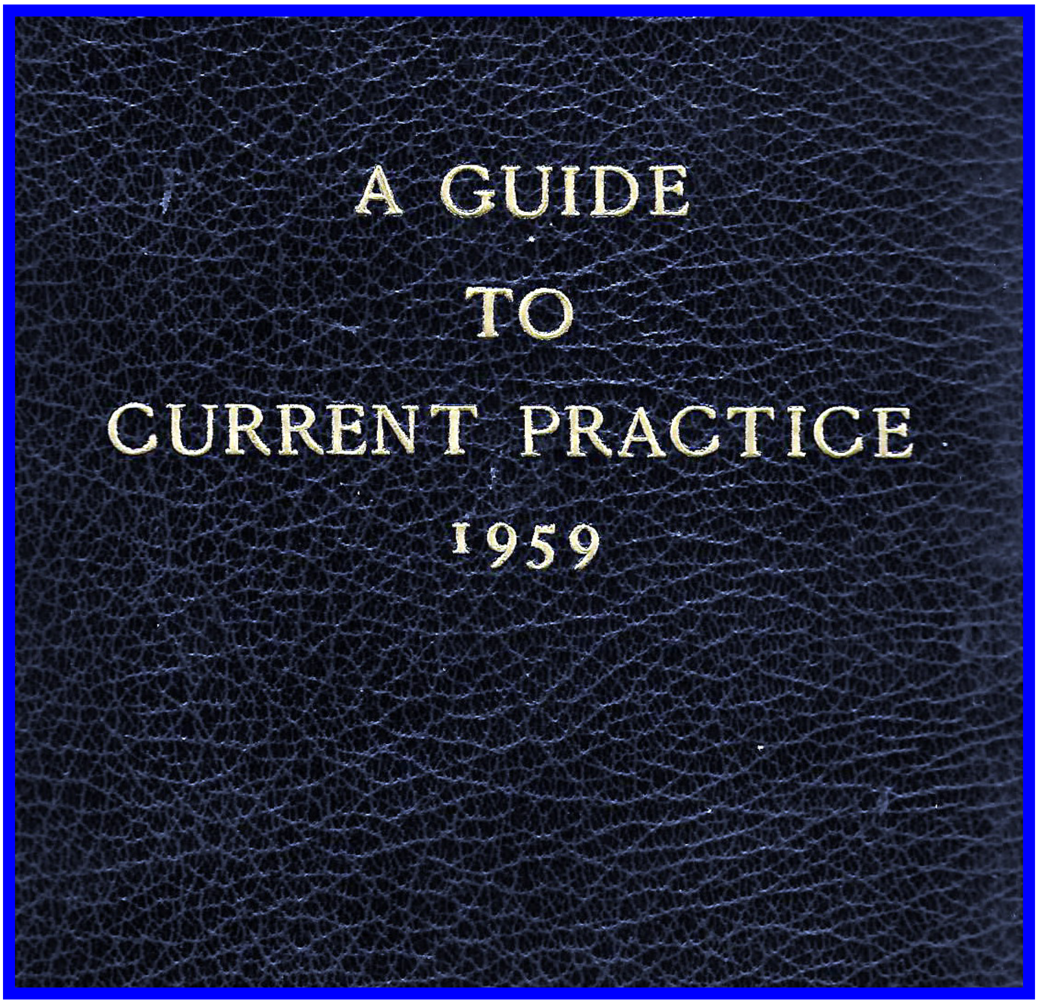
Chapter 4 Ventilation, Infiltration and Fabric Losses



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:

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

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.

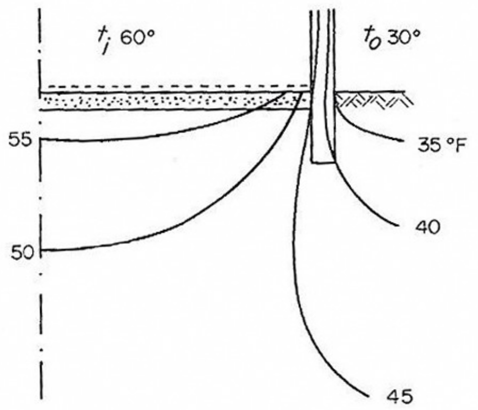


Fig. 13.2. Ground isotherms.

