Lesson

30

Psychrometry Of Air Conditioning Systems

The specific objectives of this lecture are to:

1. Purpose of psychrometric calculations (*Section 30.1*)

2. Analysis of a simple, summer air conditioning system with 100% re-circulated air (*Section 30.2.1*)

3. Analysis of a summer air conditioning system with outdoor air for ventilation and with zero by-pass factor (*Section 30.2.2*)

4. Analysis of a simple, summer air conditioning system with outdoor air for ventilation and with non-zero by-pass factor (*Section 30.2.2*)

5. Analysis of a summer air conditioning system with re-heat for high latent cooling load applications (*Section 30.2.3*)

6. Selection guidelines for supply air conditions (*Section 30.3*)

At the end of the lesson, the student should be able to:

1. Estimate the load on the cooling coil and fix the supply conditions for various summer conditioning systems, namely:

a) Systems with 100% re-circulation
b) Systems with outdoor air for ventilation with zero by-pass factor
c) Systems with outdoor air for ventilation with non-zero by-pass factor
d) Systems with reheat for high latent cooling load applications

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30.1. Introduction:

Generally from the building specifications, inside and outside design conditions; the latent and sensible cooling or heating loads on a building can be estimated. Normally, depending on the ventilation requirements of the building, the required outdoor air (fresh air) is specified. The topic of load estimation will be discussed in a later chapter. From known loads on the building and design inside and outside conditions, psychrometric calculations are performed to find:

1. Supply air conditions (air flow rate, DBT, humidity ratio & enthalpy)

2. Coil specifications (Latent and sensible loads on coil, coil ADP & BPF)

In this chapter fixing of supply air conditions and coil specifications for summer air conditioning systems are discussed. Since the procedure is similar for winter air conditioning system, the winter air conditioning systems are not discussed here.

30.2. Summer air conditioning systems:

30.2.1. Simple system with 100 % re-circulated air:

In this simple system, there is no outside air and the same air is recirculated as shown in Fig.30.1. Figure 30.2 also shows the process on a psychrometric chart. It can be seen that cold and dry air is supplied to the room and the air that leaves the condition space is assumed to be at the same conditions as that of the conditioned space. The supply air condition should be such that as it flows through the conditioned space it can counteract the sensible and latent heat transfers taking place from the outside to the conditioned space, so that the space can be maintained at required low temperature and humidity. Assuming no heat gains in the supply and return ducts and no energy addition due to fans, and applying energy balance across the room; the Room Sensible Cooling load ($Q_{s,r}$), Room Latent Cooling Load ($Q_{l,r}$) and Room Total Cooling load ($Q_{t,r}$) are given by:

$$Q_{s,r} = m_s C_{pm}(t_i - t_s) \tag{30.1}$$

$$Q_{l,r} = m_s h_{fg}(W_i - W_s) \tag{30.2}$$

$$Q_{t,r} = Q_{s,r} + Q_{l,r} = m_s(h_i - h_s) \tag{30.3}$$

From cooling load calculations, the sensible, latent and total cooling loads on the room are obtained. Hence one can find the Room Sensible Heat Factor (RSHF) from the equation:

$$RSHF = \frac{Q_{s,r}}{Q_{s,r} + Q_{l,r}} = \frac{Q_{s,r}}{Q_{t,r}} \tag{30.4}$$

