Unit 4: Indoor Thermal Comfort

Unit objectives

- Be familiar with the basic thermal system of a human body
- Understand the major ways of heat exchange between a human body and its surrounding environment
- Understand the basic requirement of thermal comfort
- Be familiar with the measurable variables that affect human thermal comfort
- Understand the indoor design criteria
- Understand the major ways that an air conditioning system used to maintain indoor thermal comfort in bother warm and cold seasons.

Unit topics

- Thermal comfort
    - Thermoregulation system of a human body
    - Heat exchanges
- Factors affecting thermal comfort
    - Environmental factors
    - Personal factors
- Thermal indices
- Indoor design criteria
- Tutorial questions

Suggested reading

- Jones W P, *Air Conditioning Engineering,* 5th Edition, Edward Arnold. Chapters 4 & 5.
- CIBSE Guide Book A. Sections A1 & A2.
- Legg R, *Air Conditioning Systems,* Batsford, 1991. Chapters 3 & 4.

4.1 Human thermal comfort

4.1.1 Thermoregulation of a human body

The thermoregulation system of a healthy human body is to maintain the body core temperature very closely around 37^{o}C, as its vital organs can only function properly within this narrow range of temperature.

As the human body gains energy from foods, and the body heat has to dissipate into its surrounding environment.

Figure 4.1 shows that body temperature varies a little under various ambient temperatures. It also shows that the body temperature rises slightly when the body is carrying a heavy activity. All these attempts are to maintain thermal equilibrium with the surroundings.

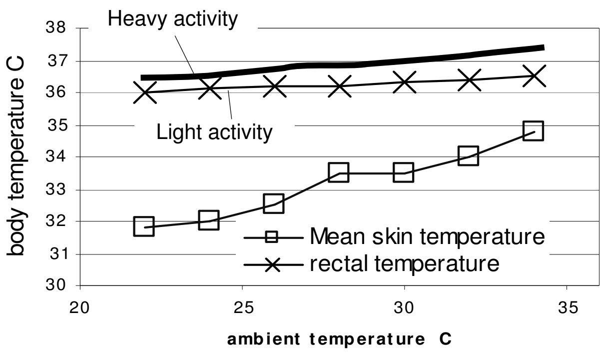


Figure 4.1 body temperature responses the change of environmental temperature.

This is achieved by the *hypothalamus* located in the brain controlling the thermoregulatory system. Under heat stress blood is circulated to vessels just under the skin surface in order that heat can be more effectively liberated at the body surface. This process is known as *Vasodilation*. Conversely, under extreme cold conditions, the blood flow to the body surface and the extremities is reduced therefore protecting the body core. This is known as *Vasoconstriction*.

4.1.2 Heat exchanges

If thermal equilibrium does not take place other involuntary responses such as shivering and sweating may also occur. The thermal balance between the body and its surroundings is controlled by means of interchanges of heat involving:

i Evaporation – moisture from skin;
ii Radiation – exchange in skin/cloth surfaces with other surrounding surfaces;
iii Convection – skin/cloth surfaces with the surrounding surfaces;

In addition to these three major ways of heat exchanges, the heat exchange also takes place in forms of conduction via cloths, and, most significantly in forms of sensible and latent heat exchanges by respiration.

Hence it is possible to represent the thermal balance by a simple equation:

$$S = M - W + K + R + C - E - RES \qquad (4.1)$$

where:

S = heat storage in body, W
M = the metabolic rate, W
W = the useful rate of work, W
K = the conduction heat exchange, W
R = the radiation heat inter change, W
C = the convective heat inter change, W
E = the evaporative heat inter change, W
RES = heat loss by respiration, W

S = 0 means that no heat accumulation inside the human body. This is the essential condition for thermal comfort.

However, to maintain zero heat storage in the body does not necessarily mean that thermal comfort has achieved. As mentioned earlier, *Vasoconstriction* can reduce heat lost to a certain degree; whilst sweating can release a large amount of heat. Accelerated breath rate, often as a result of exercise, can also increase both sensible and latent heat losses. Although all of these measures can, to some degree, maintain zero storage inside the body, it is not comfortable.

