CHAPTER 10

Energy Efficient Heating

This chapter outlines good practice in the design and operation of building heating systems. It includes a discussion of the types of heating system that should be selected for particular applications. In addition, the concept of building heat loss is discussed, and methodologies presented for predicting heating energy costs and optimizing plant utilization.

10.1 Introduction

For those who live in temperate, cool or cold climates, the provision of adequate heating is essential for large parts of the year in order to maintain comfort and good health. While some may consider good 'central heating' to be a luxury, a recent epidemiological study undertaken in several European countries revealed that death rates amongst those over 50 years of age are proportional to the degree of coldness experienced [1]. In fact, in England and Wales there is a 2% increase in mortality for every 1°C below an outside temperature of 19°C [2]. As a result, weather-related mortality rates in the UK are amongst the highest in Europe, with an estimated 40,000 additional deaths occurring during the winter months each year [1]. One major contributory factor to this sad state of affairs is the poor state of much of the housing stock in the UK [1,3].

In addition to preserving the good health of building occupants, adequate heating of public, commercial and industrial buildings is essential to promote efficient and productive work practices. If people are cold and uncomfortable, they will not perform well and so output will fall. Clearly it is false economy to save a little on fuel costs and lose much in productivity.

Although a building may be heated to a comfortable level, it does not always mean that it is efficiently heated. In many buildings large quantities of heat energy are wasted daily due to a combination of poor design and poor operating and maintenance practices. Broadly speaking, heat can be wasted by any combination of the following:

- Poorly designed heating systems, which are often wrongly selected for particular applications.
- Poorly designed and insulated building envelopes.
- Poorly insulated heating systems.
- Poorly commissioned and maintained boiler plant.
- Poor controls.
- Poor operating practices.

To understand the reasons why heat is so easily wasted in buildings, it is worth considering the case of an old, poorly maintained church hall with a high roof, which is heated by an old low temperature hot water (LTHW) radiator system, served by an oil-fired boiler. Let us assume that the building is used for meetings and social events, mainly in the evenings and at weekends. Without going into too much detail, there are a number of possible ways in which the heating system might be considered to be inefficient:

- The building envelope is probably poorly insulated so heat will be easily lost to the outside by conduction, especially on cold winter nights.
- The building envelope is probably not very well sealed and so large amounts of the warm heated air will be lost through cracks around doors and window frames.
- In the church hall the warm air heated by the radiators will rise up into the high roof space where it will stratify. Consequently, any pigeons that might be in the roof space will be warm, while the occupants at floor level will probably feel cold and uncomfortable.
- The pipework from the boiler to the radiators may not be very well insulated. This will result in major heat losses if the pipes run through unheated spaces, such as floor voids or in the boiler room.
- The boiler will probably be poorly maintained, with the result that some of the heat meant to warm up the water will be lost up the flue with the combustion gases.
- Since the system is old, it is probable that its controls will be inadequate. For example, the system might have a time clock which turns the boiler and pump on at the same time each day, whether or not the hall is occupied. Alternatively, the thermostat in the room space may be set at too high a temperature, so that the hall overheats and the occupants become uncomfortable. It is not uncommon for the occupants of buildings to open windows rather than turn down thermostats. After all, it is often the case that the occupants of a building are not the ones paying the fuel bills.

The illustration of the poorly heated church hall demonstrates four important points:

1. It is essential to take a holistic view when designing a heating system. The building fabric and the mechanical heating system are equally important.
2. It is important to select the correct heating system for the particular application.
3. It is important to have an adequate control system.
4. It is no use designing and installing an excellent heating system, if the building operatives and occupants do not understand how to use it correctly.

10.2 Thermal Comfort

When an individual is in a room, his/her vital organs are maintained at a temperature of 37.2°C through a complex set of heat-transfer mechanisms. The person will lose heat to the surrounding air by convection and to any cold surfaces within the room space by radiation. However, hot surfaces within the room will cause the person to gain heat by radiation. Consider a room in which the air is maintained at 21°C by wall-mounted radiators which have a surface temperature of 70°C. The occupants of the room will lose heat by convection to the air, because the surface of their clothes will be at approximately 30°C. They will also lose heat by radiation to many of the cool surfaces in the room, many of which will be at a temperature of less than 21°C. At the same time they will gain heat by radiation from the hot radiators. The occupants will also lose heat by evaporation through exhalation and perspiration. A small amount of heat will also be lost through their feet by conduction. Consequently, a complex heat balance is set up, which maintains the core temperature of the occupant's bodies at 37.2°C. If for any reason a person becomes hot or cold, in other words uncomfortable, then their body will take involuntary action in order to maintain the core temperature. One such mechanism is perspiration, which increases under warm conditions, so that increased evaporative cooling occurs. If a person is uncomfortable, he/she can also take voluntary steps to rectify the situation. For example, extra/excess clothing can be worn or removed. Equation (10.1) expresses the heat balance between the human body and the surrounding environment [4]. It should be noted that the term for conduction has been omitted from eqn (10.1) because in most normal situations it is negligible:

$$M - W = Q_e \pm Q_c \pm Q_r + S \qquad (10.1)$$

where M is the metabolic rate (W), W is the rate at which energy is expended in mechanical work (W), Q_c is the rate of heat transfer by convection (W), Q_r is the rate of heat transfer by radiation (W), Q_e is the rate of heat loss by evaporation (W), and S is the rate at which heat is stored in the body (W).

Food that is digested by the body is converted into energy. Some of this energy is used to perform mechanical work, but most of it produces heat. In fact, under normal conditions, more heat is produced by the body than it actually requires to maintain its core temperature. Therefore heat is lost from the body by convection, radiation and evaporation. The human body can also gain heat by convection and radiation under hot conditions. It is, however, impossible to gain heat by evaporation, since this is always a cooling mechanism. If a person performs exercises or mechanical work, then the metabolic

TABLE 10.1 Sensible heat outputs and recommended dry resultant temperatures [5]

Type of work	Typical application	Required room dry resultant temperature (°C)	Sensible heat output per person (W)
Light work	Office	20.0	100
Walking slowly	Bank	20.0	110
Light bench work	Factory	16.0	150*
Heavy work	Factory	13.0	200*

*Assumes a dry-bulb temperature of 15°C.

rate increases and the body is required to reject more heat, otherwise it will overheat. Consequently, in rooms such as gymnasia, where vigorous exercise is performed, it is necessary to have a lower room air temperature to compensate for the increased heat production rate of the human body. Table 10.1 shows heat outputs for a range of work rates, together with recommended room temperatures.

The thermal comfort of building occupants can be affected by personal and environmental factors. Personal factors can be defined as those variables which are directly connected with the individual. These include:

- Activity: The higher the level of personal activity, the greater the heat output.
- Clothing: The greater the amount of clothing worn, the lower the heat loss.
- Age: The metabolic rate decreases with age.
- Gender: The resting metabolic rate of women is approximately 10% lower than that of men.
- Health: Illness affects the ability of the body to maintain its core temperature at 37.2°C.

These personal comfort factors should always be considered when designing heating systems for particular applications. For example, because the metabolic rate of the elderly is lower than that of younger people, it is important to maintain air temperatures at a higher than normal level in sheltered accommodation for the elderly. Similarly, when designing the foyer of a railway station, where passengers in transit may be wearing outdoor winter clothes, it is advisable to maintain the air temperature at a lower temperature than, say, an office, where indoor clothing would normally be worn.

The environmental parameters which influence thermal comfort are: air temperature, mean radiant temperature, relative humidity and air velocity. These parameters, in different ways, influence heat transfer by convection, radiation and evaporation. Convective heat transfer is influenced by air temperature and air velocity. It takes place continually, although in most buildings it is imperceptible because air speeds are low (i.e. below 0.1 m/s). To be perceptible, air speeds must exceed approximately 0.2 m/s. Radiant heat transfer occurs between the skin/clothes and those surfaces 'seen' by the human body. It is therefore heavily influenced by the *mean radiant temperature* (t_r) of the surfaces within rooms. For cuboid shaped rooms, the approximate mean radiant temperature at the centre of the room can be determined by:

$$t_r = \frac{a_1 \cdot t_1 + a_2 \cdot t_2 + a_3 \cdot t_3 + \cdots + a_n \cdot t_n}{a_1 + a_2 + a_3 + \cdots + a_n} \tag{10.2}$$

where t_r is the mean radiant temperature (°C), a_1, a_2,..., are the room component surface areas (m^2), and t_1, t_2,..., are the room component surface temperatures (m^2).

Evaporative heat loss is governed by the relative humidity of air and air velocity. If conditions are very humid, as in tropical countries, then evaporative heat losses will be low, since any perspiration produced is unable to evaporate. However, if the air is dry then evaporation readily takes place and the body is cooled. Evaporative cooling becomes an important heat-transfer mechanism at air temperatures above 25°C. At air temperatures above 29°C almost all heat is lost from the body by evaporation [6].