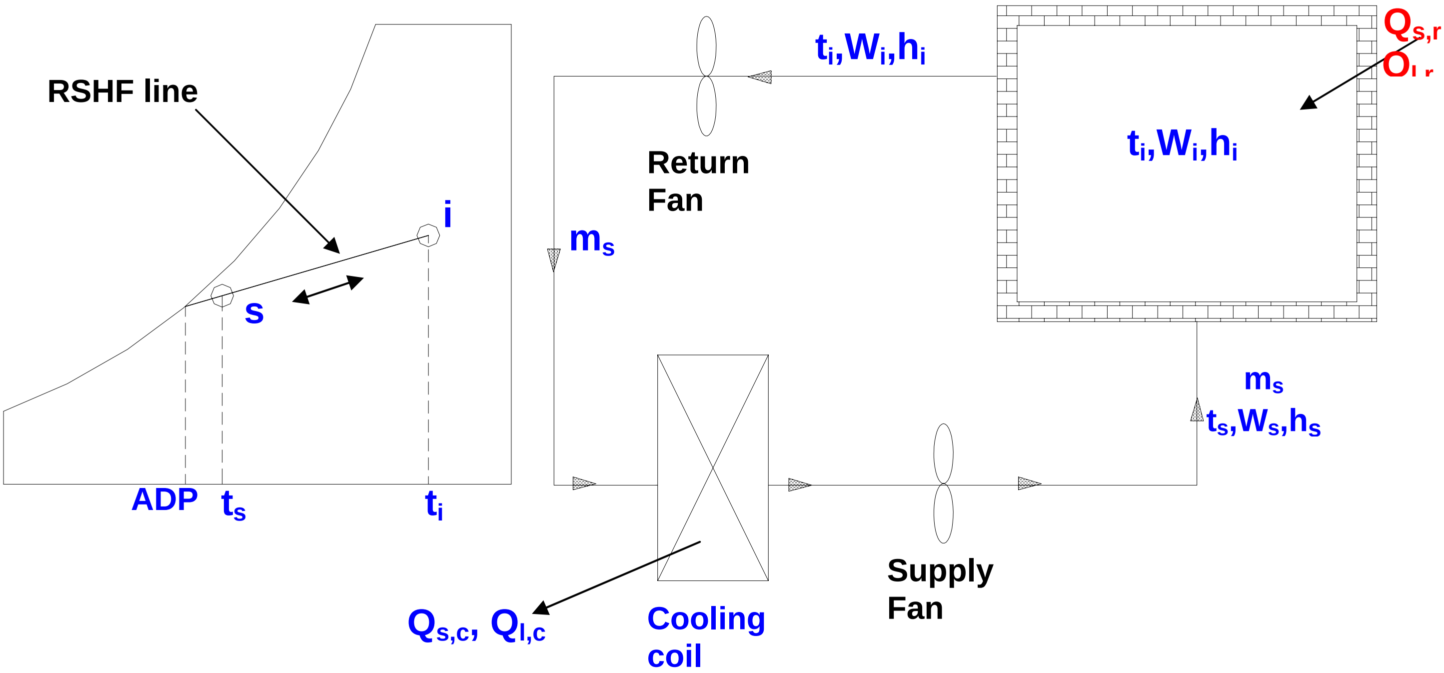


***Fig.30.1**: A simple, 100% re-circulation type air conditioning system*

From the RSHF value one can calculate the slope of the process undergone by the air as it flows through the conditioned space (process s-i) as:

$$\textbf{slope of process line s-i}, \tan\theta = \frac{1}{2451}\left(\frac{1-RSHF}{RSHF}\right) \qquad (30.5)$$

Since the condition i is known say, from thermal comfort criteria, knowing the slope, one can draw the process line s-i through i. The intersection of this line with the saturation curve gives the ADP of the cooling coil as shown in Fig.30.1. It should be noted that for the given room sensible and latent cooling loads, **the supply condition must always lie on this line so that the it can extract the sensible and latent loads on the conditioned space in the required proportions**.

Since the case being considered is one of 100 % re-circulation, the process that the air undergoes as it flows through the cooling coil (i.e. process i-s) will be exactly opposite to the process undergone by air as it flows through the room (process s-i). Thus, the temperature and humidity ratio of air decrease as it flows through the cooling coil and temperature and humidity ratio increase as air flows through the conditioned space. Assuming no heat transfer due to the ducts and fans, the sensible and latent heat transfer rates at the cooling coil are exactly equal to the sensible and latent heat transfer rates to the conditioned space; i.e.,

$$Q_{s,r} = Q_{s,c} \;\&\; Q_{l,r} = Q_{l,c} \qquad (30.6)$$

Fixing of supply condition:

The supply condition has to be fixed using Eqns.(30.1) to (30.3). However, since there are **4 unknowns** (m_s, t_s, W_s and h_s) and **3 equations**, (Eqns.(30.1) to (30.3)), one parameter has to be fixed to find the other three unknown parameters from the three equations.

If the by-pass factor (X) of the cooling coil is known, then, from room conditions, coil ADP and by-pass factor, the supply air temperature t_s is obtained using the definition of by-pass factor as:

$$X = \left(\frac{t_s - t_{ADP}}{t_i - t_{ADP}}\right) \Rightarrow t_s = t_{ADP} + X(t_i - t_{ADP}) \tag{30.7}$$

Once the supply temperature t_s is known, then the mass flow rate of supply air is obtained from Eqn.(30.1) as:

$$m_s = \frac{Q_{s,r}}{c_{pm}(t_i - t_s)} = \frac{Q_{s,r}}{c_{pm}(t_i - t_{ADP})(1 - X)} \tag{30.8}$$

From the mass flow rate of air and condition i, the supply air humidity ratio and enthalpy are obtained using Eqns.(30.2) and (30.3) as:

$$W_s = W_i - \frac{Q_{l,r}}{m_s h_{fg}} \tag{30.9}$$

$$h_s = h_i - \frac{Q_{t,r}}{m_s} \tag{30.10}$$

From Eqn.(30.8), it is clear that the required mass flow rate of supply air decreases as the by-pass factor X decreases. In the limiting case when the by-pass factor is zero, the minimum amount of supply air flow rate required is:

$$m_{s,min} = \frac{Q_{s,r}}{c_{pm}(t_i - t_{ADP})} \tag{30.11}$$

Thus with 100 % re-circulated air, the room ADP is equal to coil ADP and the load on the coil is equal to the load on the room.

30.2.2. System with outdoor air for ventilation:

In actual air conditioning systems, some amount of outdoor (fresh) air is added to take care of the ventilation requirements. Normally, the required outdoor air for ventilation purposes is known from the occupancy data and the type of the building (e.g. operation theatres require 100% outdoor air). Normally either the quantity of outdoor air required is specified in absolute values or it is specified as a fraction of the re-circulated air.

Fixing of supply condition:

Case i) By-pass factor of the cooling coil is zero:

Figure 30.2 shows the schematic of the summer air conditioning system with outdoor air and the corresponding process on psychrometric chart, when the by-pass factor X is zero. Since the sensible and latent cooling loads on the conditioned space are assumed to be known from cooling load calculations, similar to the earlier case, one can draw the process line s-i, from the RSHF and state i. The intersection of this line with the saturation curve gives the room ADP. As shown on the psychrometric chart, when the by-pass factor is zero, the room ADP is equal to coil ADP, which in turn is equal to the temperature of the supply air. Hence from the supply temperature one can calculate the required supply air mass flow rate (which is the minimum required as X is zero) using the equation:

$$m_s = \frac{Q_{s,r}}{C_{pm}(t_i - t_s)} = \frac{Q_{s,r}}{C_{pm}(t_i - t_{ADP})} \qquad (30.12)$$

From the supply mass flow rate, one can find the supply air humidity ratio and enthalpy using Eqns.(30.9) and (30.10).

