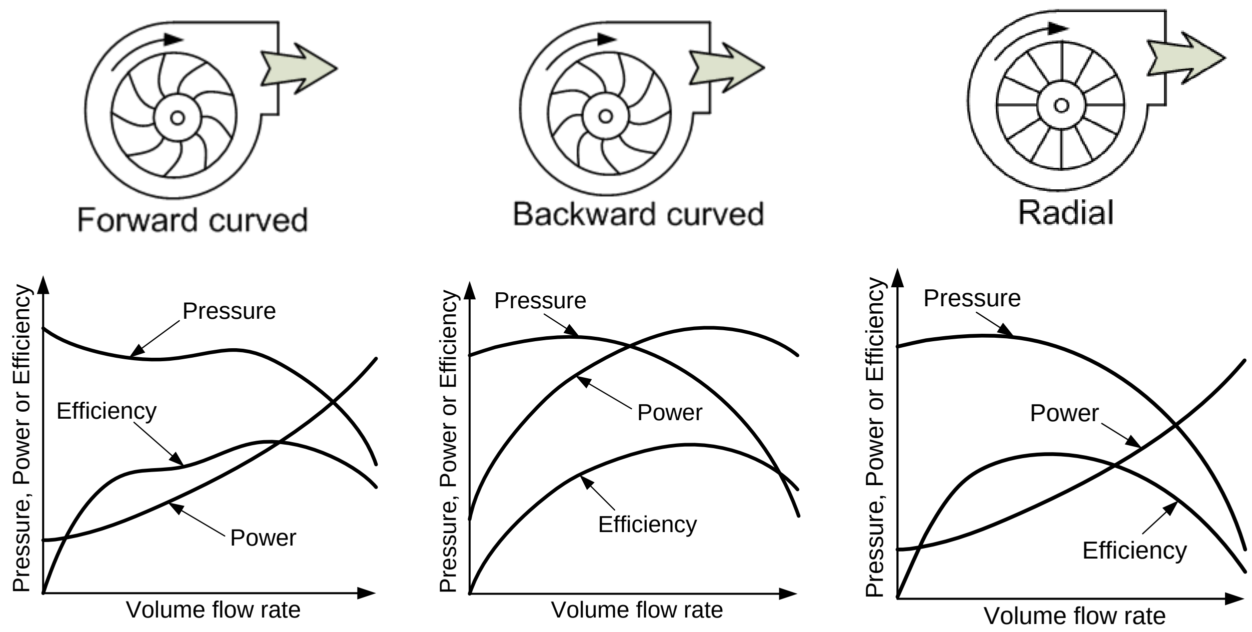
Figure 3.8 Velocity components of air for different types of fans [V = Absolute velocity of air leaving blade (Shown equal for all three blade types); V_r = Velocity of air leaving blade relative to blade; V_b = Velocity of blade tip]

Table 3.1 Typical performances of different types of fans

	Forward	Radial	Backward	Axial
Air flow	High	Medium	Low	High
Pressure	Low	Medium	High	Low
Efficiency	50 to 70%	55 to 75%	65 to 85%	60 to 80%

3.4 Fan and System Characteristic Curves

Fan Characteristic Curve: The relationship between the volume flow rate of air and the pressure developed by a fan is known as fan characteristic curve or fan curve. Fan manufacturers generally include the variation of fan input power and efficiency in the fan characteristic curve. Typical characteristic curves for the forward curved, backward curved and radial fans are shown in Figure-3.9. For axial flow fans with adjustable blade angle, characteristic curve changes with the change of blade angle. For a specific flow rate of air and pressure loss of the ducting system, characteristic curves of different types of fans should be analysed carefully to select the fan system.



(a) (b) (c)

Figure 3.9 Typical characteristic curves for (a) forward curved, (b) backward curved and (c) radial fans

Fan characteristic curve, similar to the pump characteristic curve, moves downward if the diameter or speed of the impeller is reduced as shown in Figure-3.10.

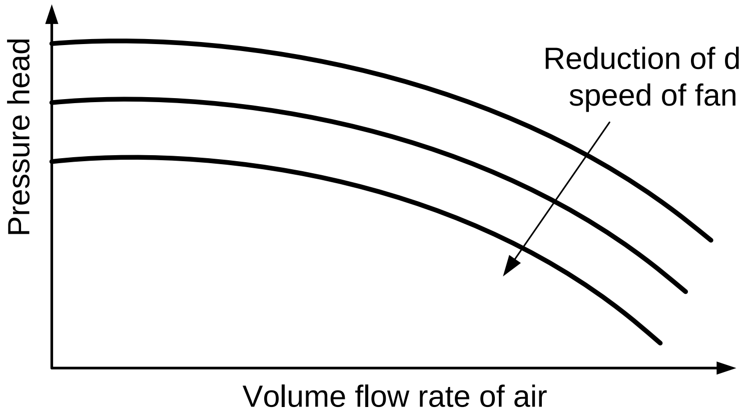


Figure 3.10 Fan characteristic curves for different diameters or speeds of impeller

Fan performance is often presented by the manufacturers in the form of a set of curves at different speeds for a particular impeller diameter as shown in Figure-3.11. The set of the performance curves changes with the change of impeller diameter.

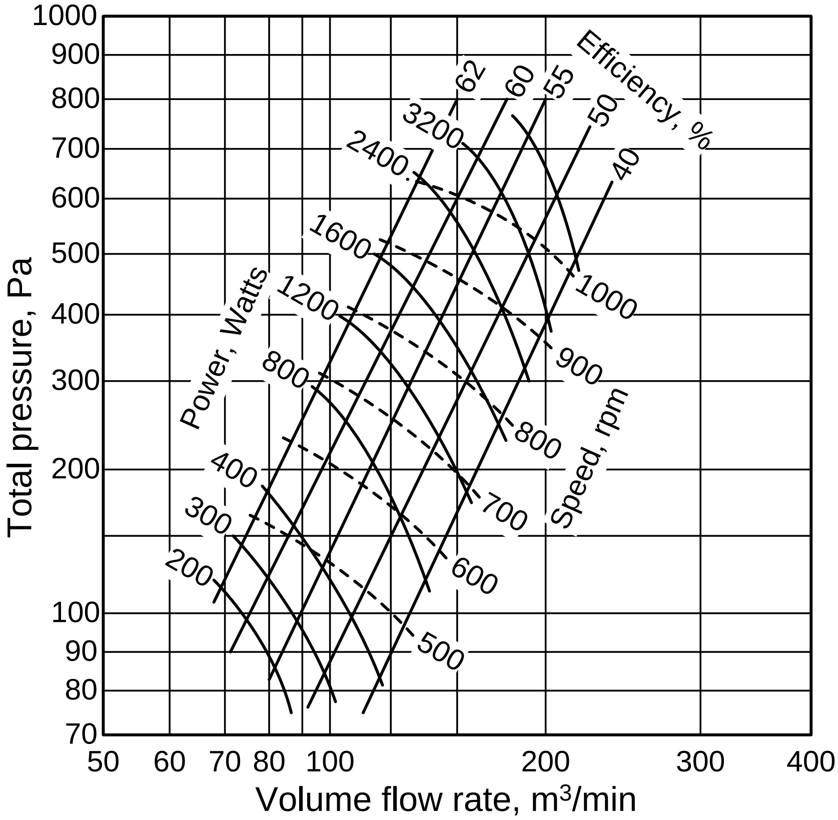


Figure 3.11 Fan performance curves at different speeds

System Characteristic Curve: When air flows through the ducting systems, there is a pressure drop due to the friction losses and the dynamic losses. The friction losses in the ducts caused by the viscous effects of the flowing air can be expressed using Darcy-Weisbach equation as:

$$\Delta P_{friction} = f \frac{L}{D} \frac{V^2}{2g} \tag{3.1}$$

where

$\Delta P_{friction}$ = friction loss in the duct, m

f = friction factor, dimensionless

L = length of the duct, m

D = hydraulic diameter of the duct, m

V = average velocity of air, m/s

g = acceleration due to gravity, 9.81 m/s^2

The hydraulic diameter of the duct is calculated as:

$$D = \frac{Cross\ sectional\ area\ of\ duct}{Perimeter} \tag{3.2}$$

The friction factor f depends on ducting material and the roughness of internal surface of the ducts. Average flow velocity of air in a duct is calculated as:

$$V = \frac{Q}{A} \tag{3.3}$$

where

Q = volume flow rate of air, m^3/s

A= cross sectional area of duct, m^2

Combining Eqs. (3.1) and (3.3) gives:

$$\Delta P_{friction} = f \frac{L}{D} \frac{Q^2}{2gA^2} \tag{3.4}$$

Dynamic losses for the fittings such as dampers, elbows, converging flow fittings, diverging flow fittings, tee-joints etc. of the ducting systems are calculated as:

$$\Delta P_{fitting} = C_o \frac{V^2}{2g}$$

$$\Delta P_{fitting} = C_o \frac{Q^2}{2gA^2} \tag{3.5}$$

where

$\Delta P_{fitting}$ = dynamic losses in fittings, m

C_o = loss coefficient of the fitting, dimensionless

V = average velocity of air, m/s

Q = volume flow rate of air, m^3/s

g = acceleration due to gravity, 9.81 m/s^2

The loss coefficient C_o depends on the type and size of the fittings. Summation of the friction and dynamic losses of the ducts represent the total pressure drop of a ducting system. If a fan is installed in the ducting system, the fan must produce pressure head equal to the total pressure drop of the ducting system to maintain the flow. Total pressure drop can be expressed in Pa as:

$$\Delta P_{total} = \left(\Delta P_{friction} + \Delta P_{fitting}\right)\rho g \tag{3.6}$$

where

ΔP_{total} = total pressure drop, Pa

$\Delta P_{friction}$ = friction losses in the duct, m

$\Delta P_{fitting}$ = fitting or dynamic losses in fittings, m

ρ = density of air, kg/m^3

g = acceleration due to gravity, 9.81 m/s^2

Based on Eqs. (3.4), (3.5) and (3.6), the friction, fitting and total pressure drop of a ducting system are proportional to the square of the volume flow rate of air. Therefore, the plot of the total pressure drop of a ducting system versus volume flow rate, known as system curve, is approximately parabolic as shown in Figure-3.12. Thus, the equation of the system curve of a ducting system can be expressed in general as:

$$\Delta P \propto Q^2 \tag{3.7}$$

$$\Delta P = CQ^2 \tag{3.8}$$

where

C = constant of proportionality

Each ducting system has its own system curve due to its own unique duct sizing, ducting geometry and fittings. The system curve of a ducting system will change, if any of these components, such as percentage opening of damper, is changed.