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).

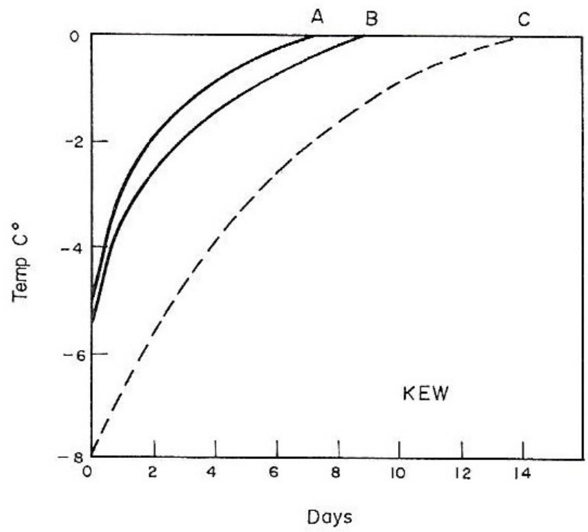


Fig. 13.3. Occurrence of cold spells.
(Courtesy, CIBS)

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 97½% of the hours in December, January, February and March (or alternatively stated, the temperature will be *below* this value for 2½% 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.

The resistance of air spaces

The thermal conductivity of air is very low, and it might be thought that a closed air space would, on this account, offer a very considerable resistance to the passage of heat across it. The actual transfer by conduction is, in fact, very small; but in addition to this, heat is radiated from one bounding surface to the other, and is also transferred by convection currents in the air itself. These latter modes of transfer are very much more important than conduction, which is generally negligible in comparison with them.

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The radiation transfer between two equal parallel planes may be computed from the formula

$$H_r = \sigma \,.\, F_\epsilon \,.\, A(T_1^4 - T_2^4) \qquad (1.13)$$

where H_r = radiation exchange (B.Th.U./hr.).
T_1, T_2 = the absolute temperatures of the surfaces (°F.).
A = the area (sq. ft.).
σ = Stefan's constant ($17{\cdot}4 \times 10^{-10}$ B.Th.U./ft.^{2} hr. °F.^{4}).
F_ϵ = an emissivity factor.

The factor F_ϵ has a maximum value $\epsilon_1\epsilon_2/(\epsilon_1+\epsilon_2-\epsilon_1\epsilon_2)$ when the planes are large in comparison with their separation; and a minimum value ($\epsilon_1\epsilon_2$) when they are widely separated.

Some typical values are:

	F_ϵ *maximum*	F_ϵ *minimum*
$\epsilon_1=\epsilon_2=0{\cdot}9$.	0·82	0·81
$\epsilon_1=0{\cdot}9$; $\epsilon_2=0{\cdot}05$.	0·05	0·045
$\epsilon_1=\epsilon_2=0{\cdot}05$.	0·025	0·0025

For air spaces as encountered in buildings, the maximum value of F_ϵ will generally be appropriate. The table shows the enormous reduction in the radiation transfer when one of the surfaces has a low emissivity. The value of the second surface also having non-radiating properties is seen to be slight. True, the radiation is halved, but it is in any case small when one surface already has a low emissivity.

When (T_1-T_2) is small, we may take as an approximation

$$H_r = h_r \,.\, F_\epsilon A(T_1 - T_2) \qquad (1.14)$$

h_r has the value 0·8 B.Th.U./ft.^{2} hr. °F. previously found (p. 21), and hence $h_r \,.\, F_\epsilon = 0{\cdot}65$ B.Th.U./ft.^{2} hr. °F. when $\epsilon_1=\epsilon_2=0{\cdot}9$

The transfer of heat by convection from the hot surface to the air or from the air to the cold surface can, in theory, be calculated from the normal convection equation

$$H_c = C(\theta_1-\theta_2)^{5/4} = h_c(\theta_1-\theta_2) \qquad (1.15)$$

At ordinary temperatures and with small differences of temperature, h_c for a vertical surface has a value of about 0·50 B.Th.U./ft.^{2} hr. °F. (Table 1.9). In using these expressions to compute the heat transfer across an air space, we must put

AIR CHANGES AND TEMPERATURES FOR HEAT-LOSS CALCULATIONS

AIR CHANGE DUE TO INFILTRATION.

