Chapter 13 Part 1
HEATING AND VENTILATING DESIGN

13.1 HEAT LOSSES

The discovery that heat was a form of energy was not made until 1798 when Rumford carried out his celebrated experiment on the boring of a cannon. In the concluding part of his Essay, Rumford writes:[37]

"... these computations show how large a quantity of Heat might be produced, by proper mechanical contrivance, merely by the strength of a horse... . But no circumstances can be imagined in which this method of procuring Heat would not be disadvantageous; for more Heat might be obtained by using the fodder necessary for the support of a horse as fuel."

Joule's first direct determination of the mechanical equivalent of heat was made in 1843, and he succeeded thereby in establishing this fact. He obtained a value of 770 ft-lb/Btu at 60°F. Gradual refinements in experimental technique have given us more accurate values, but these differ only slightly from Joule's original value. The accepted figure is now 778 ft-lb/Btu. This figure is of course the basis for computing the heat gains to buildings from machinery and electrical apparatus.

Before Tredgold's time, the quantity of heat (or rather, the area of heating surface) necessary to warm a room was proportioned to the volume of the room, for example, 1 ft^2 of steam pipe per 200 ft^3, or 1 ft^3 of boiler per 2000 ft^3. Tredgold (1824) realised that there could be no universal ratio, and that heat requirements must depend on the structure, window area and ventilation. These requirements, he said[64] could always be measured in terms of a volume of air to be heated from the outdoor to the indoor temperature. His calculations were probably the first with any pretence to a rational scientific basis.

He starts from observations of the rate of cooling of a water filled vertical cylinder in air.* A cylinder contains c in^3 of water at a temperature of 180°F; it is allowed to cool in air in a room t°F (Fig. 13.1). The rate of cooling is observed to be n°F/min at the mean water temperature of T°F. The cooling effect of

---

*Although steam heating was largely used at that time, Tredgold thought the difficulty of observing the cooling of a steam pipe to be too great, and he turned to water instead.

the cylinder material is defined to be

$$\varepsilon = \frac{c\ n}{1728}\quad ft^3\ {}^{\circ}F/min$$

ε is thus the number of ${}^{\circ}F$ through which 1 ft^3 of water (or 2850 ft^3 of air)[*] would be cooled in 1 minute.
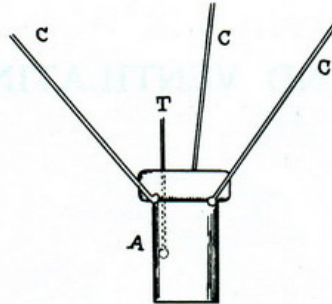

Fig. 13.1.   Tredgold's heat loss experiment (1824).

Tredgold postulated that the cooling effect was proportional to the area of the cylinder s and to the mean temperature difference, i.e.

$$\varepsilon = A.s(T-t)$$

where A is a constant depending on the material.   He found

for glass	$A = 0.000644$
cast iron	0.000738
tin plate	0.00041

(Péclet stated that Tredgold overestimated the heat loss, due to inadequate experimental precautions, though the ratios between various surface finishes were nearly correct.   Clément's were better, but still larger than Péclet's own determinations from the condensation of steam in pipes.)

Tredgold assumed that this rate of heat loss would apply to the case of a window, and to the heat loss from pipes.   He deduced that 1 ft^2 of window glass will cool $1\frac{1}{2}$ ft^3/min of air.   By this device, Tredgold was able to express heat loss through glass in terms of the equivalent in fresh air supply.

The total heat loss from a room was then set down as the sum of the necessary ventilation, the infiltration through windows and doors, and the window losses in air equivalent.   For the first, the fresh air supply per person was put at 4 ft^3/min; the infiltration was calculated by Tredgold to be 11 ft^3/min per window or door; and the window loss was equated to $1\frac{1}{2}$ ft^3/min per ft^2 of glass, say, a total of V ft^3/min.

He used the same formula $\varepsilon = A.s.(T - t_i)$ to work out the area of heating pipe needed, and to calculate the hourly quantity of steam needed, viz., 0.34 lb/h per ft^2 of heating pipe.

----

[*]This is Tredgold's figure.   A more accurate value is 3225 ft^3 of air.

To estimate the hourly and annual fuel consumption, Tredgold uses the experimental result that to supply sufficient steam to 182 ft^2 of pipe in air at 60°F would require the firing of 0.1 bushel* of coal per hour. If the internal temperature is to be 56°F, then in London heating is needed on 220 days a year, and the mean outdoor temperature during this period is 40°F — an average temperature difference of 16°. The quantity of steam needed during the heating season is found to be

$$Q = 95 \ V \ \text{lb.}$$

and this requires the burning of 0.153 V bushels of coal.

This consumption refers to continuous heating. For intermittent operation, Tredgold proposed to assume some appropriate preheating period to be added to the duration of occupation to give the daily total hours of use. The fuel consumption would be proportional to this total.