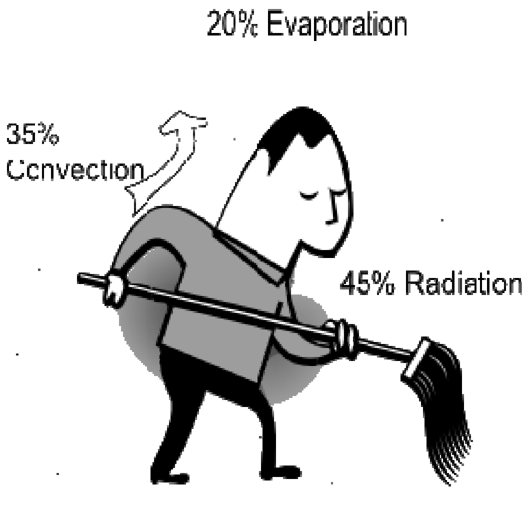


Figure 4.2 percentages of heat released

Studies have revealed that a thermally comfortable environment requires three main parts of heat release from a human body in the proportions shown on Figure 4.2. If the balance between the three parts breaks, it results in discomfort.

4.2 Factors that influence thermal comfort

4.2.1 Environmental factors

The environmental factors that influence the modes of heat transfer and, hence, thermal comfort are:

i) dry bulb temperature
ii) relative humidity
iii) air movement rate
iv) mean radiant temperature

As they can be measured by instruments, these four variables are used to assess objectively the thermal condition of an indoor environment. The measurement can be easily done by using simple thermometers, such as a mercury thermometer for air temperature, and a whirling/sling hygrometer comprising a dry bulb and a wet bulb thermometer for the relative humidity (Figure 4.3 right). Two readings from the dry bulb and wet bulb thermometers are plotted onto a psychrometric chart to find the relative humidity, or percentage saturation.

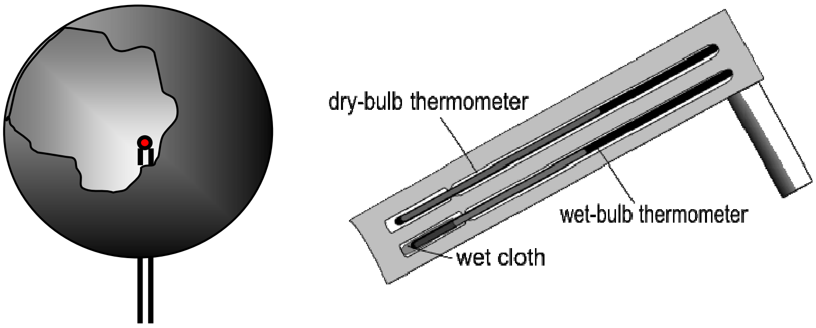


Figure 4.3 Kata thermometer(left) and Sling hygrometer

The air movement can be measured using an anemometer, which gives reading of air speed. When the speed is very low, the air movement is measured using a kata thermometer, comprising an ordinary thermometer placed at the centre of a black sphere 150mm in diameter (Figure 4.3 left). As the air movement increases heat exchange between the sphere surface and the surrounding air, the reading of the thermometer at the centre responses such changes on the surface. The reading of the temperature can be converted into the air movement rate.

As the black sphere surface is much larger than the tip of the thermometer, the Kata thermometer is also more sensitive to the changes of the radiation of surrounding surfaces, which is specified by the mean radiant temperature. Therefore the reading given by Kata thermometer really provides information for the air temperature, air movement rate and the mean radiant temperature. The following empirical formula shows how the three variables contribute to one new variable: global temperature.

$$t_g = \frac{t_r + 2.35\, t_{ai}\sqrt{v}}{1 + 2.35\sqrt{v}} \qquad (4.2)$$

where:

t_g = globe temperature, ^oC

t_r = mean radiant temperature, ^oC

t_{ai} = room air temperature, ^oC

v = air velocity, ms^{-1}

If still air conditions exist, (when $v \leq 0.1\ ms^{-1}$) the globe thermometer can be considered to measure mean radiant temperature directly.

4.2.2 Personal factors

In addition to the four environmental variables, there are two other variables define the state of the human body that are influential to thermal comfort. Known as “personal” factors, they are:

i) activity level, met
ii) clothing, clo

The activity level is a variable that is easy to specify for the occupants of a building when its main function is known. This variable has a very close correlation with the metabolic rate, which quantifies heat generated inside the body main as a result of the oxidation of food. The unit used to quantify metabolic rate is the *MET* and 1 met equals 58.2 Wm^{-2} relating to body surface area which can be considered to be approximately $2m^2$ for a typical adult.

In most buildings, a significant part of heat release from an occupant body is through clothing. Therefore this factor, specifically, its thermal resistance needs to be defined. CIBSE suggests that the thermal property of an occupant’s clothing be quantified in terms of the *CLO* and 1 clo (0.155 m^2KW^{-1}) is the equivalent of a male attired in underwear, shirt and tie and a light business suit.

activity	Wm^{-2}	met
reclining	46	0.8
Seated, relaxed	58	1.0
Standing, relaxed	70	1.2
Sedentary activity (office, dwelling, school, lab)	70	1.2
Standing activity (shopping, light work)	93	1.6
Standing activity(domestic work, machine work)	116	2.0
Medium activity (heavy machine work..)	165	2.8

Table 4.1 Common activities in buildings and the correspondent heat release and MET

4.3 Thermal comfort Indices

Six variables and various forms of heat exchanges make it difficult to assess thermal comfort of a room in engineering practise. Many attempts, therefore, have been made to combine all the measurable properties into one convenient measure of thermal comfort. These thermal indices are numerous and typically include globe temperature, equivalent temperature and effective temperature used in the USA.

