12
Low-energy air conditioning and lighting control

12.1 Displacement ventilation

Originally used for industrial buildings for ventilation and to remove air-borne contaminants from the occupied zone by natural convection, displacement ventilation is now applied to office buildings. Compared to conventional air conditioning, it reduces the concentration of polluting particles and gases generated by structural materials, furnishings and the occupants themselves. Conventional air conditioning relies on mixing the room air with the cold supply air, typically from ceiling diffusers (giving between 5 and 15 air changes per hour). Displacement ventilation uses about 2.5 to 3 air changes per hour [1] and is designed to minimize mixing in the occupied zone. Arguably this lower mixing can reduce sick building syndrome [2], but local discomfort due to draft and the vertical temperature gradient may be critical [3].

Displacement ventilation is achieved by the supply of conditioned air, with a temperature slightly lower (1 K to 3 K) than the desired room air temperature in the occupied zone. The occupied zone is often taken as up to 1.8 m above the floor level [4], but some displacement manufacturers consider it as up to 1.1 m or 1.2 m above the floor [5]. The supply outlets are at or near floor level and supply air at low velocities (typically $\leqslant 0.5$ m s^{-1}, compared to 2–6 m s^{-1} for conventional mixed-flow systems) to form a shallow layer of cool, clean air with an average speed in the occupied zone of less than 0.1 m s^{-1} and a turbulence factor of less than 5%. The turbulence factor (%) is

$$\mathrm{Tu} = s/v$$

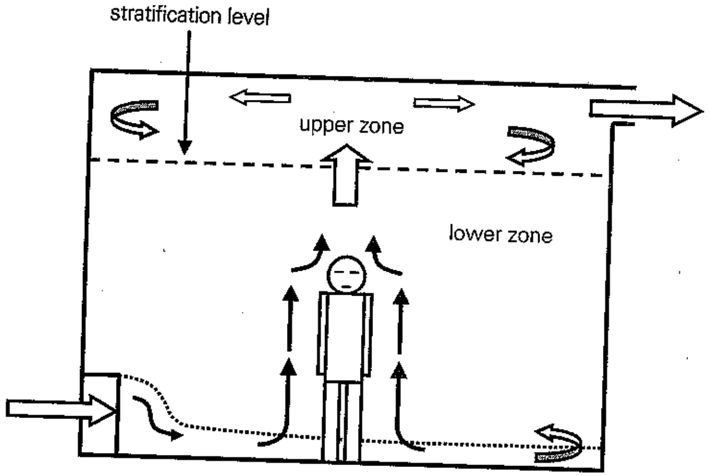

Fig. 12.1 Displacement ventilation.

where v is the average measured velocity of the air (m s^{-1}) at the point of measurement, 1.2 m above the floor, and s is the standard deviation of the air speed. The comparable values for standard mixed-flow air conditioning are v = 0.15–0.20 m s^{-1} and Tu = 25–35%.

The sea of cool supply air rises in **plumes** of warm air from heat sources, e.g. people and machines (Fig. 12.1).

12.1.1 Zones

The plume around a person achieves a velocity of 0.25 m s^{-1} at head height [6] and entrains a small amount of surrounding air to produce a plume of expanding warm air rising out of the occupied zone. This is then extracted at high level from the **upper zone** above the **stratification level** or boundary. The upper zone is characterized by recirculation of air into the plume, whereas in the **lower zone** there is entrainment of surrounding air into the rising plume. One could also consider a separate **pool zone** just above the floor, where the pool of cool air gathers before being entrained. To understand fully the air movement in a room, one would have to model it with **computational fluid dynamics** (CFD). However, the concept of zones does emphasize the stable, vertically stratified layers that are essential for the displacement ventilation to function effectively.

It is reported that an opened door can upset the plumes [7], and one could imagine other disturbances such as people's movement in the room also having an influence on the plumes. The control of displacement ventilation is to ensure that the pool zone does not occupy too much of the occupied zone or that the upper zone does not penetrate the occupied zone.

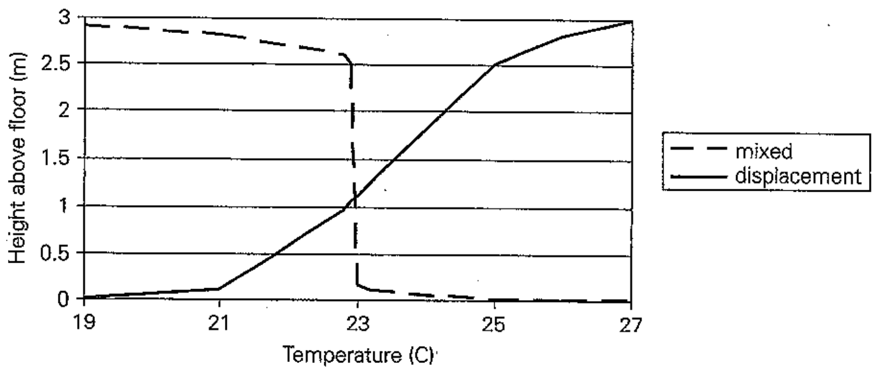

Fig. 12.2 Temperature gradients for displacement and mixed ventilation.

With displacement ventilation and its different zones it is not surprising that the vertical temperature gradient is markedly different from a conventional mixed-flow temperature profile (Fig. 12.2).

12.1.2 Mixing compared to displacement and temperature gradients

For mixed-flow ventilation the room air temperature is fairly constant throughout the room (23°C in Fig. 12.2), but for displacement ventilation the room temperature changes with height, so a design room temperature has to be defined with reference to height. Often this is the temperature at 1.1 m from the floor, $t_{1.1}$ (typically neck to head height for a seated person). Some designers use $t_{1.2}$ as the design room temperature. These heights are also sometimes used to define the height above floor level of the occupancy zone.

For comfort it is suggested [8] that the vertical air temperature gradient in the occupied zone should be less than or equal to 3 K m^{-1}, which corresponds to an air temperature difference of 3 K between the 1.1 m reference height and the ankle (about 0.1 m from the floor). Other sources recommend a lower value of 2 K m^{-1} (in Germany and Switzerland, for instance [5]), although 3 K m^{-1} has recently been confirmed as satisfactory for displacement ventilation [9]. It is also recommended that there are minimum values of $t_{0.1}$ between 19°C and 21°C in winter and between 21°C and 22°C in summer [5].

Besides air temperature, air movement is important for comfort, and the velocity at people's ankles should not be more than 0.15 m s^{-1}. Near the displacement diffusers there is an **adjacent zone** (defined as the length and width of the envelope at ankle level containing air moving at greater than 0.2 m s^{-1}) where discomfort will probably be experienced. This means that occupants' working positions should be away from floor-standing diffusers near walls or away from diffusers in the floor itself. The recommended upper limit on the air thermal gradient has implications for the design and control of displacement ventilation as Example 12.1 shows.

Example 12.1

Consider a room 3 m high. What is the likely sensible heat extraction by displacement ventilation? How does this compare with conventional mixed-flow air conditioning?

Solution

Assuming the maximum temperature gradient of $3\ \mathrm{K\,m^{-1}}$ throughout the height of the room, then for a room of height 3 m this produces a temperature difference of 9 K between supply and extract (assuming the extract is at the top of the room). The maximum air change rate [1] is between $2.5\ \mathrm{h^{-1}}$ and $3\ \mathrm{h^{-1}}$. So for 3 air changes per hour the air supply rate for $1\ \mathrm{m^2}$ of floor area is $9\ \mathrm{m^3\,h^{-1}}$ ($0.0025\ \mathrm{m^3\,s^{-1}}$ or $2.5\ \mathrm{l\,s^{-1}}$). This gives a sensible cooling capacity of

$$(\dot{V}\rho)C_p\Delta t \qquad (12.1)$$

where $\dot{V}$ = volume flow rate ($\mathrm{m^3\,s^{-1}}$)
ρ = density of air ($1.2\ \mathrm{kg\,m^{-3}}$)
C_p = specific heat capacity of the air ($1.02\ \mathrm{kJ\,kg^{-1}\,K^{-1}}$)
Δt = supply to extract air temperature difference (K)

So the cooling capacity is

$$0.0025 \times 1.2 \times 1020 \times 9 = 27.5\ \mathrm{W\,m^{-2}}$$

This figure is not unreasonable, as it has been suggested [10] that the maximum convective load which can be dealt with by displacement ventilation in offices is limited to $25\ \mathrm{W\,m^{-2}}$. If a floor-mounted induction unit is used, inducing room air 1 m above floor height, then the primary air can be reduced depending on the induction ratio. If an equal volume of room air is induced, then the primary air can be reduced to about 7 K below the room temperature at 1.1 m and the cooling capacity of the system raised to $50\ \mathrm{W\,m^{-2}}$.

For mixed-flow air conditioning there can be up to 15 air changes per hour. For 15 air changes per hour the air supply rate for $1\ \mathrm{m^2}$ of floor area is $45\ \mathrm{m^3\,h^{-1}}$ ($0.0125\ \mathrm{m^3\,h^{-1}}$ or $12.5\ \mathrm{l\,s^{-1}}$). With a typical temperature difference of 8 K between the supply air and the extract air (the same as the room temperature with good mixing), this gives a sensible cooling capacity of

$$(\dot{V}\rho)C_p\Delta t = 0.0125 \times 1.2 \times 1020 \times 8$$
$$= 122.4\ \mathrm{W\,m^{-2}}$$

This is a little high; a more practical range would be 90–100 $\mathrm{W\,m^{-2}}$ [6].