The rate of air change by natural infiltration depends on the construction of the building, tightness of windows and doors, the strength of wind, exposure, height opening of doors, and other factors outside the control of the heating and ventilating engineer.

In the interests of economy of design of the heating installation and of fuel conservation, the building designer should endeavour to reduce uncontrolled air-change rate to a minimum by weather-stripping, sealing and all other steps possible.

Although in this section of the *Guide* certain standards of air changes are nominated, to cover the average case the designer will, of necessity, use his judgment in varying from these standards where such appears to be warranted.

AIR CHANGE FOR VENTILATION.

During occupation, infiltration air change may not be adequate for good ventilation. Ventilation may be obtained by opening windows, or by means of a natural or mechanical system.

If ventilation is by windows, or other natural means, or by mechanical extraction, heat for warming incoming air will be furnished by the heating system which must be designed accordingly.

WARM-AIR INLET.

If ventilation is by *Mechanical Warm-air Inlet*, heat for warming the incoming air will be provided for in the ventilating plant. The heating system in this case should be designed for the following :

Room or building with no external windows or doors	fabric loss only.
Room or building with exposed windows or doors	$\frac{1}{2}$ air change per hour.

TABLE OF TEMPERATURES AND RATES OF AIR CHANGE.

In the table on page 37 recommended air changes and temperatures are given.

The air-change rates are intended as a guide to normal requirements for infiltration or ventilation, but as stated previously, in view of the many variables, they may need to be varied according to conditions.

A note on the effect of exposure and occupancy is given over the page.

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Thermal transmittance of ground floors

The thermal transmittance of a ground floor beneath which is a fully ventilated air space may be computed in the usual way, assuming that still-air conditions exist on both sides of the floor

TABLE 1.27 THERMAL TRANSMITTANCE OF GROUND FLOORS

Floor construction	*Thermal transmittance* * (*B.Th.U./ft.² hr. °F.*)
Ventilated wood joist floor:	
Air brick on one side only, bare boards . .	0·30
Air brick on one side only, covered with parquet, linoleum or rubber	0·25
Air brick on more than one side, covered with parquet, linoleum or rubber	0·35
Floors in contact with earth, hardcore, etc.:	
Concrete or other dense surface finish (e.g. tiles or granolithic)	0·20
Wood block on concrete	0·15

* The temperature difference between indoors and outdoors is to be used.

and that the underside air temperature is equal to the outdoor air temperature. In some cases, as when the floor is of plain-edged boarding, an extra allowance is necessary to take account of the air infiltration through the floor into the room. This allowance may be about 20 per cent. of the transmittance computed for an air-tight floor; and it accounts for the reduction

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in U when the floor is covered with a carpet or linoleum. The data given in Table 1.27, taken from the I.H.V.E. booklet, are estimated values which are found in practice to be suitable for the purpose of determining the size of a heating plant.

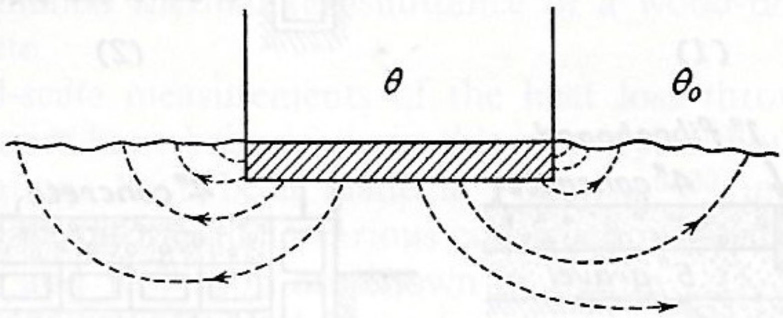

Fig. 1.14 Heat flow through a ground floor

It will be noticed that the thickness of concrete floors is not specified. The reason for this will be clear from a reference to Figure 1.14. The diagram represents the lines of heat flow

TABLE 1.28 HEAT LOSSES FROM GROUND FLOORS (DILL *et al.*)