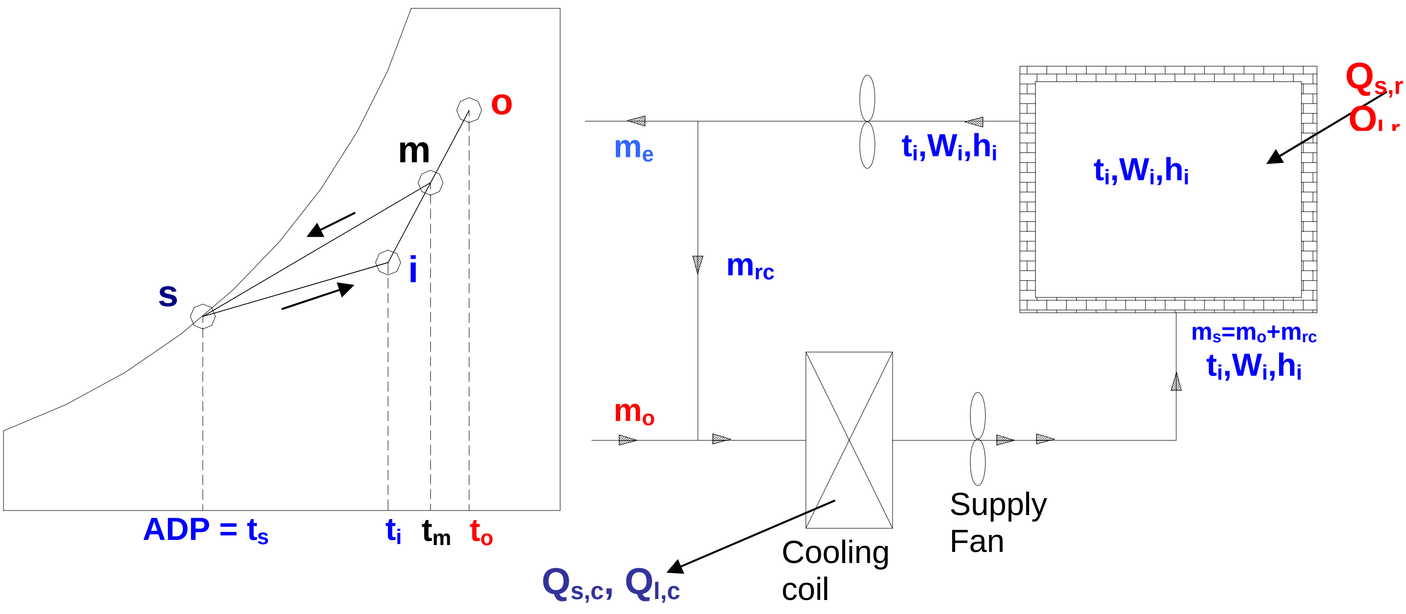


Fig.30.2: A summer air conditioning system with outdoor air for ventilation and a zero by-pass factor

From mass balance of air;

$$m_s = m_{rc} + m_o \qquad (30.13)$$

Where m_{rc} is the re-circulated air flow rate and m_o is the outdoor air flow rate. Since either m_o or the ratio $m_o : m_{rc}$ are specified, one can calculate the amount of re-circulated air from Eqn.(30.13).

Calculation of coil loads:

From energy balance across the cooling coil; the sensible, latent and total heat transfer rates, $Q_{s,c}$, $Q_{l,c}$ and $Q_{t,c}$ at the cooling coil are given by:

$$\begin{aligned} Q_{s,c} &= m_s C_{pm}(t_m - t_s) \\ Q_{l,c} &= m_s h_{fg}(W_m - W_s) \\ Q_{t,c} &= Q_{s,c} + Q_{l,c} = m_s(h_m - h_s) \end{aligned} \qquad (30.14)$$

Where 'm' refers to the mixing condition which is a result of mixing of the recirculated air with outdoor air. Applying mass and energy balance to the mixing process one can obtain the state of the mixed air from the equation:

$$\frac{m_o}{m_s} = \frac{W_m - W_i}{W_o - W_i} = \frac{h_m - h_i}{h_o - h_i} \approx \frac{t_m - t_i}{t_o - t_i} \qquad (30.15)$$

Since $(m_o/m_s) > 0$, from the above equation it is clear that $W_m > W_i$, $h_m > h_i$ and $t_m > t_i$. This implies that $\mathbf{m_s(h_m - h_s) > m_s(h_i - h_s)}$, or the load on the cooling coil is greater than the load on the conditioned space. This is of course due to the fact that during mixing, some amount of hot and humid air is added and the same amount of relative cool and dry air is exhausted ($m_o = m_e$).

From Eqn.(30.1) to (30.3) and (30.14), the difference between the cooling load on the coil and cooling load on the conditioned space can be shown to be equal to:

$$\begin{aligned} Q_{s,c} - Q_{s,r} &= m_o C_{pm}(t_o - t_i) \\ Q_{l,c} - Q_{l,r} &= m_o h_{fg}(W_o - W_i) \\ Q_{t,c} - Q_{t,r} &= m_o(h_o - h_i) \end{aligned} \qquad (30.16)$$

From the above equation it is clear that the difference between cooling coil and conditioned space increases as the amount of outdoor air (m_o) increases and/or the outdoor air becomes hotter and more humid.

The line joining the mixed condition 'm' with the coil ADP is the process line undergone by the air as it flows through the cooling coil. The slope of this line depends on the Coil Sensible Heat Factor (CSHF) given by:

$$\text{CSHF} = \frac{Q_{s,c}}{Q_{s,c} + Q_{l,c}} = \frac{Q_{s,c}}{Q_{t,c}} \qquad (30.17)$$

<u>Case ii: Coil by-pass factor, X > 0:</u>

For actual cooling coils, the by-pass factor will be greater than zero, as a result the air temperature at the exit of the cooling coil will be higher than the coil ADP. This is shown in Fig.30.3 along with the process on psychrometric chart. It can be seen from the figure that when X > 0, the room ADP will be different from the coil ADP. The system shown in Fig.30.3 is adequate when the RSHF is high (> 0.75).

