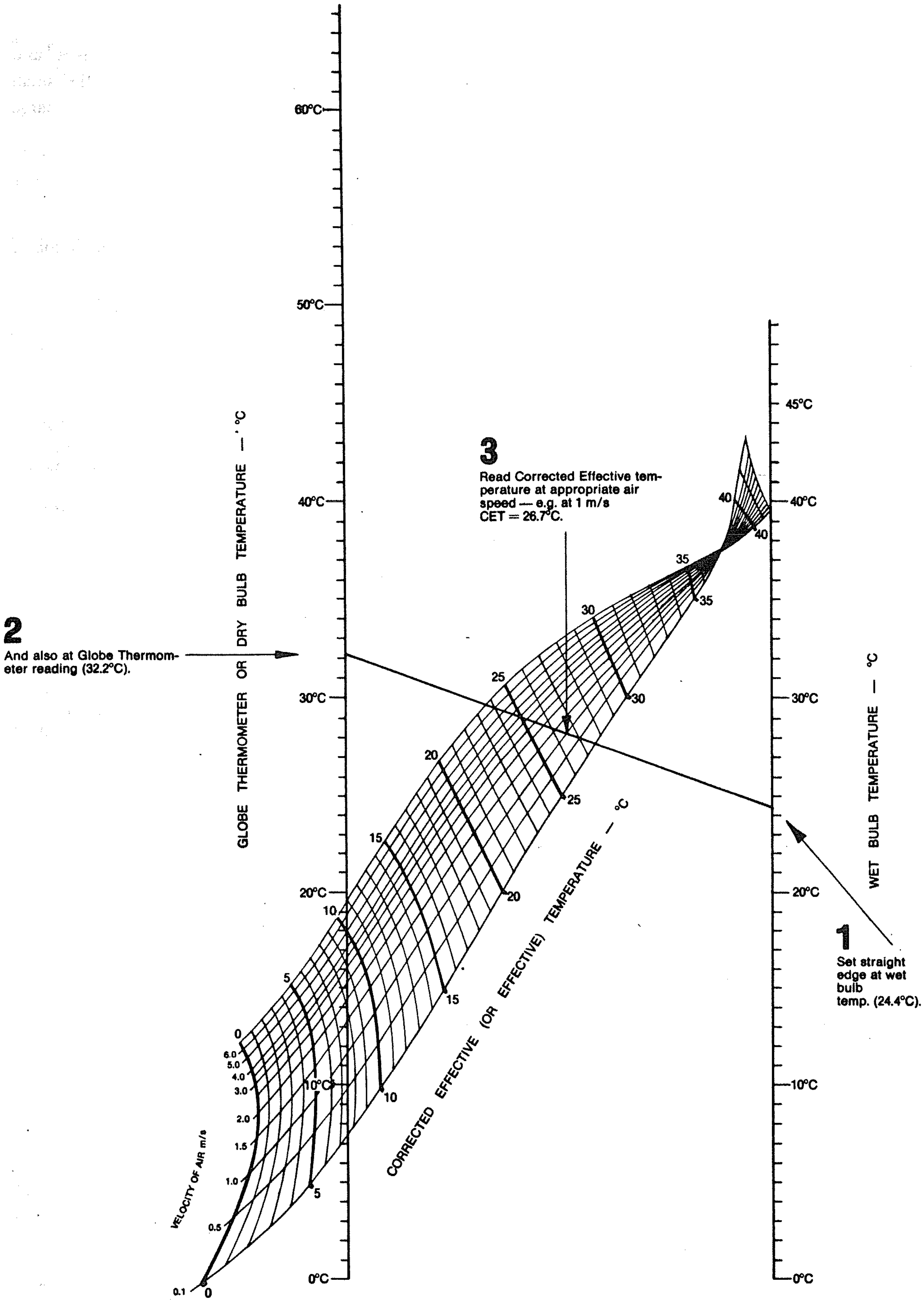


CHART SHOWING NORMAL SCALE OF CORRECTED FFFECTIVE (OR EFFECTIVE) TEMPERATURE[19]

FIG. 6.2

Radiation Conductance—h_r

The simplified formula for radiation conductance which can be used is:—

$h_r = 27 \cdot 71 \times F \times F_e$ W/(m^2.K)

Where F = Configuration Factor
= 0·22 for most building situations

and F_e = Emissivity Factor
= 0·80 for non reflective ceilings
= 0·20 for Aluminium Sheets
= 0·05 for Aluminium Foil Insulation

Notes: The Configuration Factor takes into account the spatial relationship of the small receiving surface with respect to the large emitting surface.

The Emissivity Factor allows for the emissivity of the two surfaces concerned.

A more detailed treatment is given by Billington[14].

Case History No. 1

No. 1 CASE HISTORY DATA	MEASURED				CALCULATED
ROOF TREATMENT	Dry Bulb Temp. °C	Relative Humidity %	Air Movement m/s	Ceiling Surface Temperature °C	Radiation onto top of head having temp. of 31·7°C W/m²
Outdoor Conditions	30·6	51	2·5	—	—
Aluminium Sheet Only	30·6	48	0·4	More than 50	More than 22
Painted Aluminium Sheet ..	31·1	48	0·5	50	89
Aluminium Foil Insulation ..	30·6	51	0·2	43	3·5

A wool store covering several acres, roofed with aluminium sheet, had ceiling treatment which varied from place to place and included untreated aluminium roof sheet, white paint, and aluminium foil insulation.

When moving from area to area there were marked differences in thermal comfort, to such a degree that there were serious complaints from persons employed in the store.

An examination of the table below shows that conditions in all areas were very similar except for radiation which was undoubtedly the source of trouble.

Note that the radiation from the plain aluminium roof sheet was at least 480% higher than that from the aluminium foil insulation; and that from the painted aluminium roof 2500% higher!

The problem was overcome by installing reflective foil insulation. This case history shows the necessity for designing to limit thermal radiation in hot climates by limiting overhead surface temperatures and by using insulating materials with a very low surface emissivity.

Case History No. 2

No. 2 CASE HISTORY DATA	Max. Ceiling Temperature °C	Radiation onto top of head having a temp. of 31·7°C W/m²	Comments
No Insulation	38·3	32	Excessive radiation.
With Aluminium Foil Insulation	32·2	2·4	Acceptable radiation level.

A 160 m² all brick/tiled home in Sydney had no ceiling or roof insulation and as there was no air conditioning the occupants felt the effects of heat in summer. The tiles were removed and reflective insulation was installed under the tile battens. (Blue coated material was used to minimise glare for roof tiling personnel.)

Experimental readings showed that on equivalent hot days the inside ambient was reduced by 2·8K and ceiling temperatures by 6·1K. This reduction in ceiling temperature had a very appreciable effect on indoor comfort.

Inside wall temperatures were also reduced; thus further promoting comfort. The complete results of the two year investigation have been fully reported in other publications[9].

7. INSULATION OF COLD STORES USING REFLECTIVE INSULATION

7.1 Comparison of Reflective and Mass Insulation

It has been shown that "reflective air spaces" are very effective in preventing heat flow, both in walls and in ceilings. In terms of thickness of the traditional insulating material, cork, reflective insulation has the following effectiveness:—

Vertical Spaces: 20 to 100 mm wide, equivalent to 25 mm cork.

Horizontal Spaces: 50 mm deep, equivalent to 50 mm cork.

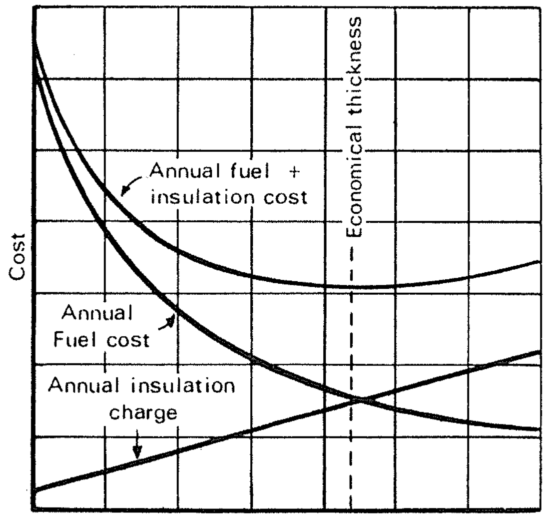
Horizontal Spaces: 100 mm deep, equivalent to 65 mm cork.

(In cold stores, heat flow through ceilings is always downwards.)

Having recognised the effectiveness of reflective insulation it is simply a matter of putting together several air spaces, separated by aluminium foil, with timber battens for spacing, so that any desired amount of insulation can be applied around a cool or cold store. The timber battens have a higher heat conductance than the reflective air space, which is allowed for by designers of reflective insulation systems; as well as which, the direction of the battens is alternated to minimise any direct heat paths. (Fig. 7.3.)

7.2 Selection of Insulation on an Economic Basis

The amount of insulation (i.e. the number of reflective layers) is usually selected on an economic basis, the aim being to strike a balance between initial cost of insulation and the cost of removing heat from the cold store over a period of years by means of refrigeration plant. Fig. 7.1. The "Insulation Quantity Chart" (Fig. 7.2) is a useful aid to designers in selecting the desired amount of insulation.



Economical thickness of insulation

INSULATION COSTS

FIG. 7.1

7.3 Designing Insulation to Prevent Surface Condensation

In selecting the amount of insulation it is important to check that the vapour barrier surface temperature is always higher than the **DEW POINT** of the air with which it is in contact, particularly in very humid areas.

Floor insulation is selected for its thermal properties and also for its many other physical properties. For this reason mass type insulation is normally used in the floors of reflectively insulated stores.

Wall/ceiling and wall/floor junctions need special care to prevent direct heat paths and cold spots on the outside of the insulation. Corners, i.e., wall/wall junctions, need careful attention also.

7.4 Vapour Barriers for Cold Stores

A vapour barrier is a membrane of very low permeability placed on the warm side of insulations, which limits the flow of water vapour into the insulations. Publication No. 759 of the American National Academy of Sciences, "Refrigerated Storage Installations" states that the single most significant factor determining the success or failure of a cold store is the quality of the vapour barrier and the care with which it is installed.

The operation of cold stores insulated with reflective insulation is not critically affected by condensation. Should it happen, it will only occur on the cold side of a cavity. However, for obvious reasons, the presence of water is undesirable and a vapour barrier is formed on the warm side of the insulations by very carefully sealing the outer layer of reflective insulation.

Insulating layers are not sealed, but are lapped approximately 50 mm, thus allowing any water vapour to migrate to the cold side of the structure.

7.5 Design Details

Drawing Number 54 (Fig. 7.3) shows typical construction details for a reflective insulation cold store at the wall/ceiling and wall/floor junctions. This drawing shows typical design for cold stores operating as low as —20°C. Experience has shown cold storage insulations of this kind to be very satisfactory.