Floor No.	*Crawl space temp. (°F.)*	*Outdoor air temp. (°F.)*	*Thermal transmittance* Indoor—outdoor (B.Th.U./ft.^{2} hr. °F.)	*Thermal transmittance* Indoor—crawl space (B.Th.U./ft.^{2} hr. °F.)	*F* (B.Th.U./ft. hr. °F.)*
1 . . .	—	30·2	0·14	—	0·81
2 . . .	—	30·2	0·12	—	0·69
3 . . .	—	30·4	0·09	—	0·55
4 . . .	—	30·7	0·13	—	0·75
5 vent. . .	38·8	31·8	0·20	0·24	—
5 unvent. .	49·9	30·0	0·12	0·24	—
6 vent. . .	40·4	31·9	0·34	0·43	—
6 unvent. .	53·7	30·2	0·19	0·46	—
7 vent. (a) .	41·5	32·1	0·30	0·46	—
(b) .	38·4	31·9	0·26	0·32	—
7 unvent. (a)	52·9	29·6	0·18	0·42	—
(b) .	53·3	31·7	0·16	0·36	—
8 vent. . .	38·0	32·2	0·24	0·28	—
8 unvent. .	49·5	30·8	0·17	0·31	—

* F=Heat loss per ft. of exposed edge, and per °F. difference of temperature between indoors and outdoors.

(a) Bare floor; (b) Carpeted floor.

13.6 WILD HEAT

Heating engineers have, as a class, been concerned to avoid the risk of underheating, and as a result many systems have been considerably oversized. The loads are calculated for an empty, dark building, whereas in fact, buildings are occupied and lit. The heat dissipation from people, lights and machinery, together with that from sunshine through the windows, can be quite considerable — it is indeed the *raison-d'etre* of the degree-day base.

The value of this wild heat is reducing the demand even in design weather was not often appreciated. However, Hoffman and Raber (US, 1913) said that heat gains from lights, and metabolic heat, could be credited to the heating system.(29) Fowler(24) on the basis of observed energy consumptions in similar rooms, drew attention to this contribution, but he did not go so far as to advocate any reduction in the design load. It remained for the electrical industry to capitalise on these gains, albeit unwittingly, in their design techniques for storage heating. HVRA(1) through measurements in a number of buildings, demonstrated that maximum loads were in fact reduced by the miscellaneous gains, and that it was legitimate to deduct them from computed steady-state structural and ventilation losses in order to arrive at a suitable plant size (1961).

13.7 DEGREE-DAYS

The quantity of fuel used in a heating season to maintain a room or building at a given temperature is expected to depend on the difference between the inside and outdoor temperatures and upon the duration of this difference. The concept of the "degree-day" to quantify the duration and severity of a climate originated with Sir Richard Strachey in 1878, who used it for agricultural purposes. The first application of the degree-day to heating problems was due to the American Gas Association.* They found, from a survey of a large number of buildings, that no fuel was consumed when the daily mean outdoor temperature rose above 65°F (18°C) (for an internal temperature of 70°F (21°C)), and that the fuel consumption on any one day was proportional to the difference between 18°C and the 24-h mean outdoor temperature. If this difference is, for example, 10°K, then there are 10 degree-days for that day. The annual total is the sum of the number of degree-days for each day of the year, and the annual fuel usage is proportional to the annual degree-day total.

The temperature above which no fuel is needed for heating — in this case 18°C — is termed the "base" of the degree-day; the difference between this base and the maintained indoor temperature arises from miscellaneous sources of heat within the building — lights, people and solar gains. Once the base is given, the degree-day total is easily computed from meteorological records for the locality.

Dufton(20) published a map of degree-days for the British Isles in 1934, using a base of 60°F (15°C) to correspond with the British indoor temperature of 65° (18°C). He thus continued the American assumption of a 5°F difference between base and indoor temperatures. The idea of degree-days also gained a foothold in Europe, though the basis was rather different. There, degree-days were counted only for the heating season, and this too was variously defined as the period of the year when the daily mean fell below the indoor design temperature or below some other (lower) temperature designated the heating limit.

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*In 1915, Eugene D. Milener, an engineer with the gas utility company in Baltimore, found that gas consumption of heating plants in that city varied with the number of degrees difference between 64°F (17.8°C) and the outside temperature. Later studies indicated an improved relationship if 65°F (18.3°C) was chosen.