Table 12.1 Temperatures with a selection of displacement ventilation systems

	t_{supply}	$t_{0.1}$	$t_{1.2}$	$t_{extract}$
Non-induction	16	18	22.2	24.2
Induction	16	19.8	22.1	24.1
Floor outlet	16	19.5	22.5	24.1

12.2 Control of displacement ventilation

To control a displacement ventilation system the occupant temperature has to be identified; it is usually taken as the temperature 1 m, 1.1 m or 1.2 m (t_1, $t_{1.1}$, $t_{1.2}$) above the floor. The normal criterion is that for comfort the supply air temperature should not be more than 3 K between the head and the ankle. There is also a heuristic rule that the supply temperature should not go below 16°C for sedentary workers, although it has been suggested that it should be between 18°C and 20°C [11]. With some floor-mounted diffusers discharging horizontally along the floor, the supply air can pick up heat and raise its temperature by about 1 K. So a **floor temperature**, t_{floor}, or ankle temperature, $t_{0.1}$, can be used to differentiate it from the supply temperature. Table 12.1 gives some typical temperatures at different heights in a 3 m high room.

With the criterion of maintaining a supply temperature that will not cause discomfort, the simplest form of control is proportional-only or PI control of the supply temperature. This produces open-loop control with no feedback from the room or zone. In fact, one manufacturer recommends that a displacement ventilation system should be operated throughout the year with a supply air temperature between 18°C and 20°C, although it concedes that the system can be operated economically as a VAV system [5]. With the room temperature gradient in displacement systems, and the consequent variation of temperature with height, location of the room temperature sensor can be difficult, unlike a mixed conventional system where the extract temperature is the same as the room air temperature in the occupied zone. Feedback control of displacement ventilation is considered below.

Remember that a displacement ventilation system can only cope with a small load ($27\ \mathrm{W\,m^{-2}}$ in Example 12.1) and this will probably be a reasonably constant load. It could not cope with a large solar load. Also the fresh air load will be a large part of the supply air. With the recommended [11] fresh air supply per person of $8\ \mathrm{l\,s^{-1}}$ and assuming $10\ \mathrm{m^2}$ floor area per person, this equates to an air change rate of $0.96\ \mathrm{h^{-1}}$ for a 3 m high room. The maximum air change rate is likely to be up to $3\ \mathrm{h^{-1}}$, although $3.9\ \mathrm{h^{-1}}$ was used in an experimental room [9]. However, slight changes in occupant density could mean that most of the supply air is fresh air, so volume flow control would not be very flexible. Any air control is mostly to ensure

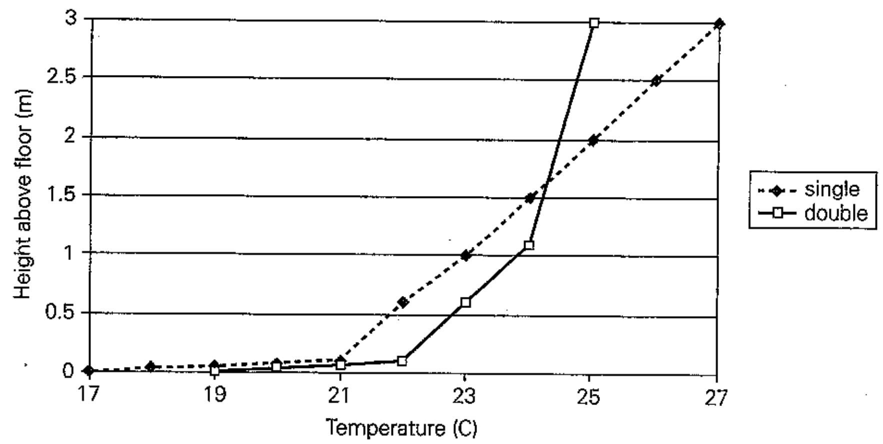

Fig. 12.3 Temperature gradients for displacement ventilation.

sufficient air is supplied and that the diffusers are balanced. Floor diffusers are commonly supplied by a floor plenum under the raised floor. In terms of network analysis (Chapter 13) this plenum has low resistance, so there is little pressure difference between the floor outlets and they are effectively balanced. For an extensive plenum, ducts may be used to help distribute the air within the plenum, with **regulators** at the ends of the ducts as they discharge into the plenum. The regulators are counterweighted dampers which can be set to the required flow rates.

Variable air volume (VAV) boxes can be used with displacement diffusers to control the volume flow (Chapter 10). These boxes can either maintain a constant flow to the room or zone at a constant supply temperature (the temperature control is at the central plant), or vary the volume flow from a room temperature signal. Volume flow variation requires care to maintain an adequate fresh air supply.

An alternative is to maintain a constant volume supply and to vary the supply temperature using the standard reset type of schedule. The reset signal can come from room air temperature sensors (with the usual problems of where to site them, especially in open-plan offices, and how many to have) or from an extract temperature sensor. To relate the extract temperature to the occupancy temperature, use an approximation of the temperature distribution through the room. This is a dog-leg shape (Fig. 12.3) [12, 13]. The diagram shows a double dog-leg and a single dog-leg, from the two references. In the double dog-leg the supply temperature is $t_s = 19°C$ and the extract temperature is $t_e = 25°C$; in the single dog-leg $t_s = 17°C$ and $t_e = 27°C$. Both dog-legs can be used in the design of displacement ventilation systems, using the **comfort method**. An alternative is the **air quality method** [11] using Skaret's equations for the airflow in a displacement plume [14].

The single dog-leg has been verified as a reasonable approximation in a study of three sites [12]. But it is a simplification and cold surfaces (e.g. external walls and windows) and infiltration or exfiltration can influence the dog-leg significantly [12]. In Fig. 12.3 the single dog-leg gradient changes at 0.1 m above the floor – ankle height. Empirically the temperature difference $(t_{0.1} - t_s)$ is between 0.3 and 0.5 times the temperature difference $(t_e - t_s)$ for a typical 3 m high room [11]. In Fig. 12.3 it is taken as the typical design value of 0.4:

$$t_{0.1} - t_s = 0.4(t_e - t_s) \tag{12.2}$$

The important temperature gradient $(t_{1.1} - t_{0.1})/(1.1 - 0.1)$ has the maximum value of $3\ \mathrm{K\,m^{-1}}$, but when using the comfort method, $2\ \mathrm{K\,m^{-1}}$ is often recommended for sedentary workers. Kruhne has determined approximate values for the ratio $(t_{1.1} - t_{0.1})/(t_e - t_s)$ for various heat source positions and room usages [15]. For a museum with lighting near the ceiling it is 0.16; for an office area it is nearer to 0.25. For an office with a high thermal load and a chilled ceiling it is closer to 0.4 [5].

Example 12.2

A displacement ventilation system for an office is designed to provide a cooling load of $25\ \mathrm{W\,m^{-2}}$ with an occupancy design air temperature of $t_{1.1} = 23°C$. The design temperature gradient $(t_{1.1} - t_{0.1})$ is $2\ \mathrm{K\,m^{-1}}$. If the cooling load reduces to $20\ \mathrm{W\,m^{-2}}$, to what value would the supply temperature have to be raised in order for $t_{1.1}$ to be maintained at 23°C?

Solution

The straight-line gradient from 0.1 m to the ceiling height of 3 m gives

$$2 = \frac{t_e - t_{0.1}}{2.9}\ \mathrm{K\,m^{-1}} \tag{12.3}$$

Putting the design conditions into equation (12.2):

$$t_{0.1} - t_s = 0.4(t_e - t_s)$$

Rearranging gives

$$t_{0.1} - 0.6t_s = 0.4t_e \tag{12.4}$$

and given

$$2 = t_{1.1} - t_{0.1}$$

but $t_{1.1} = 23°C$ so

$$2 = 23 - t_{0.1}$$

$$t_{0.1} = 21°C \tag{12.5}$$

From equations (12.3) and (12.5) we see that $t_e = 26.8°C$ and from equation (12.4) we see that $t_s = 17.1°C$. For the removal of 25 W m^{-2} we require

$$\dot{m}C_p(t_e - t_s) = 25$$

$$\dot{m}C_p = 2.58 \qquad (12.6)$$

When the heat gains are reduced to 20 W m^{-2} we have

$$t_e - t_s = 20/2.58$$
$$= 7.75 \text{ K}$$

To keep $t_{1.1}$ at 23°C and assuming the dog-leg relationship still holds, equation (12.4) gives

$$t_{0.1} - t_s = 0.4(7.75)$$
$$= 3.1 \text{ K} \qquad (12.7)$$

For the upper line

$$\frac{t_{1.1} - t_{0.1}}{1} = \frac{t_e - t_{1.1}}{1.9} \qquad (12.8)$$

Also

$$t_e - t_{0.1} = 0.6 \times 7.75$$
$$t_e = 4.65 + t_{0.1} \qquad (12.9)$$

Substituting equation (12.9) into equation (12.8):

$$23 - t_{0.1} = \{(4.65 - t_{0.1}) - 23\}/1.9$$
$$43.7 - 1.9t_{0.1} = t_{0.1} - 18.35$$
$$t_{0.1} = 21.4°C$$

And from equation (12.7) we have

$$t_s = 18.3°C \quad \text{and} \quad t_e = 26.05 \text{ K}$$

So the changes in internal gains can be controlled by changes in the supply air temperature of the displacement ventilation. The feedback control signal would come from sensing the return air temperature. The ankle-to-head gradient reduces so comfort can be maintained.