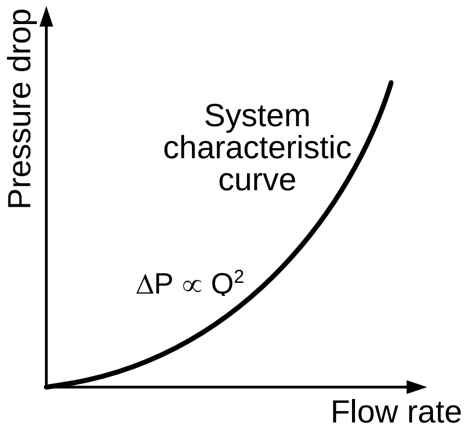


Figure-3.12 System curve for ducting system

Steps for Developing the System Curve of Ducting Systems: To develop the system curve of an existing ducting system, pressures at the inlet $P_{in,m}$ and exit $P_{out,m}$ of the fan and air flow rate Q_m should be measured as shown in Figure-3.13. As the total pressure drop of the ducting system is equal to the pressure head generated by the fan, total pressure drop of the ducting system is:

$$\Delta P_m = P_{out,m} - P_{in,m} \tag{3.9}$$

where

ΔP_m = measured pressure drop of ducting system, Pa

$P_{out,m}$ = measured pressure at the outlet of the fan, Pa

$P_{in,m}$ = measured pressure at the inlet of the fan, Pa

Putting the value of the measured total pressure drop ΔP_m and air flow rate Q_m in Eq. (3.8) gives:

$$\Delta P_m = CQ_m^2$$

$$C = \frac{\Delta P_m}{Q_m^2} \tag{3.10}$$

where

Q_m = measured air flow rate, m^3/s

Combining Eqs. (3.8) and (3.10) gives the equation of the system curve as:

$$\Delta P = \frac{\Delta P_m}{Q_m^2} Q^2 \tag{3.11}$$

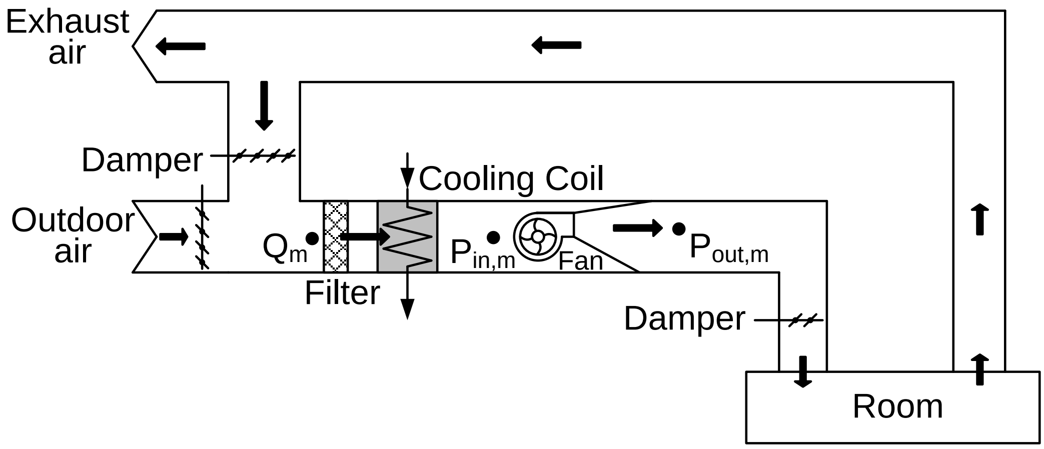


Figure 3.13 Measurement of pressure and air flow rate in ducting system

For a new ducting system, the total pressure drop for the corresponding design flow rate of air needs to be calculated. The system curve can then be determined following the same procedure presented above. If none of the ducting system parameters such as the duct size, ducting geometry, fittings, opening of the dampers etc. is changed; the equation and profile of the system curve will remain unchanged. The total pressure drop of the ducting system will change from ΔP_m to ΔP_1 if the flow rate of air through the ducting system is changed from Q_m to Q_1. The system pressure drop ΔP_1 for the air flow rate of Q_1 can be calculated using Eq. (3.11) as:

$$\Delta P_1 = \frac{\Delta P_m}{Q_m^2} Q_1^2 \tag{3.12}$$

If any of the ducting system parameters such as the opening of the damper is changed, the system curve will move either to the left or right as shown in Figure-3.12 and the equation of the new system curve needs to be redeveloped following the steps discussed above. The system curve moves to the left if the system pressure is increased due to the closing of the damper. Similarly, the system curve moves to the right if the system pressure is decreased due to the opening of the damper. The changes of the system pressure drop with the flow rate of air and the movement of system curve with the opening of the damper are shown in Figure-3.14.

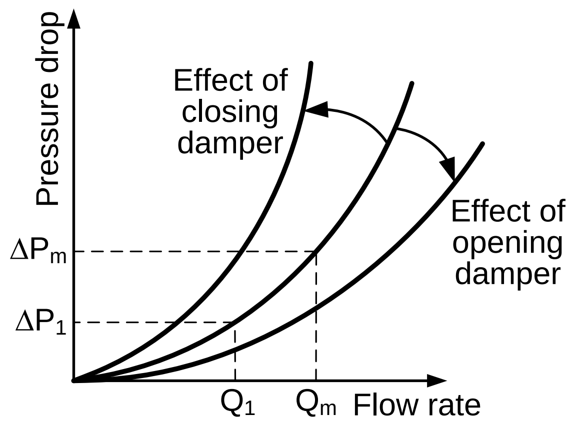


Figure 3.14 Changes of the system pressure drop with the flow rate of air and the movement of system curve with the opening of the damper

3.5 Fan Sizing

Similar to the water pumping systems, the steady and unit mass flow rate of air in a perfectly insulated ducting system as shown in Figure-3.15 is governed by the first law of thermodynamics, which leads to the equation:

$$\frac{P_1}{\rho_1 g}+\frac{V_1^2}{2g}+Z_1+\frac{\dot{W}_{fan}}{g}=\frac{P_2}{\rho_2 g}+\frac{V_2^2}{2g}+Z_2+e_{loss} \qquad (3.13)$$

where

P = static pressure, Pa

ρ = density of flowing air, kg/m^3

g = acceleration due to gravity, 9.81 m/s^2

V = average flow velocity, m/s

Z = elevation from a datum line, m

$\dot{W}_{fan}$ = fan output power to maintain unit mass flow rate of air, W/(kg/s)

e_{loss} = pressure loss in ducting system, m

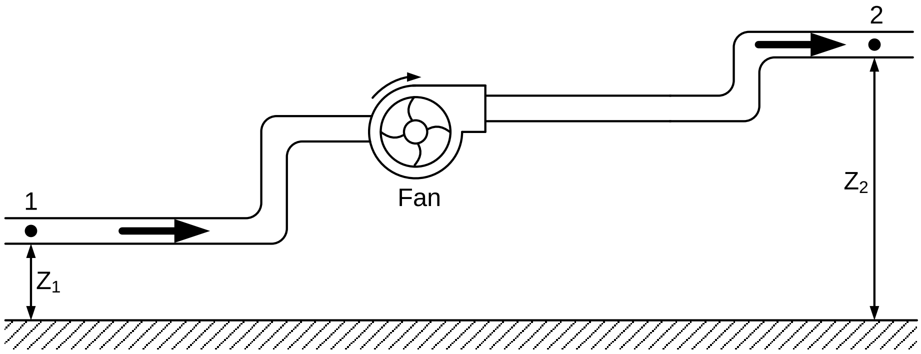


Figure 3.15 Steady flow of air in a perfectly insulated ducting system

Eq. (3.13) can be applied across the fan of AHUs to calculate the required power of the fans for the unit mass flow rate of air. Applying Eq. (3.13) across the inlet and outlet of the fan of an AHU as shown in Figure-3.16 gives:

$$\frac{P_{in}}{\rho_{in} g} + \frac{V_{in}^2}{2g} + Z_{in} + \frac{\dot{W}_{fan}}{g} = \frac{P_{out}}{\rho_{out} g} + \frac{V_{out}^2}{2g} + Z_{out} + e_{loss} \tag{3.14}$$

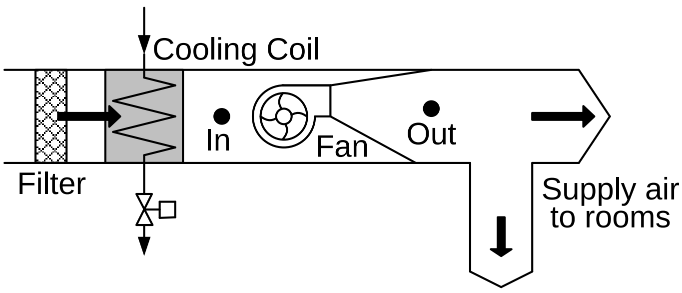


Figure 3.16 Inlet and outlet of fan of an AHU system

As the change of pressure of the flowing air is small for the fan systems, Eq. (3.14) can be simplified based on the following assumptions:

i. $\rho_{in} = \rho_{out} = \rho$ as the change of air density is negligible (McQuiston et al., 2005).

ii. $V_{in} = V_{out}$ as the diameter of the duct at the inlet and outlet of the fan is the same.

iii. $Z_{in} = Z_{out}$ as the elevation of the measurement points at the inlet and outlet of the fan is the same.

iv. $e_{loss} \approx 0$ as the measured points at the inlet and outlet of the fan are quite close.

Simplified form of Eq. (3.14) becomes:

$$\dot{W}_{fan} = \frac{P_{out} - P_{in}}{\rho} \tag{3.15}$$

Fan output power for the mass flow rate of air *m* can be expressed as:

$$W_{fan} = \frac{m(P_{out} - P_{in})}{\rho} \tag{3.16}$$

Similarly, pump output power for the volume flow rate *Q* can be expressed as:

$$W_{fan} = \frac{(\rho Q)(P_{out} - P_{in})}{\rho}$$

$$W_{fan} = Q(P_{out} - P_{in}) \tag{3.17}$$

where

P_{out} = static pressure at the outlet of the fan, Pa

P_{in} = static pressure at the inlet of the fan, Pa

m = mass flow rate of air, kg/s

Q = volume flow rate of air, m^3/s

W_{fan} = fan output power, Watts

If the input and output pressures are expressed in kPa, the unit of fan output power expressed in Eq. (3.16) and (3.17) will be kW.

Fans of AHUs are usually coupled with the motor using belt and pulley arrangement as shown in Figure-3.17. If the efficiency of the fan is η_{fan}, input power to the fan can be calculated as:

$$W_{fan,input} = \frac{Q(P_{out} - P_{in})}{\eta_{fan}} \tag{3.18}$$

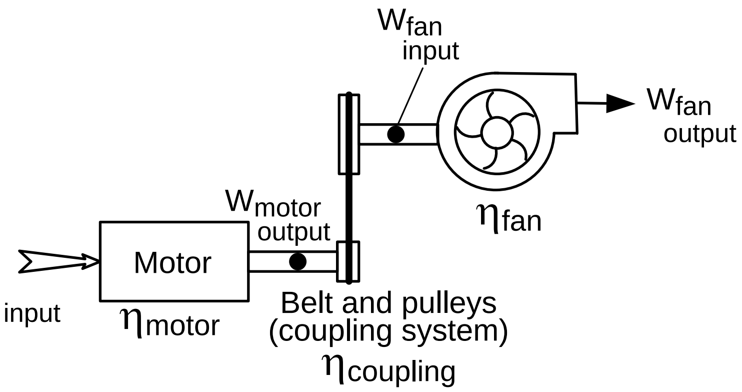


Figure 3.17 Belt and pulley system for the coupling of fan and motor

If the efficiencies of the belt and pulley coupling system and motor are $\eta_{coupling}$ and η_{motor} respectively, input power to the motor can be calculated as:

$$W_{motor,input} = \frac{Q(P_{out} - P_{in})}{\eta_{motor}\eta_{coupling}\eta_{fan}} \tag{3.19}$$

where

P_{in} = static pressure at the inlet of the fan, kPa

P_{out} = static pressure at the outlet of the fan, kPa

Q = volume flow rate of air, m^3/s

η_{fan} = efficiency of fan, dimensionless

$\eta_{coupling}$ = efficiency of coupling system, dimensionless

η_{motor} = efficiency of motor, dimensionless

$W_{fan,input}$ = fan input power, kW

$W_{motor,input}$ = motor input power, kW

3.6 Fan Operating Point

Similar to the pumping systems, the intersection of the system curve and the fan curve represents the operating point. Figure-3.18 shows the operating point of a fan system. The fan will deliver air at a rate of Q_1 with the pressure head of ΔP_1, which is equal to the corresponding total pressure drop for the ducting system. Intersections of the vertical line from flow rate of Q_1 with the efficiency curve and power curve represent the corresponding operating efficiency η_1 and input power for the fan W_1 respectively.