Fanger has derived an index using a seven-point scale that makes possible the assessment of thermal comfort response by occupants of spaces and that takes account of all the variables. Using a thermal comfort meter, expressions of predicted percentage of dissatisfied (PPD) are determined from the predicted mean vote (PMV) for a number of locations in a room.

In the UK, the dry resultant temperature was recommended by CIBSE as a temperature index for moderate indoor environment. Resultant temperature (t_{res}) is equivalent to the temperature measured by mercury in a glass thermometer at the centre of a black globe 100mm in diameter and can also be obtained from the following equation.

$$t_{res} = \frac{t_r + t_{ai}\sqrt{10v}}{1+\sqrt{10v}} \qquad (4.3)$$

In still air conditions i.e. when v = 0.1ms^{-1} this equation simplifies to:

$$t_{res} = 0.5t_r + 0.5t_{ai} \qquad (4.4)$$

Resultant temperatures that provide optimum comfort for occupants can be found in the old versions of CIBSE Guide A, section A1. It should be noted that there will be no significant increase in dissatisfaction if temperatures remain within ±1.5 ^{0}C of the selected value.

The updated CISBE Guide A, introduces a new temperature index, 'operative temperature', to be in harmony with European and American practice.

Very much like the resultant temperature, the operative temperature takes into account of both the air temperature and the mean radiant temperature and converts them into one single value for a single space. The operative temperature, t_c, (°C), is defined as:

$$t_c = (1 - H)t_r + Ht_{air}\,; \qquad (4.5)$$

where t_{ai} is the indoor air temperature (°C), t_r is the mean radiant temperature (°C), $H = \dfrac{h_c}{h_c + h_r}$; $1 - H = \dfrac{h_r}{h_c + h_r}$, h_c and h_r are the surface heat transfer coefficients by convection and by radiation respectively $W{\cdot}m^{-2}{\cdot}K^{-1}$.

Interestingly, many studies have found that there is no difference between the values of a resultant and an operative temperature when they are used for the same room.

4.4 Indoor design criteria

Recommended resultant temperatures were provided by CIBSE in their Guide A for both sedentary and active occupations (Figure 4.1). In the case of sedentary workers, the majority of people will be neither warm nor cold in winter, in rooms where the resultant temperature is between 19 and 23°C. This is when the air movement rate is less than 0.1 ms^{-1} (i.e. still air conditions) and relative humidity lies between 40% and 70%.

A1.2. Design conditions.

Country	Season	Occupancy/category	Resultant temperature/°C	Relative humidity/%	elative iidity/%
UK	Summer	Continuous	20 to 22	50	50
		Transient	23	50	50
UK	Winter	Continuous	19 to 20	50	50
		Transient	16 to 18	50	50
Tropics	Summer	Continuous (optimum)	23	50	50
		Continuous (maximum)	25 26	60 45	60 45
		Transient (humid climate)	25 26	70 50	70 50
		Transient (arid climate)	27 28	45 40	45 40
Tropics	Winter	Short winter (as in humid climate)	Generally no heating required		
		Long winter (as in arid climate)	22	45	45

Table 4.2 CIBSE criteria for indoor thermal conditions (CIBSE Guide A1)

Where physical work is carried out the resultant temperature should be 3° to 5°C below that recommended for sedentary occupations. Some leisure activities also involve a high degree of physical activity but others such as swimming are carried out in warm conditions with low air movement because people are lightly clad. In this situation resultant temperature should be between 25°C and 30°C.

Jones points out that comfort conditioning systems provide automatic control over the dry bulb temperature with air movement rate being selected to achieve good air distribution.

In summer a design temperature of 22° - 23°C is a suitable choice for long-term sedentary occupancy in the UK with humidity allowed to swing between 40% and 60%. In spaces that are occupied only for relatively short-term periods, higher temperatures e.g. 25°C when the outside air temperature is 28°C, are acceptable. During winter, indoor air temperatures could be allowed to fall to 20° - 21°C.