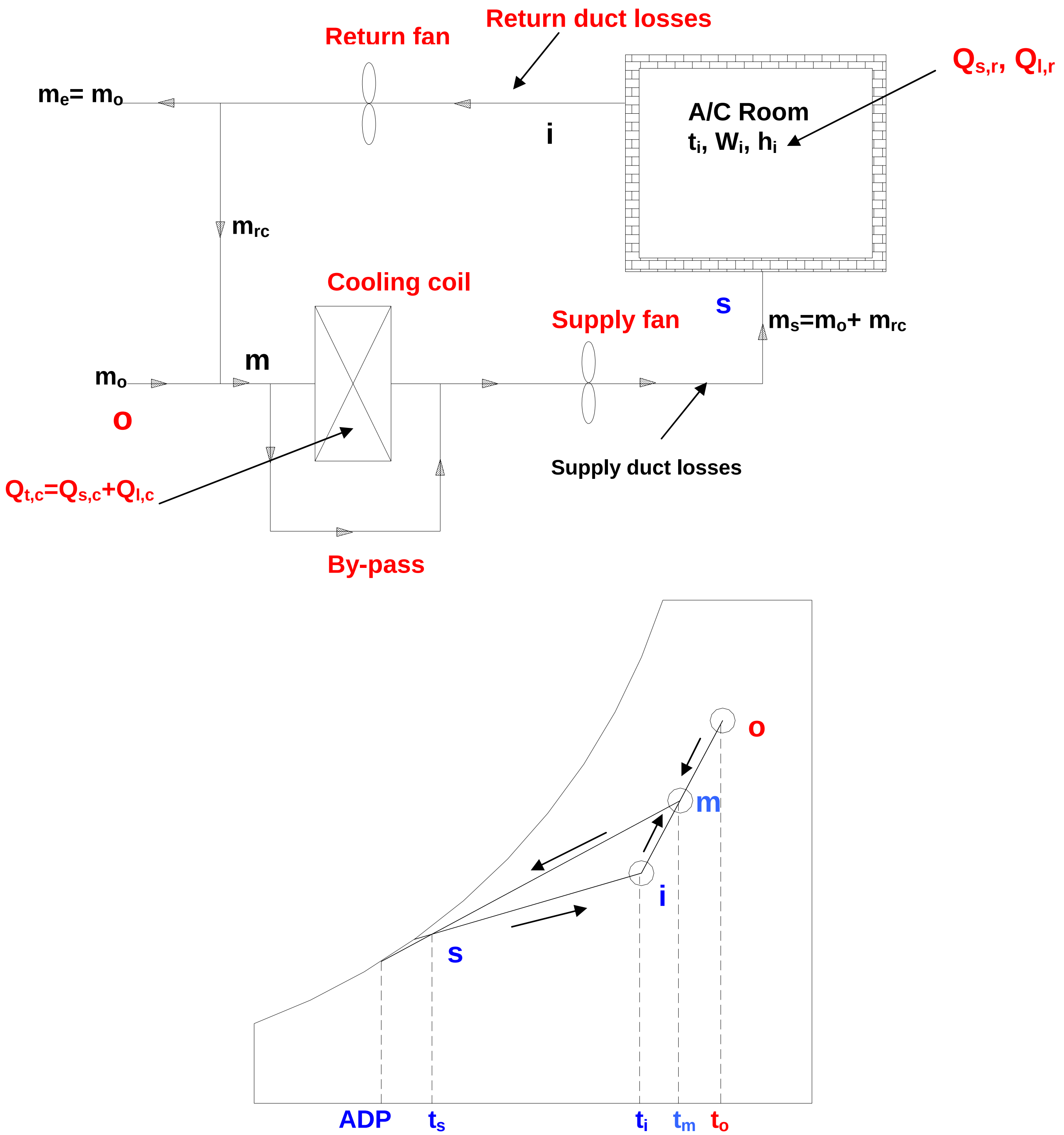


Fig.30.3: *A summer air conditioning system with outdoor air for ventilation and a non-zero by-pass factor*

Normally in actual systems, either the supply temperature (t_s) or the temperature rise of air as it flows through the conditioned space (t_i-t_s) will be specified. Then the step-wise procedure for finding the supply air conditions and the coil loads are as follows:

i. Since the supply temperature is specified one can calculate the required supply air flow rate and supply conditions using Eqns. (30.8) to (30.10).

ii. Since conditions 'i', supply air temperature t_s and RSHF are known, one can draw the line i-s. The intersection of this line with the saturation curve gives the room ADP.

iii. Condition of air after mixing (point 'm') is obtained from known values of m_s and m_o using Eqn.(30.15).

iv. Now joining points 'm' and 's' gives the process line of air as it flows through the cooling coil. The intersection of this line with the saturation curve gives the coil ADP. It can be seen that the coil ADP is lower than the room ADP.

v. The capacity of the cooling coil is obtained from Eqn.(30.14).

vi. From points 'm', 's' and coil ADP, the by-pass factor of the cooling coil can be calculated.

If the coil ADP and coil by-pass factor are given instead of the supply air temperature, then a trial-and-error method has to be employed to obtain the supply air condition.