Thermal comfort is determined by a number of environmental factors, particularly mean radiant temperature, air temperature and air velocity. It is thus important that these factors are considered when designing buildings. The concept of *dry resultant temperature* (t_{res}) was developed to make allowances for these considerations. The dry resultant temperature is defined as:

$$t_{res} = \frac{t_r + t_a \cdot \sqrt{10 v_a}}{1 + \sqrt{10 v_a}} \tag{10.3}$$

where v_a is the air velocity (m/s), and t_a is the air temperature (°C).

However, in most buildings, because the air velocity is below 0.1 m/s, eqn (10.3) can be simplified to:

$$t_{res} = 0.5\, t_r + 0.5\, t_a \tag{10.4}$$

Equation (10.4) tells us that mean radiant temperature is as important as air temperature when maintaining a comfortable environment within buildings. This explains why on returning to an unheated house after some days of absence, the rooms will feel cold and uncomfortable, even though the heating is on and the air up to temperature. The air temperature may be acceptable, but the dry resultant temperature will still be low because the fabric of the building is still cold.

10.3 Building Heat Loss

When a building is heated to a steady internal temperature, an equilibrium is established in which the heat power into the building equals the rate at which heat is lost from the building. Thus to maintain a comfortable internal environment, the output from any heating system must be equal to or greater than the combined effect of the heat loss through the building fabric and the ventilation heat losses. Fabric heat losses are those which occur primarily by conduction through walls, windows, floors and roofs. Ventilation losses are the convective heat losses which occur when warm air is lost from a building and replaced by cold air. A good approximation of the fabric heat losses can be determined using the generic equation:

$$Q_f = U \times A \times (t_{ai} - t_{ao}) \tag{10.5}$$

where Q_f is the fabric heat loss rate (W), U is the thermal transmittance (U value) (W/m^{2}K), A is the area (m^2), t_{ai} is the internal air temperature (°C), and t_{ao} is the external air temperature (°C).

Similarly, the ventilation heat loss can be determined using:

$$Q_v = 0.333 \times n \times V \times (t_{ai} - t_{ao}) \tag{10.6}$$

where Q_v is the ventilation heat loss rate (W), n is the ventilation rate (air changes per hour), and V is the volume (m^3).

From eqns (10.5) and (10.6) it can be seen that both the fabric and the ventilation heat loss rates are directly proportional to the difference between the internal and the external air temperatures. Since the internal air temperature should be maintained at a constant level during the winter, the heat loss therefore varies with the outside air temperature. Boilers and heat emitters should be sized for the 'worst-case' scenario (i.e. a very cold day) and be capable of maintaining the internal design temperature under this extreme weather condition. When operating under less extreme conditions (i.e. under part-load conditions), the output of the heating system can be reduced by using controls. Table 10.2 gives some sample winter external design conditions for various parts of the world.

The simplest way to reduce the energy consumed by a heating system is to create a building envelope which is well insulated and in which ventilation rates are controlled to the minimum required for healthy living. Achieving this usually involves care and attention during both the design and the construction stages, together with increased capital outlay. Not surprisingly, many speculative builders have little incentive to construct energy efficient building envelopes, since they usually do not pay the fuel bills. Therefore most countries have building regulations, of varying degrees of rigour, to force developers to conform to certain minimum thermal insulation standards. Usually these standards are expressed in terms of maximum permissible U values for glazing, walls, floors and roofs.

10.3.1 *U* Values

Thermal insulation standards are usually expressed in terms of U values. The U value, or thermal transmittance, is a measure of the overall rate of heat transfer, under standard conditions, through a particular section of construction. It has units W/m^{2}K and, as can be seen from eqn (10.7), is the inverse of thermal resistance. The lower the U value of an item of construction, the better its thermal insulation performance:

$$U = \frac{1}{\Sigma R} \tag{10.7}$$

where ΣR is the total thermal resistance of the construction (m^{2}K/W).

TABLE 10.2 External winter design temperatures for sample cities around the world [7]

Country	City	Winter external design temperature (°C)
Australia	Perth	6
	Sydney	6
Belgium	Brussels	−7
China	Shanghai	−3
France	Lyon	−10
	Paris	−4
Germany	Berlin	−11
	Hamburg	−9
	Munich	−13
Italy	Milan	−6
	Naples	2
	Rome	1
India	New Delhi	4
Japan	Tokyo	−2
New Zealand	Christchurch	−1
	Wellington	3
Norway	Oslo	−16
Spain	Barcelona	2
	Madrid	−2
Sweden	Stockholm	−13
United Kingdom	Birmingham	−3
	Glasgow	−2
	London	−2
	Manchester	−2.5
USA	Chicago	−20
	Dallas	−6
	Kansas City	−14
	Los Angeles	4
	Miami	8
	New Orleans	1
	New York	−9
	San Francisco	3
	Seattle	−9
	Washington DC	−8

TABLE 10.3 Thermal conductivity of various materials [8]

Material	Density (kg/m^3)	Thermal conductivity (W/m K)
Brickwork (outer leaf)	1700	0.84
Brickwork (inner leaf)	1700	0.62
Cast concrete (dense)	2100	1.40
Cast concrete (lightweight)	1200	0.38
Concrete block (heavyweight)	2300	1.63
Concrete block (medium weight)	1400	0.51
Concrete block (lightweight)	600	0.19
Fibreboard	300	0.06
Plasterboard	950	0.16
Plaster (dense)	1300	0.50
Plaster (lightweight)	600	0.16
External rendering	1300	0.50
Screed	1200	0.41
Asphalt	1700	0.50
Tile	1900	0.84
Wood–wool slab	650	0.14
Expanded polystyrene	25	0.035
Glass fibre slab	25	0.035
Phenolic foam	30	0.040

The thermal resistance of each component layer of any construction can be determined by using eqn (10.8):

$$R = \frac{l}{\lambda} \tag{10.8}$$

where R is the thermal resistance of layer (m^2 K/W), l is the thickness of layer (m), and λ is the thermal conductivity (W/m K).

The total thermal resistance of a construction is the summation of the resistances of all the individual layers, plus the resistances to heat transfer of any surfaces. Table 10.3 gives typical thermal conductivity values for a variety of building materials.

Figure 10.1 illustrates the heat-transfer mechanisms that take place in a typical external wall with an air cavity. While the heat transfer through the solid parts of any construction is by conduction, it is radiation and convection that control the heat transfer at the

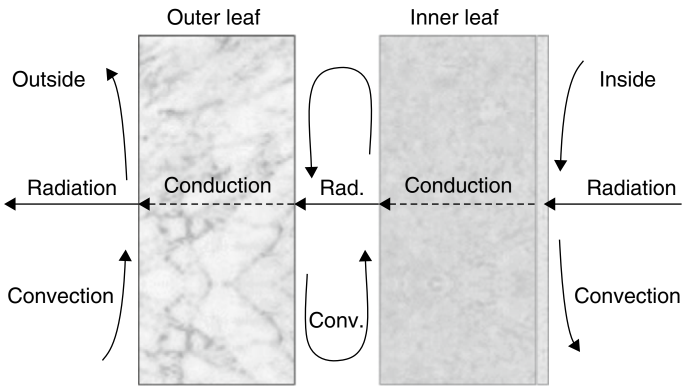

FIG 10.1 Heat transfer through a cavity wall.

surfaces. The resistance to heat transfer of a surface can be calculated using eqn (10.9), which takes into consideration the convective and radiative heat-transfer coefficients, and the emissivity of the surface:

$$R_s = \frac{1}{h_c + (E \times h_r)} \tag{10.9}$$

where R_s is the surface resistance (m^2 K/W), h_c is the convective heat-transfer coefficient (W/m^2 K), h_r is the radiative heat-transfer coefficient (W/m^2 K), and E is the emissivity factor.

The convective heat-transfer coefficient is the rate at which heat is transferred to or from a 1 m^2 surface by convection, per 1°C temperature difference between the surface and the neighbouring fluid (i.e. air). The radiative heat-transfer coefficient is the rate of radiant heat transfer to or from a 1 m^2 surface of a *black body* divided by the difference between the mean temperatures of the radiating surface and that of the surrounding surfaces. The term *black body* refers to a body which absorbs all energy incident on it at every wavelength and conversely emits energy at every wavelength. The emissive power of a black body can be calculated using eqn (10.10):

$$E_b = \sigma \times T^4 \tag{10.10}$$

where E_b is the emissive power (W/m^2), σ is the Stefan–Boltzmann constant (i.e. 5.67×10^{-8} W/m^2 K^4), and T is the absolute temperature of the black body (K).

Of course, black bodies are theoretical and building materials do not emit and absorb energy at all wavelengths and are therefore non-black bodies. The term *emissivity* is used to describe the ratio of the emissive power of a non-black body to that exhibited by a black body. The emissivity of a black body is 1, while that of all other bodies will be less than 1. Dull surfaces, of the type exhibited by most building materials, have a high emissivity, while shiny metallic surfaces have a low emissivity. High-emissivity surfaces are good emitters and receivers of radiation, while low-emissivity surfaces are not. Table 10.4 gives examples of the emissivity values for typical building materials.