Table 1.5 Recommended comfort criteria for specific applications

Building/room type	Winter operative temp. range for stated activity and clothing levels*			Summer operative temp. range (air conditioned buildings†) for stated activity and clothing levels*			Suggested air supply rate / ($L.s^{-1}$ per person) unless stated otherwise	Filtration grade‡	Maintained illuminance¶ / lux	Noise rating§ (NR)
	Temp. / °C	Activity / met	Clothing / clo	Temp. / °C	Activity / met	Clothing / clo				
Airport terminals:										
— baggage reclaim	12–19[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	F6–F7	200	45
— check-in areas[3]	18–20	1.4	1.15	21–23	1.4	0.65	10[2]	F6–F7	500[4]	45
— concourse (no seats)	19–24[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	F6–F7	200	45
— customs area	18–20	1.4	1.15	21–23	1.4	0.65	10[2]	F6–F7	500	45
— departure lounge	19–21	1.3	1.15	22–24	1.3	0.65	10[2]	F6–F7	200	40
Art galleries — see *Museums and art galleries*										
Banks, building societies, post offices:										
— counters	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F6–F7	500	35–40
— public areas	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	300	35–45
Bars/lounges	20–22	1.3	1.0	22–24	1.3	0.65	10[2]	F5–F7	100–200[5]	30–40
Bus/coach stations — see *Railway/coach stations*										
Churches	19–21	1.3	1.15	22–24	1.3	0.65	10[2]	G4–F6	100–200	25–30
Computer rooms[6]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7–F9	300	35–45
Conference/board rooms	22–23	1.1	1.0	23–25	1.1	0.65	10[2]	F6–F7	300/500[7]	25–30
Drawing offices	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7	750	35–45

Figure 4.3 Design criteria for various indoor spaces (CISBE Guide A, Table1.5)

4.5 Maintaining indoor conditions

As summarised in Unit 3, changes of outdoor weather affect indoor thermal conditions. Also due to the casual heat gains, which will be discussed in next unit, the indoor temperature will also fluctuate and thermal discomfort may occur without service systems to provide heat supply or removal.

Provision of heating or cooling can be achieved in many ways, one of which is supplying conditioned air. In warm seasons, this is done by supplying conditioned air at a temperature a few degrees lower than the temperature to be maintained for comfort to remove the indoor heat gains, and in cold seasons, by supplying air warmer then the indoor recommended temperature.

The conditioned air is taken from outdoors. Therefore it has to be proceeded according, which determined the system and operation. Selecting a system and determine the design variables are the main task in system design.

The likely range of room and outside air conditions through the year has been outlined earlier. It is now necessary to show how these conditions determine the choice of conditioning equipment. Daily variations in room sensible and latent heat loads will result in a changing relation between the supply and room air conditions and the control of conditioning equipment to follow these variations is also covered.

4.5.1 Control of room conditions

Changing sensible and latent heat loads requires a control system to monitor and correct for changes in room dry bulb temperature and humidity. In order to form an automatic control system the instrumentation chosen must be capable of giving an electrical output that can be used, via a comparative system, to drive valves controlling the flow of the heating or cooling primary medium.

Temperature measurement can be achieved by a range of transducers employing either the expansion of an element to trigger a control sequence, or the change of resistance with temperature of an element. This will give a continuous signal in the form of, for example, a varying voltage for constant current, or vice versa.

Humidity control depends commonly on the change of electrical resistance of an element as it absorbs moisture: hygroscopic salts as lithium chloride may be used in this way to give a continuously varying voltage output.

Control monitors of this type may be mounted either in the room or in the extract duct from such rooms. Positioning in the extract duct has the advantage of yielding an average valve of room temperature and humidity.

4.5.2 The use of extract and fresh air mixtures

A minimum amount of fresh air must be employed for ventilation purposes. Beyond this the ratio used in a mixture with re-circulated air can be controlled to give the most economical system operation.

In summer when outside temperature is high only minimum fresh air should be used. When the outside wet bulb temperature falls below that of the room then full fresh air should be used to assist cooling. In winter the outdoor air has to be heated and the volume above minimum used depends on the mixing conditions required for air entering the plant.

4.5.3 Winter cycles

Here the need is to heat the outside air and to increase its moisture content. Generally the room point will be above and to the left of the supply air condition.

Air washer and re-heater

An air washer using water at a temperature above air wet bulb temperature both heats and humidifies the air passed through it. If this is followed by a heater coil to achieve control of sensible heat gain then such a system could be used in winter, employing 100% outside air as shown below.

Referring to the diagram, control of the spray water temperature via a humidistat and mixing valve $V1$ allowed the position of point P to be varied. This will also affect the dry bulb temperature on entry to the re-heater. A thermostat and control valve, $V2$ that governs the mass flow to the re-heater battery can allow for this.