Example 12.2 can be generalized to derive a control equation relating t_s to t_e but keeping $t_{1.1}$ constant. The heat load (kW) to be absorbed by the displacement ventilation is

$$Q = \dot{m}C_p(t_e - t_s)$$

or

$$(t_e - t_s) = Q/\dot{m}C_p \qquad (12.10)$$

For a room 3 m high, the gradient of the lower line is

$$t_{0.1} - t_s = 0.4(Q/\dot{m}C_p) \qquad (12.11)$$

and the gradient of the upper line is

$$\frac{t_{1.1} - t_{0.1}}{1} = \frac{t_e - t_{1.1}}{1.9}$$

Rearranging gives

$$t_{0.1} = t_{1.1} - \left(\frac{t_e - t_{1.1}}{1.9}\right) \qquad (12.12)$$

From equations (12.10), (12.11) and (12.12) we can get t_s in terms of t_e and $t_{1.1}$:

$$t_e - t_s = \frac{1}{0.4}\left[t_{1.1} - \left(\frac{t_e - t_{1.1}}{1.9}\right) - t_s\right]$$

giving

$$t_s = 2.54t_{1.1} - 1.54t_e$$

To check this with Example 12.2, when $t_e = 26.8°C$ then t_s should be 17.1°C:

$$t_s = 2.54 \times 23 - 1.54 \times 26.8$$
$$= 17.1°C$$

The control of t_s is reset control from t_e, such that when $t_e = 26.8°C$ then $t_s = 17.1°C$ and the cooler coil is at a maximum, producing 25 W m^{-2}. The cooler coil output is zero when t_s is 23°C and t_e is also 23°C.

Although temperature control can be used with displacement ventilation, care has to be taken that it does not vary too much and cause discomfort. The supply temperature should generally be between 18°C and 20°C and never below 16°C for sedentary occupants. It has been shown that up to half the temperature rise between the supply and extract air can take place in the layer 0.1 m above the floor [16].

12.2.1 Energy saving

Compared to conventional air conditioning, supplying air at 13°C, displacement ventilation, supplying air at 17°C and upwards, will save on chiller energy. In fact, the free cooling by using cool fresh air will contribute a considerable amount to the cooling, especially in the spring and autumn, so the chiller is not required except during hot outside conditions.

Only summer ventilation and removal of heat loads have been considered so far. For heating, the displacement ventilation will act rather like convector heaters. With high ceilings it may be necessary to provide ceiling destratification fans to stop too much heat accumulating at the top of the room. However, Twinn considers that displacement ventilation is not suitable for winter heating and advocates a perimeter system [7].

Note that traditional air conditioning typically maintains the room at an air temperature of 21°C whereas displacement ventilation has a typical occupancy temperature of 23°C, so this also saves thermal energy, although it does reduce comfort. But the cooling differentials also have to be considered. For conventional cooling the differential is 21 − 13 = 8 K. For the displacement systems in Fig. 12.3 the differentials are 25 − 19 = 6 K and 27 − 17 = 10 K. For the same heat load, the displacement system differentials of 6 K and 10 K will respectively require higher and lower flow rates than the conventional system. Displacement systems cannot cope with more than about 25 W m^{-2} whereas a conventional system can cope with up to 100 W m^{-2}, so the displacement ventilation will use less energy.

12.3 Chilled ceilings

The displacement ventilation systems discussed above can absorb about 25 W m^{-2} of heat load. This demands stringent design of a building to eliminate solar and other gains for displacement ventilation alone to work satisfactorily. However, chilled ceilings are often employed with displacement ventilation systems to increase the cooling capacity. In this section the ceilings alone will be considered, and their use with displacement systems will be discussed in Section 12.4. As no fans are involved, chilled ceilings are often known as **static cooling**, along with chilled beams (cooling convectors often placed at ceiling level).

Air conditioning has traditionally used ceiling diffusers to supply the air, injecting the tempered air with a velocity of 2–6 m s^{-1} to create a good mixing zone. The tempered air then diffuses into the comfort zone of the occupants. But it is cheaper and quieter to use water to transport heat and coolth (Chapter 13). The thermal power transmitted to the motive power of transmission is

$$\frac{\Delta P}{\rho C_p(t_f - t_r)} \qquad (13.16)$$

For a typical water system transmitting 1 kW of thermal power, 7.14 × 10^{-3} W kW^{-1} m^{-1} was required to transmit it. For a typical air system it needed 0.1 W kW^{-1} m^{-1} about 14 times more transmission power.

This shows there is a power advantage in using cool water in chilled ceilings rather than cooled air. The early chilled ceiling schemes used pipes

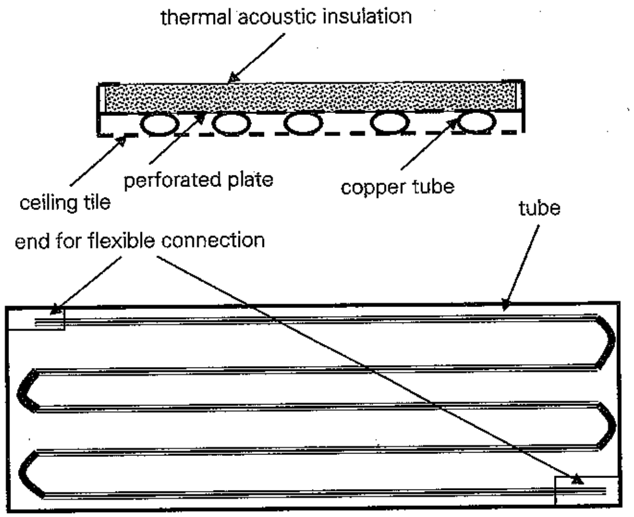

Fig. 12.4 A chilled ceiling section.

embedded in the actual floor or ceiling slab, hence they had a high capital cost. Modern systems use pipes clipped to perforated acoustic ceiling tiles made of aluminium or steel (Fig. 12.4), typically at 60 mm separation suspended from the concrete ceiling slab. The perforated tile encourages convection around the chilled pipes. A layer of thermal (and acoustic) insulation is placed above the tiles and pipes; this directs the heat transfer down into the room. These modern systems are cheaper than the embedded pipe systems and are proving more popular. However, good control of the flow water temperature is required to prevent condensation on the pipes and tiles, but there are fewer worries now about leaks and the flexibility of the modern systems. An auxiliary air supply system is essential to deliver cooled, dehumidified air to the occupied space. Often this is by a displacement ventilation system (Section 12.2). It is recommended that variations in sensible gain are dealt with by the ceiling, and the constant sensible heat gain and all the latent heat gains are dealt with by the auxiliary air supply system [17]. This implies control of the chilled flow water and/or its flow temperature.

For offices the chilled ceiling typically has a mean water temperature of approximately 17°C (16°C flow and 18°C return) and a room air temperature of 24–26°C. The flow can go down to 14°C to increase the output. With these temperatures the occupant perceives the room temperature (dry resultant temperature) as being 1 K to 1.5 K lower than the air temperature. Note that the average surface temperature of the panel depends on the spacing of its pipes and the insulation behind the pipes. From data for an aluminium ceiling panel with pipes spaced at 300 mm centres [6], the typical temperature difference between the panel surface temperature and the average water temperature is

$$t_s - t_w = 0.107q$$

where q is the cooling capacity of the ceiling (W m^{-2}).

The panels cool primarily by radiation, so the occupants perceive the room temperature as 1 K to 1.5 K less than the air temperature. The amount of radiant cooling is limited by condensation considerations as the panels and pipes may approach the dew point of the room air.

With a chilled ceiling using 8 mm copper tubes spaced 60 mm apart on aluminium or steel perforated ceiling panels, a 17°C mean water temperature in the pipes and a room air temperature of 27°C, then according to one manufacturer, the ceiling will provide 70–80 W m^{-2} cooling. If there are ceiling diffusers then the air movement can increase the panel's cooling effect by an extra 10%. Care should be taken to avoid laminar water flow in the pipes above the panels as this greatly reduces the heat transfer compared to turbulent flow. Small-bore polypropylene pipework (1.9 mm internal bore) is also used to form a lattice of cooled pipes in the ceiling panel. These are reported to produce 70 W m^{-2} [18].

Example 12.3

Consider a chilled ceiling with a mean water temperature of 17°C and a room air temperature of 27°C, as quoted above. Assume a mean radiant temperature of 25°C. Does theory confirm the output given?