30.2.3. High latent cooling load applications (low RSHF):

When the latent load on the building is high due either to high outside humidity or due to large ventilation requirements (e.g. hospitals) or due to high internal latent loads (e.g. presence of kitchen or laundry), then the simple system discussed above leads to very low coil ADP. A low coil ADP indicates operation of the refrigeration system at low evaporator temperatures. Operating the system at low evaporator temperatures decreases the COP of the refrigeration system leading to higher costs. Hence a reheat coil is sometimes used so that the cooling coil can be operated at relatively high ADP, and at the same time the high latent load can also be taken care of. Figure 30.4 shows an air conditioning system with reheat coil along with the psychrometric representation of the process. As shown in the figure, in a system with reheat coil, air is first cooled and dehumidified from point 'm' to point 'c' in the cooling coil and is then reheated sensibly to the required supply temperature t_s using the reheat coil. If the supply temperature is specified, then the mass flow rate and state of the supply air and condition of the air after mixing can be obtained using equations given above. Since the heating process in the reheat coil is sensible, the process line c-s will be horizontal. Thus if the coil ADP is known, then one can draw the coil condition line and the intersection of this line with the horizontal line drawn from supply state 's' gives the condition of the air at the exit of the cooling coil. From this condition, one can calculate the load on the cooling coil using the supply mass flow rate and state of air after mixing. The capacity of the reheat coil is then obtained from energy balance across it, i.e.,

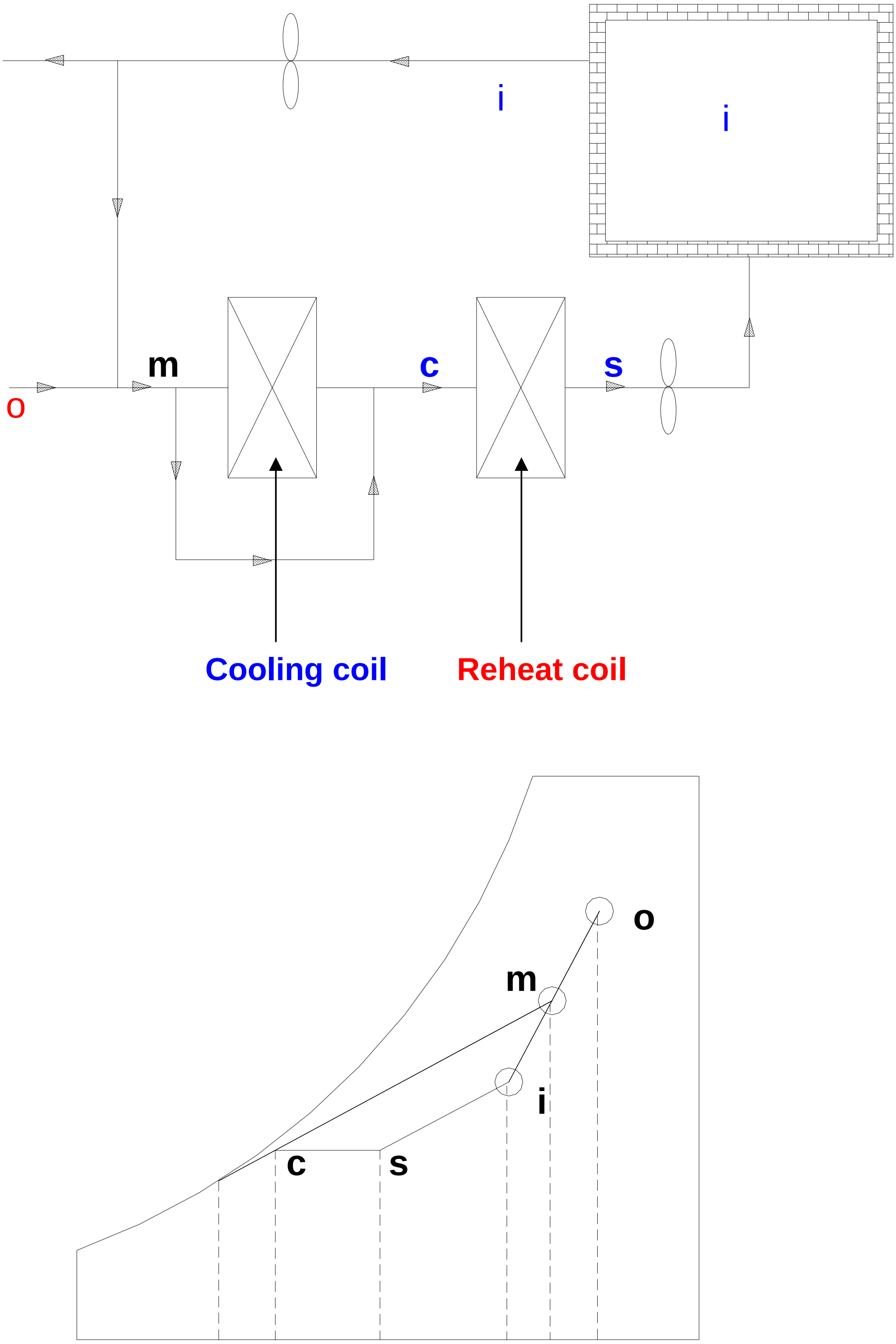


__Fig.30.4__: A summer air conditioning system with reheat coil for high latent cooling load applications

Advantages and disadvantages of reheat coil:

a) Refrigeration system can be operated at reasonably high evaporator temperatures leading to high COP and low running cost.
b) However, mass flow rate of supply air increases due to reduced temperature rise (t_i-t_s) across the conditioned space
c) Wasteful use of energy as air is first cooled to a lower temperature and then heated. Energy is required for both cooling as well as reheat coils. However, this can be partially offset by using waste heat such as heat rejected at the condenser for reheating of air.

Thus the actual benefit of reheat coil depends may vary from system.