TABLE 10.4 Typical values of emissivity [9]

Surface	Emissivity
Black body	1
Black (non-metallic)	0.90–0.98
Concrete	0.85–0.95
White paint	0.85–0.95
Aluminium (dull)	0.20–0.30
Aluminium (polished)	0.02–0.05

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TABLE 10.5 Typical internal surface resistances [8]

Building element	Direction of heat flow	Surface resistance ($m^2 K/W$)	
		High emissivity factor ($E = 0.97$)	Low emissivity factor ($E = 0.05$)
Walls	Horizontal	0.12	0.30
Ceiling or roofs	Upward	0.10	0.22
Ceiling or floors	Downward	0.14	0.55

The emissivity factor (E) referred to in eqn (10.9) allows for the emissivity and geometrical relationship of both the emitting and the receiving surfaces. The emissivity factor varies with the specific geometrical arrangement of the surfaces involved, but for most building applications it can be taken as being approximately 0.9.

The thermal resistance of any surface is strongly influenced by air velocity. However, for most building applications, room air velocities are not greater than 0.1 m/s and so it is possible to determine typical values for a variety of surfaces (as shown in Table 10.5). Table 10.5 presents typical internal surface resistances for both high- and low-emissivity surfaces.

The resistance of external building surfaces is heavily influenced by wind speed. Table 10.6 gives typical external surface resistances under sheltered, normal and severe conditions.

Unventilated air cavities in walls and roofs offer resistance to heat transfer. For cavities in walls the thermal resistance increases with thickness up to approximately 25 mm. Thereafter the thermal resistance of the air cavity is virtually constant despite any further increase in thickness. Heat transfer across an air cavity is by convection and radiation. Thus the emissivity of the cavity surfaces significantly influences the heat transfer. Table 10.7 gives typical thermal resistances for a variety of air cavities.

Example 10.1 illustrates how the above equation and data can be used to determine the U value of an external wall.

TABLE 10.6 Typical external surface resistances [8]

Building element	Emissivity of surface	Surface resistance ($m^2 K/W$)		
		Sheltered	Normal	Severe
Wall	High	0.08	0.06	0.03
	Low	0.11	0.07	0.03
Roof	High	0.07	0.04	0.02
	Low	0.09	0.05	0.02

Sheltered: Applies up to the third floor in city centres.
Normal: Applies to most suburban and rural areas, and from the fourth to the eighth floors on tall buildings in city centres.
Severe: Applies to coastal and hill sites, above the fifth floor in suburban and rural districts and above the ninth in city centres.

TABLE 10.7 Typical resistances of unventilated air cavities [8]

Cavity thickness	Emissivity of surface	Thermal resistance ($m^2 K/W$) for stated heat flow direction		
		Horizontal	Upward	Downward
5 mm	High	0.10	0.10	0.10
	Low	0.18	0.18	0.18
25 mm or greater	High	0.18	0.17	0.22
	Low	0.35	0.35	1.06

Example 10.1

An external wall has the following construction:

Element	Thickness (mm)	Thermal conductivity (W/m K)
Plaster	13	0.500
Concrete block (inner leaf)	100	0.200
Air cavity	50	n.a.
Brick (outer leaf)	102	0.840

Assuming that the air cavity has a thermal resistance of 0.18 $m^2 K/W$, the internal surface resistance is 0.123 $m^2 K/W$ and the external surface resistance is 0.055 $m^2 K/W$, determine:

(i) The *U* value of the wall.
(ii) The *U* value of the wall if the cavity is filled with insulating foam having a thermal conductivity of 0.036 W/m K.

Solution

Using the above data and eqn (10.8), it is possible to determine the total thermal resistance of the cavity wall as follows:

Element	Thickness (m)	Thermal conductivity (W/m K)	Thermal resistance (m^2 K/W)
Internal surface resistance	n.a.	n.a.	0.123
Plaster	0.013	0.500	0.026
Concrete block (inner leaf)	0.100	0.200	0.500
Air cavity	0.050	n.a.	0.180
Brick (outer leaf)	0.102	0.840	0.121
External surface resistance	n.a.	n.a.	0.055
			Total resistance = 1.005

(i) Therefore:

$$U \text{ value (existing wall)} = \frac{1}{1.005} = 0.995 \text{ W/m}^2 \text{ K}$$

(ii) If the cavity of the wall is filled with foam, then:

$$\text{Thermal resistance of foam} = \frac{0.05}{0.036} = 1.389 \text{ m}^2 \text{ K/W}$$

However, the thermal resistance of 0.18 m^2 K/W for the air cavity no longer applies, therefore:

$$\text{New thermal resistance of wall} = 1.005 + 1.389 - 0.18 = 2.214 \text{ m}^2 \text{ K/W}$$

Therefore:

$$\text{New } U \text{ value} = \frac{1}{2.214} = 0.452 \text{ W/m}^2 \text{ K}$$

It can be seen from Example 10.1 that by filling the cavity with insulating foam, it has been possible to reduce the *U* value of the wall by 54.6%.

10.3.2 Heat Loss Calculations

If the *U* values for the various component parts of a building's fabric are known, then it is possible using eqns (10.5) and (10.6) to determine, with relative accuracy, the wintertime design day heat loss rate and thus ultimately size boiler plant and heat emitters. Example 10.2 illustrates how this calculation should be performed.

Example 10.2

The surface areas and U values of the elements of a building are as follows:

Element	Area (m^2)	U value ($W/m^2 K$)
Floor	200	0.45
Roof	200	0.28
Single glazing	16	5.60
External doors	8	2.00
External walls	216	0.60

If the internal design temperature is 21°C and the external design temperature is −1°C, determine the design day heat loss rate. (Assume that the building experiences three air changes per hour and that its volume is 800 m^3.)

Solution

Under the wintertime design condition, the temperature difference between the inside and outside is 22. Given this, and by applying eqn (10.5) (i.e. $Q_f = UA(t_{ai} - t_{ao})$), it is possible to calculate the heat loss rate through each component element of the building fabric:

Element	Area (m^2)	U value ($W/m^2 K$)	Temperature difference (°C)	Heat loss (W)
Floor	200	0.45	22	1980.0
Roof	200	0.28	22	1232.0
Single glazing	16	5.60	22	1971.2
External doors	8	2.00	22	352.0
External walls	216	0.60	22	2851.2
				Fabric heat loss = 8386.4

By applying eqn (10.6) it is possible to determine the ventilation heat loss:

$$\text{Ventilation heat loss} = 0.3333 \times 3 \times 800 \times (21 - (-1)) = 17{,}600 \text{ W}$$

Now:

$$\text{Total heat loss} = \text{Fabric loss} + \text{Ventilation loss}$$

Therefore:

$$\text{Total heat loss} = 8386.4 + 17{,}600 = 25{,}986.4 \text{ W}$$

From an energy conservation point of view, one of the advantages of performing a design day heat loss calculation is that it gives an elemental breakdown of the relative heat losses from the building, enabling them to be quickly and easily evaluated. From Example 10.2 it can be seen that 1971.2 W is lost through the single glazing (i.e. 7.6% of the total heat loss). However, the ventilation loss of 17,600W represents 67.7% of the total heat loss. Given this, it would be folly to install double glazing without first reducing the ventilation heat loss.

While the use of eqns (10.5) and (10.6) gives a relatively accurate value of the winter design day heat loss, it can be inaccurate, especially in applications where radiant heating is used. A superior model, which fully takes into account the radiant heat transfer which occurs within a room space, is described by eqns (10.11) and (10.12):

$$Q_f = F_1 \times \sum (AU) \times (t_c - t_{ao}) \tag{10.11}$$

and

$$Q_v = F_2 \times 0.333 \times N \times V \times (t_c - t_{ao}) \tag{10.12}$$

and

$$Q_p = Q_f + Q_v \tag{10.13}$$

where Q_p is the heating plant output (W), t_c is the dry resultant temperature in the centre of the room (°C), and F_1, F_2 are the characteristic temperature ratios.

The temperature ratios, F_1 and F_2, are defined as:

$$F_1 = \frac{(t_{ei} - t_{ao})}{(t_c - t_{ao})} \tag{10.14}$$

and

$$F_2 = \frac{(t_{ai} - t_{ao})}{(t_c - t_{ao})} \tag{10.15}$$

where t_{ei} is the internal environmental temperature (°C).

The internal environmental temperature is a theoretical construct which is used to calculate the radiative and convective heat transfer to the inside surface of an external wall from the other surfaces in a room. In temperate and hot climates, it can be defined as:

$$t_{ei} = \frac{1}{3} \cdot t_{ai} + \frac{2}{3} \cdot t_m \tag{10.16}$$

where t_m is the mean surface temperature of the room (°C).

The CIBSE publish tables of values for F_1 and F_2 for various heating systems [10]. Tables 10.8, 10.9 and 10.10 show values of F_1 and F_2 for a forced warm air system, a panel radiator system and a high temperature radiant strip system. In each table values of F_1 and F_2 are presented against two variables, $\Sigma(AU)/\Sigma(A)$ and $NV/3\Sigma(A)$.