Solution

Radiation heat loss was discussed in Chapter 7. There the radiation heat exchange between two surfaces, 1 and 2, was given as

$$Q_{12} = A_1 \sigma F_a F_e (T_1^4 - T_2^4) \tag{12.13}$$

where σ is the Stefan–Boltzmann constant (5.67×10^{-8} W m^{-2} K^{-4}), F_a is a form factor for the relative geometries of the two surfaces and F_e is a factor accounting for the emissivities and absorptivities of the two surfaces. ASHRAE [19] suggests that Hottel's equation can be used for heating and cooling panels in a boxlike room with the panel at a uniform temperature, the other surfaces at another temperature and all surfaces perfectly diffusing:

$$F_a F_c = \frac{1}{\dfrac{1}{F_{12}} + \left(\dfrac{1}{\varepsilon_1} - 1\right) + \dfrac{A_1}{A_2}\left(\dfrac{1}{\varepsilon_2} - 1\right)} \tag{12.14}$$

where F_{12} = view factor = 1.0
$\varepsilon_1, \varepsilon_2$ = emissivities of the surfaces
A_1, A_2 = areas of the surfaces (m^2)

The ASHRAE Handbook goes on to say that in practice the emissivity of non-metallic or painted metal non-reflecting surfaces is about 0.9 and that when used in equation (12.14) $F_a F_c$ is about 0.87 for most rooms. Substituting this into equation (12.13) gives

$$q = 4.93\left(\left[\frac{T_r}{100}\right]^4 - \left[\frac{T_p}{100}\right]^4\right) \tag{12.15}$$

where q = heat transferred from the room to the panel per unit area of panel (W m^{-2})
T_r = area-weighted surface temperature of the room's surfaces excluding the ceiling (K)
T_p = panel temperature (K)

The temperatures are divided by 100 to account for the 10^{-8} index of the Stefan–Boltzmann constant. The area-weighted surface temperature is taken as a practical approximation to the mean radiant temperature, as the radiant temperature varies with position in the room and is difficult to measure, whereas the surface temperature does not vary with position and is easier to measure [20, 11].

Using equation (12.15) with the mean water temperature as an approximation to the panel surface temperature gives the ceiling panels' output as

$$4.93\left(\left[\frac{298}{100}\right]^4 - \left[\frac{290}{100}\right]^4\right) = 40 \text{ W m}^{-2} \tag{12.16}$$

There is also natural convection from the ceiling panels [18], for which the heat absorption is

$$q = 2.18(t_{ai} - t_p)^{1.31} \quad \text{W m}^{-2} \tag{12.17}$$

where t_p = panel temperature (°C)
t_{ai} = room air temperature (°C)

In this example the output is

$$q = 2.18(27 - 17)^{1.31}$$
$$= 44 \text{ W m}^{-2}$$

This gives a total cooling output of 84 W m^{-2}, a little above the manufacturer's values. But remember that the average water temperature was used and this will be slightly lower than the panel surface temperature, giving an overestimate of the heat transfer. The room air temperature and mean radiant temperature are rather high for comfort. Combining these two values to give the dry resultant comfort temperature, t_c:

$$t_c = \frac{1}{2}t_{ai} + \frac{1}{2}t_r$$
$$= 26°\text{C}$$

This would be a little uncomfortable and a better temperature would be between 23°C and 25°C. A temperature of 25°C is the upper limit for assessing the comfort of natural ventilation before cooling is required [21]. A rule of thumb for determining the output of chilled ceilings is that the cooling effect is 10 W for each 1 m^{-2} of ceiling for each 1 K of temperature difference between the ceiling and the room.

12.3.1 Control of chilled ceilings

Compared to hot water radiators for heating, chilled ceilings have a greater surface area but a lower surface-to-room temperature difference. The driving force of the heat transfer for the ceiling is less than for a radiator, hence there is more self-regulation with the chilled ceiling and less need for control.

Basic control of chilled ceilings is either by varying the water inlet temperature or by varying the water flow rate with a constant inlet temperature. On/off control is also used. Water temperature and flow rate are varied using a three-port mixing valve and secondary pump, as with compensator control (Chapter 8), or by a diverting valve circuit. The variable temperature/constant volume mixing circuit will give a better distribution of chilled water through the panel, whereas a diverting constant temperature/variable volume circuit will have a larger temperature difference from flow to return and hence a more variable panel temperature. Two-port valves can be used instead of three-port mixing and diverting valves but the control will not be as good. Figure 12.5 shows a two-port variable temperature circuit but two-port variable flow circuits can also be used [18].

If the ceiling panels are supplied with chilled water from a primary chilled water circuit, perhaps supplying a cooling coil, then the flow temperature is likely to be about 6°C and an **injection circuit** will be required. As explained in Chapter 13, the injection circuit enables the secondary chilled ceiling circuit to have a higher flow water temperature, about 16°C.

The feedback control from the room is often just for on/off control. Although this is a rather basic form of control, even this is not very easy as the location of the room temperature sensor is sometimes difficult, especially in open-plan offices. The sensor should measure both air temperature and radiant temperature, although the radiant temperature will vary around the room. The air temperature varies more quickly than the radiant temperature as the radiant temperature relates to the fabric temperature, which can vary slowly. Hence a more responsive control comes from air temperature control. In one installation it was considered that a black bulb temperature sensor would give the best control signal.

Reset, proportional-only or PI control can be used for the feedback control instead of on/off, but the crucial element of chilled ceiling control is the **dew point control**. This is vital to stop any condensation forming on the chilled ceiling and falling as 'rain' on the occupants. For a typical zone at 22°C dry

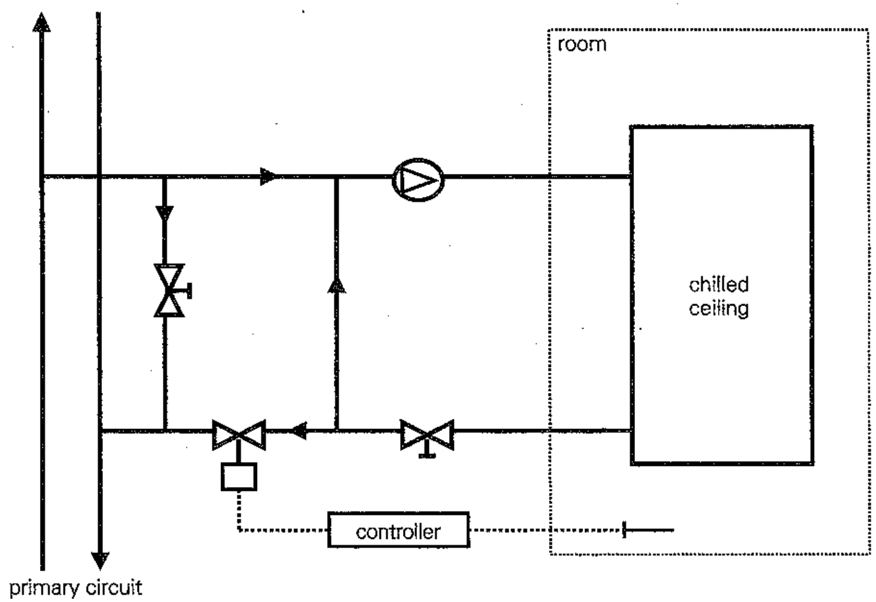

Fig. 12.5 Variable temperature control of a chilled ceiling panel with two-port valves.

bulb room air temperature and 50% RH the room dew point temperature is just above 11°C. As a safety measure, due to the variation in panel temperature between flow and return, 13°C would be considered a sensible lower limit for a chilled ceiling [6]. The chilled ceiling can only deal with sensible cooling loads. The latent loads in the zone have to be dealt with by the air supply, which is often a displacement ventilation system. A moisture sensor on the flow water pipe to the panel can be used for controlling the condensation. Although the room air temperature may not cause condensation on the ceiling, often the windows are openable and this may allow more humid air to enter and cause problems. Zone controls that close down the water supply to the relevant panels can alleviate this problem [18].

Hydronic control and balancing will often be necessary as the individual panels may be independently controlled. Network analysis will be necessary to design for adequate flows. There is also the possibility of using free cooling techniques to produce the chilled water, such as evaporative cooling towers [22]. Cooling towers can operate for a substantial period in moderate European climates and so save on mechanical cooling.

12.3.1.1 Self-balancing control

As with hot water radiators, to an extent, there is self-balancing with chilled ceilings. When the cooling output of the ceiling panel increases, the room temperature drops and reduces the temperature difference between room and panel, typically 25°C – 17°C = 8 K. The temperature difference is the driving force of heat transfer, so as it gets smaller the cooling effect is reduced. Ultimately, with little cooling load, the panel would reduce the room temperature to 16°C, the panel's flow water temperature. The ceiling panel is considered more self-balancing than a radiator, with a mean radiator-to-room temperature difference of 75°C – 20°C = 55 K. The radiator, with a small heat load, would drive the room temperature up to 75°C, hence it needs good control.

Example 12.4
A chilled ceiling system has been installed with a polypropylene pipework lattice behind the ceiling tiles. The output of the tiles is given by the equation in DIN 4715 [18]:

$$q = C(t_{mw} - t_{ai})^n \qquad (12.18)$$

For this system, $C = 5.56$ and $n = 1.105$; t_{mw} is the mean water temperature in the pipes and q is the heat absorbed by the panel (W m^{-2}). The flow water temperature is designed to be 14°C and the return 17°C with a maximum air temperature at high level of 25.5°C. If the flow water temperature is controlled at 14°C but the heat gain falls to half the design value, what does the room air temperature become if there is no control on the panel? Assume the heat gain is equally radiant and convective.