7.6 Conclusion

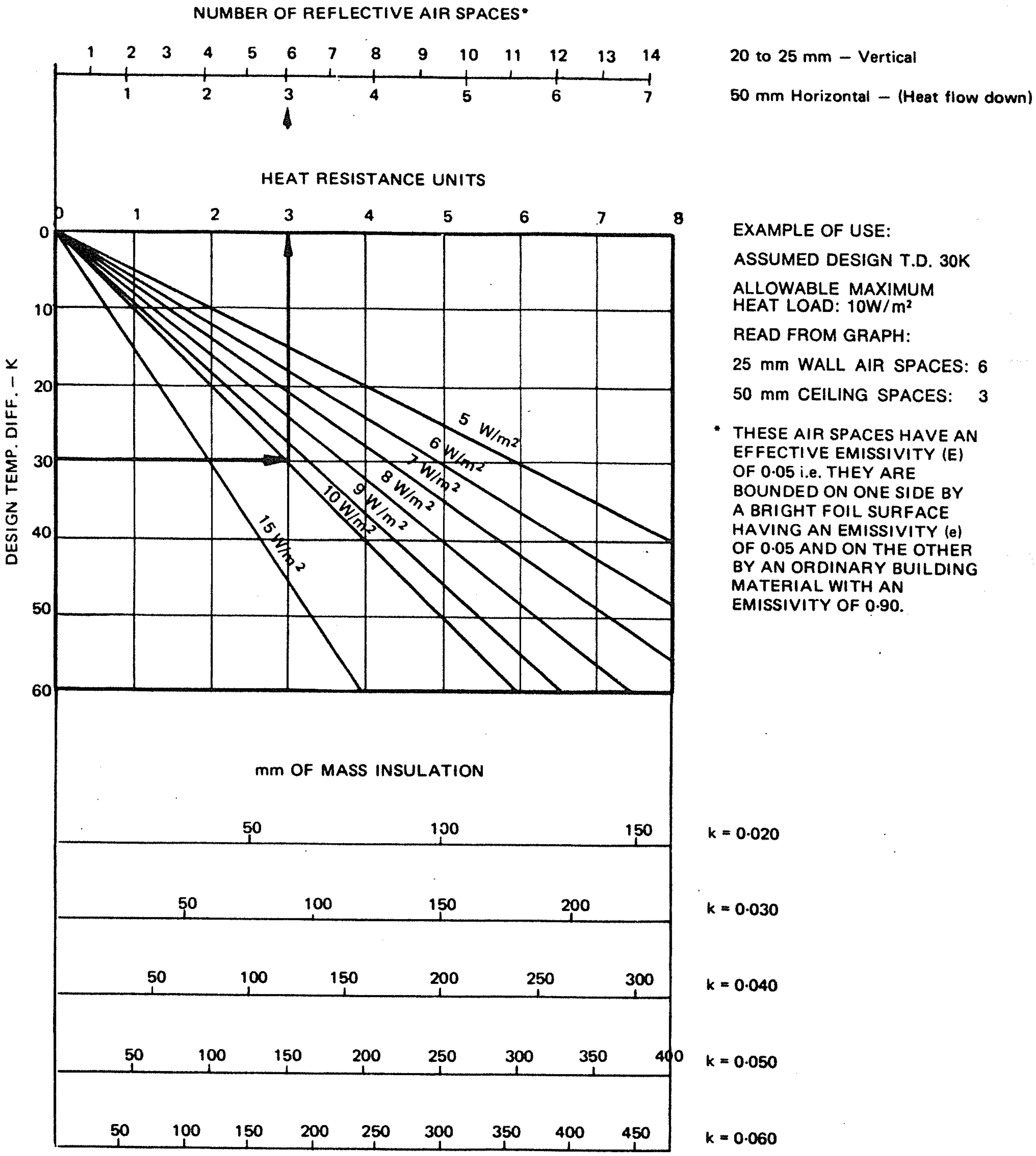
Any medium which is effective in slowing down the transfer of heat is termed an INSULATOR. Traditionally, mass insulation such as cork and mineral wool have been used. Closer examination of these traditional materials shows that **IT IS THE TRAPPED AIR WHICH INSULATES** and not the solid material.

For example, glass-wool has a k value of approximately 0·034 W/(m.K), whereas solid glass has a k value of about 1·050 W/(m.K), i.e., 30 times greater.

Reflective insulation forms air spaces in which convection is controlled, and radiation practically eliminated. The heat flow across reflective air spaces has been extensively studied, both theoretically and under laboratory and field conditions. As a result any desired amount of insulation can be provided by selecting the correct number of reflective layers and installing in accordance with established procedures.

INSULATION QUANTITY CHART

KNOWING DESIGN CONDITIONS, THE REQUIRED AMOUNT OF MASS OR REFLECTIVE INSULATION IS READ FROM CHART



EXAMPLE OF USE:

ASSUMED DESIGN T.D. 30K

ALLOWABLE MAXIMUM HEAT LOAD: 10W/m²

READ FROM GRAPH:

25 mm WALL AIR SPACES: 6

50 mm CEILING SPACES: 3

* THESE AIR SPACES HAVE AN EFFECTIVE EMISSIVITY (E) OF 0·05 i.e. THEY ARE BOUNDED ON ONE SIDE BY A BRIGHT FOIL SURFACE HAVING AN EMISSIVITY (e) OF 0·05 AND ON THE OTHER BY AN ORDINARY BUILDING MATERIAL WITH AN EMISSIVITY OF 0·90.

INSULATION QUANTITY CHART

FIG. 7.2

8. INSULATION FOR WINDOWS

8.1 The Need for Windows

People have a deep seated psychological need to gaze out of windows in order to "relate" themselves to their environment by keeping visual contact with the outside world from within their modern centrally heated or air-conditioned confinement!

The glazed window is the traditional means of providing light during the day and in general, the greater the glass area, the broader the view and the better the lighting.

But glass can admit too much solar heat in summer, creating hot-house conditions and can lose an excessive amount of heat in winter. In both extremes the result is human discomfort and excessive cooling or heating costs.

Some have advocated reduction in window size and even elimination of windows. But this is not acceptable, as the British Building Research Station is finding out with current research. People need a continuity of outside vision, with as wide an arc of skyline as possible with some sky above and ground below the "horizon".

The ideal solution to the problem is to have full sized windows for outside vision, but which will limit the passage of heat energy in either direction. This solution can be achieved by the use of **reflective insulation** on glass—applied when the glass is being manufactured or later as a reflective metallic film of molecular thickness on a polyester plastic sheet substrate. One such commercial product is REFLECTO-SHIELD which is available in two grades, namely RSL20 and RSL40 which, when applied to 3 mm clear glass, transmit directly 14% and 30% respectively of the solar energy.

8.2 Solar Radiation

(a) **Direct Radiation.** The sun's radiation in space (i.e. before it hits the earth's atmosphere) varies from 1351 to 1442 W/m² depending on the distance of the earth from the sun.
In passing through the earth's atmosphere it is scattered and absorbed by dust, gas molecules, ozone and water vapour.
Figure 8.1 shows the wavelengths of solar radiation in space and on earth. For comparison it also shows, to a different scale, the radiation emitted by a black body at 35°C.

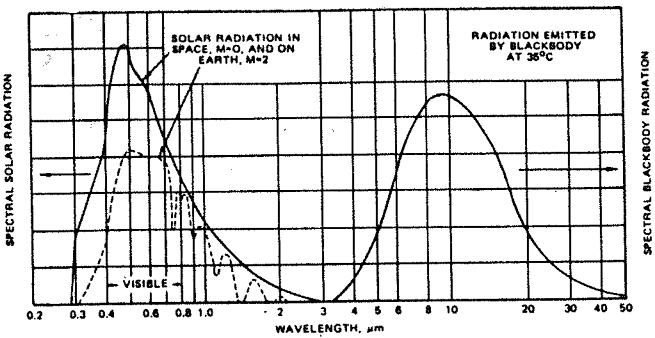


Fig. 8.1

Solar and Blackbody Radiation at Various Wavelengths

Note that the Black Body Radiation is to a different scale – is very much less than Solar Radiation

Fig. 8.1

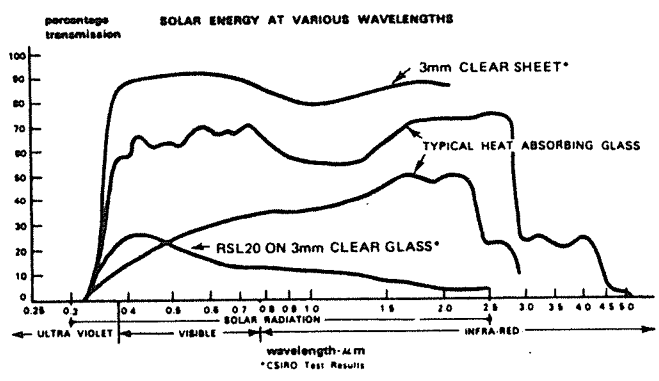


Fig. 8.2

The maximum intensity of solar energy at the earth's surface occurs at 0·05μm (green portion of spectrum). There is relatively little solar energy at wavelengths of less than 0·29μm due to the absorption by ozone in the upper atmosphere. (Sun "tan" is produced best by Ultra-violet radiation around 0·33μm and skin cancer is probably produced by Ultra-violet below 0·31μm with a peak effect at 0·30μm.)

Fading and deterioration of curtains and fabrics is caused by Ultra-violet and also by visible radiation.

(b) **Diffuse Radiation.** This is solar radiation which is scattered by the air and water vapour molecules, dust, etc., and which also penetrates glass areas and causes heat and glare problems even in the absence of direct radiation. It can be of the order of 25% of the direct solar radiation and therefore has also to be accounted for.

(c) **Reflected Solar Energy.** Quite often, a substantial amount of solar energy is reflected from surfaces of large bodies of water and also from nearby walls, roofs and roadways. This should always be allowed for when designing for Solar Control. The C.S.I.R.O. is currently publishing new radiation tables[26] which include figures for radiation reflected from the ground.

8.3 Solar Radiation and Glass

When the sun's energy hits glass, some is reflected, some is absorbed and some is transmitted, depending on angle of incidence, reflective treatments, chemicals in glass, etc. Absorbed heat raises the temperature of the glass until a temperature balance is achieved, in that absorbed heat is re-transported (by low temperature re-radiation, conduction and convection) to both sides of the sheet, at the same rate as solar radiation is absorbed.

Experimentally determined Solar-Optical properties of 3 mm clear sheet glass for normal incidence are:—

REFLECTANCE	·08
ABSORPTANCE	·05
TRANSMITTANCE	·87
TOTAL	1·00

The total will always be 1·00 or 100%.

3 mm soda lime sheet is used as a reference STANDARD as described in the A.S.H.R.A.E. Handbook of Fundamentals, 1967.

Figure 8.10 shows that the greatest heat gain is when the solar beam is "normal" to the glass but for practical purposes it is fairly constant within the incident angle range 0° to 30°.

Figure 8.3 shows the simplified basic geometry needed when analysing the effect of solar radiation on glass. Note that when the Aspect Angle is 0° (e.g. a vertical, north facing window at solar noon), the Incident Angle (I) is equal to Altitude.

If the angle of incidence is required it can be formed using the following expression:—