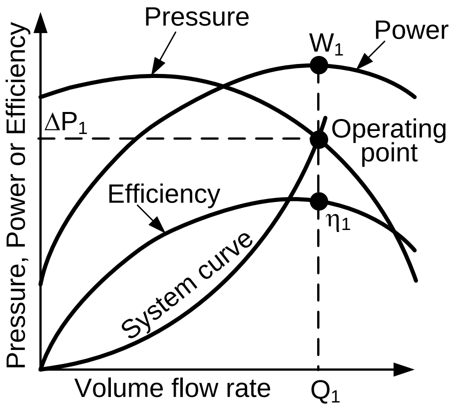


Figure 3.18 Fan characteristic curves, system curve and operating point

If fan performance curve for different speeds of impeller is obtained from the manufacturer, the design flow rate of air and the corresponding total design pressure losses of the ducting systems can be plotted on the fan performance curve as shown in Figure-3.19 to obtain required speed of the impeller and the fan power consumption.

For the air flow rate of 200 m^3/min and duct pressure losses of 500 Pa, the fan needs to be operated at 1000 rpm (operating point A of Figure 3.19) and corresponding power consumption and efficiency of the fan will be about 3500 Watts and 47 percent, respectively.

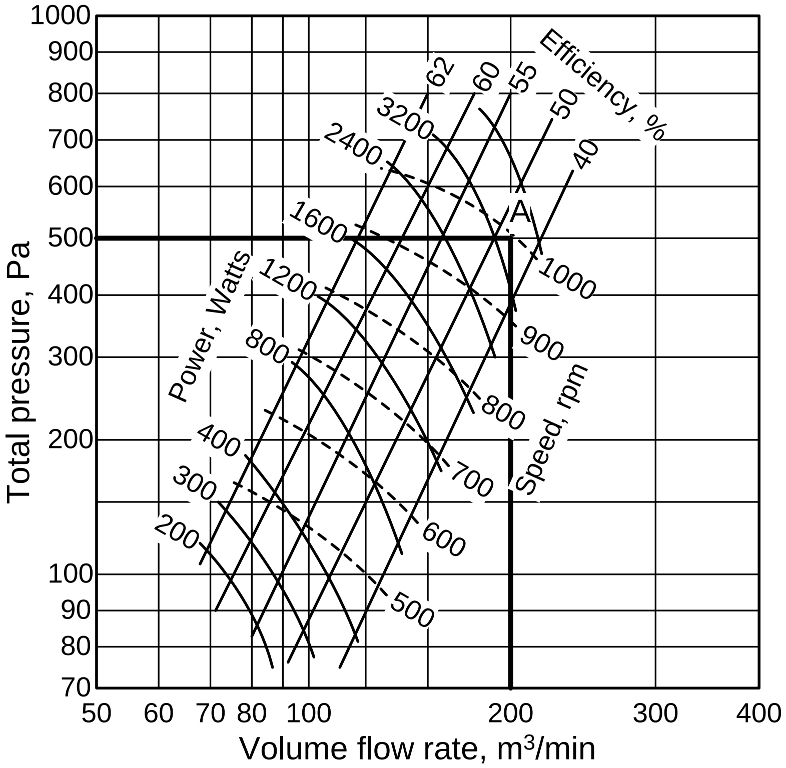


Figure 3.19 Fan performance curves and operating point

Effect of Partial Closing of Damper: The effect of closing dampers of ducting systems is similar to the closing of flow modulating valves of piping systems. Suppose the fan of a ducting system delivers air at a rate of Q_1 and the corresponding total pressure losses of the ducting system is ΔP_1 as shown in Figure-3.20, when the damper is fully open. Under this operating condition, the power consumption of the fan is:

$$W_{fan,input} = \frac{Q_1 \Delta P_1}{\eta_{fan}} \qquad (3.20)$$

If the flow rate of air Q_1 is higher than the required value, the flow rate of air can be reduced by partially closing the damper of the ducting system. Figure-3.14 shows that the system curve moves to the left if the damper is partially closed. The system curve as well as the operating point-1 of Figure-3.20 will therefore move to the left to point-2 due to the partial closing of the damper. Fan curve will not change its position as the speed of the fans is not changed. The flow rate of air at point-2 will reduce from Q_1 to Q_2. However, the total pressure losses of the ducting system will increase from ΔP_1 to ΔP_2. The power consumption of the fan at operating point-2 (Figure-3.20) will be:

$$W_{fan,input} = \frac{Q_2 \Delta P_2}{\eta_{fan}} \tag{3.21}$$

Even though the flow rate of air is decreased, the total pressure loss of the ducting system is increased due to the closing of the damper. Hence, the overall power consumption of the fan will remain almost the same and no significant energy savings is achieved due to the reduction of the flow rate from Q_1 to Q_2.

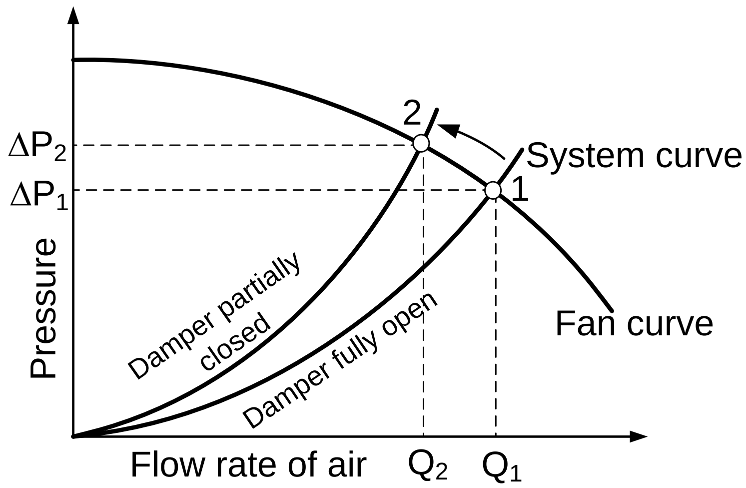


Figure 3.20 Effect of closing damper of the ducting systems

Effect of Fan Speed Reduction: If the flow rate of air Q_1 shown in Figure-3.21 is higher than the required value, the flow rate of air can also be reduced by reducing the speed of the impeller. As presented in Figure-3.10, the fan curve moves downwards if the speed of the impeller of the fan is reduced. The fan curve as well as the operating point-1 of Figure-3.21 will therefore move down to point-3 due to the reduction of the speed of the impeller. In this case, system curve will not change its position as the opening of the damper is not changed. The flow rate of air at point-3 will reduce from Q_1 to Q_2. At the same time, the total pressure drop of the ducting system will decrease from ΔP_1 to ΔP_3 as shown in Figure-3.21. The power consumption of the pump at operating point-3 will be:

$$W_{fan,input} = \frac{Q_2 \Delta P_3}{\eta_{fan}} \tag{3.22}$$

As the flow rate of air as well as the total pressure drop of the ducting system drop due to the reduction of the speed of the impeller, overall power consumption of the fan will drop remarkably.

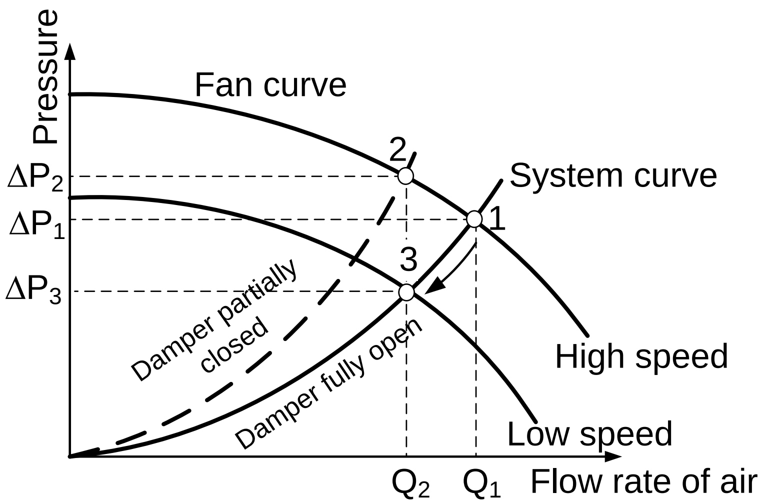
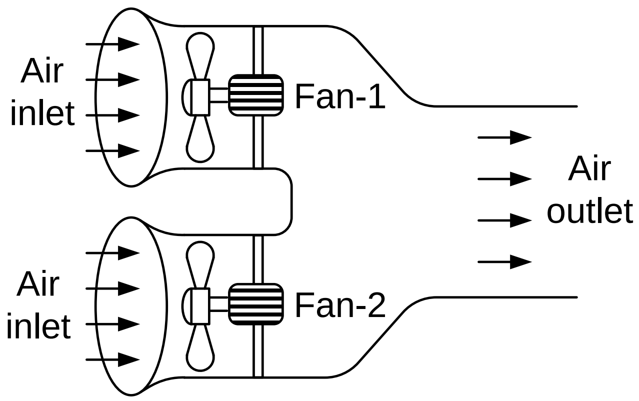
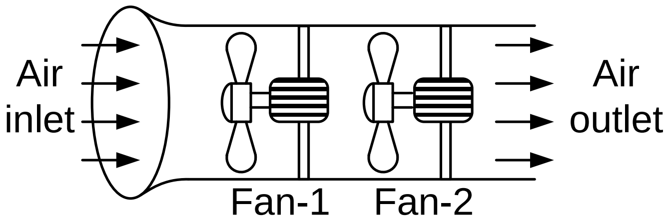


Figure 3.21 Effect of closing damper of ducting systems

Fans in Parallel and Series: Fans can be connected in parallel or series as shown in Figure-3.22 to accommodate variable flow of air and total pressure head requirements of the ducting system or to provide redundancy in case of any fan failure. Capacity of the fans could be the same or different. Variable speed drives (VSDs) can also be used in conjunction with the parallel or series fan configuration to provide more flexibility in operation.