Washers and sprayed cooler coils are not used nowadays because of the attendant risk of Legionnella. It is still a useful academic exercise though, to consider the psychrometric processes associated with these items of equipment.

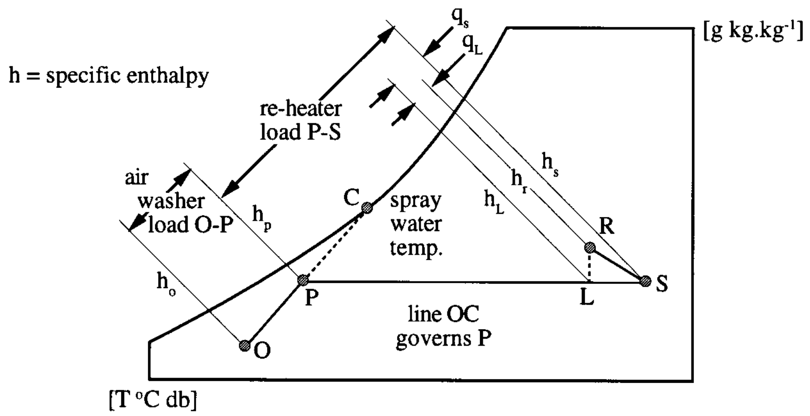


Fig 4.4

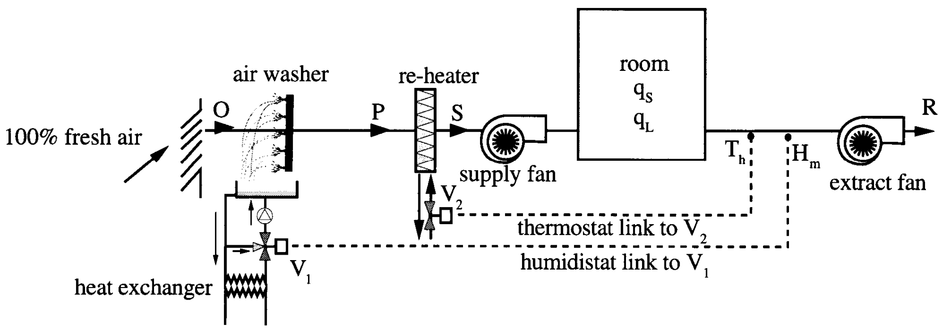


Fig 4.5

Pre-heater, air washer and re-heater

To avoid the passage of very cold air through the plant a preheated system followed by an adiabatic saturation humidifier process may be used to replace the air washer considered above. Use of this system also allows separate control of humidity and dry bulb temperature to be dealt with separately, which can have advantages in part load operation.

The position of point *P*, the Air State on entry to the humidifier can be controlled by a humidistat linked to a flow control valve on the preheated supply. This in turn affects the final moisture content achieved by passage through the air washer, to point *W*. Finally the sensible load is controlled by a thermostat linked to the flow control valve supplying the re-heater battery.

It should be noted that these controls only react to changes in the room loads i.e. occupancy, fabric losses or gains, lighting etc. Any variations in outside conditions or in the heating medium state has to be sensed by thermostats and humidistats in the supply duct which are not shown here but can also be linked to the control. Valves V_1 and V_2 below.