Solution
From the design conditions and the DIN 4715 equation, the design heat gain is

$$q = 5.56(15.5 - 25.5)^{1.105}$$
$$= 70.8 \text{ W m}^{-2}$$

and the heat extracted by the chilled water is

$$70.8A = \dot{m}C_p(t_r - t_f) \qquad (12.19)$$

where $\dot{m}$ is the mass flow of chilled water, C_p is the specific heat capacity of the water, t_r is the return water temperature, t_f is the flow water temperature and A is the chilled ceiling area. From the design conditions we have

$$\frac{\dot{m}}{A}C_p = 23.6 \text{ W m}^{-2}\text{ K}^{-1}$$

The only control equation we have is that the flow water temperature, t_f, is controlled at a constant value of 14°C. So when the load halves to 35.4 W m^{-2} then from equations (12.18) and (12.19) we have

$$35.4 = 5.56\left(\frac{t_r + 14}{2} - t_{ai}\right)^{1.105} \qquad (12.18)$$

$$35.4 = 23.6(t_r - 14) \qquad (12.19)$$

From equation (12.19) $t_r = 15.5$°C and substituting into equation (12.18) gives

$$6.37 = (14.75 - t_{ai})^{1.105}$$
$$5.34 = 14.75 - t_{ai}$$
$$t_{ai} = 20.09°\text{C}$$

There is a degree of self-control in that this temperature is not uncomfortable, but cooling is unlikely to be required continuously at this temperature, so further control from the room is required, preferably to modulate the ceiling temperature rather than simple on/off control.

12.4 Chilled ceiling and displacement ventilation

As displacement ventilation systems, through floor or side wall mounted units, can only cope with about 25 W m^{-2} of heat gain, chilled ceilings are often used with them to increase the cooling capacity to about 90 W m^{-2} with room temperatures of about 24–26°C. Correspondingly, chilled ceilings cannot cope with latent heat gains and need a separate air supply. So the two systems are complementary.

However, the displacement ventilation system uses less fan power than a conventional system and relies on the stack effect to move the air through the room. The chilled ceiling has a detrimental effect upon the displacement ventilation flow, suppressing the stratified boundary layer at ceiling temperatures of 18–21°C and destroying displacement flow altogether at ceiling temperatures of 14–16°C [23]. This work was done in a laboratory test room. The room had a ceiling with six circuits in parallel, providing chilled water to the ceiling panels. The circuits could be individually activated, each circuit comprising four or five panels connected in series. Control of the system was not discussed. To assess the effect of the chilled ceiling on the displacement ventilation, the vertical temperature profiles were measured for different ceiling temperatures between 14°C and 21°C and with the ceiling turned off.

A constant heat load of 62 W m^{-2} was used and the air change rate was fixed at 3.9 h^{-1}. The air supply temperature for the displacement system was 19°C. With just the displacement ventilation, the ceiling turned off, the profile was approximately a dog-leg with a floor temperature (0.1 m) of about 23°C and a 2.5 m high temperature of about 28.5°C. As the ceiling temperature reduced, the profile got steeper; the 14°C ceiling had a floor temperature of about 21°C and a 2.5 m high temperature of 21.5°C. These findings are in general agreement with those of Alamdari and Eagles [24]

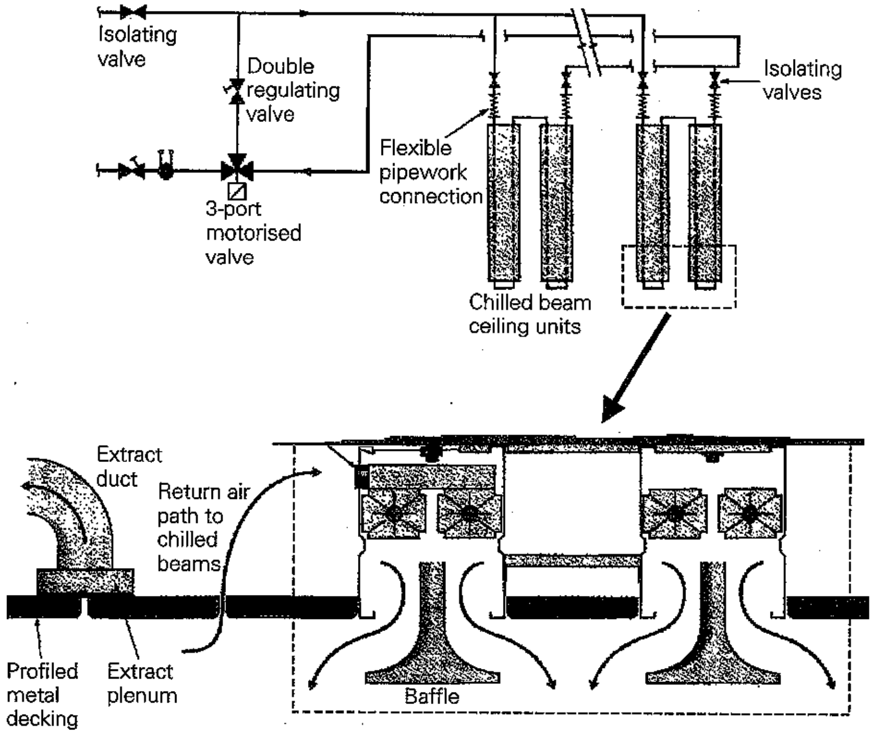

Fig. 12.6 A chilled beam. (Reprinted, with permission, from [28])

that the displacement ventilation, as identified by its temperature gradient, is suppressed by increasing output from the chilled ceiling.

There is a danger that the ventilation effectiveness can be greatly impaired by the chilled ceiling. It is therefore recommended [5] in the design of combined chilled ceiling and displacement ventilation systems that the ventilation system should cater for at least 30% of the total cooling load, Q_T, i.e. $Q_D \geqslant 0.3Q_T$. This is because there is a possibility of designing the combined system so that displacement ventilation deals with just the fresh air requirements for the occupants, which equates to 8 l s^{-1} per person [25, 11]. For an occupancy density of 1 person per 10 m^2 this is 0.8 l s^{-1} m^{-2}. For a temperature difference between the supply air and extract air of 10 K this yields

$$\dot{V}\rho C_p \times 10 = 0.0008 \times 1.2 \times 1.02 \times 10$$
$$= 9.8 \text{ W m}^{-2}$$

A small cooling load and therefore a small displacement flow is dominated by the ceiling cooling. Effectively there would be a mixed ventilation system with the displacement stratification destroyed by the ceiling.

Swirl-type diffusers can also reduce the stratification of displacement ventilation as they introduce the air at higher velocities than standard displacement ventilation. This can promote mixing in the occupied zone and so reduce the buoyancy-driven plumes of the displacement ventilation [7].

12.4.1 Chilled beams

Passive chilled beams, effectively inverted finned-tube convectors sited in ceiling recesses, are also used in conjunction with displacement ventilation systems. Active beams combine the air supply in the ceiling unit, so they do not require a floor displacement system. The downward plumes from the passive beams hit the floor and spread out [26] with a 'filling-box' regime [27] forming a layer of cool air which gradually expands upwards from the floor. The beam plume upsets the buoyancy for the displacement system over time and the combined system can be regarded as a mixing system [26]. A chilled beam is shown in Fig. 12.6 with its diverting control valve and reverse return pipework to the beams to aid balancing [28].

12.5 Mixed mode

Many buildings have natural ventilation from windows that the occupants can open as well as mechanical ventilation and/or mechanical cooling to help maintain comfort. They are called mixed-mode air-conditioned buildings or simply mixed-mode buildings. The combination of systems and control strategies for mixed-mode systems is quite enormous but some refer to **zonal mixed mode**, where certain zones of the building are naturally ventilated and others are mechanically air conditioned. Another version is **seasonal mixed mode**, where the mechanical ventilation or air conditioning can be switched on when required, primarily in the summer and probably with the windows shut.

An example of a simple mixed-mode building is where fan coil units are installed as well as openable windows. Trip switches on the windows simply shut off the fan coils when the window is open. However, there are more sophisticated buildings, some of which are discussed below.

12.5.1 The Elizabeth Fry Building, University of East Anglia

The Elizabeth Fry Building at the University of East Anglia uses Termodeck integrated heating and ventilating, where the supply air is passed through the hollow cores of the concrete floor slabs (Fig. 12.7) [29]. Termodeck was devised in the 1970s by the Swedish engineer Loa Andersson. It aims to achieve a higher heat transfer efficiency into the building fabric due to the air supplied to the rooms or zones flowing within the core of the slab.