30.3. Guidelines for selection of supply state and cooling coil:

i. As much as possible the supply air quantity should be minimized so that smaller ducts and fans can be used leading savings in cost of space, material and power. However, the minimum amount should be sufficient to prevent the feeling of stagnation. If the required air flow rate through the cooling coil is insufficient, then it is possible to mix some amount of re-circulated air with this air so that amount of air supplied to the conditioned space increases. This merely increases the supply air flow rate, but does not affect sensible and cooling loads on the conditioned space. Generally, the temperature rise (t_i-t_s) will be in the range of 8 to 15^{o}C.

ii. The cooling coil should have 2 to 6 rows for moderate climate and 6 to 8 rows in hot and humid climate. The by-pass factor of the coil varies from 0.05 to 0.2. The by-pass factor decreases as the number of rows increases and vice versa. The fin pitch and air velocity should be suitable.

iii. If chilled water is used for cooling and dehumidification, then the coil ADP will be higher than about 4^{o}C.

Questions and answers:

1. State which of the following statements are TRUE?

a) The purpose of psychrometric calculations is to fix the supply air conditions
b) The purpose of psychrometric calculations is to find the load on the building
c) In a 100% re-circulation system, the coil ADP is equal to room ADP
d) In a 100% re-circulation system, the coil ADP is less than room ADP

Ans.: a) and c)

2. State which of the following statements are TRUE?

a) In a 100% re-circulation system, the load on coil is equal to the load on building
b) In a system with outdoor air for ventilation, the load on building is greater than the load on coil
c) In a system with outdoor air for ventilation, the load on building is less than the load on coil
d) In a system with outdoor air for ventilation, the Coil ADP is less than room ADP

Ans.: a), c) and d)

3. Which of the following statements are TRUE?

a) Systems with reheat are used when the Room Sensible Heat Factor is low
b) Systems with reheat are used when the Room Sensible Heat Factor is high
c) When reheat coils are used, the required coil ADP can be increased
d) When reheat coils are used, the required supply airflow rate increases

Ans.: a), c) and d)

4. A 100% outdoor summer air conditioning system has a room sensible heat load of 400 kW and a room latent heat load of 100 kW. The required inside conditions are 24^{o}C and 50% RH, and the outdoor design conditions are 34^{o}C and 40% RH. The air is supplied to the room at a dry bulb temperature of 14^{o}C. Find a) the required mass flow rate of air b) moisture content of supply air, c) Sensible, latent heat loads on the coil, and d) The required cooling capacity of the coil, Coil Sensible Heat Factor and coil ADP if the by-pass factor of the coil is 0.2. Barometric pressure = 1 atm. Comment on the results.

Ans.: The psychrometric process for this system is shown in Fig.30.5.

The psychrometric properties at inside and outside conditions are:

Inside conditions: t_i =24^{o}C(DBT) and RH_i = 50%

From psychrometric chart or using psychrometric equations; the moisture content and enthalpy of inside air are:

$$\mathbf{W_i = 0.0093\ kgw/kgda,\ h_i = 47.66\ kJ/kgda}$$

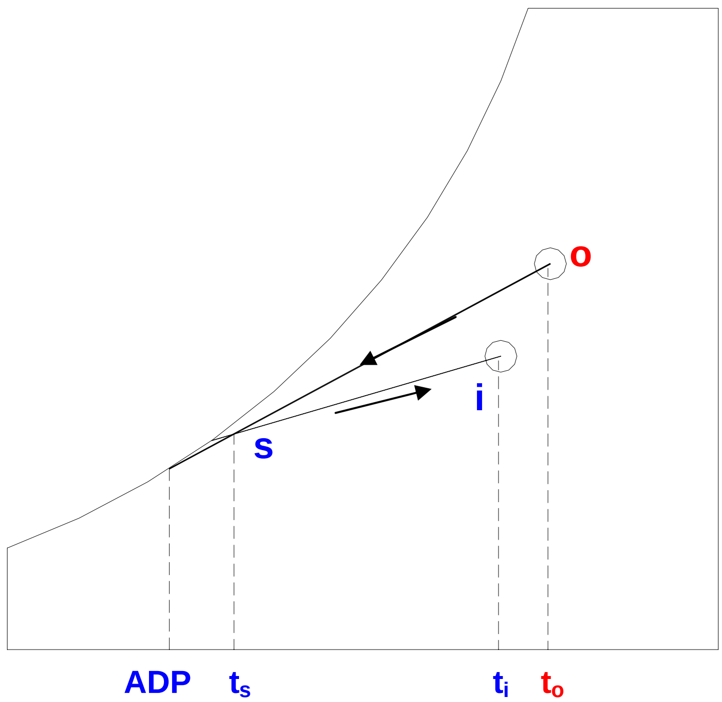


Fig.30.5: *A summer air conditioning system with 100% outdoor air*

<u>outside conditions</u>: $t_i = 34^oC$(DBT) and RH_i = 40%

From psychrometric chart or using psychrometric equations; the moisture content and enthalpy of inside air are:

$$W_o = 0.01335 \text{ kgw/kgda}, h_1 = 68.21 \text{ kJ/kgda}$$

a) From sensible energy balance equation for the room, we find the required mass flow rate of air as:

$$m_s = \frac{Q_{s,r}}{C_{pm}(t_i - t_s)} = \frac{400}{1.0216(24-14)} = 39.154 \text{ kg/s} \qquad \textbf{(Ans.)}$$

b) The moisture content of supply air is obtained from latent energy balance of the room as:

$$W_s = W_i - \frac{Q_{l,r}}{m_s h_{fg}} = 0.0093 - \frac{100}{39.154x2501} = 0.0083 \text{ kgw/kgda} \qquad \textbf{(Ans.)}$$

c) From energy balance, the sensible and latent loads on the coil are obtained as:

$$\begin{aligned} Q_{s,c} &= m_s C_{pm}(t_o - t_s) = 39.154 \,x\, 1.0216 \,x\, (34-14) = 800 \text{ kW} \\ Q_{l,c} &= m_s h_{fg}(W_o - W_s) = 39.154 \,x\, 2501 x\, (0.01335 - 0.0083) = 494.5 \text{ kW} \end{aligned} \qquad \textbf{(Ans.)}$$

d) The required cooling capacity of the coil is equal to the total load on the coil, $Q_{t,c}$:

$$Q_{t,c} = Q_{s,c} + Q_{l,c} = 800 + 494.5 = 1294.5 \text{ kW} \qquad \textbf{(Ans.)}$$