Once the values of F_1 and F_2 have been established for any system, it is possible to calculate the internal environmental temperature and the mean surface temperature of a room by using eqns (10.17) and (10.18):

$$t_{ei} = (F_1 \times (t_c - t_{ao})) + t_{ao} \tag{10.17}$$

TABLE 10.8 F_1 and F_2 values for 100% convective, 0% radiant heating (i.e. forced warm air heating) [10]

$NV/3\Sigma(A)$	$\Sigma(AU)/\Sigma(A)$							
	0.2		0.4		0.6		0.8	
	F_1	F_2	F_1	F_2	F_1	F_2	F_1	F_2
0.1	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
0.2	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
0.4	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
0.6	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
0.8	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
1.0	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
1.5	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
2.0	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
3.0	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13
4.0	0.99	1.03	0.98	1.07	0.97	1.10	0.96	1.13

TABLE 10.9 F_1 and F_2 values for 70% convective, 30% radiant heating (i.e. panel radiators) [10]

$NV/3\Sigma(A)$	$\Sigma(AU)/\Sigma(A)$							
	0.2		0.4		0.6		0.8	
	F_1	F_2	F_1	F_2	F_1	F_2	F_1	F_2
0.1	1.00	1.01	0.99	1.03	0.98	1.05	0.98	1.06
0.2	1.00	1.00	0.99	1.02	0.99	1.04	0.98	1.06
0.4	1.00	0.99	1.00	1.01	0.99	1.02	0.99	1.04
0.6	1.01	0.97	1.00	0.99	1.00	1.01	0.99	1.03
0.8	1.01	0.96	1.01	0.98	1.00	1.00	1.00	1.01
1.0	1.02	0.95	1.01	0.96	1.01	0.98	1.00	1.00
1.5	1.03	0.92	1.02	0.93	1.02	0.95	1.01	0.97
2.0	1.04	0.89	1.03	0.90	1.03	0.92	1.02	0.93
3.0	1.06	0.83	1.05	0.85	1.05	0.86	1.04	0.88
4.0	1.07	0.78	1.07	0.80	1.06	0.81	1.06	0.83

$$t_{ai} = (F_2 \times (t_c - t_{ao})) + t_{ao} \tag{10.18}$$

Rearranging eqn (10.16) gives:

$$t_m = \frac{3}{2} \cdot t_{ei} - \frac{1}{2} \cdot t_{ai} \tag{10.19}$$

Example 10.3 illustrates how eqns (10.11)–(10.13) can be applied to determine the plant output for the building illustrated in Example 10.2.

TABLE 10.10 F_1 and F_2 values for 10% convective, 90% radiant heating (i.e. high temperature radiant systems) [10]

$NV/3\Sigma(A)$	$\Sigma(AU)/\Sigma(A)$							
	0.2		0.4		0.6		0.8	
	F_1	F_2	F_1	F_2	F_1	F_2	F_1	F_2
0.1	1.01	0.97	1.02	0.95	1.02	0.94	1.02	0.93
0.2	1.02	0.95	1.02	0.93	1.03	0.92	1.03	0.91
0.4	1.03	0.91	1.03	0.90	1.04	0.88	1.04	0.87
0.6	1.04	0.87	1.05	0.86	1.05	0.85	1.05	0.84
0.8	1.05	0.84	1.06	0.83	1.06	0.82	1.06	0.81
1.0	1.06	0.81	1.07	0.80	1.07	0.79	1.07	0.78
1.5	1.09	0.74	1.09	0.73	1.09	0.72	1.10	0.71
2.0	1.11	0.68	1.11	0.67	1.11	0.66	1.12	0.65
3.0	1.14	0.59	1.14	0.58	1.14	0.57	1.14	0.57
4.0	1.16	0.52	1.16	0.51	1.17	0.50	1.17	0.50

Example 10.3

For the building described in Example 10.2, determine the heating plant output if the building is heated by:

(i) A forced warm air heating system.
(ii) A LTHW panel radiator system.
(iii) A high temperature radiant strip system.

Assume that the internal design dry resultant temperature is 21°C and that in all other respects the data are unchanged from that shown in Example 10.2.

Solution

In order to quantify F_1 and F_2 the values of the following parameters are determined:

$$\sum(A) = 640 \text{ m}^2$$

$$\sum(AU) = 381.2 \text{ W/K}$$

$$NV/3 = 800 \text{ W/K}$$

Therefore

$$\frac{\sum(AU)}{\sum(A)} = 0.60 \text{ W/m}^2\text{ K}$$

and

$$\frac{NV}{3\sum(A)} = 1.25 \text{ W/m}^2\text{ K}$$

By looking up Tables 10.8, 10.9 and 10.10 the values of F_1 and F_2 are found to be:

Option	Heating type	F_1	F_2
(i)	Forced warm air heating	0.970	1.010
(ii)	LTHW panel radiator system	1.015	0.965
(iii)	High temperature radiant strip system	1.080	0.755

Therefore, using eqns (10.11) and (10.13):

(i) Forced warm air heating:

$$Q_f = 0.97 \times 381.2 \times (21 - (-1)) = 8134.8 \text{ W}$$

and

$$Q_v = 1.10 \times 0.333 \times 3 \times 800 \times (21 - (-1)) = 19,360.0 \text{ W}$$

Therefore

$$Q_p = 8134.8 + 19,360 = 27,494.8 \text{ W}$$

Using eqns (10.17)–(10.19) it is possible to determine the internal air and environmental temperatures, and the mean internal surface temperature of the building:

$$t_{ei} = [0.97 \times (21 - (-1))] + (-1) = 20.3°\text{C}$$

$$t_{ai} = [1.01 \times (21 - (-1))] + (-1) = 23.2°\text{C}$$

$$t_m = \frac{3}{2} \times 20.3 - \frac{1}{2} \times 23.2 = 18.9°\text{C}$$

Similarly the plant output for options (ii) and (iii) can be established using the above methodology. A summary of the results of the winter design calculation for options (i), (ii) and (iii), together with the results from Example 10.2, is presented in the following table.

Option	Heating type	Q_f (W)	Q_v (W)	Q_p (W)	t_{ai} (°C)	t_m (°C)	t_{ei} (°C)	t_c (°C)
Example 10.2	Non-specific	8386.4	17,600.0	25,986.4	21.0	n.a.	n.a.	n.a.
(i)	Warm air	8134.8	19,360.0	27,494.8	23.2	18.9	20.3	21.1
(ii)	Panel radiators	8512.2	16,983.8	25,496.0	20.2	21.9	21.3	21.1
(iii)	High temp. radiant strip	9057.3	13,287.9	22,345.2	15.6	26.3	22.8	21.0

Comparison of the results from Examples 10.2 and 10.3 indicates that the use of the dry resultant temperature and the F_1 and F_2 coefficients has some effect on the calculated overall plant output. It can be seen that the required output of the high temperature radiant strip system is approximately -5 kW lower than that of the forced warm air system, with the radiator system in the middle. It should also be noticed that the simple method, based on the indoor air temperature, produces a plant output which was similar in magnitude to that required for the LTHW panel radiator system. The margin of error for the less accurate 'air temperature' method was −5.8% when compared with the warm air system, and +14.0% when compared with the radiant strip system. It can therefore be concluded that while the simple 'air temperature' method is acceptable for sizing standard wall radiator heating systems, the 'F_1 and F_2' method should be used when designing systems which are either 100% convective or almost 100% radiative.

10.4 Heating Energy Calculations

Equations (10.5), (10.6), (10.11) and (10.12) demonstrate that building heat loss is directly proportional to the difference in temperature between the internal and external environments. It follows therefore that buildings which experience harsher winters will, not surprisingly, consume more heating energy during the winter months. It is therefore possible to use the degree day method (see Appendix 1 for a detailed explanation of degree days) to predict annual heating costs.

For a building which is continuously heated and which experiences no substantial continuous heat gains, the annual heating energy consumption can be determined by using eqn (10.20) or (10.21):

$$E = \frac{Q_p}{(t_c - t_{ao})} \times D_{15.5} \times 24 \times \frac{1}{\eta} \qquad (10.20)$$

where E is the energy consumed (kWh), Q_p is the heating plant output (kW), $D_{15.5}$ is the number of standard degree days (i.e. to the base temperature 15.5°C), and η is the seasonal efficiency of heating system.

Typical values of η for various types of boiler plant are shown in Table 10.11. As an alternative to eqn (10.20), it is possible to use eqn (10.21) to predict heating energy consumption. However, because eqn (10.21) uses internal air temperature and represents a non-specific heating system, the results calculated are likely to be less accurate than those determined using eqn (10.20). Notwithstanding this, eqn (10.21) is reasonably accurate if buildings are well insulated and a predominantly convective heating system is being used.

$$E = \frac{Q_p}{(t_{ai} - t_{ao})} \times D_{15.5} \times 24 \times \frac{1}{\eta} \qquad (10.21)$$

When designing building heating systems it is standard practice to ignore any internal heat gains in the winter design day calculation. In this way the heating system is sized to meet the 'worst-case' scenario (i.e. when the internal heat gain is not present). In reality, if the heating plant is oversized, the heating controls should modulate down the flow water temperature and prevent the building from overheating. However, when performing energy prediction calculations, it may be important to allow for continuous internal heat gains (e.g. from lighting and equipment) in the degree day calculation. This can be achieved by altering the degree day base temperature from 15.5°C to an

TABLE 10.11 Seasonal heating plant efficiencies [11]

Type of system	Seasonal efficiency (%)
Continuous space heating	
Condenser and conventional boilers with weather compensated system	85
Fully controlled gas- or oil-fired boiler with radiator system	70
Fully controlled gas- or oil-fired boiler with radiator system (multiple modular boilers used with sequence controller)	75
Intermittent space heating	
Condenser and conventional boilers with weather compensated system	80
Fully controlled gas- or oil-fired boiler with radiator system	65
Fully controlled gas- or oil-fired boiler with radiator system (multiple modular boilers used with sequence controller)	70

TABLE 10.12 $D_d/D_{15.5}$ ratios for various base temperatures [12]

Base temperature (°C)	$D_d/D_{15.5}$
10	0.33
12	0.57
14	0.82
15	0.94
15.5	1.00
16	1.06
17	1.18
18	1.30

appropriate level. Table 10.12 shows various $D_d/D_{15.5}$ correction factors for various base temperatures.