$$\cos I = \cos Alt \times \cos H.S.A.$$

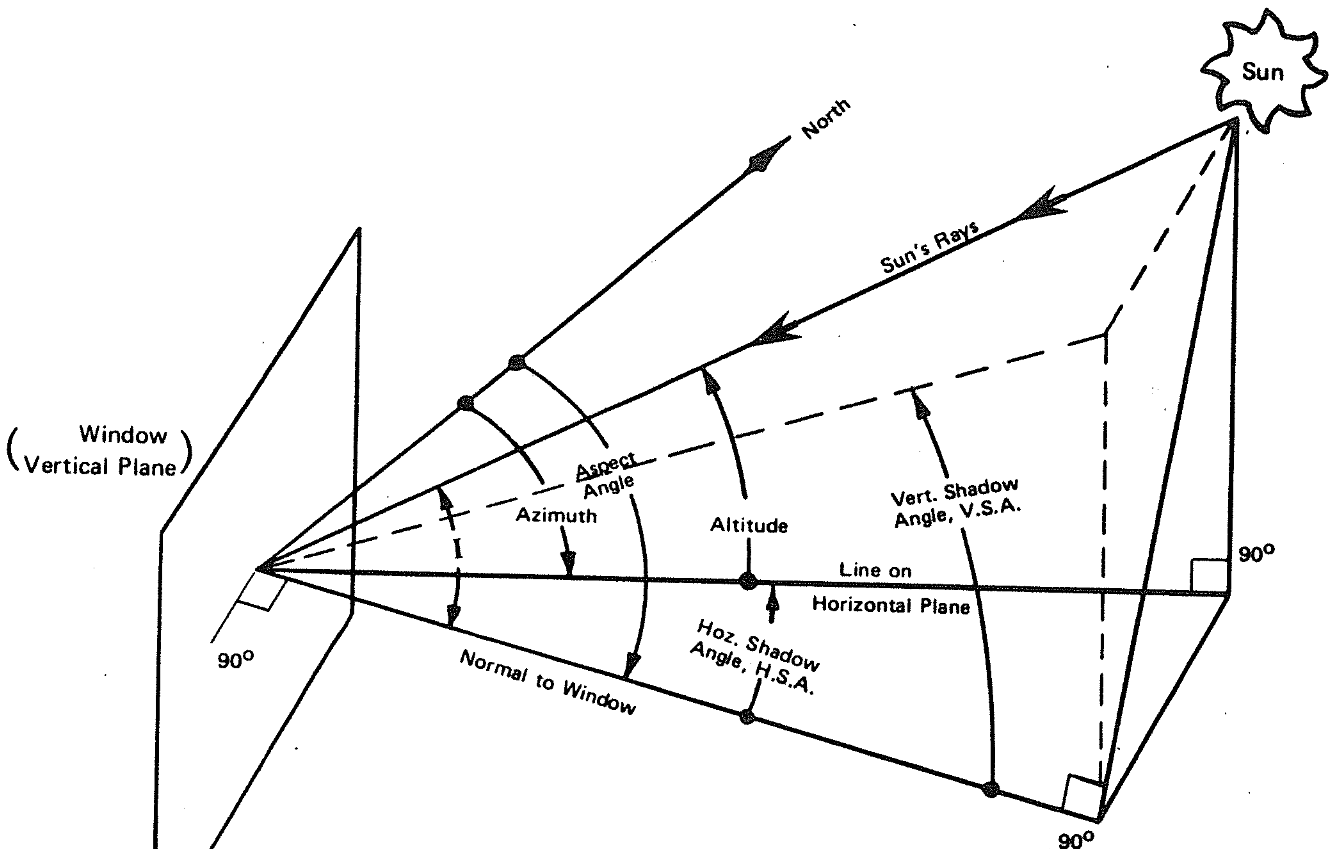


Fig. 8.3

GEOMETRY OF SOLAR RADIATION

8.4 Heat Transfer Through Standard 3 mm Clear Glass

The **TOTAL HEAT GAIN** (see right hand side of Fig. 8.4) includes both SOLAR HEAT GAIN and CONDUCTED HEAT due to the temperature difference across the glass. The latter value can be either positive or negative and applies whether or not the sun is shining. For simplification it will not be considered until the Co-efficient of Thermal Transmittance (U value) of single and double glazing has been examined in a later section.

In the case of the STANDARD 3 mm glass, the heat gain is called the **SOLAR HEAT GAIN FACTOR** (SHGF). Tables for its value have been published by ASHRAE[5] as well as the Australian C.S.I.R.O.[26]. The latter set of tables will be found more useful for Australian conditions as they are for latitudes specific to major Australian cities and they also include an allowance for extra radiation reflected from surrounding ground, assuming a reflectance of 0·2 (green lawn).

Further allowances should be made for heat reflected from other unusual surfaces such as large bodies of water or nearby large buildings.

The term **SOLAR HEAT GAIN** is used when fenestration other than the standard is involved and its value is related to the SOLAR HEAT GAIN FACTOR for standard glass using the SHADING CO-EFFICIENT (see next section).

The SHGF applies for clear skies and its value depends on:

Aspect of window (assumed vertical)
Latitude
Time of year
Time of day

Heat gain can be assessed on an **instantaneous** or **day long** basis.

The absorbed heat increases the glass temperature until a balance is attained in that the heat is "re-transported" to the exterior and interior of the building by convection, conduction and radiation at the same rate as it is absorbed.

8.5 Solar-Optical Properties of Glass

The Solar-Optical properties of clear and treated glass for radiation having approximately normal incidence are determined experimentally. Table 8.1 includes figures obtained by C.S.I.R.O. and also others quoted by manufacturers.

TABLE 8.1

Solar-Optical Properties Glass with Various Treatments		Solar-Optical Properties			Transmission of	
		Reflectance r	Absorptance a	Transmittance t	Visible Light	Ultra-Violet
	ASHRAE Standard	.08	.05	.87		
3 mm	Clear Sheet Glass					
	– Untreated *	.09	.05	.86	approx. 86%	80%
	– with RSL20 *	.56	.30	.14	17%	25%
	– with RSL40 *	.38	.32	.30	37%	45%
6 mm	Clear Plate Glass *					
	– Untreated *	.06	.20	.74	approx. 80%	67%
	– with RSL20 *	.44	.44	.12	15%	19%
	– with RSL40 *	.29	.44	.27	36%	38%
6 mm	Heat Absorbing Grey Glass *	.055	.56	.385	39%	
6 mm	Heat Reflecting 15/23 Gold "Solarshield"	.48	.43	.09	15%	
6 mm	"Coldlite" Heat Absorbing Glass	.05	.65	.30	64%–70%	

* C.S.I.R.O. Test Results

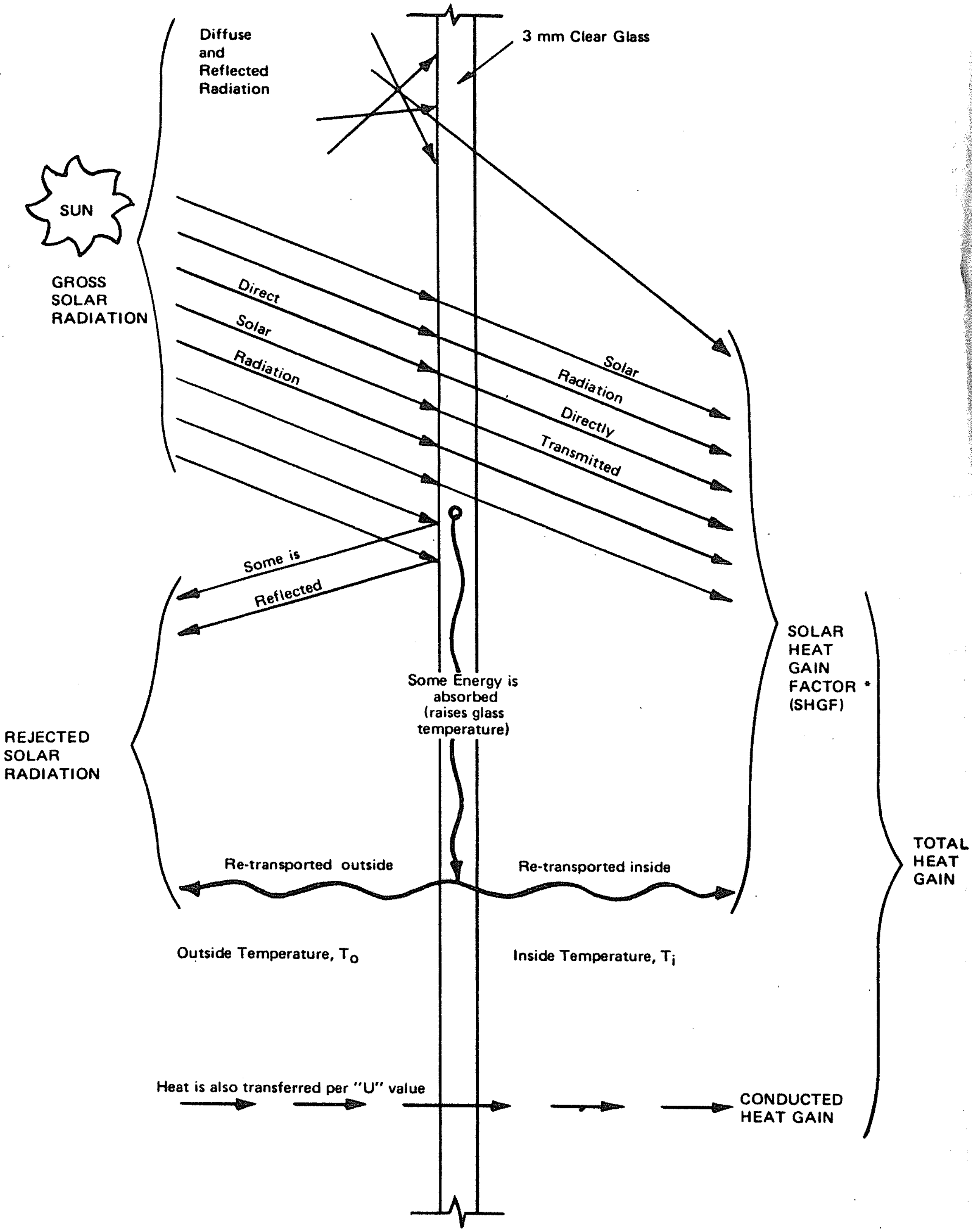


FIG. 8.4

REFLECTED, ABSORBED, TRANSMITTED AND CONDUCTED
ENERGY THROUGH CLEAR GLASS

8.6 Shading Co-efficient

The Shading Co-efficient—is the ratio of the **solar heat gain** through any window to the **solar heat gain factor** for standard glass under exactly the same conditions. Values of Shading Coefficient for various windows are given in Table 8.2.

i.e. SHADING COEFFICIENT =

$$\frac{\text{SOLAR HEAT GAIN (any glass)}}{\text{SOLAR HEAT GAIN FACTOR (3 mm GLASS)}}$$

Explanation of "Standard" Conditions:—We have previously quoted the Solar-Optical properties of the Standard sheet of glass as being—

Reflectance (r) — 0·08

Absorptance (a) — 0·05

Transmittance (t) — 0·87

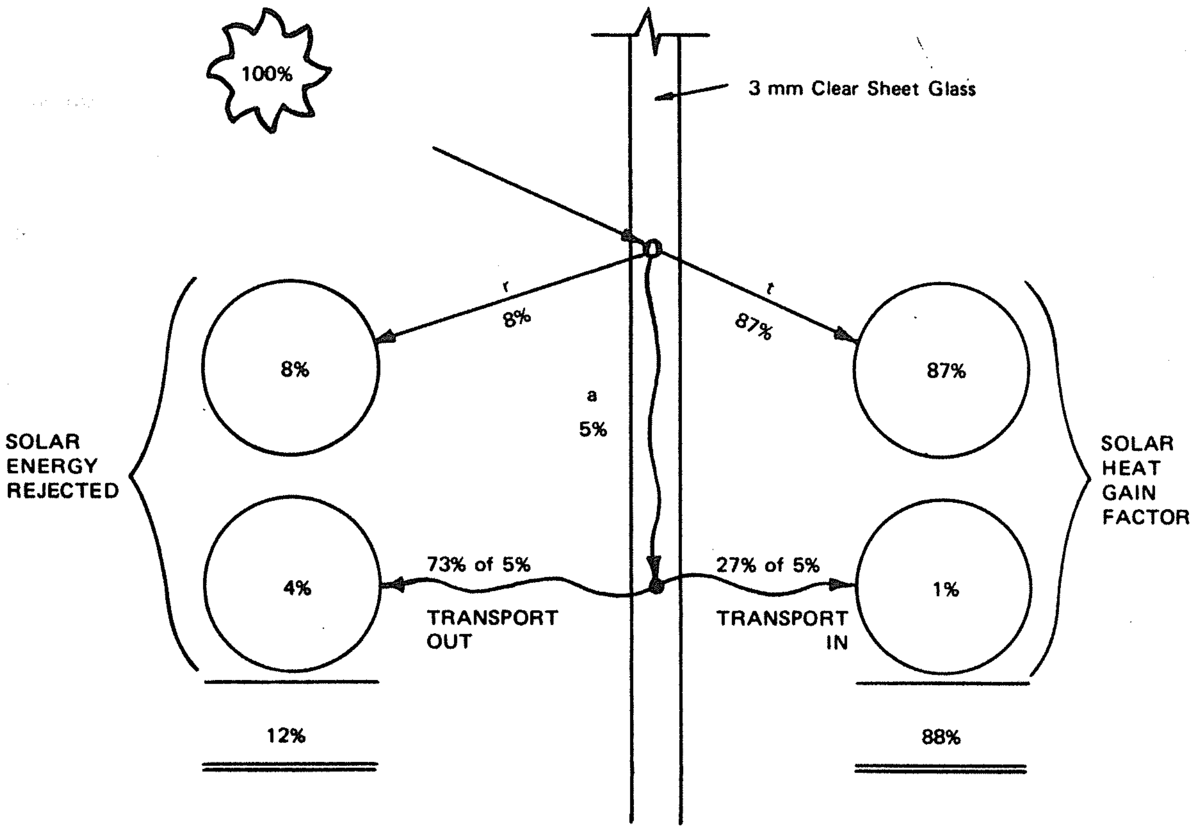


Fig. 8.5

HEAT BALANCE DIAGRAM: Standard 3mm Clear Sheet Glass

(Note that the figures in this diagram are percentages of the gross solar radiation).