He did in fact attempt to compute the preheating period, though his argument was unsound. He illustrated this by reference to a church of 100000 ft^3 volume, for which the heat loss equivalent was 4200 ft^3/min. With this rate of heat input, the time taken to heat the air from cold to the desired indoor temperature would then be 100000/4200 = 24 min. He also applied his theory to the ventilation of glasshouses to prevent overheating in summer.

Tredgold was dismayed that his rational approach was not adopted by his contemporaries. In a letter to Francis Bramah on December 31st 1827, he wrote:

> "I know there are few capable of discarding the idea of proportioning the heat
> to the space, instead of the cooling surface of glass — but if John Bull does
> not so readily comprehend this, his descendant Marcus Bull has got the right
> end of the story, and plumes himself not a little on the subject. Marcus built
> a place on purpose to experiment in.** He consulted all the wise heads of the
> Yankee colleges, and in fact did the thing in grand style. Faith, these
> spirited fellows will be giving us the go-by some of these days if we do not
> return to a more sober and frugal system."

In 1855, Hood wrote:[30]

> "No experiments on cooling are extant, that appear to be suitable to the
> present purpose (i.e. to the determination of heating surface), except some
> that were made by Tredgold, and these are erroneous in the applications he has
> made of them."

It is not, however, very clear why Hood discredited Tredgold's application, for his own approach was very similar. The only correction Hood made was to calculate the water equivalent of the vessel at its mean temperature, instead of the maximum temperature as Tredgold did.

Hood describes his own experiments on the rate of cooling of water in a 3-in o.d. horizontal cast iron pipe. He also studied the effect of surface finish on the rate of cooling. He deduced that the quantity of heat emitted by one foot of 4 in pipe at $\Delta T = 140^\circ$F was sufficient to heat 222 ft^3 of air at the rate of 1°F/min (the temperature difference being 1°F). This figure is equivalent to an

---

*An English measure of capacity. 1 bushel = 8 Imp. gallons, or 36.4 litres.

**According to Dufton, the experimental chamber built by Bull in America was for the purpose of comparing the performance of different stoves and fireplaces.

emission of 222 x 0.02 x 60 = 266 Btu/h. ft run; and may be compared with the presently accepted figure of *ca.* 400 Btu/h for a 4-in o.d. pipe.

In modern terms, Hood's results gave the following values for the surface coefficients:

rusty iron	1.64 Btu/ft^{2}h^{o}F
varnished iron	1.59
white painted iron	1.55
glass	1.59

Hood went on to show that the heat loss from a pipe was proportional to the square of the air velocity over it.

It was originally assumed that the losses through the walls, floor and roof were all negligible or zero. Although the fallacy of this assumption must have been suspected, no mention of the possibility of losses through these parts of a structure is made by either Tredgold or Hood. Tredgold's estimate of the losses through glass have already been given, and Hood too attempted to make a rough estimate of the loss of heat through glazed surfaces which he called the cooling effect of glass. He found that 1 ft^2 of glass would cool 1.279 ft^3 of air at the rate of 1°F/min, when the air temperature on the two sides of the glass differed by one degree. (This corresponds to a thermal transmittance of 1.54 Btu/ft^{2}h^{o}F.) The experiments were conducted in still air; but Hood thought the data were generally applicable since high winds are not concurrent with very low outdoor temperature.

The procedure, then, in the mid-19th century, was to work out the total area of glass and evaluate the volume of air which this glass would cool. To this was added the requirement for ventilation and/or leakage; and the area of heating surface or length of pipe calculated. It was perhaps fortunate that the transmission through glass was over-estimated, thus affording some sort of correction for the neglect of other structural losses.

The design outdoor temperature was chosen to correspond with the usage of the building. In the case of buildings for daytime use only, 25°F was recommended; but where buildings were to be heated both by night and by day, as for a forcing-house for instance, "perhaps 10°F will not be too low to calculate from". On very exposed sites, a temperature of 0°F was recommended by Hood.

What is possibly the first attempt at a guide to current practice is given by Hood, when he suggests values for the heating surface required for various types of building:

Building	Inside temperature ^{o}F	Heating surface (ft of 4-in pipe per 1000 ft^3)
Church	55	5
Large public rooms	55	5
Dwelling rooms	65	10
-do-	70	12-14
Halls, shops, waiting rooms	55-60	7-8
Factories and workrooms	50-55	5-6
Greenhouses	55	35
Graperies	65-70	45
Pineries and hothouses	80	55

Tomlinson[63] writing after Hood, quotes Arnott's estimates of the heat requirements of buildings. He assumed an inside temperature of 60°F, an outdoor temperature of 22°F, and steam heating surface at 200°F. One ft² of heating surface was allowed for each 6 ft² of window or for 120 ft² of ceiling, wall or roof constructed of ordinary materials, and for each 6 ft³/min of ventilating air. Arnott[3] also noted that double windows had only one-quarter of the heat loss of a single window.