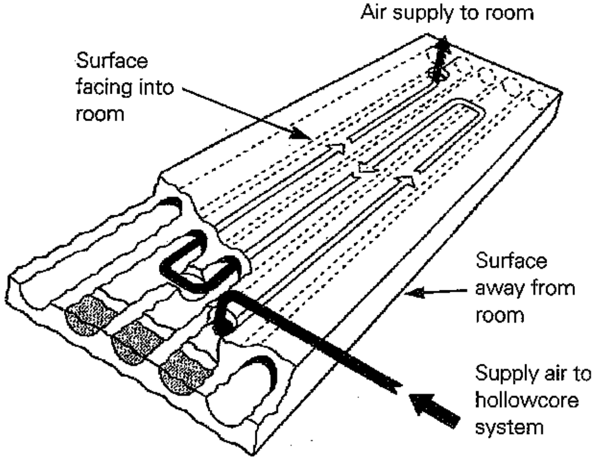

Fig. 12.7 Ventilation through a Termodeck slab. (Courtesy Termodeck)

Table 12.2 Thermal values for the Elizabeth Fry Building

Element	U-*value* ($W\,m^{-2}\,K^{-1}$)
Walls	0.2
Roof	0.13
Windows	1.3

Generally the slabs are 1.2 m wide and 0.3 m deep and they have five smooth-faced cores, three of them normally used for the air [30]. The slabs can absorb 10–40 $W\,m^{-2}$ of internal gains, assuming that 5–9 $m^3\,h^{-1}\,m^{-2}$, (1.4–2.5 $l\,s^{-1}\,m^{-2}$) of air flows through the hollow cores of the slabs. With these low loads, good insulation and reduced solar gains are essential. Table 12.2 shows the insulation values for the Elizabeth Fry Building.

The storage efficiency [31] of a slab is given by

$$\frac{t_{in} - t_{out}}{t_{in} - t_{slab}}$$

where t_{in} is the inlet air temperature, t_{out} is the outlet air temperature and t_{slab} is the slab temperature. The slab accepts 75% of potential heat available from the air. The effective volume of the slab is 0.36 m^3 about one-third of the slab's volume of 1 m^3. The ventilation through the hollow cores is by fans; the fan power varies as the third power of the flow rate but the cooling varies directly with the flow rate. Off-peak cooling will be an advantage.

The Elizabeth Fry Building has four storeys [32], the top two having 50 cellular offices for about 70 staff. The lower ground and ground floors contain lecture theatres and seminar rooms. It is of narrow plan with the offices

and meeting rooms being less than 6 m deep. The occupants can open the windows. The building was highly insulated, as Table 12.2 shows, with triple glazing and an integral sunblind. Infiltration was kept to a minimum with the building being specified and tested to have less than one air change per hour at the infiltration test pressure of 50 Pa. There is a small atrium, containing a staircase to all floors, with a glass roof and louvres to control solar gain.

Air for heating and ventilation comes from the hollow cores in the slabs through small circular ceiling diffusers. During occupancy there is a continuous supply of fresh air. In the main lecture theatres the ceiling air is ducted down to wall-mounted displacement ventilation units. As the cores can only supply one-third of the design maximum air to the lecture theatre, the rest is supplied from under the floor. The lecture theatre ventilation fans were of variable speed, controlled with a signal from a CO_2 level sensor. These maintained the CO_2 level at 800–1000 ppm, rising to 1300 ppm during periods of high occupancy.

Near the end of construction the BEMS was omitted to save money but the management team found that it needed data from the BEMS to run the building efficiently, so a BEMS was installed one year after the building's completion.

Summer night ventilation is possible with the fans blowing cool night air through hollow slabs. Extensive monitoring was conducted and a number of meetings were held between the designers and the client to ensure the building operated as intended. From the monitoring it was found that the internal temperature was maintained at a fairly steady value (Fig. 12.8). When the

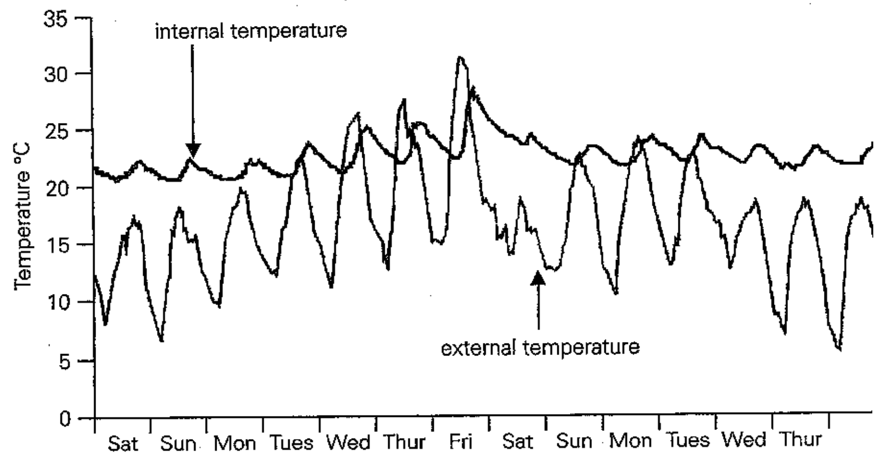

Fig. 12.8 Half-hourly temperatures in a south-facing office of the Elizabeth Fry Building for the period 1 June to 13 June 1996. (Adapted, with permission, from [29])

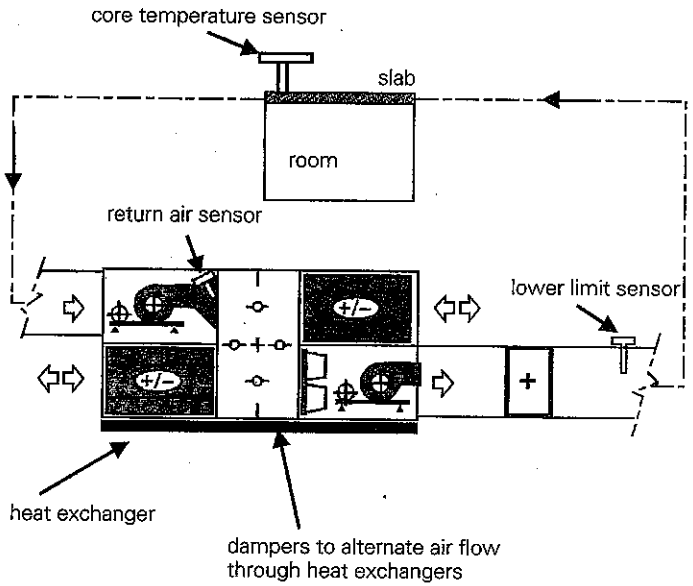

Fig. 12.9 Elizabeth Fry Building: Termodeck system and plant. (Reprinted, with permission, from Termodeck (UK) Ltd)

outside temperature rose to over 30°C on the Friday, the inside temperature did rise to 27°C but it occurred at 7 pm. Notice that the internal temperature peaks are significantly shifted compared to the external peaks just after midday.

Figure 12.9 shows the slab system and plant for the building. The plant shown is one of the four air-handling units, which have high-efficiency heat recovery units. These units use reversing regenerator heat exchangers with metal packs to absorb the heat from the exhaust airstream. The airflow between the fresh air duct and the extract duct is mechanically reversed once per minute to exchange the heat [32].

The control strategy is to maintain the core temperature, as measured by a sensor near the outlet of a hollow core, at 22°C during winter and summer. Initially the core sensor (an air temperature sensor suspended in the core) was placed near the air entry point to the core instead of near the exit. It was later repositioned and the control improved.

The heating strategy is very simple:

If t_{core} < 21.5°C then heating comes on
If t_{core} > 22°C then heating switched off

During occupied hours the system uses full fresh air, but during the unoccupied time after 2200 in the summer the cooling strategy is also simple:

If t_{core} > 23°C and $t_{ao} < t_{core} + 2$ then the fans are switched on
If t_{core} < 22°C then the fans are switched off

It is interesting to compare this simple control strategy with the more complex strategy for the Weidmuller building in West Malling, Kent [33]. This is a similar Termodeck system to the Elizabeth Fry Building, except there is an evaporative cooler before the air-handling unit; the air-handling unit contains the heat exchanger and dampers as before. The control strategy is designed to maintain the slab and room temperatures within the range 20–22°C. The night-time (off-peak) control schedule is as follows:

If t_{ao} > 20°C and room or slab > 23°C
then evaporative cooling until both < 20°C
If t_{ao} > 10°C and room or slab > 22°C
then free cooling until both < 20°C
If t_{ao} < 6°C and room or slab < 20°C
then heating until both > 22°C

And here is the daytime control schedule, 0700 to 1800:

If t_{ao} > 10°C, until t_{ao} < 6°C and extract > 23°C
then evaporative cooling until extract < 22°C
If t_{ao} > 10°C, until t_{ao} < 6°C and extract > 22°C
then direct ventilation until extract < 22°C or supply < 12°C
If t_{ao} > 10°C, until t_{ao} < 6°C and extract < 22°C then heat recovery
If t_{ao} < 6°C until t_{ao} > 10°C and extract > 22°C and supply > 15°C
then direct ventilation until extract < 22°C or supply < 12°C
If t_{ao} < 6°C until t_{ao} > 10°C and extract < 22°C then heat recovery

There were some flaws in the initial control of the plant with this more complex strategy [33]. For a large part of the monitored period, the evaporative cooled return air was actually heating the supply air rather than cooling it. The control strategy failed to differentiate between weekdays and weekends.

In the Elizabeth Fry Building the artificial lighting was measured to be 310 lx. The total electrical consumption in 1997 was 61 kWh m^{-2} y^{-1} compared to a good practice figure for academic buildings of 75 kWh m^{-2} y^{-1} [34]. The total normalized gas consumption for 1997 was 37 kWh m^{-2} y^{-1}, lower than the previous two years and reflecting the fine tuning of the system.