SHADING COEFFICIENT = $\frac{88}{88}$ = 1.00

— (Take in Fig. 8.5 and Note) —

$$\text{SHADING COEFFICIENT} = \frac{88}{88} = 1\cdot 00$$

Heat Balance for 3 mm Clear Sheet—Reference Glass. (Refer Fig. 8.5).

The ASHRAE standard calculation method assumes 3·35 m/s air outside. i.e. a surface resistance of 0·044 HRU and still air inside, i.e. a surface resistance of 0·120 HRU. Thus it can be shown that 73% of the absorbed heat is re-transported to the outside of the building.

The standard calculation method results in a solar heat gain factor which is 88% of the Gross Solar Radiation. This is illustrated in the heat balance diagram for 3 mm standard glass (Fig. 8.5).

It will readily be seen, that once the Solar-Optical properties of a glass are known, the Shading Coefficient can be calculated as follows:—

$$S\ C = \frac{t + 27\%\ \text{of}\ a}{\cdot 88}$$

Shading Coefficients can also be determined experimentally using a "Solar Calorimeter"

Under non-standard conditions, with still air outside a window and moving air inside, it is possible for only 40% of absorbed heat to be re-transported outside and as much as 60% to be re-transported to the inside. This will adversely effect all Shading Coefficients but will have least effect on reflective glass as "t" is high, relative to "a".

Shading Coefficients for a number of glass treatments are listed in table 8.2.

Table 8.2

SHADING COEFFICIENTS FOR VARIOUS WINDOW TREATMENTS

3 mm Clear Glass (Reference Standard)	1·00
3 mm Clear Glass with Reflecto-Shield RSL40	0·44
3 mm Clear Glass with Reflecto-Shield RSL20	0·25
6 mm Clear Glass	0·90
6 mm Clear Glass with Reflecto-Shield RSL40	0·45
6 mm Clear Glass with Reflecto-Shield RSL20	0·27
6 mm Clear Glass with Typical Green Flow Coat	0·70
6 mm Typical Green Heat Absorbing Glass	0·67
6 mm Heat Absorbing Grey Glass	0·61
6 mm Typical Heat Reflecting Glass (Solarshield 15/23 — Gold)	0·24
Double Glazing (Clear + Clear)	0·90
Double Glazing (Clear + H.A.)	0·56
Venetian Blinds — Light Colour	0·55
— Medium Colour	0·64
Roller Shade — White	0·40
— Medium	0·62
Net Curtains with folds (fairly dark)	0·75
White Curtain lining with folds	0·45
Heavy Curtains with white lining and folds	0·35
Miniature Louvres — dark 17/inch	0·49–0·13