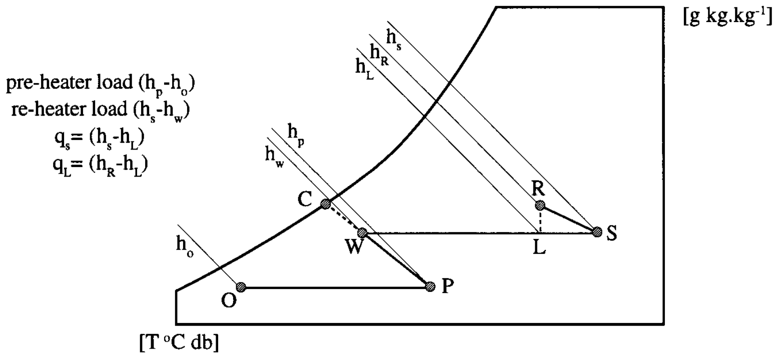


Fig 4.6

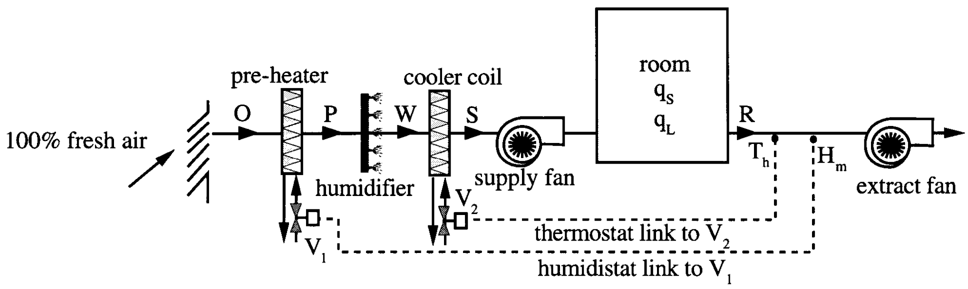


Fig 4.7

Re-circulation

Fresh and re-circulated air can be used to advantage in winter cycles. For the air washer, re-heater circuit mixture ratios can be arranged to give point M below, allowing adiabatic saturation to W. If M comes too close to WS then the spray water must be cooled, conversely if M^1 only is reached, the spray water is heated.

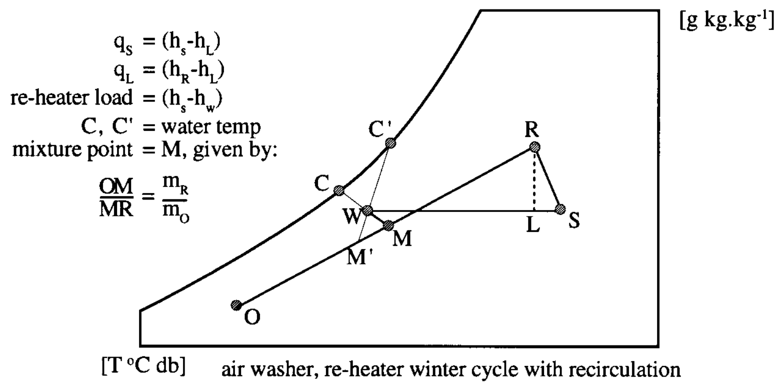


Fig 4.8

If the outside temperature is very low, below 0˚C then a pre-heater is used as shown below to take the mixture condition from *M* to *N* thus avoiding the possibility of freezing in the spray chamber. Adiabatic saturation can be used to humidify the air to point *W*.

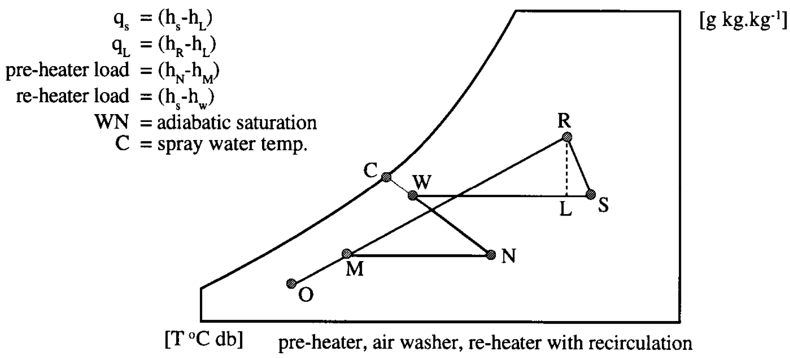


Fig 4.9

4.5.4 Summer cycles

Air washer or cooling coils and re-heater

In summer cycles, the air supplied to the room will usually be of a lower dry bulb temperature, i.e. to the left of room point on the psychrometric chart.

As the outside air temperature is likely to be high, it will be necessary to cool and dehumidify the outside air prior to supply to the room conditioned.

Summer cycle system should use full fresh air, if the outdoor wet bulb temperature is below that of the room, but above that of the supply air. i.e. point S below.

When the outdoor wet bulb temperature is above that of the room then minimum fresh air should be used to reduce the cooling load, shown by the enthalpy differences h_o - h_w and h_m - h_w.