Coil Sensible Heat Factor, CSHF = $Q_{s,c}/Q_{t,c}$ = 0.618 **(Ans.)**

Coil ADP is obtained by using the definition of by-pass factor (X) as:

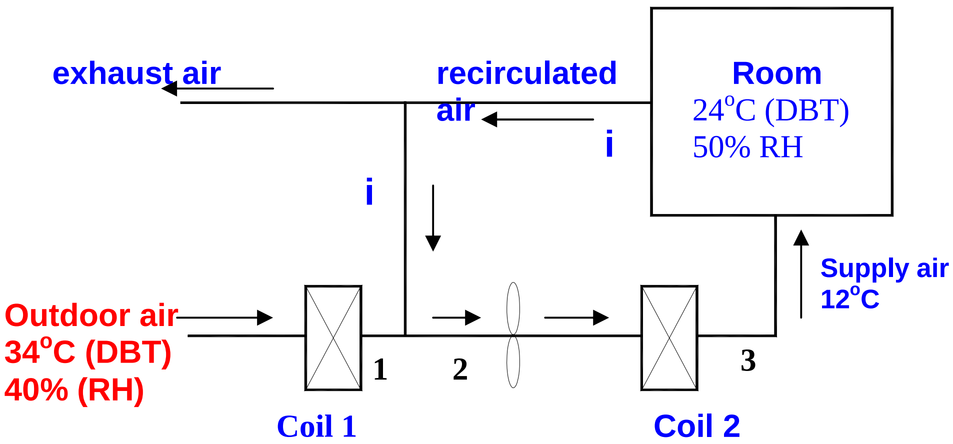
$$t_{ADP}(1 - X) = t_S - X.t_O$$

$$\Rightarrow t_{ADP} = (t_S - X.t_O)/(1-X) = (14 - 0.2 \times 34)/(1 - 0.2) = 9^oC \qquad \textbf{(Ans.)}$$

Comments:

1. It is seen that with 100% outdoor air, the load on the coil (or required cooling capacity of the coil) is much higher compared to the cooling load on the building (Required coil capacity = 1294.5 kW whereas the total load on the room is 500 kW). Since 100% outdoor air is used, the relatively cold and dry indoor air is exhausted without re-circulation and the hot and humid air is conditioned using the coil coil. Thus the required cooling capacity is very high as the cooling coil has to cool and dehumidify outdoor air.
2. It is observed that the CSHF (0.618) is much smaller compared to the room SHF (0.8), hence, the coil ADP is much smaller than the room ADP.

5. A room is air conditioned by a system that maintains 25°C dry bulb and 50 % RH inside, when the outside conditions are 34°C dry bulb and 40% RH. The room sensible and latent heat gains are 60 kW and 12 kW respectively. As shown in the figure below, The outside fresh air first flows over a first cooler coil and is reduced to state 1 of 10°C dry bulb and a relative humidity of 85%. It is then mixed with re-circulated air, the mixture (state 2) being handled by a fan, passed over a second cooler coil and sensibly cooled to 12°C dry bulb (state 3). The air is then delivered to the room. If the outside fresh air is used for dealing with the whole of the room latent heat gain and if the effects of fan power and duct heat gains are ignored, find: a) mass flow rates of outside fresh air and supply air; b) DBT and enthalpy of the air handled by the fan (state 2); and c) required cooling capacity of first cooler coil and second sensible cooler coil.



Ans.: From psychrometric chart, the following properties are obtained:

<u>Inside conditions:</u> t_i =24°C(DBT) and RH_i = 50%

$$W_i = 0.0093 \text{ kgw/kgda}, h_i = 47.66 \text{ kJ/kgda}$$

Outside conditions: t_o =34^{o}C(DBT) and RH_o = 40%

$$W_o = 0.01335 \text{ kgw/kgda}, h_i = 68.21 \text{ kJ/kgda}$$

At state 1: t_1 = 10^{o}C(DBT) and RH_1 = 85%

$$W_1 = 0.00647 \text{ kgw/kgda}, h_1 = 26.31 \text{ kJ/kgda}$$

a) Since the air is supplied to the room at 12^{o}C, the mass flow rate of supply air m_3 is obtained from sensible energy balance across the room, i.e.,

$$m_3 = \frac{Q_{s,r}}{C_{pm}(t_i - t_3)} = \frac{60}{1.0216(24-12)} = 4.894 \text{ kg/s} \qquad \textbf{(Ans.)}$$

The moisture content of supply air is obtained from latent energy balance across the room as:

$$W_s = W_i - \frac{Q_{l,r}}{m_3 h_{fg}} = 0.0093 - \frac{12}{4.894x2501} = 0.0083 \text{ kgw/kgda}$$