8.7 Heat Balance Diagrams

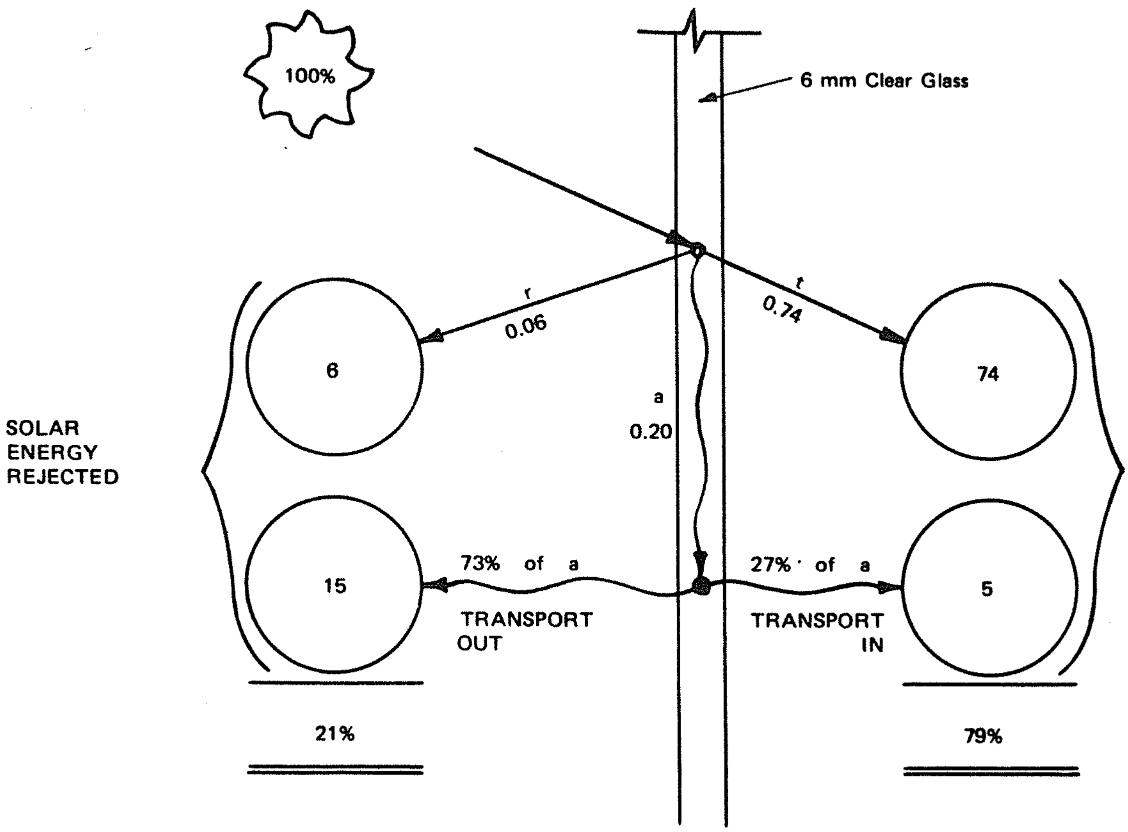


Fig. 8.6

HEAT BALANCE DIAGRAM: 6 mm Clear Plate Glass

SHADING COEFFICIENT $= \frac{79}{88} = 0.90$

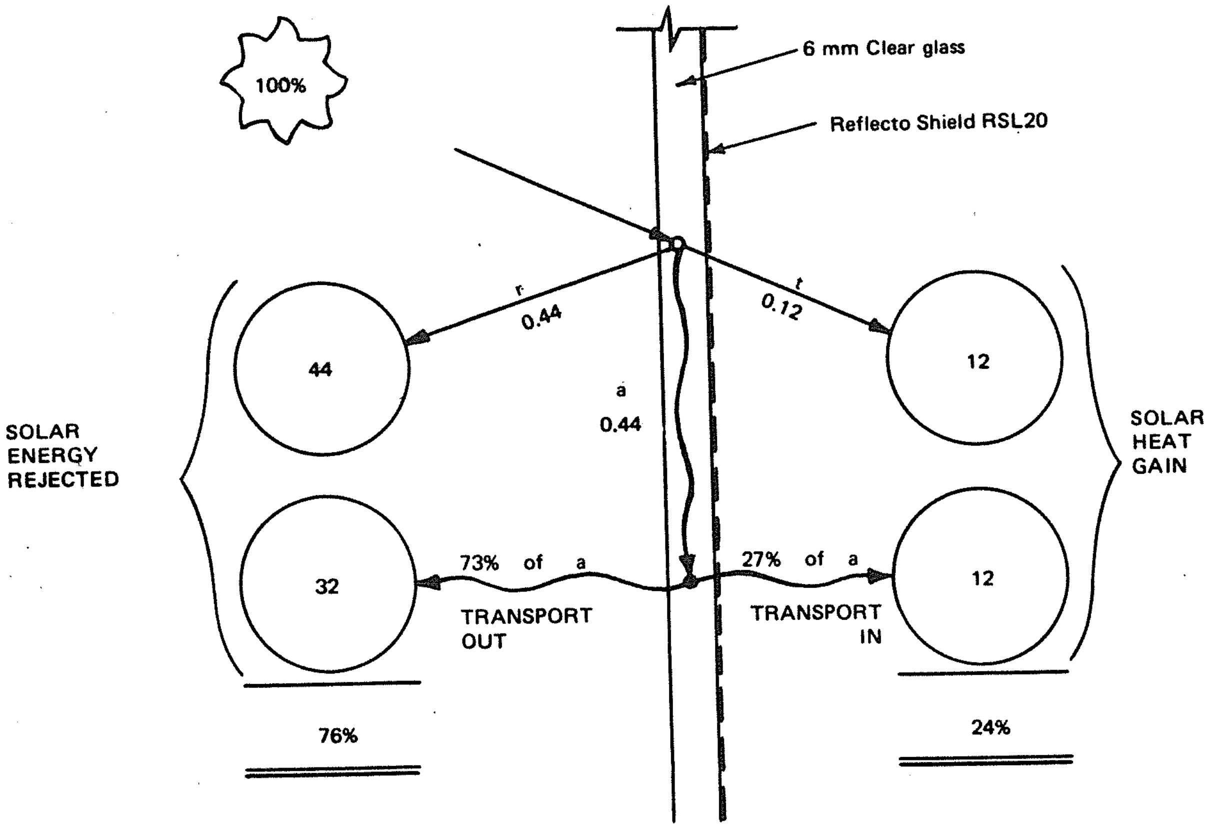


Fig. 8.7

HEAT BALANCE DIAGRAM: 6mm Clear Glass + Reflecto-Shield RSL20

SHADING COEFFICIENT $= \frac{24}{88} = 0.27$

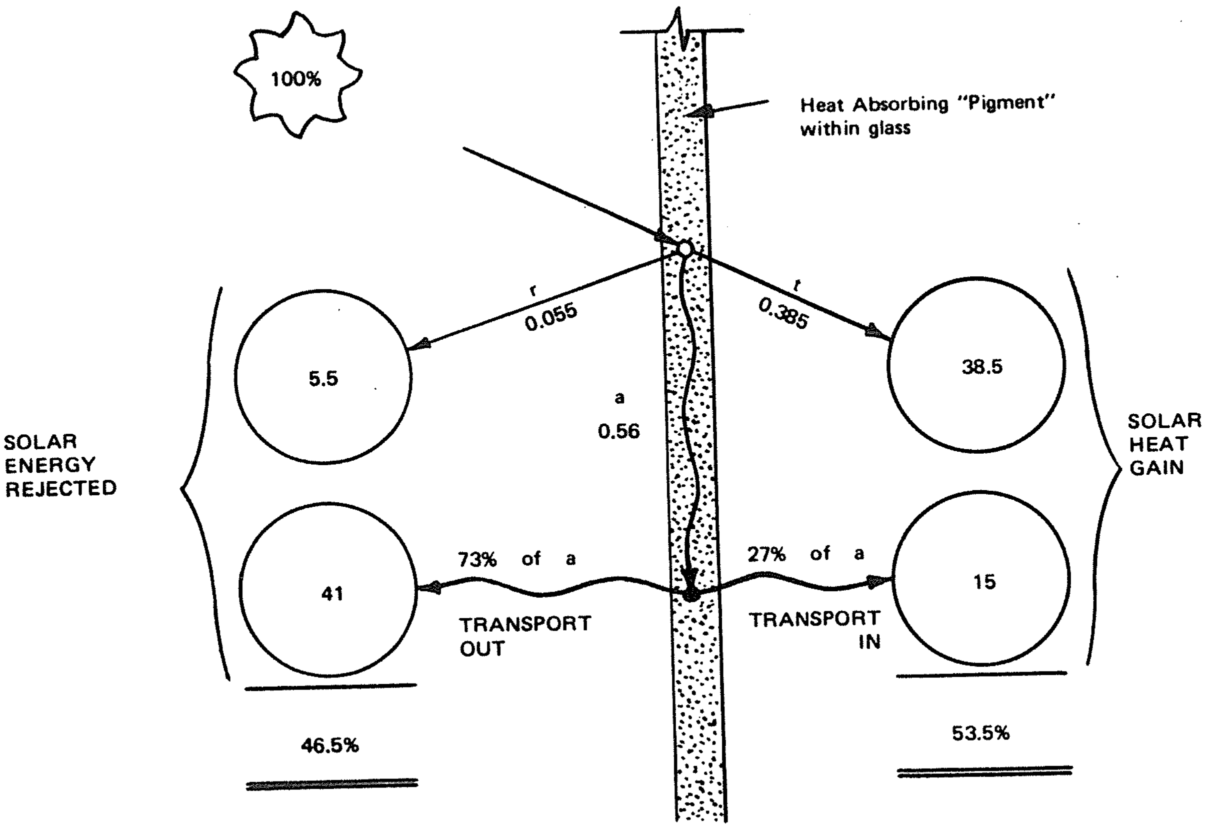


Fig. 8.8

HEAT BALANCE DIAGRAM: 6mm Heat Absorbing Glass

SHADING COEFFICIENT $= \frac{53.5}{88} = 0.61$

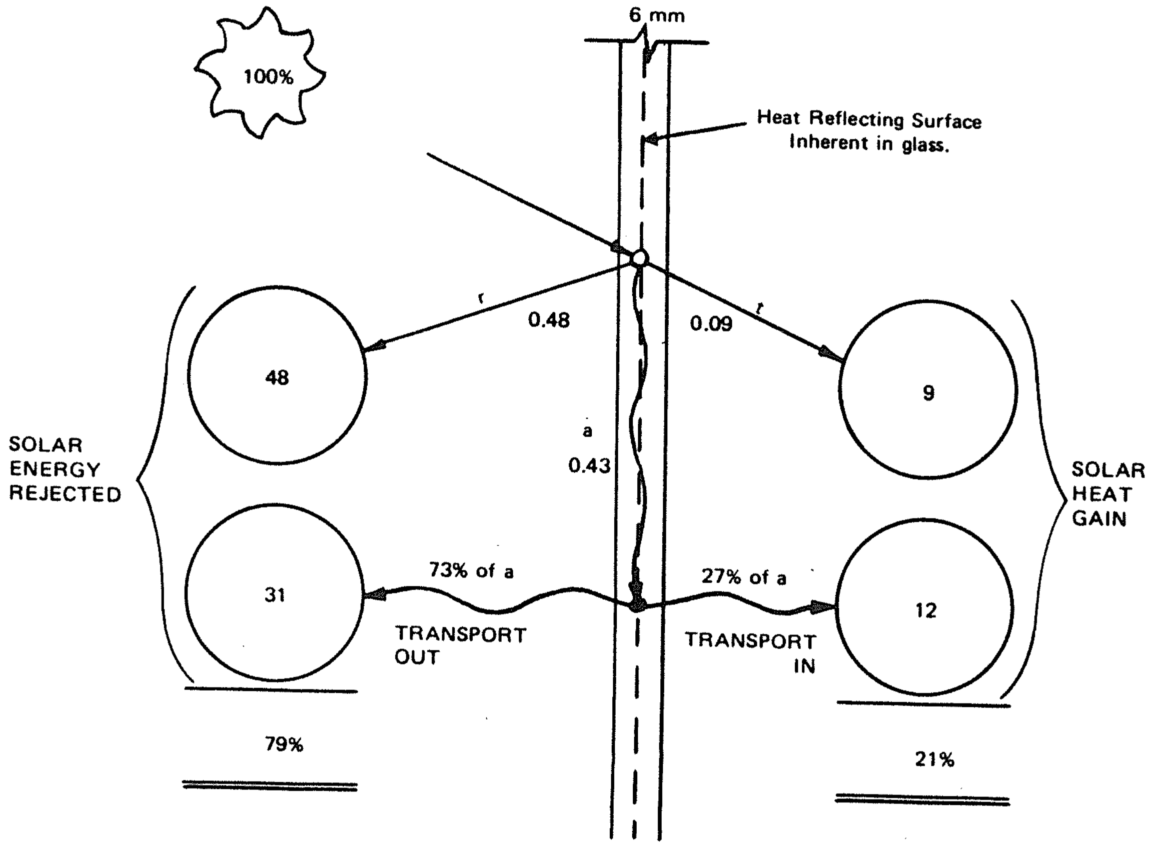


Fig. 8.9
HEAT BALANCE DIAGRAM: 6mm 15/32 Gold "Solarshield" – Pilkington's Heat Reflecting Glass

SHADING COEFFICIENT $= \frac{21}{88} = 0.24$

8.8 Angle of Incidence and its Effect on Heat Transmission Through Glass

(a) **SHGF.** Figure 8.10 shows how the Solar Heat Gain Factor, i.e. the percentage of Gross Solar Radiation which penetrates 3 mm Standard glass, reduces with increase of Incident Angle.

(b) **SHG.** Figure 8.11 shows how the Solar Heat Gain (for all window treatments other than the Standard) reduces with increase of I.

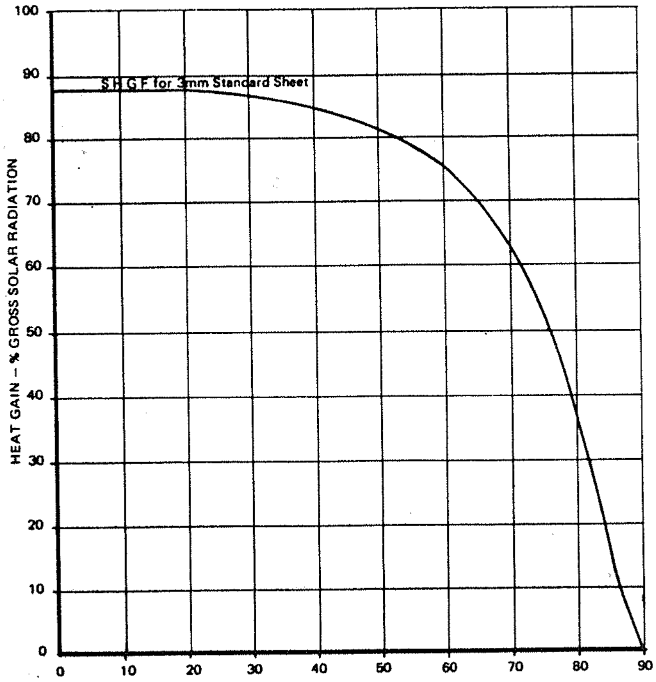


Fig. 8.10

SOLAR HEAT GAIN FACTOR (SHGF)
For 3mm STANDARD CLEAR GLASS

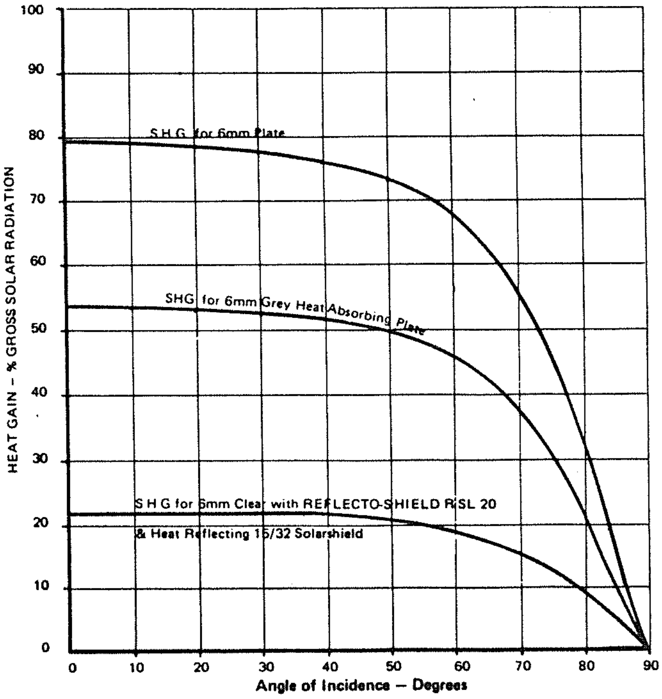


Fig. 8.11

SOLAR HEAT GAIN (SHG) FOR VARIOUS
GLASS AND GLASS TREATMENTS

8.9 Using Radiation Mask to find SHGF and SHG

If radiation tables are not available the Solar Heat Gain through a window at any instant of the day or year can be estimated using the Solar Charts in "Sunshine and Shade in Australasia"[25] and the transparent plastic "Radiation Mask" which is available with the publication.

The procedure is as follows:—

(a) **Mark the sun's position** on the appropriate Solar Chart.

(b) **Read ALT** (Altitude) **and HSA** (Horizontal shadow angle), and calculate I (angle of incidence) from the expression

$$\cos I = \cos ALT \times \cos HSA.$$

(c) **Use Radiation Mask** on Solar Chart. Read Density Factor for radiation **on** a vertical surface. Multiply Factor so found by 1000 to obtain the approximate Gross Solar Radiation incident **on the vertical surface**—W/m^2.

(d) **Find SHGF** from figure 8.10 for the particular value of I determined in (b) above. (This is the amount of radiation **penetrating** 3 mm standard glass—W/m^2.).

(e) **Use SC** (Shading Coefficient). Multiply SHGF by SC. This given the Instantaneous SHG for the type of window treatment being considered.

The above procedure can also be applied on a day-long basis to find the SHG in Wh or J per day.

Note that the "Total Heat Gain" (not yet determined) will also include an addition or subtraction to account for the "air to air" conducted heat which depends on the "U" value of the glass and its treatment.

8.10 "U" Values for Single and Double Glazing

Before calculating the "Total Heat Gain" through glass, the "U" values for air to air conduction of heat must also be known. "U" is the "Coefficient of Thermal Transmittance" and should not be confused with Solar Energy Transmittance of glass (t). The units of "U" are $W/m^2.K$.

Clear untreated glass and heat absorbing glass has a reflectivity for low temperature radiation of 0·10 approximately whereas glass treated with REFLECTO-SHIELD is partly reflective to low temperature radiation. The C.S.I.R.O. has determined a reflectivity of 0·35 for Reflecto-Shield RSL20 and this figure is used in Fig. 8.12 together with values from ASHRAE (Table 4.3) to establish the surface resistance of glass used in the ensuing calculations.

In the presence of intense solar radiation the difference in "U" value due to a partly reflective inner surface, is of little importance, but at night or under conditions of heat loss, it can be significant.