Tomlinson also used Hood's data on the heating effect of 4-in pipe. In habitable buildings, he assumed that the ventilation requirements amounted to $3\frac{1}{2}$ ft³/min per person. To this was added $1\frac{1}{4}$ ft³/min as being the equivalent of each ft² of glass. The total length of 4-in pipe required was then

$$\frac{125\ T}{t_p - t_i} \times \frac{n}{222} \quad \text{ft}^*$$

Here t_p is the temperature of the pipe, t_i the indoor temperature, T the desired temperature difference between indoors and outdoors, and n the total number of ft³ of air to be warmed (not the volume of the space). For churches, where the heat output of the occupants is relatively large, Tomlinson gave an empirical rule:

$$\text{ft of 4-in pipe} = \text{Volume}/200$$

Constantine (1881) was quite satisfied with such empirical procedures:[17]

"Of late years, the makers of boilers, pipes, bends and other fittings for the circulating system have exercised considerable ingenuity in providing everything that is necessary for reducing or enlarging the flow pipes, turning corners, dipping or ascending, so that the execution of this class work has been greatly simplified, and now almost any respectable plumber can do hot-water fitting, and they all know that one foot of 4-in pipe warms 100 ft of air; there is, therefore, no necessity for giving elaborate tables as was the custom in books written 20 years ago. Another simple rule should never be lost sight of, that is, always have a good margin of heating power both in boiler and pipes."

Perkins' high pressure systems were also designed on an empirical basis. In 1904, Walter Jones was:[33]

"not aware of any work yet published that gives the length of pipes and the boiler power required to obtain various temperatures by high-pressure heating".

He suggests that, based on experience, for offices to be warmed to 60-65°F (outdoor temperature 30°F), 18-22 feet of $\frac{7}{8}$-in bore pipe are required per 1000 ft³ of room volume. The furnace coil is to be equal to one-tenth of the room pipe length.

Nevertheless, the theory of high temperature hot water heating had been given by Einbeck in 1877, and Rietschel[52] gives detailed calculations for the furnace coil and for the room heating surface. On the assumption that the furnace temperature is 1000°C and the combustion gases leave at 200°C, the length of the furnace coil is to be

$$l = 0.002 \quad C \quad \text{metre},$$

where C is the design heat loss, kcal/h. The length of the room heating surface is calculated from

---

*The figure 222 is Hood's value for the volume of air which can be warmed by 1 ft run of pipe.

$$l_2 = \frac{10\ C}{11.5\ (\frac{1}{2}\ (t' + t'') - t_{air})}$$

t' and t'' being the flow and return temperatures (150° and 80°C), the factor 11.5 being the pipe emission (the value given here is appropriate only to the temperature quoted).

13.2 THE DEVELOPMENT OF THERMAL TRANSMITTANCE

By the time Péclet wrote his *Traité de Chaleur*, the physical principles were better understood, mainly owing to the work of French physicists, including Péclet himself. Péclet[46] was well aware that the solid parts of the wall as well as the glass transmit heat from the inside to the outside of a building, although according to him "most designers size their equipment on the basis of the volume to be heated — an obvious error" (this in spite of Tredgold's treatise). He realised that the inside surface receives heat by convection from the air and by radiation from the surrounding surfaces; and that the loss from the outside surface occurs by the same processes. The laws of heat conduction between the two surfaces were known, and Péclet applied them to this problem.

The radiation loss from a surface was initially thought to follow Dulong and Petit's law, which they deduced in 1817. Their somewhat complex formula is now known to hold only approximately over the range of temperature covered by Dulong and Petit's experiments. The dependence of the radiation emitted upon the nature of the surface was studied by Leslie in his pioneer work on the absorption and emission of radiation. The relative radiating powers given by him are little different from present-day data. Leslie's results show clearly that the radiation is independent of colour as such. He was able to show that for radiation for these same wavelengths, the absorptive power was equal to the emissive power.

Sir Humphry Davy (*ca.* 1834) showed that the absorptive powers of different colours was different when the surfaces were exposed to sunlight. He found the order of decreasing absorption to be: black, blue, green, red, yellow, white. Hood was aware of Leslie's work, and dimly appreciated the dependence of the absorption on the kind of radiation received by a surface. He did not, however, apply it to considerations of the heat loss from buildings. Powell thought that sunshine contained no "simple heat" (i.e. low temperature radiation). "It was also established by the experiments of Melloni and Nobili that the radiating powers of surfaces, for simple heat, are in the inverse order of their conducting powers." This almost led Hood to conclude that poorly conducting materials should be used for heating surfaces; but he escaped this error by good fortune and a curious argument.

The convection loss was also investigated by Dulong and Petit, who found that this loss was independent of the nature of the surface, but did depend on its size and geometry.