Internal heat gains can be allowed for in the degree day calculation by determining the temperature rise due to internal gains using eqn (10.22):

$$d = \frac{Q_g}{Q_p} \times (t_c - t_{ao}) \tag{10.22}$$

where d is the average temperature rise which can be maintained by internal heat gains alone (K), and Q_g is the internal heat gain (W).

TABLE 10.13 CIBSE classification of structures by thermal inertia [12]

Weight	Building description
Very heavy	Multi-storey buildings with masonry or concrete curtain walling and sub-divided within by solid partitions.
Heavy	Buildings with large window areas but appreciable areas of solid partitions and floors.
Medium	Single-storey buildings of masonry or concrete, sub-divided within by solid partitions.
Light	Single-storey buildings of a factory type, with little or no solid partitions.

The new base temperature, t_b, can then be determined using eqn (10.23):

$$t_b = t_c - d \tag{10.23}$$

Equations (10.20) and (10.21) only apply to continuously heated buildings. Most buildings are, however, intermittently occupied and are not heated continuously. When buildings are intermittently occupied it is necessary to allow for the additional heat energy to bring the building structure up to temperature. The amount of preheating required depends on the thermal capacity of the building.

Heavy structures require long preheat periods and, once heated, retain heat well, while lightweight structures tend to heat up and cool quickly. It is therefore impossible to fully consider the impact of intermittent occupation on energy consumption without considering the thermal capacity of the building. The CIBSE classification of structures by thermal inertia is presented in Table 10.13. To allow for intermittent heating when using the degree day method it is necessary to introduce correction factors for:

- The length of the working week.
- The length of the working day.
- The response of the building and plant.

Tables 10.14, 10.15 and 10.16 set out values for these correction factors. Example 10.4 illustrates how the annual heating costs for a building can be calculated.

Example 10.4

A three-storey office building, with a total floor area of 2400 m^3, is occupied for 5 days per week and for 8 hours per day. The design day plant output (i.e. heat loss) is calculated to be 190.0 kW when the external temperature is −3.0°C and the dry resultant temperature is 21°C. The building is heated by a series of gas-fired modular boilers connected to a responsive warm air heating system with a seasonal efficiency of 70%.

Given that the building is located in a region which experiences 2354 degree days per year and that the cost of natural gas is 1.5p/kWh, determine the annual heating fuel cost:

(i) Ignoring any internal heat gains (i.e. assuming a base temperature of 15.5°C).
(ii) Allowing for an internal heat gain (from lights and equipment) of 20 W/m^2.

TABLE 10.14 Correction factor for length of working week [13]

Occupied days per week	Lightweight building	Heavyweight building
7 days	1.0	1.0
5 days	0.75	0.85

TABLE 10.15 Correction factor for length of working day, applies to intermittent use only [13]

Occupied period	Lightweight building	Heavyweight building
4 hours	0.68	0.96
8 hours	1.00	1.00
12 hours	1.25	1.02
16 hours	1.40	1.03

TABLE 10.16 Correction factor for the response of building and plant [13]

Type of heating	Lightweight	Medium weight	Heavyweight
Continuous	1.0	1.0	1.0
Intermittent – responsive plant	0.55	0.70	0.85
Intermittent – plant with a long time lag	0.70	0.85	0.95

Solution

(i) *Ignoring any internal heat gains*: From Table 10.14, the correction factor for length of working week is 0.85. From Table 10.15, the correction factor for length of working day is 1.00. From Table 10.16, the correction factor for response of building and plant is 0.85.

Therefore:

$$\text{Annual heating energy consumption} = \frac{190}{(21-(-3))} \times 2354 \times 24 \times \frac{(0.85 \times 1.00 \times 0.85)}{0.7}$$
$$= 461{,}636.21\,\text{kWh}$$

and

$$\text{Annual energy cost} = \frac{461{,}636.21 \times 1.5}{100} = £6924.54$$

(ii) *Allowing for an internal heat gain of* $20\,\text{W/m}^2$:

$$\text{Total heat gain} = \frac{2400 \times 20}{1000} = 48.0\ \text{kW}$$

Using eqn (10.22), the temperature rise due to heat gains is:

$$d = \frac{48}{190} \times (21 - (-3)) = 6.06°\text{C}$$

Therefore, the new base temperature is:

$$t_b = 21 - 6.06 = 14.94°\text{C}$$

From Table 10.12 this corresponds to a $D_d/D_{15.5}$ value of 0.932. Therefore,

$$\begin{aligned}\text{Annual heating energy consumption (allowing for heat gains)} &= 461{,}636.21 \times 0.932 \\ &= 430{,}244.95 \text{ kWh}\end{aligned}$$

Therefore,

$$\text{Annual energy cost} = \frac{430{,}244.95 \times 1.5}{100} = £6453.67$$

As well as predicting heating energy costs, it is also possible to use the degree day method to evaluate proposed energy-saving measures. Example 10.5 illustrates how this can be achieved.

Example 10.5

An office building has a roof which has a U value of 1.1 W/m^2 K. It is proposed that additional insulation be installed in the roof to bring its U value down to 0.25 W/m^2 K.

The office building is located in a region which experiences an annual total of 2350 degree days. Assuming that the efficiency of the building heating system is 70%, the cost of fuel is 1.5p/kWh and the capital cost of installing the roof insulation is £2.00 per m^2, determine the payback on the investment.

Solution

$$\begin{aligned}\text{Annual energy saving} &= (1.1 - 0.25) \times \frac{2350}{100} \times \frac{24}{0.7} \\ &= 68.486 \text{ kWh/m}^2\end{aligned}$$

Therefore,

$$\text{Annual energy cost saving} = \frac{68.486 \times 1.5}{100} = £1.03 \text{ per m}^2$$

$$\text{Payback period} = \frac{2.00}{1.03} = 1.94 \text{ years}$$

10.5 Intermittent Heating

The degree day calculations in Section 10.4 indicate that the intermittent use of heating plant results in higher energy consumption compared with continuous heating. This is primarily because intermittently occupied buildings, such as office buildings, require the structure to be warmed up after it has been allowed to cool overnight and at weekends. Additional heat energy is therefore required to 'preheat' the building in the mornings, so that its fabric is brought up to a temperature which will be comfortable for the occupants. Buildings with a high thermal mass require greater preheating than buildings with a low thermal mass. However, once heated to the required temperature, heavyweight structures retain their heat for much longer than lightweight ones.

The preheating of a building structure is achieved by running the heating system at full capacity for a preheat period prior to the building's occupation. The heavier the building structure, the longer the preheat period. The preheat period can be reduced in length by oversizing the boiler plant and, to some extent, the heat emitters. It is generally considered more energy efficient to increase the plant margin (i.e. oversize the boiler plant) in order to reduce the preheat period [10]. Table 10.17 gives recommended plant ratios for intermittent heating [10], which should be used in conjunction with eqn (10.24) to determine the intermittent peak heating load. It should be noted that the preheat times in the table assume the use of plant with a short response time, such as a warm air heating system. For slow response heating systems such as underfloor heating the preheating period will be longer:

$$Q_{pb} = F_3 \times Q_p \tag{10.24}$$

where Q_{pb} is the intermittent peak heating output (W or kW), and F_3 is the plant ratio (i.e. maximum heat output/design day heat output).

When oversizing plant, it is important to consider both boilers and heat emitters. While increased boiler capacity may reduce the preheat time, it can result in poor part-load performance and low seasonal boiler efficiency. It should be remembered that for most of the heating season, the external air temperature will be well above the winter design condition, so for much of the year the boilers will have plenty of excess capacity. It is therefore wise to consider the use of modular boilers.

While increased boiler capacity may be advisable, it is not always necessary to increase individual heat emitters. This is because the following alternative strategies can be employed:

- In buildings which are unoccupied, natural and mechanical ventilation rates can be reduced during the night-time. Reduced ventilation rates will occur naturally during the night-time because doors and windows usually remain closed. With

TABLE 10.17 Recommended plant ratios for intermittent heating [10]

Plant ratio (F_3)	Lightweight building preheat time (hours)	Heavyweight building preheat time (hours)
1.0	Continuous	Continuous
1.2	6	Very long
1.5	3	7
2.0	1	4
2.5	0	2
3.0	0	1

mechanical ventilation systems it is also possible to reduce the ventilation load by fully recirculating the air (i.e. reducing the outside air component of the supply air to 0%) during the preheat period.