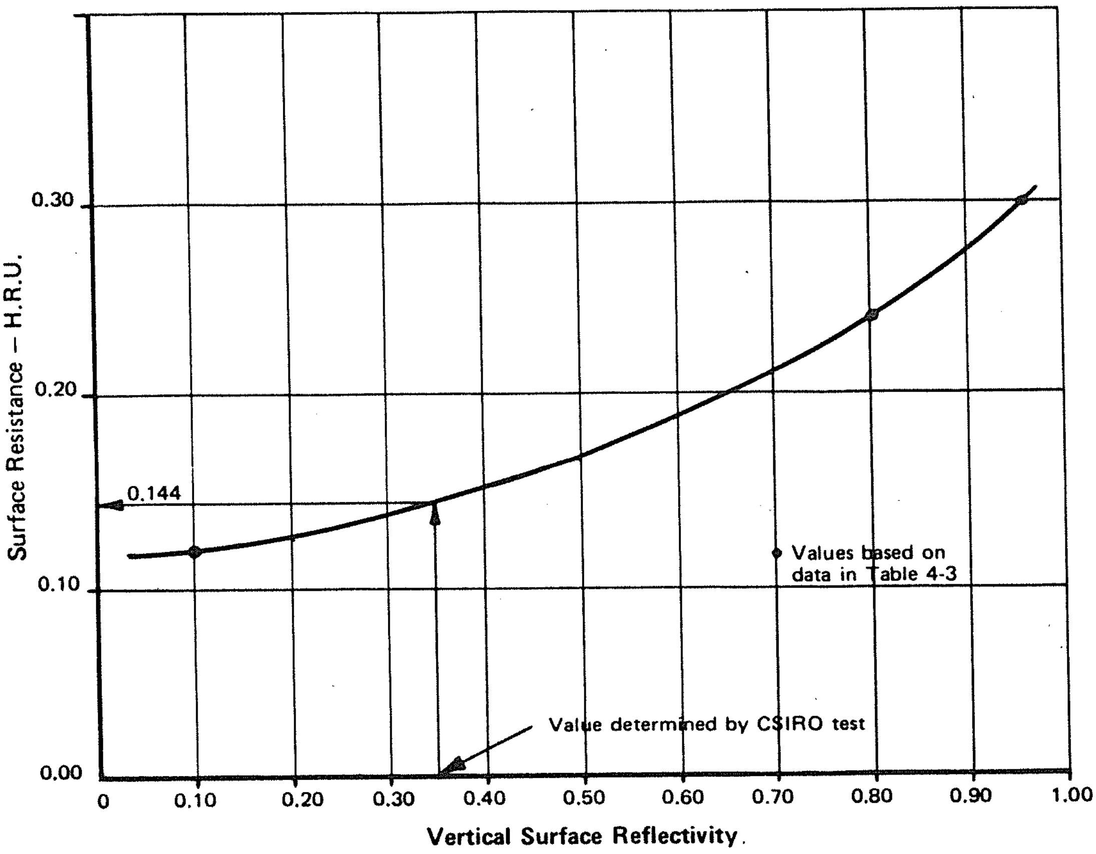


Fig. 8.12

Method of Determining Thermal Resistance of Surface treated with Reflecto-Shield RSL20

U Value Calculations.
(a) **Single Glazing.**

Table 8.3 **Thermal Resistance Values**

	3 mm. Plain Glass (Standard)	6 mm. Clear Plate or Heat Absorbing	6 mm. Clear + RSL20
Outside air layer — 3·35 m/s	·044	·044	·044
Glass (k = 1·050)	·003	·006	·006
Inside air layer—still air	·120	·120	·144
TOTAL R H.R.U.	·167	·170	·194
U, W/m².K	5·98	5·88	5·15
% Reduction in heat flow.	—	—	13%

Thus it is seen that the use of REFLECTO-SHIELD and heat reflecting glass can reduce heat loss under winter conditions by up to 13%.

(b) **Double Glazing** (20 to 100 mm between sheets).

Table 8.4 **Thermal Resistances**

Outer Glass	3 mm Clear	6 mm Clear or 6 mm H.A.	6 mm + RSL20 (inner surface)	6 mm Clear
Inner Glass	3 mm Clear	6 mm Clear	6 mm Clear	6 mm Clear + RSL20 (inner surface)
Outside Air Layer (3·35 m/s)	·044	·044	·044	·044
Glass Sheet	·003	·006	·006	·006
20 mm Air Space	·169	·169	·176 (Interpolation)	·169
Glass Sheet	·003	·006	·006	·006
Inside Air Layer (still air)	·120	·120	·120	·144
TOTAL R	0·339	0·345	0·352	0·369
$U = \frac{1}{R}$, W/(m²K)	2·95	2·90	2·85	2·72
Percent reduction in heat flow compared with using 2 sheets 3 mm clear glass	—	1·7%	3·4%	8%

8.11 Total Heat Gain

(i) **Instantaneous Total Heat Gain.**

The Total Heat Gain is the sum of the Solar Heat Gain and the Air/Air Conducted Heat.

i.e.

THG = SHG + U × Temp. Diff.

or THG = SHGF × SC + U × T.D.

EXAMPLE:

Consider an east window in Brisbane at 8 a.m. on 22nd January. The window is single glazed with 6 mm clear glass. What is the instantaneous Total Heat Gain without and with Reflective Window Insulation (Reflecto-Shield RSL20)?

DATA:

SHGF from C.S.I.R.O. Tables	— 720 W/m^2
SC from table 8.2, 6 mm clear glass	— 0·90
same with RSL20	— 0·27
U from Section 8.10, 6 mm clear	— 5·88 $W/(m^2K)$
same with RSL20	— 5·15 $W/(m^2K)$
T.D. 30°C-22°C	— 8K

(a) **6 mm Clear Glass.**

THG = 720 × 0·90 + 5·88 × 8
= 695 W/m^2 or 0·695 kW/m^2

(b) **6 mm Clear Glass + Reflecto-Shield RSL20.**

THG = 720 × 0·27 + 5·15 × 8
= 241 W/m^2 or 0·241 kW/m^2

Thus the instantaneous reduction in Total Heat Gain due to the use of reflective glass treatment is, in this example, 0·454 kW/m^2 or 65%.

(ii) **Day Total Heat Gain.**

This can also be illustrated by applying the same example quoted in (i) above.

Additional Data:

Day Total SHGF (from C.S.I.R.O. tables)	3890 Wh/m^2
Assume average temperature difference	5K
Length of day	12h

(a) **6 mm Clear Glass.**

THG = (3890 × 0·90) + (5·88 × 5 × 12) = 3873 Wh/m^2 or 3·873 kWh/m^2

(b) **6 mm Clear Glass + Reflecto-Shield RSL20.**

THG = (3890 × 0·27) + (5·15 × 5 × 12) = 1558 Wh/m^2 or 1·558 kWh/m^2

Thus the reduction in "Day-Long" Heat Gain is 2·315 kWh/m^2 or 60%.

Similar calculations can be performed for any type of glass and glass treatment and any design temperature difference, simply by using the appropriate Shading Coefficients "U" values and temperature differences.

8.12 Reduction of Air Conditioning Costs

Reflective window insulation can reduce air conditioning costs in two ways:—

(i) By reducing Plant Requirements, thus reducing space, equipment, and capital charges.

(ii) By reducing running costs.

(i) **Plant Requirements.** In the example quoted in Section 8.11 the reduction in heat ingress through the glass at a time of peak solar loading was 0·454 kW/m^2.
If the Capital Cost of Installed plant is for example $500 per kW of capacity, the cost saving is (500 × 0·454) $227 per m^2 of glass treated with RSL20 or of heat reflective glass.
Therefore the peak need for air conditioning plant is reduced depending on the area of reflected glass.

(ii) **Running Costs.** Refer again to Section 8.11. These are also reduced. If, for example, every kW of installed capacity had a power requirement of at least 1·0 kW and the power cost were 3·31 cents per kWh then, the day-long saving would be:
2·315 × 3·31 cents or 7·67 cents per day for every m^2 of reflective glass.

8.13 Sundry Features of Reflective Glass Insulation

(i) **Glass Temperature and Thermal Comfort.** It is a fact that heat absorbing glass, because of its mode of operation, increases in temperature until a balance with the surrounding environment is established. Van Straaten[16] found that, generally speaking, the higher the absorptance of the glass, the higher the temperature it attained, (see fig. 8.13). He states, "Another aspect of the use of special glasses is the alarmingly high temperatures to which they can rise. According to the test results, elevations of 22K and more above the outdoor air temperature are not uncommon, and radiation from such surfaces can result in serious thermal discomfort."
Reflective window insulation, which has a high reflectance and a relatively low absorptance does not suffer from this disadvantage.

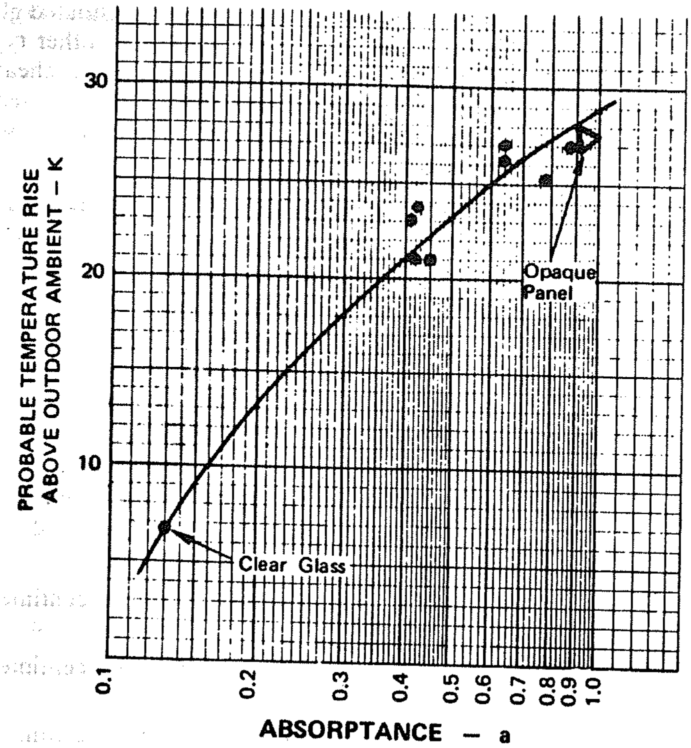


Fig. 8.13

Graph shows relationship between Absorptance of glass and Temperature rise under hot conditions.

- LOTZ & van STRAATEN [16]: Glass temperature elevations represent the maximum values recorded during the course of actual experiments under conditions of relatively high solar radiation intensity and low wind speed.
- ▲ COTTONY & DILL [21]: Temperature rise of black opaque panel.

(ii) **Reduction of Light and Visual Acuity.**
Light is essential for seeing. However, the reduction of light due to reflective window insulation does not reduce visual acuity in the same proportion as heat penetration is reduced, because the eyes response to light is logarithmic and not linear. This is illustrated in figure 8.14 which shows that a light reduction of the order of 80%, only results in a reduction of about 10% in visual acuity.

Any light reduction must be considered together with not only the heat reduction but also with the corresponding beneficial **glare reduction** due to the reduction of contrast between the window light intensity and that of the interior surfaces, particularly the "deep" spaces. Extra artificial lighting may also be desirable.

NATURAL LIGHTING MAINTAINED

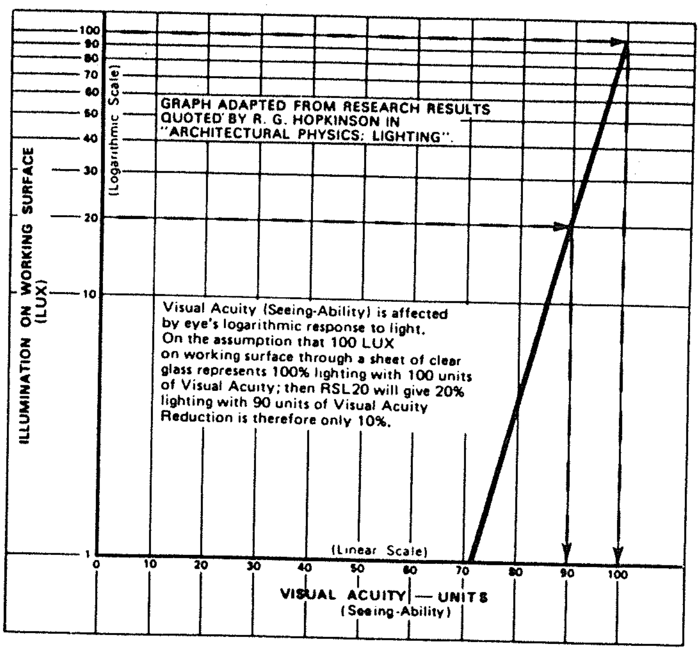


Fig. 8.14

(iii) **Luminous Efficacy.**

The term "luminous efficacy"[27] refers to the relationship between the light generated and the heat produced by a light source, whether it be an ordinary incandescent globe or a sunlit pane of glass.