Since the fresh air takes care of the entire latent load, the heat transfer across coil 2 is only sensible heat transfer. This implies that:

$$W_2 = W_3 = 0.0083 \text{ kgw/kgda}$$

Applying mass balance across the mixing of re-circulated and fresh air (1-2), we obtain:

$$m_1W_1+(m_2-m_1)W_i = m_2.W_2$$

From the above equation, we get m_1 as:

$$m_1= m_2(W_i - W_2)/(W_i - W_1) = 1.73 \text{ kg/s}$$

Hence the mass flow rate of re-circulated air is:

$$m_{rc} = m_2 - m_1 = (4.894 - 1.73) = 3.164 \text{ kg/s}$$

b) From energy balance across the mixing process 1-2, assuming the variation in cpm to be negligible, the temperature of mixed air at 2 is given by:

$$t_2 = (m_1t_1 + m_{rc}t_i)/m_2 = 19.05^oC \qquad \textbf{(Ans.)}$$

From total enthalpy balance for the mixing process, the enthalpy of mixed air at 2 is:

$$h_2 = (m_1h_1 + m_{rc}h_i)/m_2 = 40.11 \text{ kJ/kgda} \qquad \textbf{(Ans.)}$$

c) From energy balance, cooling capacity of 1st cooler coil is given by:

$$Q_{c,1} = m_1(h_o - h_1) = 1.73 \times (68.21 - 26.31) = 72.49 \text{ kW} \qquad \textbf{(Ans.)}$$

From energy balance across the 2nd cooler coil, the cooling capacity of the second coil is given by:

$$Q_{c,2} = m_2.c_{pm}(t_2 - t_3) = 4.894 \times 1.0216 \times (19.05 - 12.0) = 35.25 \text{ kW} \qquad \textbf{(Ans.)}$$

Comment: It can be seen that the combined cooling capacity (**72.49 + 35.25 = 107.74 kW**) is larger than the total cooling load on the building (**60 + 12 = 72 kW**). The difference between these two quantities (**107.74 – 72 = 35.74 kW**) is equal to the cooling capacity required to reduce the enthalpy of the fresh air from outdoor conditions to the required indoor conditions. This is the penalty one has to pay for providing fresh air to the conditioned space. Larger the fresh air requirement, larger will be the required cooling capacity.

6) An air conditioned building has a sensible cooling load of 60 kW and latent load of 40 kW. The room is maintained at 24^{o}C (DBT) and 50% RH, while the outside design conditions are: 34^{o}C (DBT) and 40% RH. To satisfy the ventilation requirement, outdoor air is mixed with re-circulated air in the ratio of 1:3 (by mass). Since the latent load on the building is high, a reheat coil is used along with a cooling and dehumidifying coil. Air is supplied to the conditioned space at 14^{o}C (DBT). If the by-pass factor of the cooling coil is 0.15 and the barometric pressure is 101.325 kPa, find: a) Mass flow rate of supply air, b) Required cooling capacity of the cooling coil and heating capacity of the reheat coil

Ans.: From psychrometric chart, the following properties are obtained:

<u>Inside conditions:</u> t_i =24^{o}C(DBT) and RH_i = 50%

$$W_i = 0.0093 \text{ kgw/kgda}, \; h_i = 47.66 \text{ kJ/kgda}$$

<u>outside conditions</u>: t_o =34^{o}C(DBT) and RH_o = 40%

$$W_o = 0.01335 \text{ kgw/kgda}, \; h_i = 68.21 \text{ kJ/kgda}$$

Since the air is supplied to the room at 12^{o}C, the mass flow rate of supply air m_3 is obtained from sensible energy balance across the room, i.e.,

$$m_3 = \frac{Q_{s,r}}{C_{pm}(t_i - t_s)} = \frac{60}{1.0216(24-14)} = 5.873 \text{ kg/s} \qquad \textbf{(Ans.)}$$

The moisture content of supply air is obtained from latent energy balance across the room as:

$$W_s = W_i - \frac{Q_{l,r}}{m_3 h_{fg}} = 0.0093 - \frac{40}{5.873 \times 2501} = 0.0066 \text{ kgw/kgda}$$

Since 25% of the supply air is fresh air, the mass flow rates of fresh and re-circulated air are:

m_o = 0.25 x 5.873 = 1.468 kg/s and m_{rc} = 0.75 x 5.873 = 4.405 kg/s (Ans.)

b) From sensible energy balance for the mixing process of fresh air with re-circulated air (Fig.30.4), we obtain the mixed air conditions as:

$$t_m = (m_o.t_o + m_{rc}.t_i)/(m_o+m_{rc}) = 26.5^oC$$

$$W_m = (m_o.W_o + m_{rc}.W_i)/(m_o+m_{rc}) = 0.0103 \text{ kgw/kgda}$$

$$h_m = (m_o.h_o + m_{rc}.h_i)/(m_o+m_{rc}) = 52.75 \text{ kJ/kgda}$$

Since heating in the reheat coil is a sensible heating process, the moisture content of air remains constant during this process. Then from Fig.30.4., writing the by-pass factor in terms of humidity ratios as:

$$X = \frac{(W_s - W_{ADP})}{(W_m - W_{ADP})} = \frac{(0.0066 - W_{ADP})}{(0.0103 - W_{ADP})} = 0.15$$

From the above expression, the humidity ratio at coil ADP condition is found to be:

$$W_{ADP}=(W_s - X.W_m)/(1-X) = (0.0066 - 0.15x0.0103)/(1.0-0.15) = 0.00595 \text{ kgw/kgda}$$

The Coil ADP is the saturation temperature corresponding to a humidity ratio of W_{ADP}, hence, from psychrometric chart or using psychrometric equations, it is found to be:

$$t_{ADP} = 6.38^oC$$

Hence, the temperature of air at the exit of the cooling coil (t_C in Fig.30.4) is obtained from the by-pass factor as:

$$t_c = t_{ADP} + X\,(t_m - t_{ADP}) = 9.4^oC$$

From W_c (= W_s) and t_c, the enthalpy of air at the exit of the cooling coil is found from psychrometric chart as:

$$h_c = 26.02 \text{ kJ/kgda}$$

Hence, from energy balance across cooling coil and reheater:

Required capacity of cooling coil, $Q_c = m_s(h_m - h_c) = 157.0$ kW (Ans.)

Required capacity of reheat coil, $Q_{rh} = m_s c_{pm}(t_s - t_c) = 27.6$ kW (Ans.)