Péclet carried out an extremely careful determination of radiation and convection losses, by observing the cooling of cylinders. He verified Dulong and Petit's laws of cooling (whose form he accepted without question). The cylinders were provided with a variety of surfaces (metal, textile, paper, paint). He also gave approximate expressions for the convection coefficient for surfaces of different shapes and dimensions.

Using his data, Péclet gave the total surface conductance of a masonry wall as 5.6 kcal/m²hC, a value which he assumed to hold for both inside and outside surfaces, and for the interior of a cavity. He was aware that the surface

coefficient should depend on the surface temperatures of the walls to which the surface could radiate: if these were the same as that of the surface being considered (as in a room with all surfaces equally exposed) then the radiation transfer is zero. The surface conductance is that due to convection alone. He thus demonstrated that for a room constructed entirely of glass the transmittance would be 1.45–1.65 kcal/m²hC, whereas for a room with only one glass wall exposed to outside, the transmittance would be 2.48–2.65 kcal/m²hC.

Although Box[12] repeated this calculation and Carpenter was also aware of it, the conclusion was ignored until Dufton (*ca*. 1935) revived it by proposing the use of equivalent temperature to estimate heat loss. Even then, no practical application was made until the 1970 issue of the IHVE Guide.

Picard[47] used Péclet's data, modified in respect of the convection coefficient. He chose values $h_c = 4$ for inside surfaces, and $h_c = 5$ for outside surfaces. The surface conductance was also studied by Rietschel and by Grashof. Rietschel quotes the formulae:

$$h_o = h_c + h_r + (0.0075 \, h_c + 0.0056 \, h_r) \, (t_{so} - t_o)^*$$

$$h_i = h_c + h_r + 0.0075 \, (t_i - t_{si})$$

but he noted also that the terms with the multiplier involving temperature difference could usually be ignored. He used Péclet's value of $h_r = 2.91$ kcal/m²hC. The values of h_c determined by Grashof were:

air at very low speeds (inside surface)	3.94 kcal/m²hC
air at low speeds	4.90
air moving at moderate speed (outside surface)	5.91

These values seem probably to be the basis for Picard's and Rietschel's choice of convection coefficients.

The experiments of Griffiths and Davis in 1922 have provided the most accurate data for the loss of heat by convection and radiation. The existence of a power law for convection transfer was shown theoretically by Nusselt, who proved that the index should be 1.25 instead of the 1.233 given by Dulong and Petit. Griffiths and Davies[27] were able to show that theory and experiment agreed in this respect.

It is now known that the outer surface coefficient for heat transfer is by no means a constant, and can take zero or even negative values under certain conditions. The conditions under which low values may occur are those in which the surface is able to radiate to the night sky, and its temperature may fall below the outdoor air temperature. This difficulty is circumvented, at least to some extent, if sol-air temperature (which includes radiation effects) is employed instead of air temperature, analogously to the use of equivalent temperature instead of the indoor air temperature.

The thermal conductivity of materials was but imperfectly known. Péclet was once again the forerunner in the determination of this property.

Apart from the value for cork, his figures are of the same order as more recent determinations. Péclet's conductivity data were used by Carpenter (1910) and by Barker, and were quoted even as late as 1928 in Poynting and Thomson's *Textbook of*

---

*t_o, t_i are outdoor and indoor air temperatures;

t_{so}, t_{si} are corresponding surface temperatures.

Physics,[48] though with the comment that "they probably need revision".

Péclet calculated the total heat loss through a wall by equating the surface transfers to the conduction through the solid. Thus he did not make use of the idea of resistance, although it is implicit in his final formula, which was identical with modern practice. Box[12] gave the symbol U to the quantity we now call thermal transmittance: this appears to be the first (and for many years, the only) use of this letter in this connection.

This method of computing heat losses seems to have been adopted in France and Germany (cf. Picard 1897 and Rietschel 1893-1911), though not in Britain or America, in spite of Box's famous *Treatise* which drew heavily on Péclet's work. Picard, for instance, gives a short table of transmittance, which he says was used in the European countries:[47]

25-cm brick wall, rendered	1.58 kcal/m^{2}hC
Single glass	3.66
Double glass	1.76
Roof	0.65
Wood floor	0.60
Metal roof	2.12

Rietschel's table was much more comprehensive.

We see that, by this time, the need to consider losses through the floor had been appreciated, though of course, in Picard's wood floor, the underside was not in contact with the ground, but the outdoor air.

Péclet had assumed that the floor and roof losses were negligible. He knew that the earth temperature at a depth of 8 m was constant throughout the year, at the mean annual temperature of the locality. Box goes on to state that beneath a building, the ground, being protected from the diurnal variations of atmospheric temperature, will take up the (constant) earth temperature. In Britain (and France), this is "pretty nearly the average temperature of our dwellings in the cold season" and the heat loss to the ground will be nothing.[12]

Rietschel made special additions to the computed or tabulated values of U to allow for variations in orientation or exposure to sun and wind. Debesson,[18] too, used thermal transmittance, and an addition of 2 or 3°C to the usual temperature difference for north-facing walls, and even up to 20-50° in very exposed cases.