- It is possible to elevate the water supply temperature to the heat emitters during the preheat period.

Although a modest oversizing of heat emitters is recommended by the CIBSE, there is no strong economic case for considerable oversizing (i.e. in excess of 25%) of emitter surfaces [10].

10.6 Radiant Heat

The importance of radiant heat transfer in buildings is often misunderstood with the result that potential 'radiant' energy-saving measures are often ignored. It is therefore worth investigating some of the 'radiant' energy-saving techniques that exist.

10.6.1 Radiant Heating

Equation (10.4) shows that the comfort of building occupants is as dependent on the mean radiant temperature as it is on air temperature. This fact can be used to great advantage in applications where a building is poorly insulated and in which ventilation rates are high, such as in old factories or workshops. In such applications, it is often prohibitively expensive to heat up large volumes of air, which are then quickly lost to the outside. It is much better to use some form of radiant heating to warm up the occupants. By using a high temperature radiant heat source it is possible to create a heat balance which enables the occupants to feel comfortable whilst still maintaining the air and fabric at a low temperature.

Radiant heating is particularly well suited to applications in which occupancy is very intermittent and in which the occupants are located in relatively fixed positions. A church building is a classic example of such an application. Such a building is occupied for a relatively short period in every week. Because radiant heating systems react very quickly and warm the occupants rather than the air, they can achieve a good comfort level without any preheating of the building. For this and the other reasons mentioned, radiant heating systems are generally considered to incur lower capital costs and lower operating costs than other comparable systems [13].

In order to achieve high levels of radiant heat transfer it is necessary to have emitters which are at a high temperature, well above 100°C. For safety and comfort reasons, these heat-emitting panels must be placed at high level, well out of the reach of any of the occupants.

10.6.2 Low-Emissivity Glazing

Glazing is often viewed as a thermal 'weak link', because heat is easily conducted through glass from the inside to the outside. This perception is broadly, but not wholly, true. It is often forgotten that much of the heat which is lost through windows occurs because they are, in effect, large flat high-emissivity surfaces which are cold in relation to the other surfaces in a room. All the other surfaces in the room, especially heated surfaces such as radiators and warm bodies, emit long-wave radiation which is readily absorbed by the cool glass. In this way much heat is lost from buildings. It is possible to minimize this problem by installing low-emissivity glazing which reduces the absorption of long-wave radiation. The low-emissivity effect is achieved during the manufacturing process by applying a microscopically thin (i.e. 0.3–0.4 m thick) coating of tin oxide doped with fluorine atoms [14] to the cavity-facing surface of the inner pane of a double-glazed unit. This significantly reduces the radiative heat loss to the cavity and thus reduces the *U* value of a typical double-glazed unit from approximately 3 W/m^2 K to 1.8 W/m^2 K.

10.7 Underfloor and Wall Heating

It is possible to heat buildings efficiently by using systems which utilize very low water temperatures. Two effective low temperature systems which can be used are:

- Underfloor heating, which utilizes flow water temperatures in the range 35–50°C.
- Wall heating, which utilizes flow water temperatures in the range 30–40°C.

The low water temperatures involved in these systems enable alternative heat sources to be utilized, such as ground source heat pumps or even solar energy. Because both wall and underfloor heating systems involve large heated surfaces, the room mean radiant temperature is increased. This makes it possible to reduce the air temperature within the room space without altering the dry resultant temperature, enabling energy to be saved whilst still maintaining a comfortable environment.

The maximum permissible surface temperature governs the maximum water flow temperature which can be used in both wall and underfloor heating installations. In the case of underfloor heating, occupants feel uncomfortable if the floor surface temperature is above 29°C. For walls the maximum safe surface temperature is about 43°C, since the disassociation temperature for plaster is approximately 45°C.

Underfloor heating systems are best suited to tall spaces, where the use of a conventional warm air or radiator system may lead to stratification of the air (i.e. the warm air becoming trapped at the top of the room space). The use of underfloor heating overcomes this problem and ensures that the air is warmest at ground level, where the occupants are likely to be located.

Underfloor heating systems often consist of a continuous cross-linked polyethylene or polypropylene flexible pipe loop embedded in a floor screed on top of a structural floor [15]. The screed is usually isolated from the structural floor by rigid insulation slabs, which reduce the heat transfer through the slab and help to maintain the screed temperature. The nature of the screed used in underfloor heating systems is of particular importance, since the screed acts as a thermal resistance to the heat transfer from the pipe to the room space and also provides the system with thermal inertia. Screeds can be either cementitious in nature and approximately 75 mm thick or an anhydrite flowing screed which enables the thickness to be reduced to approximately 50 mm. The thermal storage capacity and inertia of a floor screed depend on a number of parameters such as the specific heat capacity, screed thickness, screed density, thermal conductivity, pipe spacing and the water flow and return temperatures. It is therefore difficult to predict exactly how any given floor will perform in practice. However, for a standard 65 mm thick concrete screed floor, with a flow water temperature of 60°C, it has been estimated that the floor screed will take approximately 3 hours to charge and 3 hours to discharge. At lower water temperatures the charging process is considerably extended. The long hysteresis effect of underfloor heating has the disadvantage of being slow to respond and makes it difficult to adjust to sudden changes in the internal environment. Underfloor heating systems are therefore best suited to applications in which occupancy is continuous or well defined and predictable.

Wall heating operates in a similar manner to underfloor heating, with the exception that the pipe coils are generally not located in a material which has a high thermal mass, such as a floor screed. In a typical wall heating system, cross-linked polyethylene pipes are located on the air cavity side of a dry lined plastered wall. The system can be used either with a wet plastered board, or alternatively the pipes can be mechanically bonded to plasterboard using notched battens to make a rigid unit which can then be fixed to the wall. A flexible insulation quilt, such as rock wool, should be placed between the pipes and the structural wall, so that conduction to the building structure is minimized. It is advisable where possible to install the wall heating on internal walls, so that heat losses are minimized.

10.8 Pipework Insulation

Considerable amounts of heat energy can be lost through uninsulated or poorly insulated pipework. It is therefore important to ensure that hot water and steam pipework is properly insulated. A range of insulating materials is available and these can be either inorganic, based on crystalline or amorphous silicon, aluminium or calcium, or organic, based on hydrocarbon polymers in the form of thermosetting/thermoplastic resins or rubbers [16]. They can be either flexible or rigid, both types being available in preformed pipe sections. Table 10.18 lists some of the common types of insulation along with some of their thermal properties.

10.8.1 Pipework Heat Loss

In Section 10.3.1 heat loss through flat surfaces such as walls and roofs is discussed. The geometry of pipework is different from that of flat surfaces, requiring an alternative approach when calculating the heat loss from sections of pipework. The heat transfer through the wall of a pipe can be calculated using eqn (10.25):

$$Q = \frac{2\pi\lambda(t_1 - t_2)}{\ln(r_2/r_1)} \text{ (W per metre length)} \tag{10.25}$$

where λ is the thermal conductivity of pipe wall (W/m K), r_1 is the internal radius of pipe (m), and r_2 is the external radius of pipe (m).

TABLE 10.18 Thermal conductivities of insulating materials [16]

		Thermal conductivity (W/m K)		
Material	**Density (kg/m³)**	**50°C**	**100°C**	**300°C**
Calcium silicate	210	0.055	0.058	0.083
Expanded nitrile rubber	65–90	0.039	–	–
Mineral wool (glass)	16	0.047	0.065	–
Mineral wool (rock)	100	0.037	0.043	0.088
Magnesia	190	0.055	0.058	0.082
Polyisocyanurate foam	50	0.023	0.026	–

The thermal resistance of the pipe wall (per unit length of pipe) can be determined by:

$$R = \frac{\ln(r_2/r_1)}{2\pi\lambda} \text{ (m K/W)} \tag{10.26}$$

As hot fluid flows along a pipe, heat is transferred to the pipe wall. The rate at which this heat is transferred depends on the thermal resistance of a thin stationary layer of fluid on the pipe wall surface. The heat-transfer rate across this internal surface boundary layer can be expressed as:

$$Q = h \times A \times \Delta t \tag{10.27}$$

where h is the surface heat-transfer coefficient (W/m^2 K), A is the surface area (m^2), and Δt is the temperature difference between the surface and the bulk fluid (°C).

Equation (10.27) can also be applied to the heat transfer across the external surface of a pipe. Consequently, the internal and external surface resistance per unit length of a pipe can be expressed as:

$$R_{\text{so or si}} = \frac{1}{h \times A} \text{ (m K/W)} \tag{10.28}$$

The overall resistance per unit length of a typical insulated pipe can therefore be represented by:

$$R_t = R_{si} + R_w + R_{ins} + R_{so} \tag{10.29}$$

where R_t is the total thermal resistance of pipework per unit length (m K/W), R_w is the thermal resistance of pipe wall per unit length (m K/W), R_{ins} is the thermal resistance of insulation per unit length (m K/W), and R_{si} and R_{so} are the internal and external surface thermal resistances of insulation per unit length (m K/W).