(a)



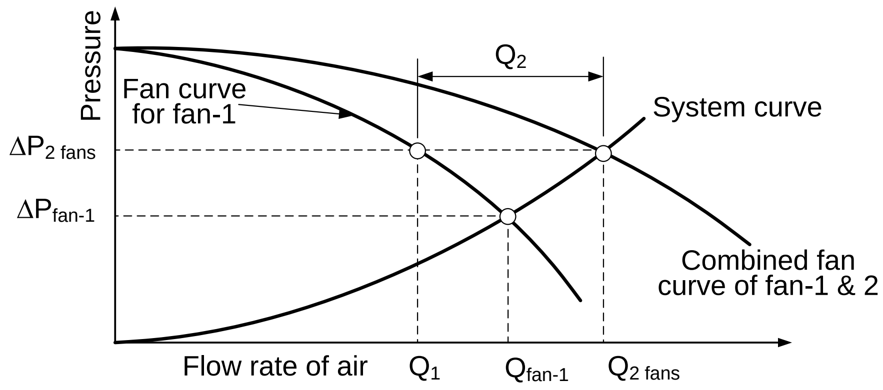
(b)

Figure 3.22 Configuration of fans (a) Fans in parallel and (b) Fans in series

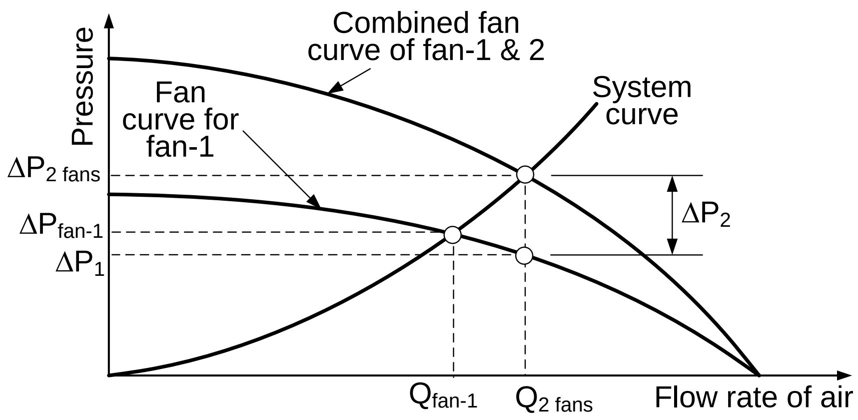
Similar to the pumping systems, if fans are connected in parallel configuration, each fan will generate equal pressure head. Total flow rate of air will be equal to the summation of flow rate of the operating fans. Therefore, parallel configuration is suitable for high flow rate applications. On the other hand, if fans are connected in series configuration, total flow rate of air will be equal to the flow rate of each fan. However, pressure of the flowing air will increase in each fan. Hence, series configuration is suitable for ducting systems of high pressure loss.

Interactions of the fan curve and system curve for the parallel and series configurations of the fans are shown in Figure-3.23. For the parallel configuration, when two fans are operated (Figure-3.23.a), total flow rate of air through the ducting system becomes $Q_{2\ fans}$, which is equal to (Q_1 + Q_2) and the corresponding total pressure loss of the ducting system reaches to $\Delta P_{2\ fans}$. To overcome the total pressure loss of the ducting system, both fans will generate equal pressure head of $\Delta P_{2\ fans}$. If only fan-1 is operated, flow rate of air through the ducting system will drop to $Q_{fan\text{-}1}$ and the corresponding pressure loss of the ducting system will drop as well to $\Delta P_{fan\text{-}1}$. Note that the flow rate of fan-1 is increased from Q_1 to $Q_{fan\text{-}1}$ when only fan-1 is operated due to the reduction of pressure loss of the ducting system from $\Delta P_{2\ fans}$ to $\Delta P_{fan\text{-}1}$.

Similarly, for the series configuration, when two fans are operated (Figure-3.23.b), flow rate of air through each fan is $Q_{2\ fans}$ (= Q_1) and the total pressure head generated by the fans is $\Delta P_{2\ fans}$. Pressure head generated by fan-1 and fan-2 are ΔP_1 and ΔP_2, respectively. If only fan-1 is operated, flow rate of air through the ducting system will drop to $Q_{fan\text{-}1}$ but the generated head by fan-1 will increase to $\Delta P_{fan\text{-}1}$. Note that the pressure head generated by fan-1 is increased from ΔP_1 to $\Delta P_{fan\text{-}1}$ when only fan-1 is operated due to the reduction of flow rate from $Q_{2\ fans}$ to $Q_{fan\text{-}1}$.



(a)



(b)

Figure 3.23 Fan and system characteristic curves (a) fans are connected in parallel and (b) fans are connected in series

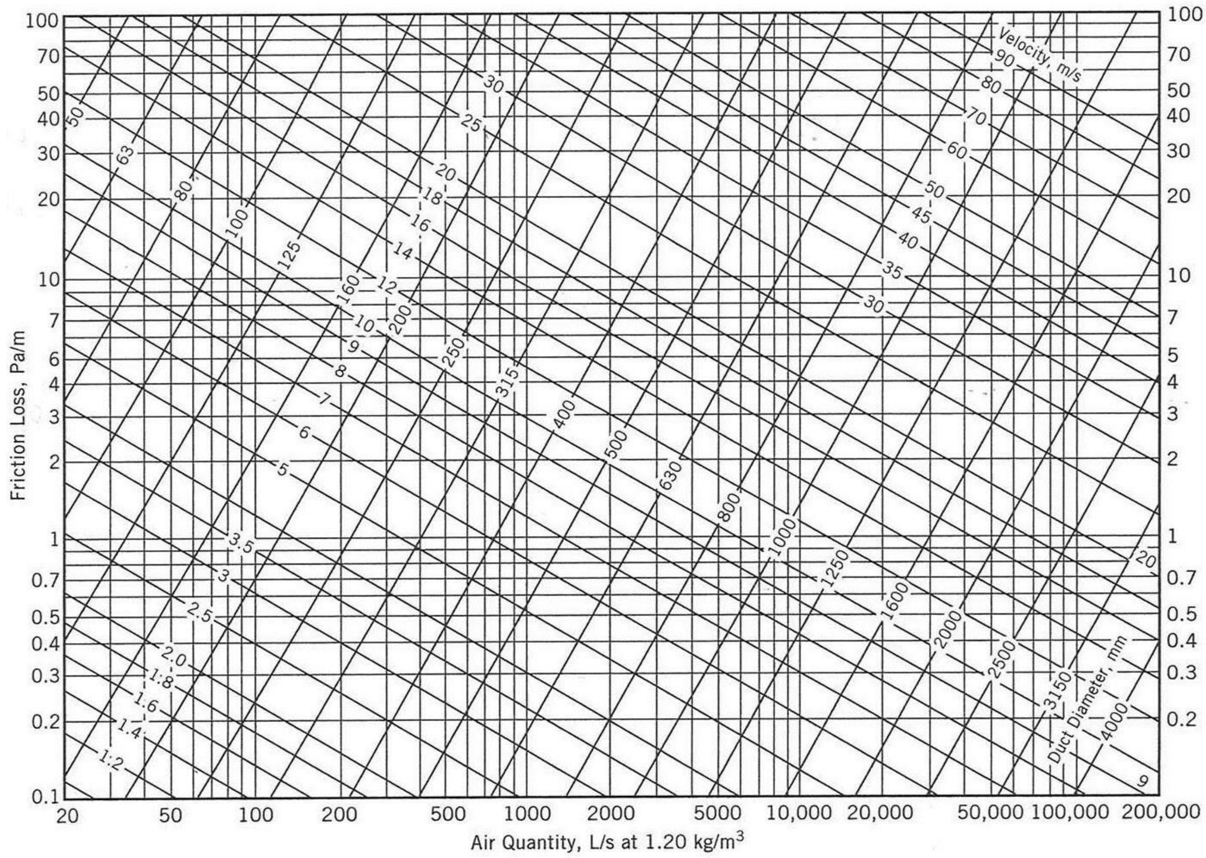
3.7 Losses in Ducting Systems

Losses in ducting systems are broadly classified as:

i. Friction loss and
ii. Dynamic loss

Friction loss: Pressure loss in the straight duct due to the friction between the flowing air and the wall of the duct is known as friction loss. Similar to the piping systems, friction loss of ducting systems mainly depends on (a) surface roughness of internal surface of the duct, (b) duct length, (c) duct diameter and (d) volume flow rate of air.

Surface roughness depends on the material of the duct. Friction loss can be calculated using Darcy-Weisbach Eq. (3.1). To facilitate the computation of friction loss and the duct sizing, friction loss charts for ducts of different materials have been developed. Friction loss chart for galvanized steel ducts of circular cross sections is shown in Figure-3.24. Frictional head loss per unit length of the duct can be obtained directly from the chart corresponding to the flow rate of air and duct diameter. Similarly, when the frictional head loss and flow rate of air are specified, a duct diameter and air flow velocity may conveniently be obtained using the chart. It is obvious from the chart that if duct of larger diameter is selected for a specific flow rate of air, pressure loss per unit length of the duct and consequently fan power consumption will drop. However, ducting cost will increase due to the selection of duct of larger diameter. Usually, friction loss of 1 to 2 Pa/m is used to select the duct diameter.



(Source: ASHRAE Handbook, Fundamentals Volume SI, 1997)

Figure 3.24 Friction loss chart for galvanized steel ducts of circular cross sections

Friction loss charts are developed for ducts of circular cross sections. However, rectangular ducts are quite commonly used to fit the ducts in the confined spaces. The dimensions of rectangular ducts of equivalent friction loss to the circular ducts as shown in Figure-3.25 can be calculated by:

$$D_{equivalent} = \frac{1.3(ab)^{0.625}}{(a+b)^{0.25}} \tag{3.23}$$

where

$D_{equivalent}$ = equivalent diameter of round duct, mm

a = dimension width of the rectangular duct, mm

b = dimension height of the rectangular duct, mm

Either the dimension a or b of the rectangular duct is first fixed based on the available space and then the other dimension is calculated using Eq. (3.23).

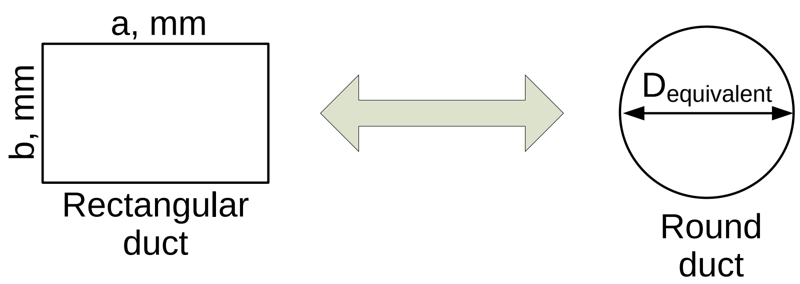
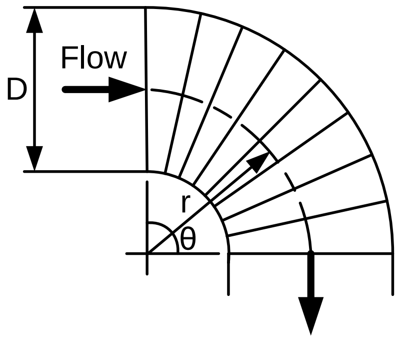


Figure 3.25 Rectangular and circular ducts of equivalent friction loss

Dynamic Losses: Head losses due to the resistance offered to the flowing air by different fittings of the duct such as dampers, elbows, converging flow fittings, diverging flow fittings, tee-joints etc. are known as the dynamic losses which can be calculated using Eq. (3.5). The loss coefficient C_o use in Eq. (3.5) depends on the type and size of the fittings. The loss coefficient C_o for few selected fittings, such as pleated elbows, round to round transitions and round diverging tees are presented in Table-3.2, 3.3 and 3.4, respectively. A complete list of the loss coefficients for other types of fittings can be obtained in the handbooks on duct design. Dynamic losses are directly proportional to the loss coefficients. One has to be careful when selecting fittings for different applications to minimise the total pressure head and associated fan power consumption.