The small attention paid to Box's book or to Péclet and Rietschel may be seen from Dye's *Plumbing and Sanitation* of 1897. He discussed a number of empirical rules for estimating heat losses and heating surface. Carpenter[14] suggested, for low-pressure hot water heating:

Heating surface = 0.4 (glass + ¼ wall + 2/55 volume).

Lawler (USA) went so far as to say that it was "next to impossible to figure out the precise surface required". Dye himself proposed one ft^2 of surface for every 3 ft^2 of glass and 6 ft^2 per 1000 ft^3 of volume. He added that the wall area and the cubic capacity never both enter into the calculations; and of the two, the cubic capacity is the more reliable. It was also said that when low pressure steam was used, the heating surface was to be between 5/8 and ¾ of that for hot water (because of the higher temperature). For gravity warm-air systems, no data were available to compute the effectiveness of the furnace surface. It was customary to allow 10 ft^2 of heating surface to each 1000 ft^3 of room volume, or for each 4000 ft^3/h of ventilating air.

In 1904, Jones made a rather detailed comparison of the various proposed (empirical) methods of evaluating heat loss and radiator surface.[33] He found very large discrepancies between them. He admits that to ignore glass area, and to base heat requirements on room volume alone must be wrong; but nevertheless he gives a table based on volume only.

Jones goes on to give a later rule based on glass and wall areas, and on the volume of the room, similar in form to Carpenter's proposal:

$$\text{Radiating surface} = \left(\frac{\text{glass area}}{6} + \frac{\text{wall area}}{12} + \frac{\text{volume}}{120} \right) \text{ ft}^2$$

This applied to a hot water system with a mean water temperature of 170°F, designed to give an indoor temperature of 60°F when it is 30° outside. Two air changes per hour is allowed for ventilation. This formula appears to give an excessively large radiator surface, but Jones says that the formulae of Baldwin, Carpenter, Dye, etc. give much less, and claims that tests (which are not described) verified his rule.

The confusion which reigned at that time is well exemplified by Thomas, who, in 1906, wrote:[61]

"There are quite enough books which give all the formulae for calculating the heating surface required as well as other figures and equations. The unfortunate part of these formulae and calculations is that they are absolutely beyond the brain power of the average hot-water engineer, while the difference between theory and practice makes it unwise to rely on such calculations."

By 1910, however, the present method was coming into use in the English speaking countries which were thus falling into line with continental Europe. Carpenter quotes some test data obtained by Wolff for the German Government, which are probably the first experimental determinations of thermal transmittance:

	U
Single window	1.09 Btu/ft²h°F
Single skylight	1.118
Double window	0.518
4 in brick	0.68
8 in brick	0.46
12 in brick	0.32

Eventually Carpenter, after reviewing all these data, concluded that for all practical purposes, the loss through a window should be taken as 1 Btu/ft²hF and that through a wall, one quarter of this. Ceilings with attics over were to be treated as walls of one-third the area. Floors were ignored. This was, in fact a reversion to earlier practice in America.

Carpenter undertook two full-scale trials, and the results are of some interest:

	Trial A	Trial B
Area of glass, ft²	96	9281
Area of wall, ft²	246	31644
Temperature difference, °F	27–28	31
Measured loss, Btu/h	4247–4240	547200
Calculated loss, Btu/h	4253–4410	532952

These results were held by Carpenter to "indicate the substantial accuracy of the rule just quoted".[14] This must have been fortuitous, for Carpenter does not appear to have measured the ventilation loss. Neither did he compute the heat loss by the more correct method of which he was aware. Nevertheless, he apparently believed that "in a few years, the rule of thumb at present in use will drop out of use entirely and be replaced by rational scientific method."

By 1912, the design of US Federal buildings was based on U-values, with allowances for exposure and orientation.[62] The U-values were to be chosen according to the number of exposed walls (cf. Box). However, a rule-of-thumb method, like Carpenter's, could be used where windows were weatherstripped.

Barker, in 1912, also used the transmittance, and showed how it might be computed, drawing heavily on Rietschel's text. He writes of a material offering a "resistance to the flow of heat".[7] This seems to be the first use of the concept of thermal resistance, although it is not defined as such. Terminology was still confused; and Barker used the symbol K to denote almost every kind of heat transfer coefficient.

Barker also noted the influence of weather upon heat loss, and felt that the heating engineer needed exposure factors and weather factors for use with heat loss coefficients. He was less sanguine than Carpenter about full-scale trials, for he wrote:

"Because of variations of outdoor temperature, it is and always must be difficult, if not impossible, to obtain a thoroughly satisfactory experimental verification on a large practical scale of the theory on which the calculations (of heat loss) are founded".