Once the overall resistance of the pipework is determined, the total heat loss per metre run can be calculated by dividing the temperature difference between the fluid and ambient air by the total resistance:

$$Q = \frac{\Delta t}{R_t} \text{ (W/m)} \tag{10.30}$$

Example 10.6

A pipe carries wet steam at 200°C through a building which has an ambient air temperature of 20°C. The pipe has an internal diameter of 53.5 mm, a wall thickness of 3.7 mm and is insulated to a thickness of 25 mm. The thermal conductivity of the pipe material is 46 W/m K and the thermal conductivity of the insulating material is 0.033 W/m K. Assuming that the inside and outside surface heat-transfer coefficients are 10,000 W/m^2 K and 10 W/m^2 K respectively, determine:

(i) The heat transfer per metre length of pipe.
(ii) The temperature of the outside surface of the insulation.
(iii) The heat loss from an uninsulated pipe, assuming that the external heat-transfer coefficient remains unchanged.

Solution

(i) *The total pipework resistance per metre length is*:

$$R_t = R_{si} + R_w + R_{ins} + R_{so}$$

and

$$R_{si} = \frac{1}{10,000 \times \pi \times 0.0535} = 0.00059 \text{ m K/W}$$

$$R_{so} = \frac{1}{10 \times \pi \times 0.1109} = 0.287 \text{ m K/W}$$

$$R_w = \frac{\ln(30.45/26.75)}{2 \times \pi \times 46} = 0.00045 \text{ m K/W}$$

$$R_{ins} = \frac{\ln(55.45/30.45)}{2 \times \pi \times 0.033} = 2.891 \text{ m K/W}$$

From the above calculation it can be seen that the resistance of the pipe wall is negligible compared with that of the insulation and the outside surface resistance. In addition the heat-transfer coefficient for the internal surface is very high and hence the internal surface resistance is negligible. Therefore, the total thermal resistance can be assumed to be:

$$\begin{aligned} R_t &= R_{ins} + R_{so} \\ &= 3.178 \text{ m K/W} \end{aligned}$$

Therefore,

$$Q = \frac{(200 - 20)}{3.178} = 56.64 \text{ W/m}$$

(ii) *The surface temperature can be found by applying the following equation*:

$$\begin{aligned} \text{Temperature of the outside of the insulation} &= t_a + (R_{so} \times Q) \\ &= 20 + (0.287 \times 56.64) = 36.3^\circ\text{C} \end{aligned}$$

where t_a is the room air temperature (°C).

(iii) *For the uninsulated pipe*:

$$\begin{aligned} R_t &= R_{si} + R_w + R_{so} \\ &= 0.00059 + 0.00045 + 0.287 = 0.288 \text{ m K/W} \end{aligned}$$

Therefore,

$$Q = \frac{(200 - 20)}{0.288} = 625 \text{ W/m}$$

As well as illustrating the mechanics of a pipework heat loss calculation, Example 10.6 illustrates the great benefit to be derived from insulation, since the 25 mm insulation layer reduced the heat loss from 625 to 56.6 W/m.

10.8.2 Economics of Pipework Insulation

It is well known that one of the simplest and most cost-effective ways of preventing energy wastage is to insulate pipework runs. Nevertheless it is not easy to determine to what extent pipework should be insulated. The capital cost of insulation increases with its thickness and the financial saving must be offset against the capital cost. The economic thickness of insulation should therefore be governed by the payback period that is required on the investment, as illustrated in Example 10.7.

Example 10.7

A steel pipe carries high pressure hot water at 120°C around a factory building. The owners of the factory propose to insulate the pipe using rock wool. Given the data below, determine the optimum thickness of the insulation and the simple payback period for that thickness.

Data:

Pipe outside diameter = 76.6 mm
Heat-transfer coefficient for outside surface insulation = 10 W/m^2 K
Thermal conductivity of insulation = 0.037 W/m K
Water temperature = 120°C
Temperature of air in factory = 15°C
Boiler efficiency = 70%
Unit price of gas = 1.52 p/kWh
Boiler operates for 2500 hours per annum

Insulation costs							
Thickness of insulation (mm)	[20]	[25]	[32]	[38]	[50]	[60]	[75]
Cost per metre length (£/m)	5.00	5.58	6.64	8.01	10.57	13.44	16.68

Assume that the write-off period for the insulation is 5 years.

Solution

Assuming that the thermal resistances of the pipe wall and of the inside surface of the pipe are both negligible, let

$$x = \text{thickness of insulation (mm)}$$

Now

$$R_{\text{ins}} = \frac{\ln(r_2/r_1)}{2\pi\lambda} = \frac{\ln((38.3 + x)/38.3)}{2 \times \pi \times 0.037}$$

and

$$R_{\text{so}} = \frac{1}{h \times A} = \frac{1}{10 \times \pi \times ((76.6 + 2x) \times 10^{-3})}$$

$$R_{\text{t}} = R_{\text{ins}} + R_{\text{so}}$$

and

$$Q = \frac{\Delta t}{R_{\text{t}}}$$

therefore

$$Q = \frac{(120 - 15)}{R_{\text{ins}} + R_{\text{so}}}$$

and

$$\text{Annual operating cost} = \frac{Q \times 2500 \times 1.52}{0.7 \times 100}$$

and

$$\text{Total annual cost} = \text{annual fuel cost} + \frac{\text{capital cost}}{\text{write-off period}}$$

The following table can be produced from the above equations:

Insulation thickness (mm)	Insulation resistance (m K/W)	External resistance (m K/W)	Heat loss (W/m)	Annual fuel cost (£/m)	Total annual cost (£/m)
0	0.00	0.42	252.68	13.72	13.72
20	1.81	0.27	50.47	2.74	3.74
25	2.16	0.25	43.52	2.36	3.48
32	2.61	0.23	36.99	2.01	3.34
38	2.96	0.21	33.09	1.80	3.40
50	3.59	0.18	27.83	1.51	3.62
60	4.05	0.16	24.90	1.35	4.04
75	4.67	0.14	21.85	1.19	4.52

It can be seen that the most economic thickness of insulation is 32 mm, since this has the lowest annual cost.

The payback period for the 32 mm insulation is:

$$\text{Payback period} = \frac{6.62}{(13.72 - 3.34)} = 0.638 \text{ years}$$

10.9 Boilers

Most heating systems, although not all, employ boilers to produce hot water or steam. Boiler efficiency therefore has an important influence on heating-related energy costs. The cost savings that can be achieved by improving overall boiler efficiency can be substantial. Essentially a boiler is a device in which a fossil fuel is burnt and the heat produced is transferred to water. The more effective this heat-transfer process, the more efficient the boiler. It is therefore important to maximize the heat transfer to the water and minimize boiler heat losses. Heat can be lost from boilers by a variety of methods, including flue gas losses, radiation losses and, in the case of steam boilers, blow-down losses. Although all these various losses have a significant effect on boiler energy consumption, the major reason for poor boiler performance occurs at the design stage, where the capacity of boilers is usually oversized and inappropriate boilers are often selected.

Boiler plant which is oversized will operate at part-load for most of the time, resulting in low seasonal efficiency and high operating costs. It has been estimated that a 15% increase in energy consumption can occur if a conventional boiler plant is oversized by 150% [17]. Boiler plant should be considered oversized if under the winter design condition the boilers are able to maintain an internal air temperature well above the design temperature (e.g. 21°C). Evidence of oversizing can be manifested in a number of ways: fuel bills may be higher than expected; excessive cycling of boiler plant may be experienced; and in installations equipped with modular boilers, a large proportion of the boilers may never be used.

Given that for much of the heating season external air temperatures are usually well above the winter design condition, it is important that any boiler installation be designed so that it operates efficiently at part-load. For most types of conventional boilers, efficiency falls dramatically below about 30% of rated capacity [18]. Large boilers are therefore at a disadvantage since for most of the time they will be operating well below their rated capacity. One simple way of overcoming this problem is to install a large number of small modular boilers with a sequence controller in preference to a few large boilers. This ensures that under part-load conditions, boilers which are always operating near their maximum efficiency will provide the bulk of the heating. With such a multi-boiler plant installation, it is wise to install a boiler sequence control system. This is a fully automatic microprocessor-controlled system which monitors and sequences on/off operations of boiler plant according to the demand for heat. This avoids running too many boilers on part-load and minimizes the number of boilers in operation at any one time.

Another technique which can be employed to ensure good part-load efficiencies is to use boilers with modulating burners. These burners modulate the fuel and air to provide a variable output from 20%–30% to 100%. With large modulating boilers it is possible to make substantial energy savings by installing variable speed drives on the combustion air fans. Variable speed drives reproduce the operating characteristics of fixed speed combustion air fans and adjustable dampers, whilst reducing the average electrical demand of the fan motor by approximately 60% [19].