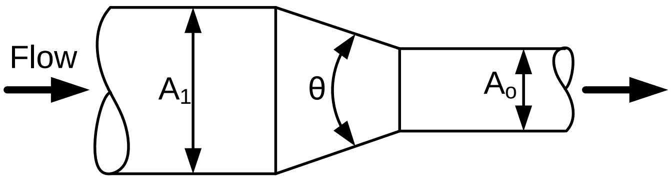
Table 3.2 Loss coefficients for pleated elbow (r/D = 1.5)



Angle, θ	C_o at D, mm						
	100	150	200	250	300	350	400
90	0.57	0.43	0.34	0.28	0.26	0.25	0.25
60	0.45	0.34	0.27	0.23	0.20	0.19	0.19
45	0.34	0.26	0.21	0.17	0.16	0.15	0.15

(*Source: ASHRAE Duct Fitting Database 1992)*

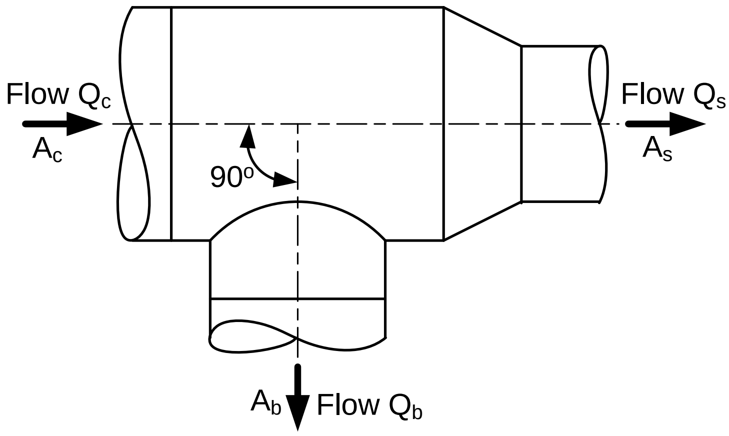
Table 3.3 Loss coefficients for round to round transition



A_o/A_1	C_o						
	θ = 10°	20°	45°	90°	120°	150°	180°
0.10	0.05	0.05	0.07	0.19	0.29	0.37	0.43
0.17	0.05	0.04	0.06	0.18	0.28	0.36	0.42
0.25	0.05	0.04	0.06	0.17	0.27	0.35	0.41
0.50	0.05	0.05	0.06	0.12	0.18	0.24	0.26
1.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2.00	0.44	0.76	1.32	1.28	1.24	1.20	1.20
4.00	2.56	4.80	9.76	10.24	10.08	9.92	9.92
10.00	21.00	38.00	76.00	83.00	84.00	83.00	83.00
16.00	53.76	97.28	215.04	225.28	225.28	225.28	225.28

(*Source: ASHRAE Duct Fitting Database 1992)*

Table 3.4 Loss coefficients for round diverging tee



A_b/A_c	Branch, $C_{o,b}$								
	Q_b/Q_c = 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	1.20	0.62	0.80	1.28	1.99	2.92	4.07	5.44	7.02
0.2	4.10	1.20	0.72	0.62	0.66	0.80	1.01	1.28	1.60
0.3	8.99	2.40	1.20	0.81	0.66	0.62	0.64	0.70	0.80
0.4	15.89	4.10	1.94	1.20	0.88	0.72	0.64	0.62	0.63
0.5	24.80	6.29	2.91	1.74	1.20	0.92	0.77	0.68	0.63
0.6	35.73	8.99	4.10	2.40	1.62	1.20	0.96	0.81	0.72
0.7	48.67	12.19	5.51	3.19	2.12	1.55	1.20	0.99	0.85
0.8	63.63	15.89	7.14	4.10	2.70	1.94	1.49	1.20	1.01
0.9	80.60	20.10	8.99	5.13	3.36	2.40	1.83	1.46	1.20

(*Source: ASHRAE Duct Fitting Database 1992)*

A_s/A_c	Main, $C_{o,s}$								
	Q_s/Q_c = 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	0.13	0.16							
0.2	0.20	0.13	0.15	0.16	0.28				
0.3	0.90	0.13	0.13	0.14	0.15	0.16	0.20		
0.4	2.88	0.20	0.14	0.13	0.14	0.15	0.15	0.16	0.34
0.5	6.25	0.37	0.17	0.14	0.13	0.14	0.14	0.15	0.15
0.6	11.88	0.90	0.20	0.13	0.14	0.13	0.14	0.14	0.15
0.7	18.62	1.71	0.33	0.18	0.16	0.14	0.13	0.15	0.14
0.8	26.88	2.88	0.50	0.20	0.15	0.14	0.13	0.13	0.14
0.9	36.45	4.46	0.90	0.30	0.19	0.16	0.15	0.14	0.13

(*Source: ASHRAE Duct Fitting Database 1992)*

Problem 3.1

Air is flowing at a rate of 750 CMH through a 250 mm 90º pleated elbow. The ratio of turning radius to diameter is 1.5. Compute the pressure loss for the pleated elbow. Assume standard air.

Solution

Air flow velocity: $V = \dfrac{Q}{A} = \dfrac{Q}{(\pi/4)D^2} = \dfrac{750/3600}{(\pi/4)(250/1000)^2} = 4.24 m/s$

Using Table 3.2 for pleated elbow and r/D = 1.5 table: pressure loss coefficient C_o = 0.28

Pressure loss: $\Delta P = C_o\left(\dfrac{\rho V^2}{2}\right) = 0.28\left(\dfrac{1.2x4.24^2}{2}\right) = 3.02 Pa$

Problem 3.2

Compute the straight-through and branch pressure losses for a 90º round diverging tee. The diameters of the common section, straight-through and branch path are 300 mm, 250 and 150mm respectively. Air flow rate through common section and branch path are 1870 CMH and 425 CMH, respectively. Assume standard air.

Solution

Air flow velocity through common section:

$$V_c = \frac{Q_c}{A_c} = \frac{Q_c}{(\pi/4)D_c^2} = \frac{1870/3600}{(\pi/4)(300/1000)^2} = 7.35 m/s$$

Air flow velocity through branch path:

$$V_b = \frac{Q_b}{A_b} = \frac{Q_b}{(\pi/4)D_b^2} = \frac{425/3600}{(\pi/4)(150/1000)^2} = 6.68 m/s$$

Air flow velocity through straight-through section:

$$V_s = \frac{Q_s}{A_s} = \frac{Q_s}{(\pi/4)D_s^2} = \frac{(1870-425)/3600}{(\pi/4)(250/1000)^2} = 8.18 m/s$$

Ratio of air flow through branch path and common section: $\dfrac{Q_b}{Q_c} = \dfrac{425}{1870} = 0.23$

Ratio of cross-sectional area of branch path and common section:

$$\frac{A_b}{A_c} = \left(\frac{150}{300}\right)^2 = 0.25$$

Ratio of air flow through straight-through and common section:

$$\frac{Q_s}{Q_c} = \frac{1870 - 425}{1870} = 0.77$$

Ratio of cross-sectional area of straight-through and common section:

$$\frac{A_s}{A_c} = \left(\frac{250}{300}\right)^2 = 0.69$$

From Table 3.4 for Q_b/Q_c = 0.23 and A_b/A_c = 0.25, using double interpolation, loss coefficient for branch $C_{o,b}$ =1.55.

Similarly, from Table 3.4 for Q_s/Q_c = 0.77 and A_s/A_c = 0.69, using double interpolation, pressure loss coefficient for straight-through $C_{o,s}$ =0.14.

Pressure loss for branch path: $\Delta P_b = C_{o,b}\left(\frac{\rho V_b^2}{2}\right) = 1.55\left(\frac{1.2x6.68^2}{2}\right) = 41.5Pa$

Pressure loss for straight-through: $\Delta P_s = C_{o,s}\left(\frac{\rho V_s^2}{2}\right) = 0.14\left(\frac{1.2x8.18^2}{2}\right) = 5.6Pa$

3.8 Affinity Laws

The *affinity laws for fans* are similar to those for pumps discussed in Chapter 2. The *affinity laws* relate the flow rate of air, pressure developed across the fan and shaft power of the fan to the new and old speeds or impeller diameters. Within a given fan casing / housing, effects of changing the fan rotational speed *N* and impeller diameter *D* are presented in Table 3.5.

Table 3.5 Affinity laws for specific fan casing / housing

Characteristic	Constant impeller diameter, D	Constant impeller speed, N
Flow rate, Q	$Q \propto N$	$Q \propto D^3$
Head, Pressure, ΔP	$\Delta P \propto N^2$	$\Delta P \propto D^2$
Power, W	$W \propto N^3$	$W \propto D^5$

Based on Table 3.5, for constant impeller diameter:

i) Change of air flow rate with the change of rotational speed is given by

$$Q_2 = Q_1\left(\frac{N_2}{N_1}\right) \tag{3.24}$$

ii) Change of developed pressure across the fan with the change of rotational speed is:

$$\Delta P_2 = \Delta P_1 \left(\frac{N_2}{N_1} \right)^2 \tag{3.25}$$

iii) Change of shaft power or input power of the fan with the change of rotational speed is:

$$W_2 = W_1 \left(\frac{N_2}{N_1} \right)^3 \tag{3.26}$$

iv) Change of fan efficiency with the change of rotational speed can be obtained as:

Efficiency of the fan at the old speed N_1 is

$$\eta_1 = \frac{Q_1 \Delta P_1}{W_1} \tag{3.27}$$

Efficiency of the fan at the new speed N_2 is

$$\eta_2 = \frac{Q_2 \Delta P_2}{W_2} \tag{3.28}$$

Combining Eqs. (3.27) and (3.28) gives:

$$\frac{\eta_2}{\eta_1} = \frac{Q_2 \Delta P_2}{W_2} \frac{W_1}{Q_1 \Delta P_1}$$

$$\frac{\eta_2}{\eta_1} = Q_1 \left(\frac{N_2}{N_1} \right) \Delta P_1 \left(\frac{N_2}{N_1} \right)^2 \frac{1}{W_1} \left(\frac{N_1}{N_2} \right)^3 \frac{W_1}{Q_1 \Delta P_1} = 1 \tag{3.29}$$

Therefore, the efficiency of the fan will remain constant if the rotational speed of the fan is changed from N_1 to N_2.

3.9 Constant and Variable Air Volume AHU Systems

The operating principles of the constant air volume (CAV) and the variable air volume (VAV) systems are discussed in Section 3.2.1. Schematic diagram of a typical CAV system without reheat coil is shown in Figure-3.26. Fan is operated at constant speed and constant volume flow rate of air to the air-conditioning spaces is maintained irrespective of the space cooling load. The temperature of the supply air and the flow rate of the chilled water through the AHU coil are modulated in response to the cooling load changes of the spaces by sensing the return air temperature. Hence, fan speed and fan power consumption remain the same even when space cooling load is low. The CAV systems are suitable for large and open spaces where the variation of the cooling load is relatively uniform. As less chilled water is supplied to the AHU coil at low part-load conditions of the air-conditioning spaces, less dehumidification of supply air may occur in the AHU coil leading to the increase of space relative humidity.