The more accurate, if more laborious, method of computing heat losses proposed by Péclet, and urged by Barker and Carpenter, was still not universally adopted. Dye, writing in 1917, gives it cautious approval:

"Until recent years, it was practice to... allow a certain quantity of radiating surface per 1000 cubic feet of space... . The simple rule can seldom work out correctly. The coefficient method is now almost universally employed, and while this aims at meeting all varying conditions, there is a feeling that some improvement on this will presently be possible. For those who have not a suitable office staff, the method is distinctly tiresome, if not impossible in some cases... . While there is a feeling that the coefficients for heat losses from walls, roofs, glass, etc., are now fairly accurate, there always remains an uncertainty as to the changes of air."[23]

Raynes (1921) suffered from some confusion between the two quantities now termed transmittance and conductance.[51] He used Box's symbol U to denote the total hourly heat loss from a building. This was to be further modified by factors to take account of the height, aspect, exposure and intermittent heating.

Between 1910 and 1925, Barker in the UK and Harding and Willard in America made attempts to determine the thermal transmittance of a wall experimentally, using a hot-box method. Barker's values were:

	U
9-in London stock brick, unplastered	0.43 Btu/ft²h°F.
4½-in London stock brick, unplastered	0.57
12-in Cavity brick wall, unventilated, unplastered	0.33
12-in Cavity brick wall, ventilated, unplastered	0.42
4½-in Ballast concrete	0.61
6-in Ballast concrete	0.57

His results were adopted as standard by the IHVE in Britain.

In 1917, the German central heating industry published the first edition of its *Rules for Calculating Heat Losses*, and this was probably the first attempt to ensure uniformity of practice. The first DIN Rules, which superseded it, were published in 1929. The first ASHVE Guide appeared in 1922, while that of the IHVE was published in 1935. The first (1929) edition of the DIN Rules resulted in oversized plant; the 1944 version divided Germany into several "climate zones"; the 1959 edition used small "Zuschlage" to cater for a variety of circumstances, such as corner rooms.

In carrying out the calculation of hourly heat loss rate, it was usual to use the air temperature difference between indoors and out. It was assumed that the air temperature was that corresponding to comfort, and since it was known that lower air temperature could be used with radiant heating, an arbitrary reduction in the calculated heat loss was applied for this mode of warming. Around 1931, Dufton had suggested that the indoor equivalent temperature would be a sounder basis, and would eliminate the need for empirical corrections, either for cold walls or for different modes of heating. Although this was never embodied in the Guides, some designers used it in practice. The 1965 IHVE Guide, for example, referred to room temperature, and not air temperature, as a quasi-official recognition of the idea.

Between 1965 and 1970, workers at Building Research Station (England) turned again to Box, under the stimulus of preparing a code for the calculation of thermal transmittance, and introduced "environmental" temperature. With this index, U is a true constant property, and no corrections for number of exposed walls or mode of heating are necessary. (Exposure still remains, since it affects the outside surface resistance.) Dufton's proposal was seen to be valid on theoretical grounds, and has been incorporated in the latest IHVE (now CIBS) Guide.

13.3   VENTILATION AND INFILTRATION LOSSES

Ventilation losses remain, as always, the great uncertainty. Two different approaches have been in use. The first, and oldest, is wholly empirical, stating a ventilation rate to be used in design — one that, hopefully, will both satisfy human need and yet not under- or over-estimate the winter infiltration. Many determinations of actual ventilation by natural means have been made (Haldane, Bedford, Masterman, Dick *et al.*) but the results are so diverse as to afford little guidance to a design engineer for his calculations.

The second and more recent is an apparently more rational method based on the length of cracks round doors and windows and the rate of air flow through them.

Tredgold was the originator of this method, though it does not appear to have been adopted at all widely for many years. He himself assumed that all doors and windows were roughly similar, and calculated an average infiltration of 11 ft^3/min (5 l/s) for each unit, on the basis of the air flow through the cracks.[64] Hoffman and Raber[29] thought that natural infiltration was "impossible to determine", and recommended that 2 room volumes/h be assumed.

The crack method was used in the USA from about 1930. With the advent of tall buildings, neither method was really satisfactory. There was no experience to guide the choice of values for the first method, and the significant stack effect and the varying pressure over the height of a tall building were factors which neither method could cope with.

About the turn of the century, we see the study of meteorology being combined with the art of ventilation. J. W. Thomas was clearly aware of the significance of wind. He noted that 'recent experiments with kites' had shown that wind speeds at 300 m are about twice those near the ground, in open country, and that these were again about twice those in large towns. Wind pressure distribution over building facades had been studied in some detail — Thomas himself seems to have made experiments which he reported to the British Association in 1900 or 1901; and he knew of the existence of a pressure on windward sides and a suction on lee facades.[61] He appreciated that these pressure differences would influence natural infiltration or ventilation.