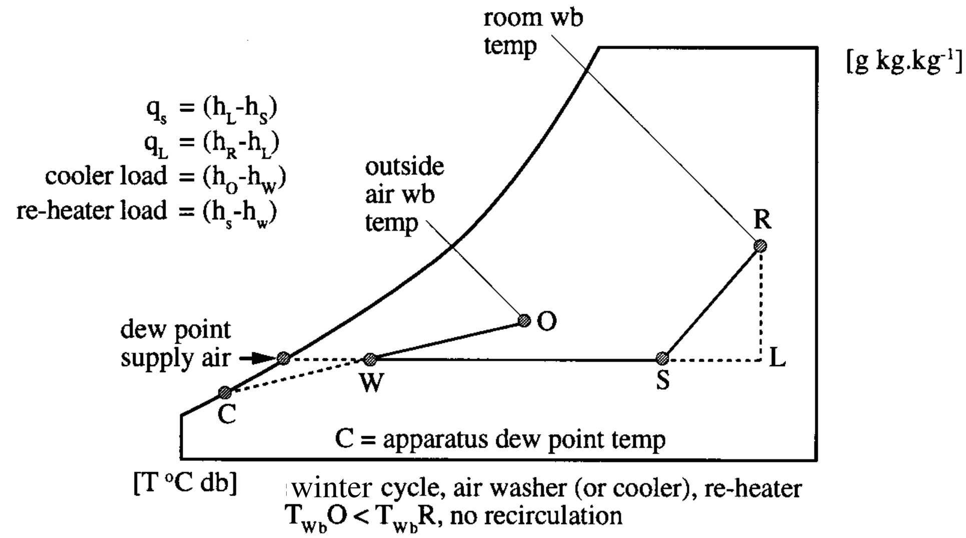


Fig 4.11

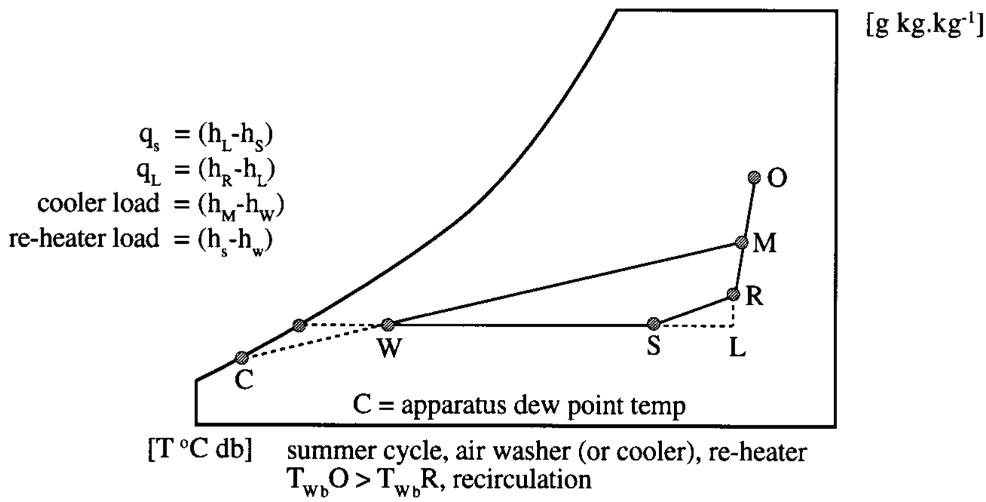


Fig 4.12

4.6 Some concluding comments

The slope of the room ratio line indicates the ratio of sensible to total heat load. There can be a number of combinations of sensible and latent gain and losses. The chart below indicates the effect on RRL of some of these combinations.

RRL_1 represents the situation when both sensible and latent gains are occurring simultaneously. RRL_2 is associated with a typical winter situation when sensible heat losses and latent gains occur in together Finally, RRL_3 represents the case of sensible and latent losses occurring concurrently.

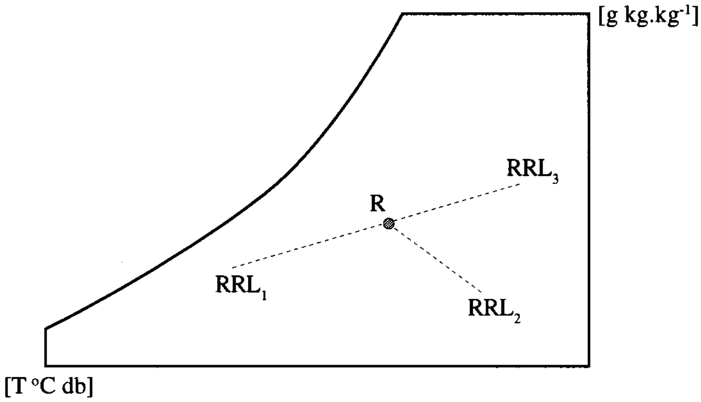
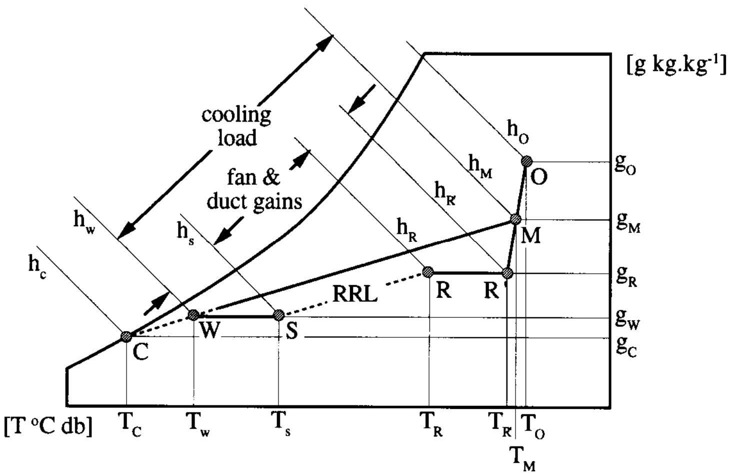
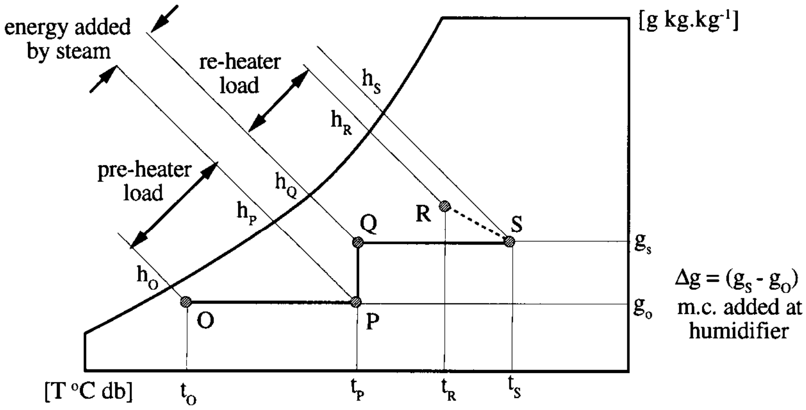