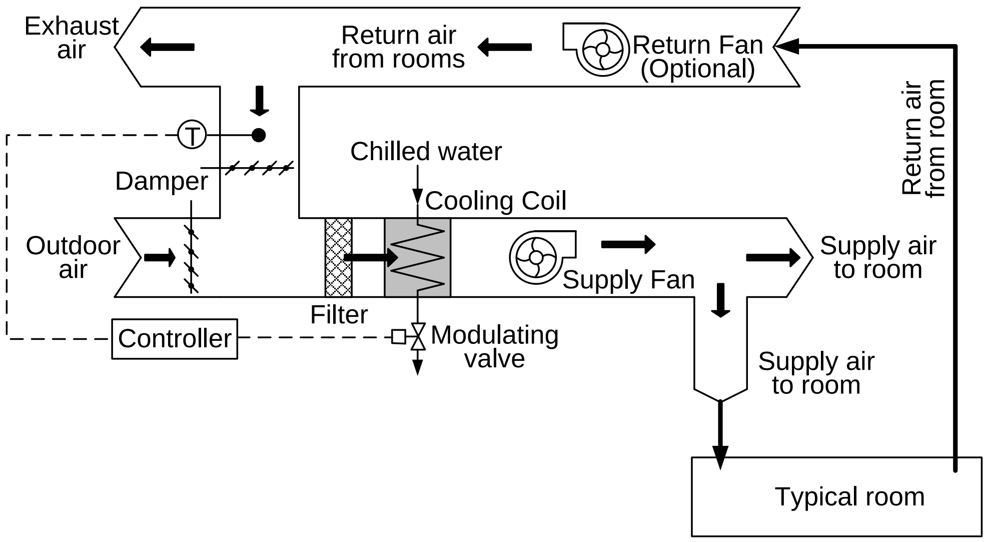


Figure 3.26 Schematic diagram of a typical constant air volume (without reheat coil) system

In the variable air volume (VAV) systems, volume flow rate of air to the air-conditioning spaces is modulated based on space cooling load. Schematic diagram of a typical VAV system is shown in Figure-3.4. The temperature of the supply air is maintained constant irrespective of the space cooling load. The flow rate of the chilled water through the AHU coil is modulated based on space cooling load. As the flow rates of the supply air and the chilled water are modulated together at low part-load, the required dehumidification of the supply air takes place in the AHU coil. If the cooling load fluctuates unevenly in different spaces, the VAV systems can conveniently maintain the required space conditions. Fan could be operated at constant or variable speed. The flow rate of air in the air-conditioning spaces is modulated in response to the change of the space cooling load using: (a) discharge dampers, (b) inlet guide vanes (IGVs) and (c) variable speed drive (VSD). Hence, fan power consumption changes with the variation of the space cooling load. Typical power consumption of the fan for different variable air volume systems is shown in Figure-3.27. Under part load conditions of the air-conditioning spaces, the reduction of fan power consumptions using discharge dampers and inlet guide vanes are much lesser in comparison to the variable speed drive systems, which are able to follow closely the affinity law of “cubic” fan power relationship.

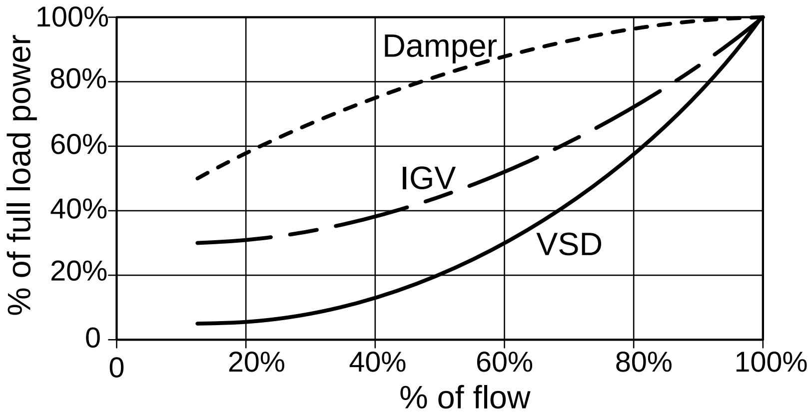


Figure 3.27 Fan power consumption for different VAV systems

3.10 Losses in Filter and Cooling Coils

Filter Systems: Depending on the indoor air quality requirements for the commercial buildings and industrial plants, various types of air filters are used in the air handling units to remove suspended solid or liquid materials from the supply air. The most common type of filters is known as media filters, which are made of fibrous material. High-Efficiency Particulate Air (HEPA) filters are generally used in cleanrooms. Photographs of the media and HEPA filters are shown in Figure-3.28. The pressure drop across the filter depends on (a) the type of filter, (b) air flow rate through the filter and (c) the amount of accumulated dust in the filter. When the filters are clean, the pressure drop across the filters is low and air can flow easily through the filters. As dust is accumulated in the filters, the pressure drop across them increases, which leads to the reduction of air flow. Generally, filters are selected to work up to a design pressure drop. If the pressure drop across the filters is higher than the design value due to the accumulation of dust, the filters need to be cleaned or replaced. As shown in Figure-3.29, the air flow rate through the filters Q_1 is higher than the design value when the filters are clean. As dust gradually accumulates in the filters, pressure drop across the filters increases. Hence, the system curve moves to the left and the flow rate of air drops to the design value Q_2. Filters should be cleaned or replaced at this stage. If the filters are not cleaned or replaced, the pressure drop across the filters will further increase and the flow rate of air will drop to Q_3, which is lower than the design value.

(a) Media filter (b) HEPA filter

Figure 3.28 Photographs of the media and HEPA filters

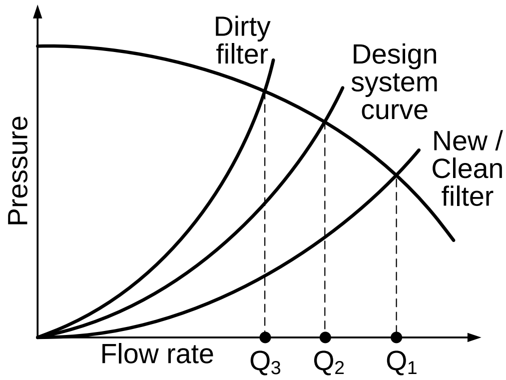


Figure 3.29 Effect of filter conditions on flow rate of air

When the filters are clean, air flow rate Q_1 is higher than the design value Q_2, which may result in overcooling of the spaces and higher power consumption of the fans. The flow rate of air can be reduced to the design value by partially closing the discharge dampers or reducing the speed of the fan using VSD as illustrated in Figure-3.30.

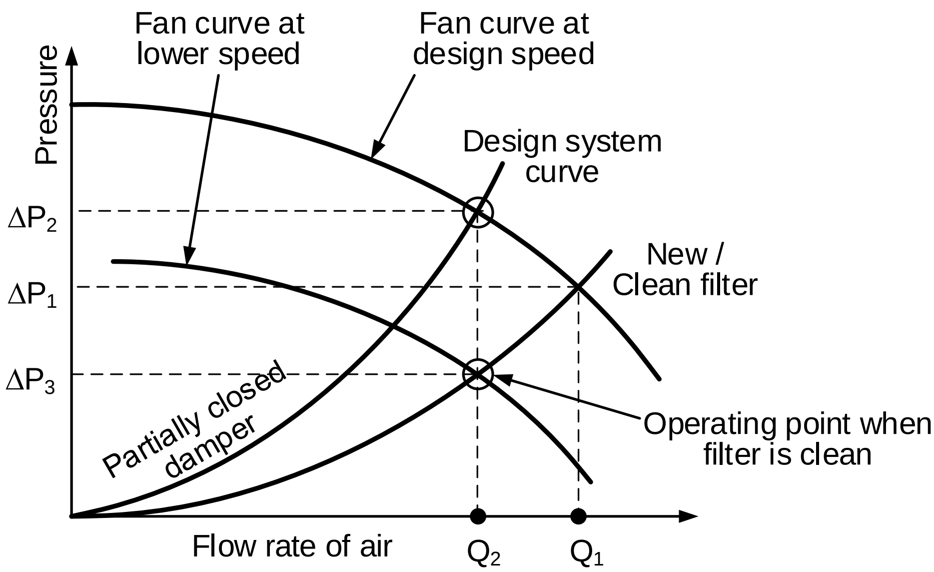


Figure 3.30 Effect of closing damper and fan speed with clean filter

When the filters are new or clean, power consumption of the fan without changing fan speed or damper position can be calculated as:

$$W_1 = \frac{Q_1 \Delta P_1}{\eta} \tag{3.30}$$

If flow rate Q_1 is reduced to Q_2 by partially closing the discharge dampers (meaning artificially adding flow resistance), the system curve of new or clean filter will move to the design system curve. Hence, flow rate of air Q_1 will drop to the design flow Q_2, but pressure drop will increase from ΔP_1 to ΔP_2. Therefore, fan power savings will be low. Power consumption of fan due to the partial closing of the discharge dampers can be calculated as:

$$W_2 = \frac{Q_2 \Delta P_2}{\eta} \tag{3.31}$$

However, if flow rate Q_1 is reduced to Q_2 by reducing the speed of the fan using VSD, fan curve will move downward as shown in Figure-3.30. Hence, flow rate of air Q_1 will drop to the design flow Q_2 and pressure drop will also reduce from ΔP_1 to ΔP_3. Therefore, significant fan power savings will be achieved. Power consumption of the fan due to the reduction of fan speed can be calculated as:

$$W_3 = \frac{Q_2 \Delta P_3}{\eta} \tag{3.32}$$

Electronic air filters, as shown in Figure-3.31, are also used in the air handling units. These filters use “electrostatic precipitation” to effectively remove dust participles as small as 0.01 microns. Pressure drop across the electronic air filter is lower in comparison to the media filters. Hence, power consumption of the fan can be reduced by replacing the media filter with the electronic air filter.

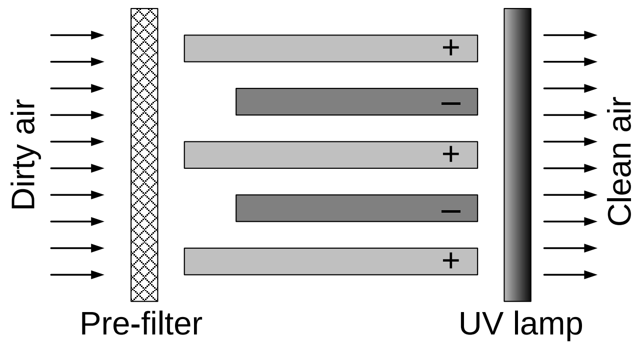


Figure 3.31 Schematic diagram of electronic air filter

Problem 3.3

The fan of an air handling unit consumes 16 kW of power and delivers air at a rate of 15 m^3/s. Presently, media filter of pressure drop 70 Pa is used in the system. The total pressure drop (including the media filter) of the air handling system is 550 Pa. Calculate the achievable fan power savings if the media filter is replaced with an electronic air cleaner which has a negligible pressure drop. Present air flow rate of 15 m^3/s is required to be maintained after replacing the media filter with the electronic air cleaner.