Perhaps the most significant advance was the publication of Napier Shaw's classic *Air Currents and the Law of Ventilation* in 1908. Following Murgues, he made use of the idea of the resistance to air flow of an opening, formulated the aerodynamic equivalent of Ohm's Law, and showed how to compute the resistance for simple plane openings. He developed a simple network theory, based on electrical analogy, and applied it to a variety of typical ventilation problems, including the aspirating chimney and mechanical ventilation.[55] The neutral zone — the plane at which internal and external pressures are equal — is implicit in Shaw's work. Its significance was realised by Rietschel, and it was applied in the 1930's by American engineers.

Shaw's work was largely neglected for more than half a century. The underlying concepts were used by Harrison (1961), by Den Ouden (1963), Jackman (1968) and by Billington (1971) to analyse specific problems in natural ventilation.

The work of Jackman at HVRA* and Den Ouden at TNO** placed ventilation estimates on a firmer footing.[65] Using the known wind pressure distribution over the face of a building, and a knowledge of the flow through cracks, they produced a computer programme for the calculation of natural infiltration. It is probably true that infiltration can now be better estimated for a large and tall block than for 1- or 2-storey structures (unless the whole network is worked out in detail). Nonetheless, the ventilation losses are still rather uncertain, and as the structural insulation is improved, and the ventilation loss becomes a greater proportion of the whole, the inherent inaccuracy of the total heat loss will increase. (None of this applies, of course, to a sealed building with mechanical ventilation, for which the air flow can be closely controlled.)

A somewhat curious method of allowing for infiltration losses was given by Dunham in 1928.[22] Structural heat losses were computed from U-values in the ordinary way, and then multiplied by a factor depending on orientation and a second factor depending on the quality (i.e., air-tightness) of the construction. Thus the total heat loss was reckoned to be:

$$H = (\Sigma A U \Delta t) \text{ (orientation factor) (leakage factor)}$$

where orientation factor = 1 for S facades
                      1.35 for N or NW facades

      leakage factor = 1.5 for buildings of good construction
                       1.8 for factory type buildings

Lang (1877) and Gosenbruch (1897) studied the passage of air through building structures. Rietschel recognised that the air flow through the materials themselves

---

*Heating and Ventilating Research Association, Bracknell, England.

**Delft, Holland.

was insignificant compared with that through cracks and joints.[52] Further studies on complete structures were carried out at the ASHVE Laboratories between 1927 and 1930, but little use seems to have been made of the data. There has been a revival of interest in this topic since *ca.* 1975, with experimental work by British Gas and by BSRIA* (where attempts are being made to correlate infiltration with pressure measurements).

13.4 BUILDING FABRIC LOSSES

13.4.1 *Air spaces*

It is not clear when the insulating value of air spaces was first recognised. Presumably double glazing was in use in Europe in the early 19th century, since in 1861 Péclet estimated the heat loss through multiple-glazed windows, and showed theoretically that the maximum resistance for double glazing was obtained with a spacing of 20 mm between the panes.[46] At this distance, the conduction through the air was greater than the surface transfers.

Rietschel and Brabbée[53] used the concept of equivalent conductivity, and gave a table based on Wierz's tests of 1921:

Width	in	$\frac{1}{2}$	$\frac{3}{4}$	1	2	3	4
Equiv. conductivity	Btu ft/ft^{2}h^{o}F	0.06	0.08	0.09	0.16	0.23	0.29
Thermal resistance	ft^{2}hF/Btu	0.69	0.78	0.93	1.04	1.09	1.15

These values agree fairly well with the most modern ASHRAE figures. It is now known, however, that the optimum width for minimum heat transfer is about 20 mm as shown by Péclet.

In addition, it has been realised that the rate of transfer depends both on the orientation of the space, and upon the direction of heat flow.

Rietschel ignored any resistance due to the air in horizontal cavities with upward heat flow:

for horizontal cavities, heat flow up, $R = 1/h_{si} + 1/h_{s2}$

for other cavities $R = (l/k_{air}) + 1/h_{si} + 1/h_{s2}$

Further, it is possible to reduce the radiation component very appreciably, by using metallic surfaces: when this is done, the heat flow across the air space is halved.

13.4.2 *Ground floors*

As has been seen, the heat loss through ground floors has been largely neglected until recent times. Péclet was aware of the constancy of the earth temperature at a depth of 8 m or so, and assumed that on this account the heat loss was negligible. Picard referred to it, and Rietschel gave the *U*-values for a number of ground and intermediate floors, though these were probably little more than intelligent guesses. Quite recently, a more scientific attack on the problem has enabled graphs and tables to be prepared which give the heat loss through solid

---

*Building Services Research and Information Association, England (formerly HVRA).

floors on ground under a variety of conditions. Interest in the problem was
triggered by difficulties arising beneath the floors of cold stores (frost heave)
or of boiler houses and kilns (drying shrinkage).