Luminous Efficacy of a light source is spoken of in **"Lumens per Watt"**, which is an important figure, particularly in air-conditioned buildings, because heat generated within a building has to be dealt with by the air-conditioning plant.

The theory will not be delved here but it is sufficient to state that for windows:
Luminous Efficacy

$$= \text{a "factor"} \times \frac{\text{Visible Light Transmittance}}{\text{Shading Coefficient}}$$

where the "factor" is 125 for direct sunlight, and 170 for diffuse light from a clear blue sky.

Table 8.5

Luminous Efficacy of Various Light Sources	Luminous Efficacy (Lumens per Watt)
Tungsten filament lamps	20
Fluorescent lamps	70
6 mm H.A. Plate Glass	78
3 mm Clear sheet with RSL20	92
3 mm Clear sheet with RSL40	95
3 mm Clear sheet untreated	110

Table 8.5 shows that reflectively insulated glass has a better luminous efficacy than other types of window treatment. It is also a cheaper form of lighting than artificial light in terms of the proportion of "heat" that is mixed with the light.

(iv) **Reflection of Light from Heat Reflecting Glass.**

It might be thought that the extra reflection of sunlight from heat reflecting glass will give troublesome glare with danger to traffic, etc.

This is not so, because solar reflection from clear, untreated glass is already blinding in its intensity, and extra reflection due to heat reflecting glass will make little difference to human eyes.

Luminous intensity figures are approximately as follows:—

Sun at meridian—
160,000 candelas per square centimeter.

Sun near horizon—
600 candelas per square centimeter.

Fluorescent Lights—
1 candela per square centimeter.

Clear Sky—
0.8 candelas per square centimeter.

The eye can tolerate a maximum intensity of just over 1 candela per square centimeter, that is, approximately the intensity of a fluorescent tube, without undue strain. If clear, untreated glass reflects a very minimum of 5% of the sun's luminance on the horizon, the luminance reflected is still 30 candelas per square centimeter (i.e. 5% of 600) which is many times that which the eye can tolerate.

Therefore the added reflection due to REFLECTO-SHIELD will make no difference to external glare.

If a case of solar reflection causes traffic problems, the remedy lies in the orientation of the windows themselves.

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22. *Pivovonsky, M. and Nagel; M. R. "Tables of Black Body Radiation Functions". New York, McMillan, 1961.*

23. *Standards Assn. of Aust. AS 1000—1970. "The International System (S.I.) Units and their Application.*

24. *Standards Assn. of Aust. AS 1155—1971. "Metric Units for Use in the Construction Industry".*

25. *Phillips, R. O. "Sunshine & Shade in Australasia" — Commonwealth Experimental Building Station Sydney 1963.*

26. *Spencer, J. W. Solar Position and Radiation Tables for Various Australian Cities. Published by C.S.I.R.O. Division of Building Research.*

27. *Stephenson, D. G. "Reflective Glazing Units", National Research Council—Canada 1968.*

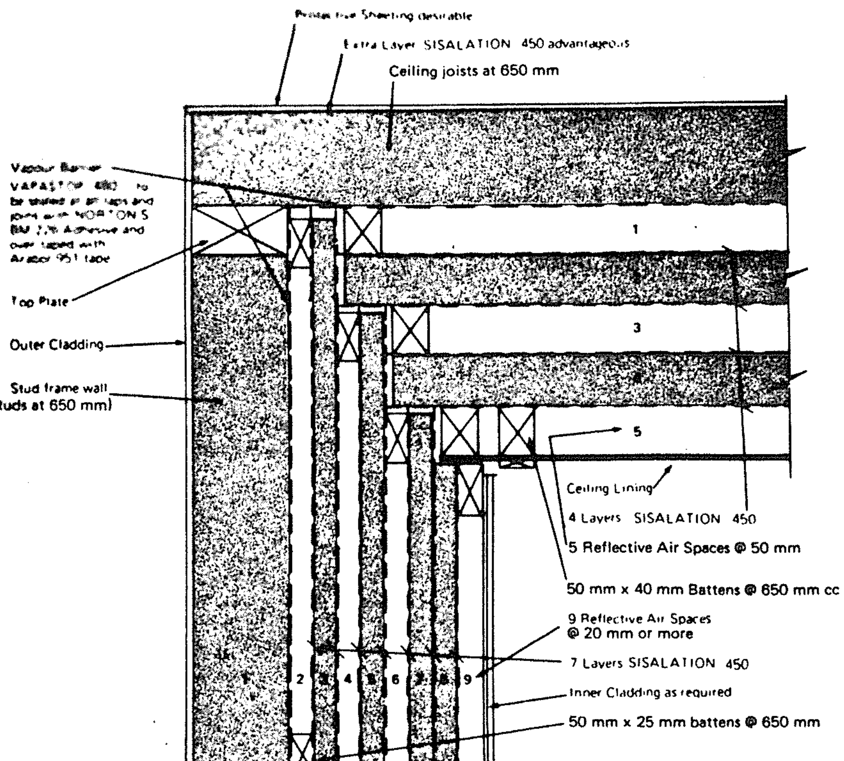
SELECTED CONVERSION FACTORS

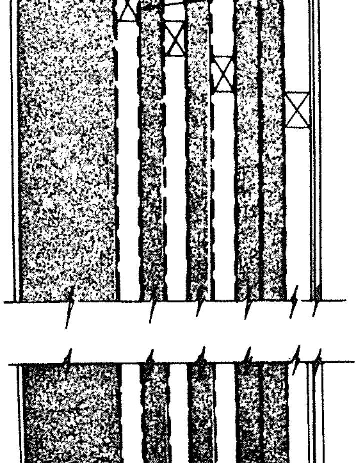
Imperial to Metric

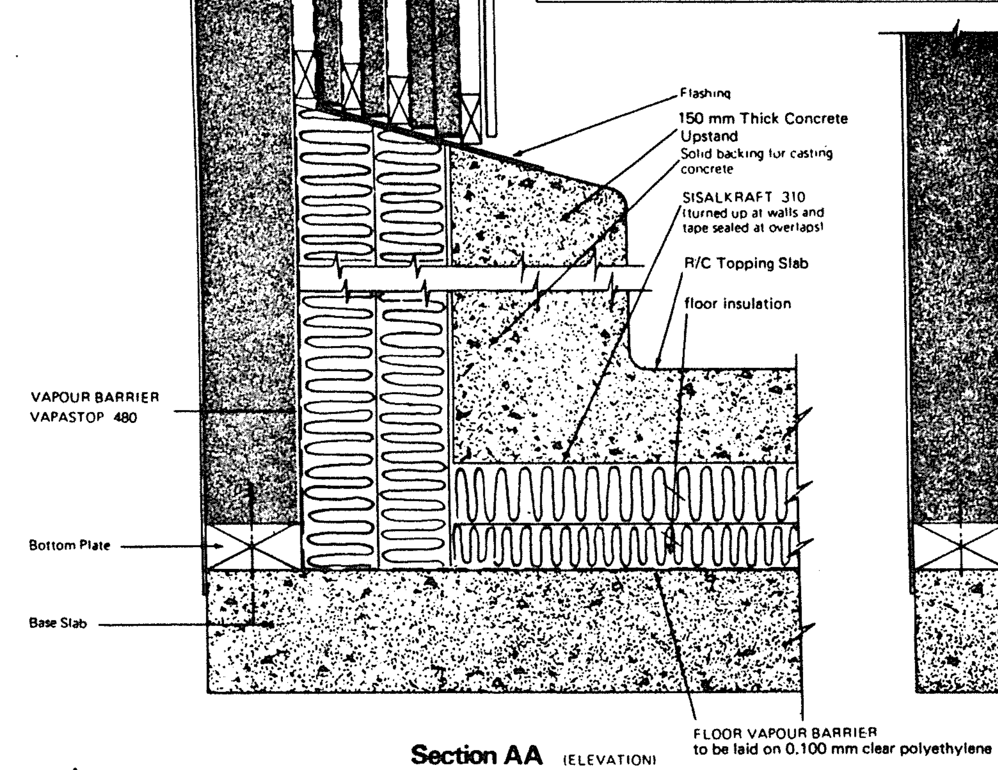
The following are some conversion factors which may be found useful in calculations involving buildings.

				ERROR
Length	1 inch	=	25·4 mm	nil
Speed	1 mile/hr	=	1·61 km/h	0·04%
	1 mile/hr	=	0·447 m/s	<0·01%
	1 ft/min	=	0·0051 m/s	<0·40%
Temperature	1°F	=	0·55K	1·00%
Weight	1 lb	=	0·454 kg	<0·10%
Heat Units	1 Btu	=	1055J	<0·01%
Heat flow rate	1 Btu/hr	=	0·293 W	<0·03%
Heat flow rate intensity	1 Btu/ft^2hr	=	3·15 W/m^2	<0·15%
Thermal conductivity	1 Btu ft/ft^2hr°F	=	1·73 W/(m.K)	<0·15%
	1 Btu in/ft^2hr°F	=	0·144 W/(m.K)	<0·16%
Thermal conductance, Radiation conductance, Thermal transmittance	1 Btu/ft^2hr°F	=	5·68 W/(m^2.K)	<0·03%
Heat Resistance Units (Reciprocal of conductance or transmittance)	1 HRU (Imperial)	=	0·176 HRU	<0·06%

NOTE:—The errors introduced by these factors are shown. If more precise values are required please consult the references. [23, 24]

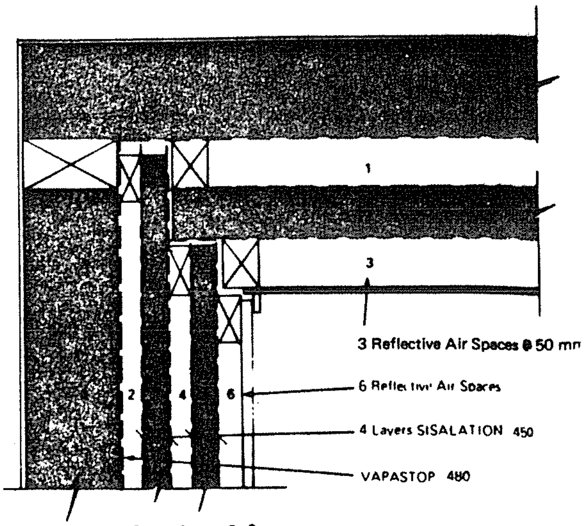






Section AA (ELEVATION)
-20°C

TYPICAL SECTION OF WALL / CEILING AND WALL / FLOOR JUNCTION



Section AA (ELEVATION)
0°C

wall : ceiling junction

SPECIFICATION:—ALUMINIUM FOIL REFLECTIVE INSU

Generally. The insulation of walls and ceiling of the cool store shall be multiple layers of SISALATION reflective foil insulation separated by 50 mm air spaces in the ceiling and 25 mm air spaces in the walls. A vapour barrier of VAPASTOP 480 shall envelope the entire store on the warm side of the insulations. The number of layers of SISALATION shall be selected so as to provide optimum THERMAL RESISTANCE (R) values. (Refer to "Insulation Quantity Chart" for R values and heat flow assessments.)