An occupant survey of just over half the staff revealed high levels of satisfaction with the internal environment, suggesting that not only was the building energy efficient but also comfortable [32]. Further details on this building and fabric thermal storage can be found in the literature [35, 36].

12.5.2 Other mixed-mode control

12.5.2.1 Inland Revenue, Nottingham

The Inland Revenue building, Nottingham, has occupant-openable windows, as well as four-speed fans to supply fresh air in summer or heated air in winter [37]. The fans are under the floor and provide displacement ventilation but without any mechanical cooling. The air is extracted via towers (Fig. 11.3). The basic control of the towers to maintain a comfortable room temperature has been discussed in Chapter 11, but a look-up table of wind speed and rain is used for additional control conditions, e.g. for worsening rain or increasing wind speed. Wind direction and outside temperature are also used to modify the control of a tower opening.

12.5.2.2 Inland Revenue, Durrington

The Inland Revenue building, Durrington, has a two-speed mechanical ventilation system providing 1.5 or 4 air changes per hour, and there is natural ventilation from automatically controlled casement vents fitted above manually openable windows. Figure 12.10 shows the daytime control schedule [37] for controlling the mechanical supply and heating.

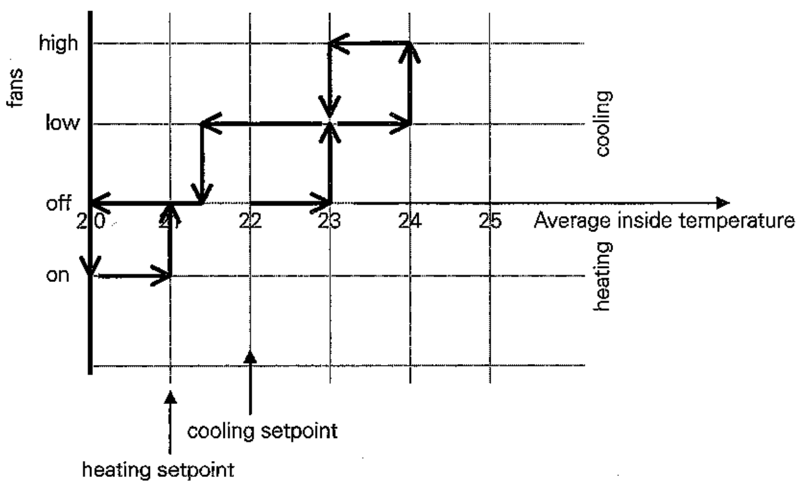

Fig. 12.10 Daytime control schedule at the Inland Revenue building, Durrington.

12.5.2.3 PowerGen, Coventry

The PowerGen building has a central atrium with an open-plan office built around it. There are three rows of windows in the perimeter, two of which can be manually operated by the occupants. The top row is controlled by a BEMS. During the occupied period there is continuous ventilation via a mechanical displacement system providing 1.5 or 3 air changes per hour with air extraction by fans in the atrium roof or via windows under automatic control at the top of the atrium. There are also blinds for the windows. When the inside temperature is greater than 23°C the automatically controlled perimeter windows are opened as well as the atrium windows on the opposite side of the building to provide crossflow ventilation. However, when the external temperature exceeds the internal temperature, the BEMS-controlled windows are made to shut.

12.5.2.4 Interlocks and occupant guidance

Interlocks are advisable on mixed-mode buildings, to prevent competition between mechanical cooling and natural cooling. The Refuge building, Wilmslow, has underfloor, four-pipe, three-speed fan coils and openable windows. During hot weather the occupants are uncertain whether to open the windows or rely on the air conditioning [38]. The natural tendency is to keep windows open on hot days, when it may well be hotter outside than in.

An interlock system was considered at the design stage but rejected on the grounds of cost and reliability. It was also felt that the style of the windows did not allow the occupants to fine-tune the ventilation.

However, at the BSI headquarters in Chiswick, the fan coils are shut off by window trip switches when the windows are opened [39]. At the Ionica building, Cambridge, which also has a Termodeck system and evaporative coolers, the design team gave seminars and distributed airline-style instruction sheets to help occupants use the window blinds and openable windows effectively. The occupants are expected to take part in controlling their environment. Note that blinds can be left down and the lights left switched on, although this is more common in older buildings where there are solar gain problems.

12.6 Lighting control

Lighting in many buildings can be a significant part of the total energy consumption, cost and CO_2 production. For an office building the installed power density can vary between 12 and 20 W m^{-2}; each year this consumes between 14 and 60 kWh m^{-2} and produces between 2 and 8.5 kg of carbon per square metre of treated floor area [40]. In cost terms this is between 15% and 31% of the total office energy costs, hence light switching, daylight and window size are important aspects of energy-efficient building design.

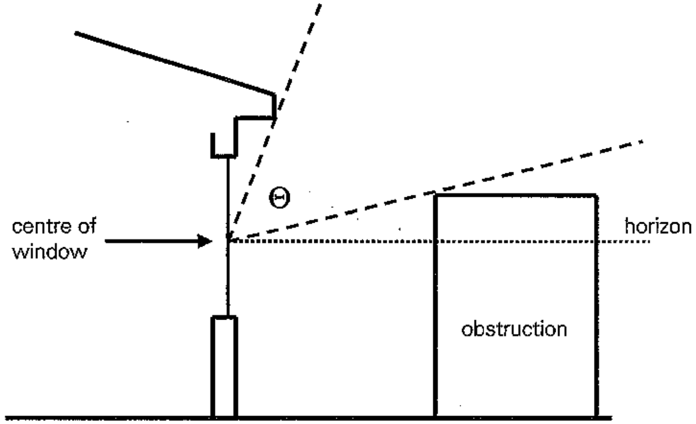

Fig. 12.11 Angle of visible sky.

Solar radiation not only gives daylight, it also generates a heat gain inside the building. The relationship between radiation power (W) and luminous flux (lm) is

$$\Phi = K\int P(\lambda)V(\lambda)\,\mathrm{d}\lambda$$

where $P(\lambda)$ = the radiation power at wavelength λ (W nm^{-1})
$V(\lambda)$ = the visibility curve
Φ = the luminous flux or visible light (lm)
λ = the wavelength of the radiation (nm)
$K = 683$ lm W^{-1}

The visibility curve, $V(\lambda)$, is the eye response to different wavelengths of light (Fig. 12.11). For instance, the eye sees yellow and green light, at the centre of the visibility curve, much better than red and blue light, at the extremes. Effectively the unit of light, the lumen, is the 'eye-corrected' watt.

The **efficacy** is the ratio of the light output of a lamp to the electrical power input. Typical values are 10 lm W^{-1} for a tungsten lamp, 95 lm W^{-1} for a high-frequency fluorescent lamp, and 145 lm W^{-1} for a monochromatic low-pressure sodium lamp. Daylight varies in its efficacy but a typical value is 100 lm W^{-1}.

12.6.1 Daylight factor

With daylight levels of up to 100 000 lx, (1 lx = 1 lm m^{-2}) on a clear summer's day, there is a great potential for reducing artificial lighting. How much depends on the **daylight factor** (DF) of the room and window. This is the ratio of the daylight at a point in the room to the daylight outside the building in an unobstructed position, excluding direct sunlight. Often the

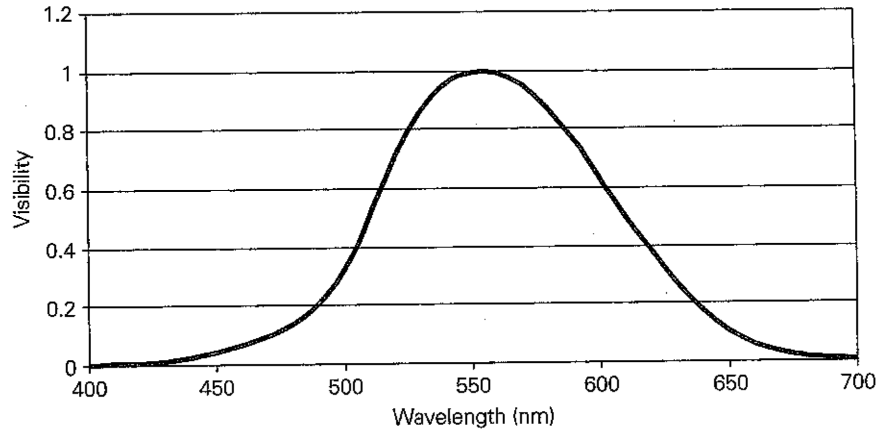

Fig. 12.12 Visibility curve for photopic vision.

average daylight factor is used; this is the average throughout the room rather than at one particular point. Here is the equation:

$$\overline{\mathrm{DF}} = \frac{A_g \theta T}{A(1 - R^2)}$$

where $\overline{\mathrm{DF}}$ = the average daylight factor (%)
A_g = glazed areas of windows (m^2)
θ = angle of visible sky (Fig. 12.12)
T = transmittance of glazing to diffuse light, including the effect of dirt
A = total area of the room surfaces including the windows (m^2)
R = mean reflectance of room surfaces (typical values: ceiling 0.7, light-coloured walls 0.5, floor 0.2, windows 0.1)

The diffuse radiation from an overcast sky, without direct sunlight, is used to calculate the daylight factor, and it can be used to assess the room appearance and the likelihood of saving energy [41]:

- $\overline{\mathrm{DF}} < 2\%$
  (1) Electric lighting is often needed during the day
  (2) Room appears gloomy with only daylight
  (3) Few savings from daylight
- $2\% \leqslant \overline{\mathrm{DF}} \leqslant 5\%$
  (1) Room is often adequately lit by daylight
  (2) Good savings possible by using daylight
- $\overline{\mathrm{DF}} > 5\%$
  (1) Room appears bright in daylight
  (2) Electric lighting rarely needed in the daytime

12.6.2 Luminaire switching

It is a requirement in UK buildings that a light switch should be no more than 8 m distant (in plan) from the luminaire it controls, or no more than three times the height of the luminaire above floor level if this is greater [42]. This will enable occupants to switch on only those luminaires that are necessary. Infrared, ultrasonic or microwave transmitters (often hand-held devices) and their receivers can also be used to switch or dim lights.