Fig 4.13

Shown below are typical summer and winter cycles for a single zone, constant volume system. The summer cycle is associated with a re-circulation system and the cycle includes fan and duct gain for both the supply and extract side of the system.

The winter cycle is associated with a full fresh air system that uses steam humidification in combination with pre-heating and re-heating processes.





Pre-heater, steam injection, re-heater winter cycle, no recirculation

Fig 4.14

The choice of a suitable supply air state is constrained by FOUR practical issues:

i) The need to minimise air quantity being handled by the system. Generally, for practical air distribution within the conditioned space, the temperature difference between room and supply air is normally limited to 8˚ - 11˚C.

ii) Practical cooler coil selection limits the coil contact factor to between 0.8 and 0.85.

iii) Fan and duct gain occurs and should be taken account of. Up to 2˚C of duct gain may arise within the system and 1˚C for each kPa of fan total pressure can also be expected. Low velocity systems operate at 600-750 Pa, high velocity systems at 2000 Pa.

iv) Water freezes at 0˚C and so this limits chilled water temperatures that are supplied to coils to between 5˚C and 6˚C with return water temperatures of between 10˚C and 11˚C.

Generally, most comfort conditioning systems are selected using volumetric flow rate rather than mass flow rate. The supply volume is established by determining the volume flow rate required to offset the sensible heat gains occurring in the conditioned space. Then, having found the required flow rate for a given supply air temperature, the moisture content of the supply air necessary to offset latent heat gains can be determined. The following equations can be used.

$$\dot{v}_t = \frac{Q_{SHG}}{(t_r - t_s)} x \frac{(273+t)}{358} \text{ or } \dot{m}_t = \frac{Q_{SHG}}{C_p(t_r - t_s)} \text{ and,} \tag{4.6}$$

$$\dot{v}_t = \frac{Q_{LHG}}{(g_r - g_s)} x \frac{(273+t)}{856} \tag{4.7}$$

Q_{SHG} the total sensible heat gains in the room, in W;

Q_{LHG} the total latent heat gains in the room, in W;

t_s supply air emperature; in ˚C;

t_r room air temperature, in ˚C;

C_p specific heat of supply air; often taken as 1.2 kJ.kg^{-1} K^{-1};

$\dot{m}_t$ mass flow rate of supply air in kgs^{-1};

$\dot{v}_t$ volumetric flow rate of supply air in m^3 s^{-1};

g_s moisture content of supply air, in g.kg^{-1}of dry air;

g_r moisture content of indoor air, in g.kg^{-1}of dry air.

Tutorial questions

4.1 List and explain the factors affecting thermal comfort and describe methods of evaluating each of these factors when attempting to assess the thermal comfort conditions within an occupied space.

4.2 Explain how a human body exchanges heat with its surrounding environment and why an indoor environment can be still thermally uncomfortable even the heat balance is already achieved.

4.3 Using the approximate method outlined in CIBSE Guide Book A, section A2, determine the design conditions for a location using the following general information:

Month with the highest average maximum dry bulb temperature July
Average maximum monthly dry bulb temperature in July 25.1^{0}C
Average maximum daily dry bulb temperature in July 18^{0}C
Average minimum relative humidity at 15.00 hours in July 64%

(Ans. 25^{0}Cdb, 17^{0}Cwb)