Grierson (1940) observed that not all buildings in Britain were warmed to 18°C, and suggested that degree-day totals were needed for a wide range of indoor temperatures.[25] He calculated figures for a 5-year period at Kew, stating only the "base" and not specifying a corresponding indoor temperature.

Knight and Cornell (1958) queried the propriety of applying conventional degree-days to buildings which were not continuously heated to a constant temperature. They proposed the use of a special degree-day for this purpose.[36] In a later paper (1966) Billington rejected the Knight-Cornell approach, preferring to retain the degree-day as a purely climatic statistic.[10] He identified the difference between the indoor temperature and the base temperature as the temperature rise caused by the miscellaneous heat gains to the building. It became clear from work at HVRA that the miscellaneous gains and their effects differed considerably from building to building, and hence that the use of a single base was no longer valid. Accordingly, Billington suggested a range of base temperatures appropriate to different structures and occupancy, while retaining the original dependence of the degree-day on outdoor temperature alone. At the time it appeared that the need for different base temperatures had also been recognised in France, but no basis for the choice of these bases had been published.

Somewhat earlier (1962) it had been proposed by Arnsted in Denmark to modify the degree-day to take account not only of temperature but also of the incidence of wind and sun, both of which affect the heat loss from a building. His corrections are to be applied only to degree-day totals for short periods such as a week or two. Arnsted states that the conventional total is adequate for the estimation of the fuel usage over long periods such as a complete heating season.

Attempts have also been made to evaluate summer (cooling) degree-days. These have so far failed because a large part of an air-conditioning load is latent heat and solar radiation through windows, and thus not directly related to the outdoor temperature.

A concept used to estimate the operating cost of refrigeration plant is that of "equivalent full-load operating hours". The same concept has been used in heating work,[10] both in Britain (chiefly at first by the electricity industry) and in Germany (often for district heating consumers). It is of course closely related to the annual degree-day total.

DEGREE DAYS

Degree days have for some time been used as a means of comparing, over different periods, the variations in load sustained by heating plants in different parts of the country. The standard method used is to assess, for monthly periods, the daily difference, in deg. F, between a base temperature of 60°F (5°F less than a typical room temperature) and the 24-hour mean outside temperature. The monthly totals may then be used to compare monthly changes in the weather factor, or be added together for the heating season, enabling the severity and duration of the winter to be compared from year to year and from place to place.

In a recent Paper to the Institution,* alternative bases of calculation of degree days were put forward to take account of the fact that buildings

* "Degree Days and Fuel Consumption for Office Buildings," by J. C. Knight and A. A. Cornell. *J.I.H.V.E.*, 1959, **26**, 309.

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are not always heated continuously to 65°F. Modern plant management is, in many cases, either to reduce inside temperatures at night and week-ends or to shut plants down altogether during those periods.

The following three tables have therefore been prepared to give, for different operating conditions, the average degree day totals and load factors from September 1st to May 31st for different meteorological stations in the country. The load factors are the relative annual heat requirements as percentages of the heat that would be required in a year if the outside temperature remained at 30°F. The stations for Table II and those for Tables III (*a*) and III (*b*) are in the same districts and the degree days and load factors are broadly comparable.

Table II

Degree days and load factors calculated by the standard method for continuous heating to 65°F, based upon records over the ten years from 1945–55. The figures are those prepared by the Gas Council.

Tables III (a) and III (b)

Degree days and load factors for the alternative method, based upon Ministry of Works figures for the eight years from 1950–58.

Table III (a) is for heating to 65°F during the daytime (6 a.m. to 6 p.m.) with a reduction to 55°F at night and at week-ends (Saturday and Sunday).

Table III (b) is for heating to 65°F during the daytime but complete shut-down at night and week-ends.

Table II (Standard Method)			**Table III (*a*)** (Alternative Method, M.O.W. Continuous)			**Table III (*b*)** (Alternative Method, M.O.W. Intermittent)	
Met. Station	Degree Days	Load Factors	Met. Station	Degree Days	Load Factors	Degree Days	Load Factors
Dundee	4469	54%	—			—	
Glasgow	4454	54%	Glasgow (Renfrew)	2580	45%	1350	52%
Durham	4327	53%	Newcastle (Tynemouth)	2449	43%	1339	51%
Wakefield	4049	49%	Hull (Spurn Head)	2488	43%	1319	50%
Manchester	4071	50%	Manchester	2498	44%	1309	50%
Birmingham (Edgbaston)	3994	49%	Birmingham (Elmdon)	2598	45%	1341	51%
Cambridge	3920	48%	Cambridge (Mildenhall)	2418	42%	1237	47%
—			Aberporth	2133	37%	1207	46%
Bristol	3736	46%	Bristol	2257	39%	1211	46%
Kew	3615	44%	Kew	2148	37%	1170	45%
Southampton	3478	42%	Bournemouth (Hurn)	2146	37%	1146	44%

TYPICAL RECORDS OF SEVERE CONDITIONS.