Solution

Air flow rate Q = 15 m^3/s

Present fan power consumption W_1 = 16 kW

Present total pressure drop ΔP_1 = 550 Pa

Pressure drop across media filter ΔP_{filter} = 70 Pa

Total pressure drop after replacing media filter $\Delta P_2 = (\Delta P_1 - \Delta P_{filter})$

= (550 – 70) = 480 Pa

Initial power consumption of fan: $W_1 = \frac{Q\Delta P_1}{\eta}$

Assuming constant fan efficiency, final power consumption of fan: $W_2 = \frac{Q\Delta P_2}{\eta}$

Therefore, $\frac{W_2}{W_1} = \frac{\Delta P_2}{\Delta P_1}$

$W_2 = W_1 \frac{\Delta P_2}{\Delta P_1}$ = 16 x (480 / 550) = 13.96 kW

Therefore, savings in fan power consumption will be = 16 – 13.96 = 2.04 kW

3.11 Air Flow Rate Optimisation

Fans of the AHUs and ventilation systems are selected to deliver the required amount of air by overcoming the system pressure losses. Required air flow rate is generally considered based on the peak cooling load and ventilation requirements. System pressure losses involve the friction and dynamic losses which depend on factors such as the diameter and length of the ducts, number and types of fittings, bending, converging and diverging sections, air flow velocity etc. Due to site constraints, ducting layout is generally changed during actual installation. Hence, a safety factor is added by the design engineer to account for the differences between the computed values and actual system losses. The value of the safety factor depends on how confident the designer is about the design. It is quite common to use high safety factors, which result in excess air flow during actual operation. Actual flow rate of air should be measured after installation of the fan systems and compared with the design values or actual requirements of the site. If the flow rate of air is found to be excessive, the flow could be reduced to the design value as shown in Figure-3.32 by partially closing the discharge dampers or reducing the speed of the fan using VSD or changing the pulley sizes. If the flow rate of air is reduced to the design value by partially closing the discharge dampers, system curve will move to the left. As a result, flow rate of the air will reduce but the system pressure loss will increase. However, if the flow rate is reduced by reducing the speed of the fan, both the flow rate of air and the system pressure loss will drop resulting in significant savings of fan power as illustrated in Figure-3.32. VSD should be installed for variable air flow rate applications. However, changing the pulley sizes is a cheaper option for constant air flow rate applications.

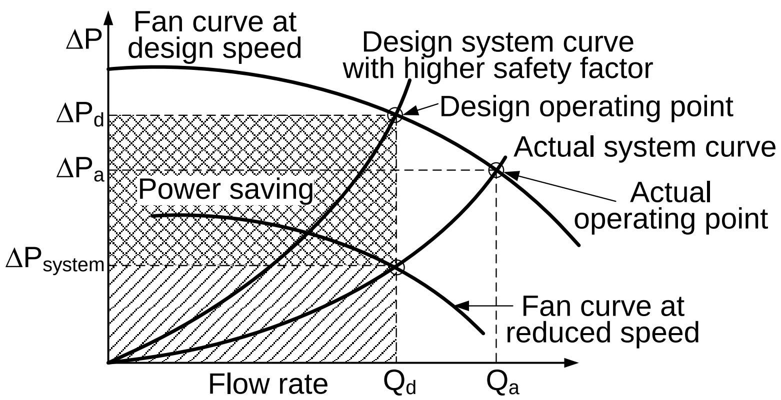


Figure 3.32 Fan performances due to use of high safety factors

Problem 3.4

A fan system is delivering 20 m^3/s of air when the actual requirement is 15 m^3/s. Present fan and motor speeds are 800 and 960 rpm respectively. The diameter of motor pulley is 200 mm. Find the new fan speed and pulley size.

Solution

Present diameter of motor pulley $D_{m,present}$ = 200 mm

Present motor speed $N_{m,present}$ = 960 rpm

Present fan speed $N_{f,present}$ = 800 rpm
Present air flow rate $Q_{present}$ = 20 m^3/s,
Required air flow rate $Q_{required}$ = 15 m^3/s,

Using affinity law, required fan speed $N_{f,required}$ to reduce the air flow rate from 20 m^3/s to 15 m^3/s:
$N_{f,required} = N_{f,present} \, x \, (Q_{required} / Q_{present})$ = 800x(15/20) = 600 rpm
Therefore, required fan speed is 600 rpm.

For linear speed of belt of the pulley system: $V = \pi D_m N_m = \pi D_f N_f$

For present pulley system: $D_{m,present} N_{m,present} = D_{f,present} N_{f,present}$

$$200x960 = D_{f,present} x800$$

Present diameter of fan pulley $D_{f,present}$ = 240 mm

For the new pulley system to reduce the fan speed from 800 rpm to 600 rpm:

$$D_{m,present} N_{m,present} = D_{f,required} N_{f,required}$$

$$200x960 = D_{f,required} x600$$

Required diameter of fan pulley $D_{f,required}$ = 320 mm
Therefore, the diameter of the fan pulley needs to be increased from 240 mm to 320 mm.

3.12 Coil Face Velocity

Pressure drop across the cooling / heating coils of AHUs depends on the number of coil rows, fin density and the velocity of air flowing through the coils known as coil face velocity. The pressure drop across a coil is proportional to the square of the coil face velocity. Reduction in coil face velocity can be achieved by selecting bigger coil. Operating cost of the fans could be reduced significantly by selecting the bigger coils due to the lower pressure drop across the bigger coils. This results in lower fan energy consumption. However, the first cost of the bigger coil is higher. Coils and AHUs should be selected based on life cycle costing.

Problem 3.5

The fan of an AHU system is delivering air at a rate of 9 m^3/s. The face area of the cooling coil of AHU is 2.5 m^2. The pressure drop across the cooling coil is 300 Pa. Calculate the pressure drop across the cooling coil and the reduction in fan power consumption if the face area of the cooling coil is increased to 4 m^2. Note that the air flow rate of 9 m^3/s needs to be maintained to support the cooling load of the spaces. Assume the efficiencies of the fan and the motor are 60% and 90% respectively.

Solution

Air flow rate Q = 9 m^3/s

Present face area of cooling coil A_1 = 2.5 m^2

Present face velocity $V_1 = Q / A_1 = 9 / 2.5 = 3.6$ m/s

Present pressure drop across the cooling coil ΔP_1 = 300 Pa

Proposed face area of cooling coil A_2 = 4 m^2

Proposed face velocity $V_2 = Q / A_2 = 9 / 4 = 2.25$ m/s

As the pressure drop across the cooling coil is proportional to the square of the coil face velocity:

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{V_2}{V_1}\right)^2$$

$$\Delta P_2 = \Delta P_1 \left(\frac{V_2}{V_1}\right)^2 = 300\left(\frac{2.25}{3.6}\right)^2 = 117 Pa$$

Therefore, pressure drop across the proposed cooling coil = 117 Pa

Reduction in pressure drop across the coil $\Delta P_3 = \Delta P_1 - \Delta P_2$ = (300 – 117) = 183 Pa = 0.183 kPa

Reduction in fan power, $\Delta W = \dfrac{Q\Delta P_3}{\eta_f \eta_m} = \dfrac{9x0.183}{0.6x0.9} = 3.05kW$

3.13 Fan Efficiency

Typical efficiencies of different types of fans are presented in Table 3.1. The efficiency of the backward curved fans is relatively higher in comparison to the other types. However, backward curved fans are normally most costly. Eq. (3.18) shows that fan power consumption is inversely proportional to the fan efficiency. Fan power consumption decreases with the increase of fan efficiency. Although, the first cost of the backward curved fans are higher, it could be more attractive as the extra cost of the fan may be recovered by the lower fan power consumption.

Moreover, fan efficiency also depends on the operating point. If a high efficiency fan is selected for the wrong operating point, it may operate at low efficiency as illustrated in Figure-3.33. If the system curve intersects the fan curve at point-1, the operating efficiency of the fan will be η_1. However, the efficiency of the same fan will improve to η_2 if the system curve intersects the fan curve at point-2. Therefore, when a fan is selected for a particular application, the operating point and the corresponding fan efficiency should be analysed carefully to ensure that the fan will be operated at the highest efficiency.

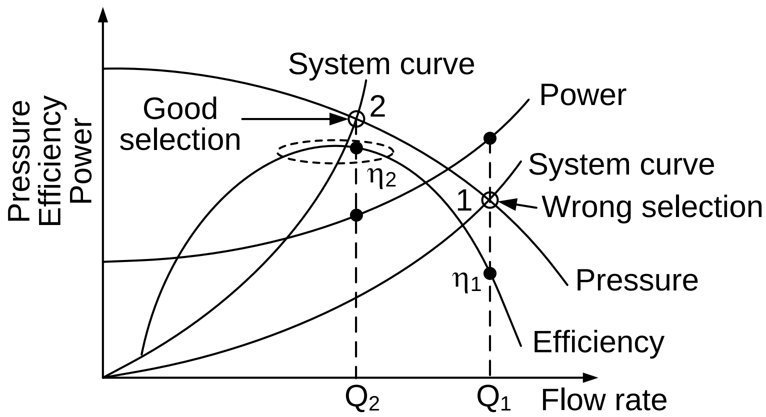


Figure 3.33 Fan efficiencies at different operating points

3.14 Reset of Fan Set-point

Similar to the pumping systems presented in Chapter 2, the energy performance of the fan systems can be improved by varying the differential pressure set point used for controlling the VSD of fans based on actual demand of the air flow rate. As the cooling load for each zone may not change by equal percentage with time, the opening of the air discharge dampers of different AHUs could be different to deliver the required amount of air. Building energy management system can be used to monitor the percentage opening of discharge dampers of all AHUs and determine the value of maximum damper opening. The value of the maximum percentage of damper opening is then compared with the pre-set limit and adjusts the differential pressure set point using appropriate control strategy as shown in Figure-3.34. For example, the pre-set limit of damper opening in Figure-3.34 is 80% to 90%. If the maximum percentage of damper opening is less than 80%, differential pressure set point will be decreased by the controller while ensuring that enough air is flowing to all the spaces. On the other hand, if the maximum percentage of damper opening is more than 90%, differential pressure set point will be increased. This continuous adjustment of the differential pressure set point results in minimum movement of the system curve as shown in Figure-3.35. The flow rate of air is varied mainly by adjusting the speed of the fans resulting in further energy saving of fan systems.