Time-operated controls, which switch off the lighting at lunchtime, at the end of occupancy and at the end of any early morning cleaning period are also good energy-saving systems. Often a **mains borne** signal is used for the switching, putting a short off pulse on the alternating mains sine wave at the zero-voltage switching point (i.e. when the voltage changes polarity). Effectively this extends the zero-voltage point by a short time without affecting the transmitted power significantly. This enables the **latching relays** in the luminaires to drop out of contact, disconnecting the luminaire's power supply. The latching relay can be reset to make contact, and so switch on the luminaire, by the occupant pulling a pull cord. Digital signals on the mains transmitted to receivers in the luminaires can also be used. In a narrow-plan, open-plan office in London, half the luminaires were fitted with latching relays and pull cords and the other half were left with the existing large panel of switches near the entrance door. Monitoring of both halves over a winter period showed electricity savings of 63% due to the pull-cord system.

Light switching is mostly through on/off signals, sometimes accompanied by a luminaire identity number. This simplicity explains why mains borne signalling is so common. Any noise or harmonics on the mains can easily be detected in the signals, which carry small amounts of information and for such a short time. For general BEMS use with HVAC plant, where perhaps temperatures and a range of control signals are constantly being sent, then the information being transmitted and the rate of transmission make mains borne signalling less attractive as it is more prone to noise and consequent errors. There is also a trend that where a bus system is used for transmission of control signals to luminaires, it is a separate bus to the HVAC BEMS as there are companies that specialize in lighting control and put in their own bus system with a rapid response and low information transmission.

12.6.3 Intelligent luminaires

Too much light switching may annoy the occupants and the energy savings may not reach expectation. However, **intelligent luminaires** employ high-frequency control circuits and lamps which can easily be dimmed sufficiently slowly that the occupant is not distracted or annoyed. The intelligent luminaire has a **light-dependent resistor** (LDR) pointing towards the working surface or the floor to sense the light reflected back. The output from the LDR is non-linear so a look-up table is used to linearize the output. A setpoint for the LDR can then be set. For instance, if 500 lx is required on the working plane, typically the plane at desk height in an office, and the desk has a reflectance of 20% then the LDR can be set at 100 lx. This is a very crude method and more exact calculations would be required to include the inverse square law of illumination, so in practice a light meter is placed on the desk and the LDR is set accordingly. The luminaire can then be controlled to this LDR setpoint. If daylight is present, the luminaire either dims to supplement the natural lighting or it switches off. High-frequency fluorescent lamps can typically be dimmed to 10%; below this they become unstable and may begin to flicker [43].

If a light sensor is used with just light switching then it is recommended that the luminaire is switched off when the combined daylight and electric lighting is at least three times the required task illuminance [40]. For most office tasks this will be between 300 lx and 500 lx, so the switching-off level will be 900 lx and 1500 lx respectively. A sensible switch-on level will be the task illuminance, i.e. 300 lx to 500 lx.

A dimming strategy is a little more complex than the simple on/off control. The control strategy for a popular intelligent luminaire varies the rate of change of light output from the luminaire in proportion to the error from the setpoint. If the control signal, u, varies from 0 V to 10 V then 10 V corresponds to the full light output of the luminaire, corresponding to an illuminance level on the working plane of E_{max}, and 0 V corresponds to a zero output (it may be offset to the practical dimming minimum of 10% output). Here the zero output will be assumed. If the setpoint illuminance is E_{set} then the error, e, is

$$e = E_{set} - E$$

and as a percentage it becomes

$$e' = \frac{E_{set} - E}{E_{set}} \times 100\%$$

Note that E is made up of the illuminance from the luminaire, E_l, and any daylight contribution, E_d. If the error is more than 50% away from the setpoint, i.e. $|e'| > 50\%$, then

$$\frac{du}{dT} = 1.0 \operatorname{sign} e'$$

$$\text{If } 10\% \leqslant |e'| \leqslant 50\% \quad \text{then} \quad \frac{du}{dT} = 0.2 \operatorname{sign} e'$$

$$\text{If } |e'| < 10\% \quad \text{then} \quad \frac{du}{dT} = 0.033 \operatorname{sign} e'$$

If e' is positive then sign $e' = 1$ and if e' is negative then sign $e' = -1$; if $e' = 0$ then sign $e' = 0$.

This is a form of integral control which effectively introduces a slowing down of the luminaire response. A proportional control would immediately change the control signal, u, which could give a noticeable change in lighting to the occupant. The sampling time of the control is typically 1 s with a dead time of 5 s before a change in the control schedule. An occupancy or presence detector can sense movement in the occupied space and switch off the luminaire if no movement is detected. Often there is a delay of about 10 min in case the occupant has been still and to avoid causing annoyance by switching off too soon.

Example 12.5

An intelligent luminaire can produce an illuminance of 800 lx on a person's desk. The desk has a reflectivity of 20%. The setpoint illuminance for the luminaire is 500 lx. The luminaire is providing a steady 300 lx on the desk when the window blind is down. The blind is then opened and daylight contributes a further 200 lx. How long does the luminaire take to adjust to this new condition with the above control schedule?

Solution

Here $E_{max} = 800$ lx which corresponds to $u_{max} = 10$ V and $E_{min} = 0$, i.e. $u_{min} = 0$. The luminaire is currently giving an illuminance of 300 lx and the daylight contribution through the blind is 200 lx. When the blind is opened there is an extra 200 lx from the daylight, so

$$e' = \frac{500 - 700}{500} \times 100\% = 40\%$$

so

$$\frac{du}{dT} = 0.2\ \mathrm{V\,s^{-1}}$$

$$= 16\ \mathrm{lx\,s^{-1}}$$

and when the luminaire is down to 150 lx (i.e. $E = 550$ lx, within 10% of the setpoint illuminance) then

$$\frac{du}{dT} = 0.033\ \mathrm{V\,s^{-1}}$$

$$= 2.67\ \mathrm{lx\,s^{-1}}$$

The time for the luminaire to reduce its output so that the desk illuminance is 500 lx is therefore

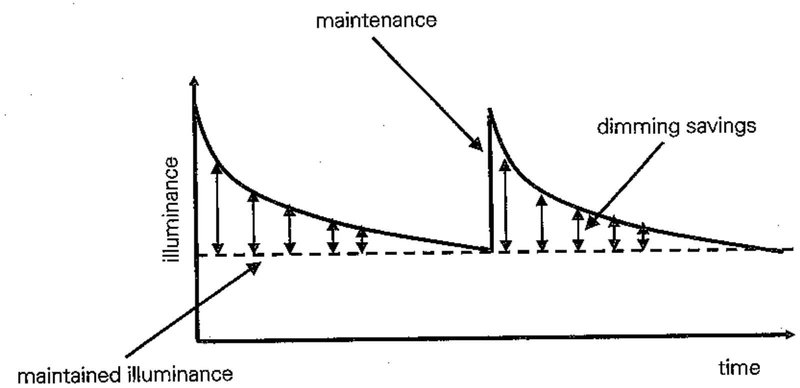

Fig. 12.13 Maintained illuminance and dimming savings.

$$\frac{150}{16} + \frac{50}{2.67} = 28.1\ \mathrm{s}$$

This does not take account of the dead time that may be present to stop excessive changes, e.g. when clouds briefly alter the daylight contribution or an occupant passes under the sensor.

12.6.3.1 Maintained illuminance

An intelligent luminaire can make savings due to dimming when daylight is present and also from its presence detector. However, it can also save by not producing its full output and therefore going above the required lighting level. This is because lighting installations are designed to a **maintained illuminance**, i.e. the illuminance at which maintenance is required. So if the maintained illuminance is 500 lx then with various maintenance factors to account for the depreciation of the lamps, and the room and luminaire surfaces getting dirty, the initial illuminance level may well be about 700–800 lx from full output, undimmed lamps. Figure 12.13 shows that the savings may be very significant. Although 500 lx is the commonly accepted and recommended maintained illuminance for offices [44], many low-energy buildings are now using values around 400 lx or even 300 lx.

References

[1] Schultz, U. (1993) A new eye for indoor climate. *Heating and Air Conditioning*, 1 July.

[2] World Health Organization (1983) Indoor air pollutants, exposure and health effects, *European Reports and Studies*, **78**.