10.9.1 Flue Gas Losses

All boilers require a minimum amount of air to ensure that complete combustion of the fuel takes place and that no carbon monoxide is produced. Yet, if too much air is added then heat is wasted in warming up the excess air, which then escapes through the flue. The amount of combustion air should therefore be limited to that necessary to ensure complete combustion of the fuel. In practice some excess air, around 15–25% for oil-fired boilers [19] and 15–30% for gas-fired boilers [20], is needed. The actual amount required to give the optimum boiler efficiency depends on the fuel used and the type of boiler and burner. If the air flow rate to a boiler is too low, then a proportion of the fuel will remain unburnt and running costs will increase. In the case of oil-fired plant, incomplete combustion will produce smoke which will be visible. For coal-fired plant, incomplete combustion results in unburnt carbon in the ash. It is therefore essential to maintain the correct fuel-to-air ratio at all times. With modern microprocessor-controlled burners, which are fitted to fuel valves and air dampers, it is possible to automatically select and maintain specific fuel-to-air ratios for a variety of fuels. These controllers continually monitor the level of oxygen in the flue gases, and alter the combustion air supply in order to maintain optimum conditions.

Flue gas losses are by far the greatest heat losses which occur from boilers. The flue gases contain considerable sensible heat and also latent heat which is 'bound up'

TABLE 10.19 Typical CO_2 and O_2 contents by volume expected in flue gas (dry basis) [18]

Fuel	Minimum fire		Full fire		
	CO_2 (%)	O_2 (%)	CO_2 (%)	O_2 (%)	CO (ppm)
Coal	11.0	8.5	14.0	5.0	2–500
Fuel oils	11.5	5.5	13.5	3.0	–
Butane	9.4	7.0	12.0	3.0	2–400
Propane	9.2	7.0	12.0	3.0	2–400
Natural gas	8.0	7.0	10.0	3.3	2–400

in water vapour. It is possible to determine the amount of heat which is being lost through the flue, by monitoring the presence of O_2 or CO_2 in the flue gases. If there is little excess air in the combustion gases, then the percentage of CO_2 will be high and the percentage of O_2 low. Conversely, if a large amount of excess air is present, the relationship will be reversed. In a typical gas-fired shell and tube boiler the flue gases should contain about 9–10% CO_2 and 3–5% O_2 [20], while for an oil-fired boiler the CO_2 content of the flue gases should be in the region of 13–14% [19]. Typical CO_2 and O_2 flue gas contents for efficient boiler operation are presented in Table 10.19.

With large boiler plant it is possible to increase boiler efficiency by preheating the combustion air. It has been estimated that the thermal efficiency of a boiler can be increased by approximately 1% if the temperature of the combustion air is raised by 20°C [19,20]. Any one of the following sources of heat can be utilized to preheat the boiler combustion air:

- Waste heat from the flue gases.
- Drawing high temperature air from the top of the boiler room.
- Recovering waste heat by drawing air over or through the boiler casing.

10.9.2 Other Heat Losses

With shell and tube boilers it is possible for the 'smoke' tubes to become fouled by soot and other deposits, resulting in a reduction in the amount of heat which is transferred from the hot flue gases to the water. This increases the temperature of the flue gases and results in greater flue gas losses. Boiler smoke tubes should therefore be cleaned regularly to minimize the flue gas temperature rise, since it has been estimated that a rise of 17°C in the flue gas temperature causes a decrease in efficiency of approximately 1% [19]. Boiler efficiency can also significantly be reduced by a build-up of scale on the water side of the smoke tubes. Water treatment should therefore be employed in order to prevent scale formation.

Heat can be lost through the surface casing of boilers. This is generally referred to as *radiation* loss, although it includes heat which is lost by convection. With modern boilers radiation losses are usually not greater than 1% of the maximum rating. On older boilers this figure may be as high as 10% where the insulation is in poor condition [19,20].

10.9.3 Boiler Blow-Down

With steam boilers it is necessary to eject a small proportion of the water regularly in order to remove sludge and to maintain acceptable levels of total dissolved solids. This process is called *blow-down* and it prevents scaling up of the tubes on the water side. Although necessary, blow-down represents a considerable energy loss and the level of blow-down should be kept to a minimum while still maintaining the recommended level of dissolved solids.

It is possible to recover waste heat from the blow-down process by collecting the flash steam which forms as the pressure falls through the blow-down valve. Because the condensate produced by the blow-down process is both hot and pure, with no dissolved solids, it can be added directly to the make-up water for the boiler, thus reducing energy consumption.

10.9.4 Condensing Boilers

Boiler flue gases are often in excess of 200°C and as such are a useful source of waste heat recovery. Heat exchangers can be installed in flues to recover both sensible and latent heat from the hot products of combustion. However, because of the corrosion problems associated with sulphur bearing fuels, such as fuel oil, flue gas heat recovery is generally only practised on gas-fired boilers. Gas-fired boilers which incorporate integral flue gas heat exchangers are known as *condensing boilers*. If used correctly, condensing boilers can achieve operating efficiencies in excess of 90% [18,21].

With a condensing boiler it is desirable to operate the system so that the return water temperature is as low as possible. This ensures that condensation of the flue gases occurs and that maximum heat recovery from flue gases is achieved. If a condensing boiler is used in conjunction with a weather compensating controller, the boiler will move into condensing mode during the milder part of the season, when return water temperatures are at their lowest. In this way, high efficiency is maintained under part-load conditions. In multiple boiler installations it is usually cost-effective to install only one condensing boiler. This should always be the lead boiler, since it exhibits the highest efficiencies. This ensures that good energy utilization will occur under part-load conditions.

References

[1] Keating WR. The Eurowinter Group. Cold exposure and winter mortality from ischaemic heart disease, cerebrovascular disease, respiratory disease, and all causes in warm and cold regions of Europe. Lancet 1997;349:1341–46.

[2] Forecasting the nation's health. Met Office Report, November; 2000.

[3] Wilkinson P. Social and environmental determinants of excess winter deaths in England, 1986–1996. Report for the Joseph Rowntree Foundation, July; 2000.

[4] Edholm OG. Man – hot and cold (Studies in Biology No. 97) (Chapter 1). London: Arnold; 1978.

[5] CIBSE Guide A1, Environmental criteria for design; 1986.

[6] Smith BJ, Phillips GM, Sweeney M. Environmental science (Chapter 2). Harlow: Longman Scientific and Technical; 1987.

[7] CIBSE Guide A2, Weather and solar data; 1982.

[8] CIBSE Guide A3, Thermal properties of building structures; 1980.
[9] Smith BJ, Phillips GM, Sweeney M. Environmental science (Chapter 1). Harlow: Longman Scientific and Technical; 1987.
[10] CIBSE Guide A9, Estimation of plant capacity; 1983.
[11] Moss KJ. Energy management and operating costs in buildings (Chapter 1). London: E & FN Spon; 1997.
[12] CIBSE Guide B18, Owning and operating costs; 1986.
[13] Economic use of fired space heaters for industry and commerce, Fuel Efficiency Booklet 3. Department of the Environment; 1993.
[14] Enhancing the built environment. Pilkington Plc. website.
[15] CIBSE Guide B1, Heating; 1986.
[16] The economic thickness of insulation for hot pipes, Fuel Efficiency Booklet 8. Department of the Environment; 1994
[17] Hendrick R, Witte MJ, Leslie NP, Bassett WW. Furnace sizing criteria for energy-efficient set back strategies. ASHRAE Trans 1992;98:1239–46.
[18] Energy audits and surveys. CIBSE Applications Manual AM5; 1991.
[19] Economic use of oil-fired boiler plant, Fuel Efficiency Booklet 14. Department of the Environment; 1993.
[20] Economic use of gas-fired boiler plant, Fuel Efficiency Booklet 15. Department of the Environment; 1994.
[21] Condensing boilers. CIBSE Applications Manual AM3; 1989.

Bibliography

CIBSE Guide A1, Environmental criteria for design; 1986.
CIBSE Guide A2, Weather and solar data; 1982.
CIBSE Guide A3, Thermal properties of building structures; 1980.
CIBSE Guide A9, Estimation of plant capacity; 1983.
CIBSE Guide B1, Heating; 1986.
CIBSE Guide B18, Owning and operating costs; 1986.
Degree days, Fuel Efficiency Booklet 7. Department of the Environment; 1995.
Eastop TD, Croft DR. Energy efficiency for engineers and technologists (Chapters 4 and 5). Harlow: Longman Scientific & Technical; 1990.
Eastop TD, Watson WE. Mechanical services for buildings (Chapter 3). Harlow: Longman Scientific & Technical; 1992.
Edholm OG. Man – hot and cold (Studies in Biology No. 97) (Chapter 1). London: Arnold; 1978.
Economic use of fired space heaters for industry and commerce, Fuel Efficiency Booklet 3. Department of the Environment; 1993.
Economic use of oil-fired boiler plant, Fuel Efficiency Booklet 14. Department of the Environment; 1993.
Economic use of gas-fired boiler plant, Fuel Efficiency Booklet 15. Department of the Environment; 1994.
Levermore GJ. Building energy management systems (Chapter 10). London: E & FN Spon; 1992.
Moss KJ. Energy management and operating costs in buildings (Chapters 1 and 3). London: E & FN Spon; 1997.
Stoecker WF, Jones JW. Refrigeration and air conditioning (Chapters 2 and 4): McGraw-Hill International Editions; 1982.
The economic thickness of insulation for hot pipes, Fuel Efficiency Booklet 8. Department of the Environment; 1994.