The data which follow in Figs. 13–17 are by way of illustration of a number of severe conditions which can occur. They are not intended to indicate extremes that are never exceeded.

Typical Thermograms of Severe Conditions.

Figs. 13–15 show the variation of dry-bulb temperature on three particular days. The differences between the three curves are due to the varying meteorological conditions which prevailed ; thus the day shown in Fig. 13 was one of clear, calm conditions with strong radiation gains by day and losses by night, while the days shown in Figs. 14 and 15 were days of cold winds and obscured skies.

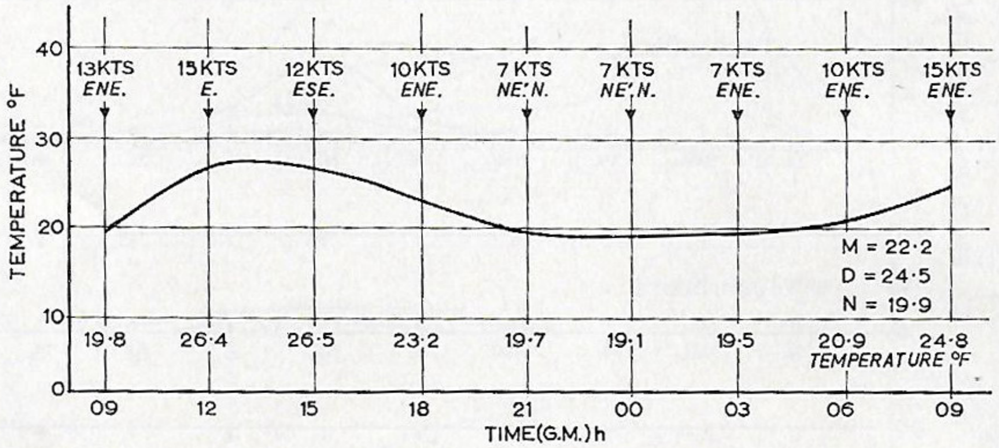

Fig. 13.—London Airport Thermogram for January 29th, 1947.

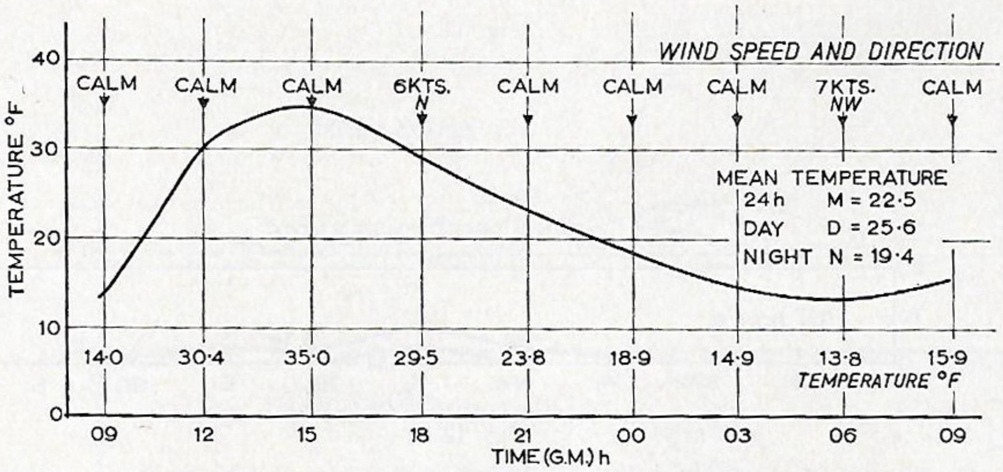

Fig. 14.—London Airport Thermogram for February 24th, 1947.