Measurements of the loss through solid floors were made by Dill *et al.* (1945)[19]
and by Bareither (1948)[6] in the USA. A theoretical analysis had been given by
Macey. In 1951, Billington made a study of the phenomena using a network analyser
(Fig. 13.2), and this served to confirm Macey's theory and agreed with the full-
scale American trials. Since flow through a ground floor is three-dimensional, the
heat loss depends very much on the size and shape of the floor. The analyser
studies enabled these effects to be determined, and quantified the concept of edge
insulation which had been used on an empirical basis by the electrical industry in
Britain for floor-heating installations. It is interesting to note that Dye's 1917
figures were not very wide of the mark for large floors, while those of the earlier
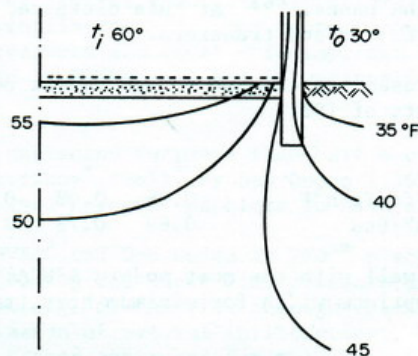IHVE Guides were approximately correct for small floors.

Fig. 13.2. Ground isotherms.

13.5 BASIC DESIGN TEMPERATURE

Early engineers chose what we would now think to be extremely low outdoor
temperatures for design purposes. Reference has already been made to Tredgold and
Hood in Britain, and to Péclet in France. They had nothing but experience to guide
them, for meteorological data were sparse. The finer aspects of climate, and the
influence of the building structure, were only vaguely understood. Little progress
was made until this century. Barker[7] was able to summarise the position in 1912:

"In this country, the conditions generally specified are an external temperature
of 30°F; in America, 0°F; and on the continent 10°F."

It was everywhere realised that these were by no means the most extreme conditions
likely to be encountered, but there was an implied understanding that extreme
conditions would be both rare and short-lived, and that to attempt to meet them
would be unduly expensive.

It was not until 1955 that the choice of 30°F in Britain was seriously questioned —
due partly to changes in building construction and partly to the introduction of
storage heating. (Brunt Report, Post-war Building Study 33, 1955). The committee
examined the intensity and frequency of cold spells of 1, 2, ... days' duration,
and deduced suitable basic outdoor design temperatures for buildings of different
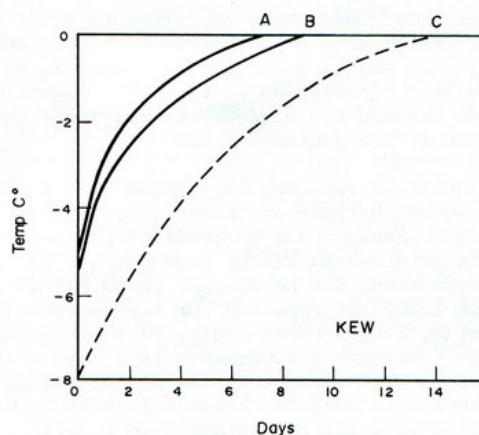thermal mass (Fig. 13.3).

Fig. 13.3.   Occurrence of cold spells.
(Courtesy, CIBS)

Shklover (1947) had already given some indication of the effect of thermal inertia
of the building, by considering the attenuation of temperature cycles through the
structure.[56]   And in 1976, Billington[9] applied the admittance procedure to the
same problem, arriving at basic design temperatures which were in substantial
agreement with the Brunt Report. He also found that the insulation of a building
(since it affects the thermal response) has an influence on the design temperatures.

Earlier American practice had been to base the choice of outdoor design temperature
on the likelihood of the occurrence of one or two short spells of cold weather
(Hoffman and Raber 1913).[29]   Current approach is based on a statistical
examination of the local climate; the design temperature is that outdoor
temperature which is equalled or exceeded during $97\frac{1}{2}$% of the hours in December,
January, February and March (or alternatively stated, the temperature will be *below*
this value for $2\frac{1}{2}$% of the time in these four months). If the low temperatures
occurred in a single spell, this would be of three days' duration. It leads to
design temperatures which are substantially above the mean annual minimum.

In Europe, it is customary to divide the country into a number of climate zones,
instead of treating each locality separately, and each zone has a different base
temperature. Britain has always been regarded as a single zone, though Billington
has suggested that a division into three zones might be more appropriate.[8]

Another factor, not normally taken into account at this time, is that of "wind
chill". It has been noted that Hood realised that high winds do not normally
coincide with low outdoor temperatures. Billington examined meteorological data
for Reading, and concluded that the maximum heat loss is likely to occur, not at
the usual design minimum temperature of $30°$F but at some higher value. This
conclusion has been confirmed by Jackman (UK) and in the USA. A similar conclusion
was reached by Anapolshaya and Gandin (USSR),[2] who considered a hypothetical
outdoor temperature T_e which gave the same heat loss as the actual temperature T
and wind speed v. They showed (as expected) that T_e is less than T (i.e., wind
increases total heat loss) but the minimum values of T_e (corresponding to maximum
heat loss) occurred not at the minimum T and maximum v, but at intermediate values.