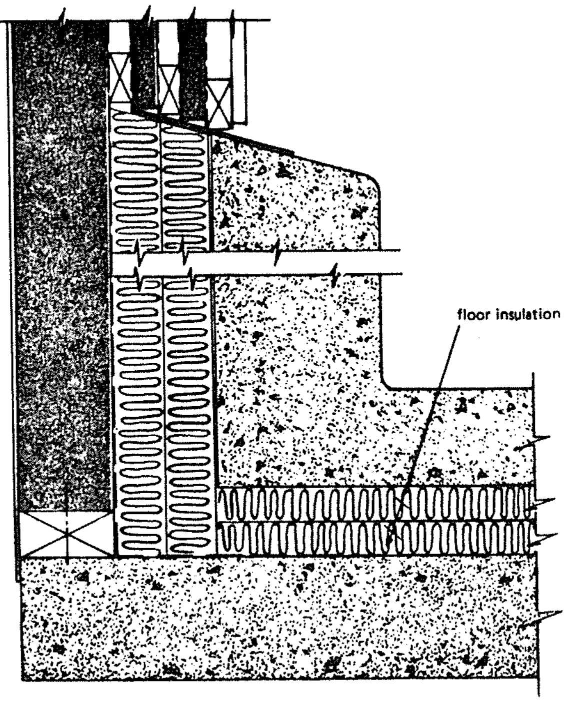
Timber "Grounds" shall be provided by spacing the wall studs and ceiling joists at 650 mm centres to provide fixing for the vapour barrier. Brick and concrete walls shall be overlaid with 75 mm x 45 mm starter battens spaced at 650 mm centres and shot fastened at intervals of 600 mm.

Ceiling and Wall Vapour Barriers shall be fixed allowing a 50 mm overlap at side laps and end joins of VAPASTOP 480. All side laps and end joins of vapour barrier shall have a solid timber backing. Alkali protection of the vapour barrier in the floor is provided by laying SISALKRAFT 310 under the VAPASTOP 480. Sheets shall be held in position on timbers with a minimum number of gun type staples having a width of at least 10 mm and a depth of at least 6 mm. Overlaps shall be sealed by applying NORTON'S BM.226 adhesive to both surfaces and adhering per adhesive manufacturer's instructions. The overlapped joins shall be wiped dry of any condensed moisture (due to chilling of foil caused by evaporation of adhesive solvents) and then taped with 50 mm wide ARABOL 951 Aluminium Pressure Sensitive Tape. All staple holes and other penetrations shall be sealed with small squares of ARABOL tape. The ceiling vapour barrier shall be turned down at the w
distance of 150 mm with a right angle bend so that the we
barrier can be overlapped and sealed against the top plate.
vapour barrier shall connect with the floor vapour barrier in
manner.

Insulation Layers shall be separated by 50 mm x 40 mm batte
ceiling and 50 mm x 25 mm battens in the wall.

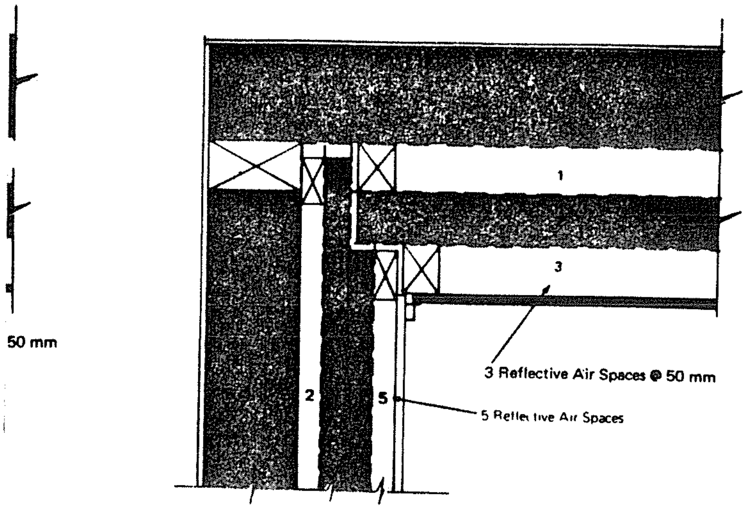
Battens shall be spaced at 650 mm centres and shall be dou
at all intersections. The first layer of battens shall be naile
"dobs" of mastic placed on the vapour barrier ensuring that a
tions are carefully located between the faces of intersecti
battens. Great care shall be taken to ensure that the length of
to fix subsequent layers of battens over the vapour barrier is
it will not penetrate the vapour barrier and that the ends of th
do not puncture the vapour barrier at corners. Foil insulatio
applied over each layer of battens until sufficient layers are
Laps of insulation layers shall be approximately 100 mm and sh
sealed. The connection between insulation layers in the ce
walls and the junction between wall and floor shall be in a
with the manufacturer's drawing detail. The ceiling finish
formed by the innermost insulation layer of SISALATION d
and held in place with "Dressed" and painted timber cover
centres, or alternative material as required.

Internal Linings to walls shall suit the particular installation, c
taken to prevent excessive reverse flow of water vapour unde
conditions. Wall linings shall be securely fixed to timber fran

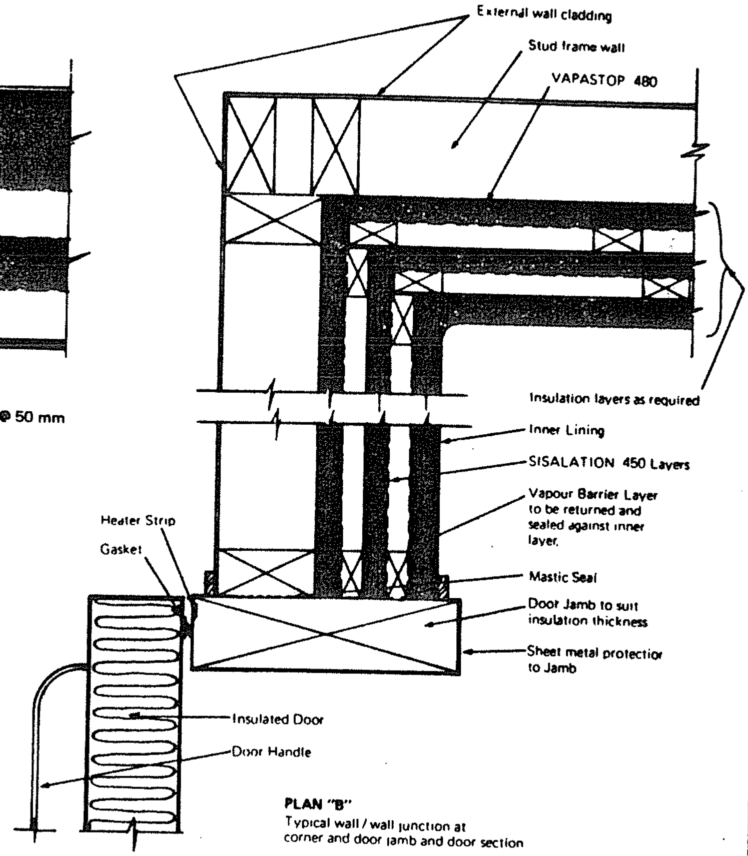


Section AA (ELEVATION)
0°C

WALL / FLOOR JUNCTION



Section AA (ELEVATION)
5°C
WALL / CEILING JUNCTION



PLAN "B"
Typical wall / wall junction at corner and door jamb and door section

ISULATION FOR COLDROOMS

t the walls for a
'he wall vapour
plate. The wall
rier in a similar

a battens in the

e double nailed
nailed through
that all penetra-
rsecting timber
gth of nail used
rier is such that
's of the battens
ulation shall be
rs are installed.
and shall not be
the ceiling and
in accordance
finish may be
ON drawn tight
cover strips at

tion, care being
r under-adverse
r framework as

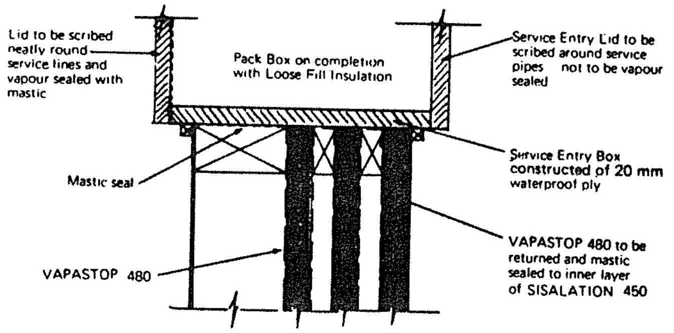
detailed and shall be complete with all necessary flashing, corner, and trimming moulds.

Materials. Reflective Foil insulation sheeting shall be double sided, reflective, fire resistant, reinforced SISALATION 450, 1350 mm wide — OR: Reflective Foil insulation shall be double sided, reflective, reinforced SISALATION 420, 1350 mm wide.

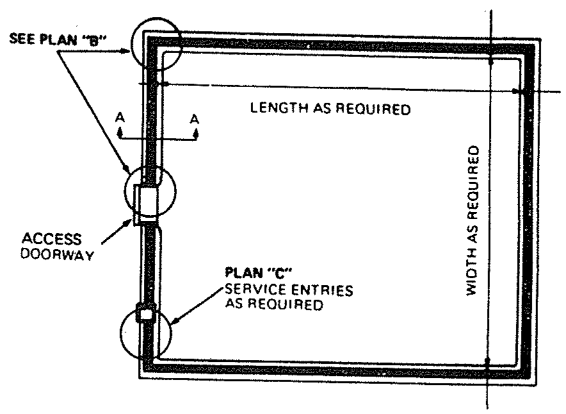
Separator battens shall be approved soft-wood dried to a moisture content of not more than 15%. Fixing of battens shall be by 45 mm galvanised nails to battens in walls and by 90 mm galvanised nails to battens on edge in ceiling. (Under no circumstances shall the nail length be greater than twice the batten thickness less 6 mm.) Bulk insulation for floors, etc., shall be selected for its physical and thermal properties to the satisfaction of the Proprietor. Staples, Foil Tape, Adhesives, etc., shall be as detailed. All materials must be new and the best quality of their respective kinds.

Inspection. During construction, the vapour carrier shall be given 100% visual inspection both before and after fixing the next layer of battens over it. Any faults or damage shall be repaired to the entire satisfaction of the Proprietor's representative. Other layers of reflective foil insulation, the wall lining, floor insulation, topping slab, concrete kerbing, etc., shall be inspected periodically and shall be, to the satisfaction of the Proprietor's representative.

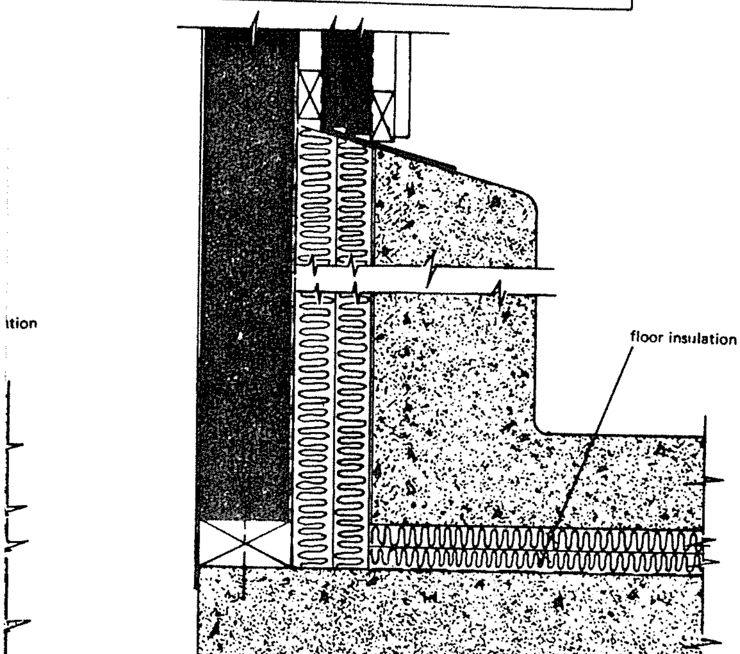
Initial Temperature Reduction shall be gradual and continuous over a period of several days, so as to facilitate escape of gaseous water vapour from the wall, ceiling and floor constructions without causing temporary condensation.



Plan "C"
TYPICAL SECTION OF WALL AT "SERVICE ENTRY BOX"



Plan of Store
DIMENSIONS AS REQUIRED



Section AA (ELEVATION)
5°C
WALL / FLOOR JUNCTION

COLDSTORES FOR OPERATION AT −20°C, 0°C 5°C		
DRAWN BY: PH.	ACI	SCALE:-
CHECKED: P.Hassall	Fibreglass Insulation	DWG. NO. 54
DATE: 1972		