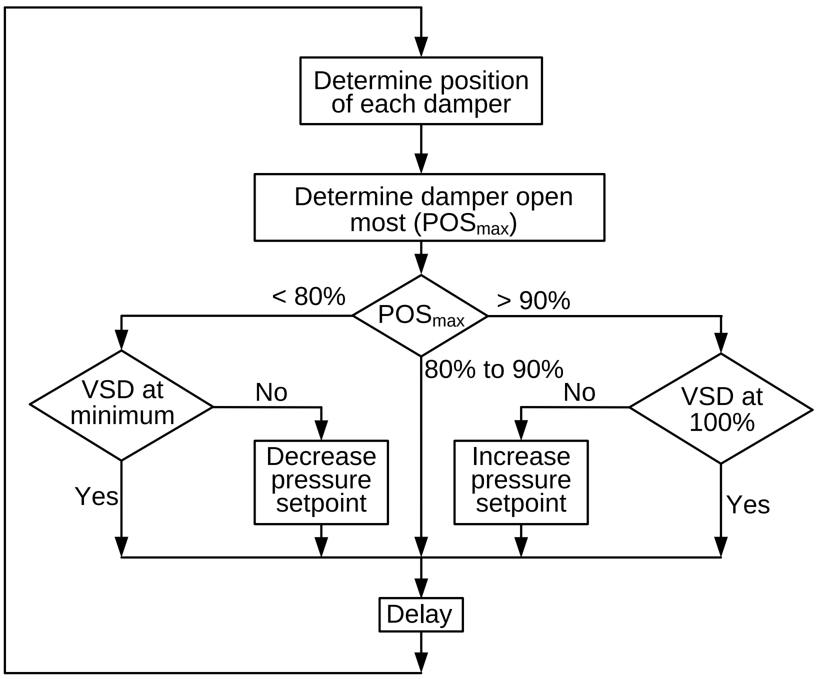


Figure 3.34 Control strategy of variable differential pressure set point for VSD of fans

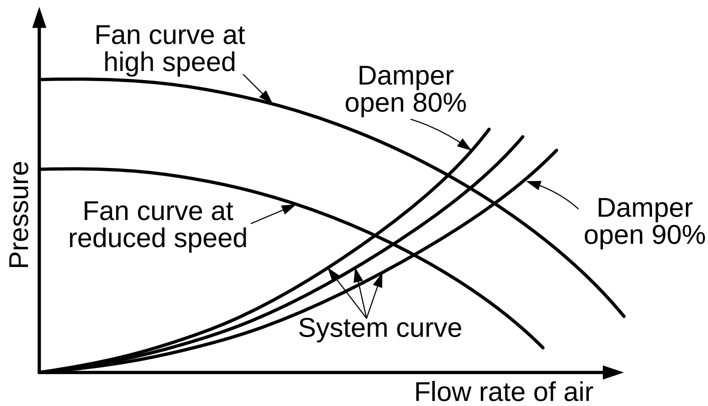


Figure 3.35 Movement of system and fan curves for fan systems using variable differential pressure set point

3.15 Modified Air Handling Systems

To meet the specific requirements of the air-conditioning spaces and operate the air handling systems in an energy efficient manner, several modified air handling systems are used in commercial buildings and industrial processes. Salient features of different modified air handling systems are discussed in the following sections:

3.15.1 Primary Air Handling Unit (PAU)

i. If fresh air requirement for a building is relatively high (such as hotel guest rooms), primary air handling units (PAUs) as shown in Figure-3.36 are generally used to cool and dehumidify outdoor fresh air.
ii. As PAUs handle 100% outdoor fresh air, which contains a lot of moisture, chilled water of relatively higher temperature can be used to cool and dehumidify the fresh air.
iii. Cold and dehumidified fresh air from the PAUs could be supplied directly to the air-conditioned spaces or to the inlet of the AHUs.
iv. For this case, AHU cooling coils are designed to handle mainly sensible cooling load and relatively small latent load of the air-conditioning spaces.
v. As chilled water of relatively higher temperature can be used with PAU systems, the lift of the chiller compressor reduces. This results in improvement of chiller efficiency.
vi. Based on AHRI design condition, chilled water supply temperature is 6.7°C. Chilled water supply temperature of about 8 to 9°C (depending on the design of the cooling coil) can be used in PAU systems.

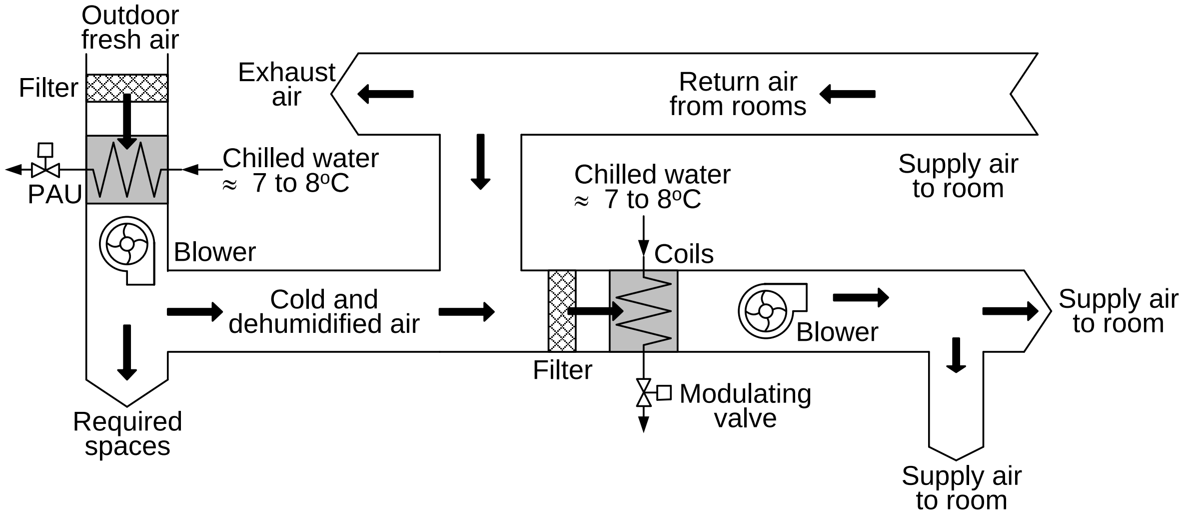


Figure 3.36 Schematic diagram of a primary air handling system

3.15.2 Dual-Path AHU

i. In the conventional AHUs, warm and humid outdoor fresh air is mixed with the return air and then the mixed air is supplied over the cooling coil of AHU for the necessary cooling and dehumidification.
ii. Outdoor air of high specific moisture content is mixed with a large volume of return air of low specific moisture content. Since the flow rate of outdoor fresh air is much lesser than the return air, the specific moisture content of the resultant mixed air remains relatively low. As a result, potential for moisture dehumidification of mixed air drops which leads to the use of low temperature chilled water and deep cooling coils of higher pressure drop in conventional AHUs.
iii. In dual-path air handling units, two separate coils are used to treat the outdoor fresh air and return air separately as shown in Figure-3.37. The outdoor air coil is designed for dehumidification as the moisture content of outdoor fresh air is high while the return air coil is designed for providing mainly sensible cooling.
iv. Similar to the PAU systems, chilled water of relatively higher temperature can be used in the cooling coils of dual-path air handling systems causing the reduction of the lift of compressor and chiller power consumption.
v. Chilled water supply temperature of about 8 to 9^{o}C (depending on the design of the cooling coil) can be used in dual-path air handling systems.

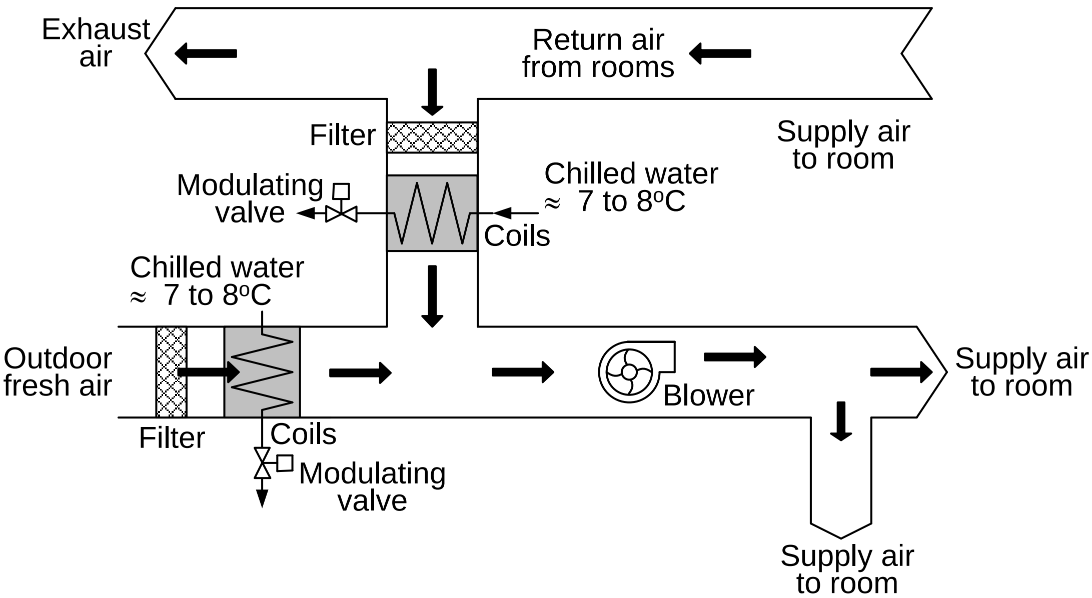


Figure 3.37 Schematic diagram of dual-path air handling system

3.15.3 Make-up Air Unit (MAU)

i. Few industrial production spaces, such as pharmaceutical plant, semiconductor fab, operation theatre, lab areas etc., require low space relative humidity (RH) and may need to supply 100% fresh air due to the generation of contaminants.

ii. For these applications, usually low temperature chilled water (about 4 to 5ºC) is supplied to the cooling coils of the air handling units called make-up air unit (MAU). Supply air is overcooled by the cooling coils to remove sufficient moisture. Thereafter, the overcooled supply air is reheated using electric duct heaters before supplying the air into the production spaces as shown in Figure-3.38.

iii. This process not only wastes electrical energy in the electric duct heaters but also increases the cooling load of the chilled water systems. Due to the operation of the chillers at low chilled water supply temperature, cooling capacity of chillers drops while the lift of chiller compressor and subsequent chiller power consumption increases significantly.

iv. Run-around coil as shown in Figure-3.39 can be used instead of the duct heater, which leads to the reduction of chiller cooling load and energy consumption. Due to the precooling of the outdoor fresh air using the run-around coil, chilled water supply temperature can also be increased as illustrated in Figure-3.39, which will eventually increase the efficiency of the chillers.

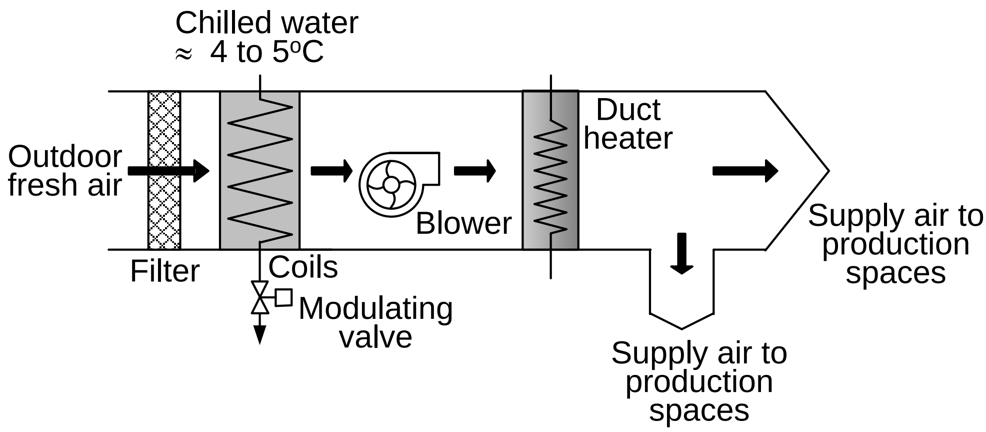


Figure 3.38 Schematic diagram of make-up air unit

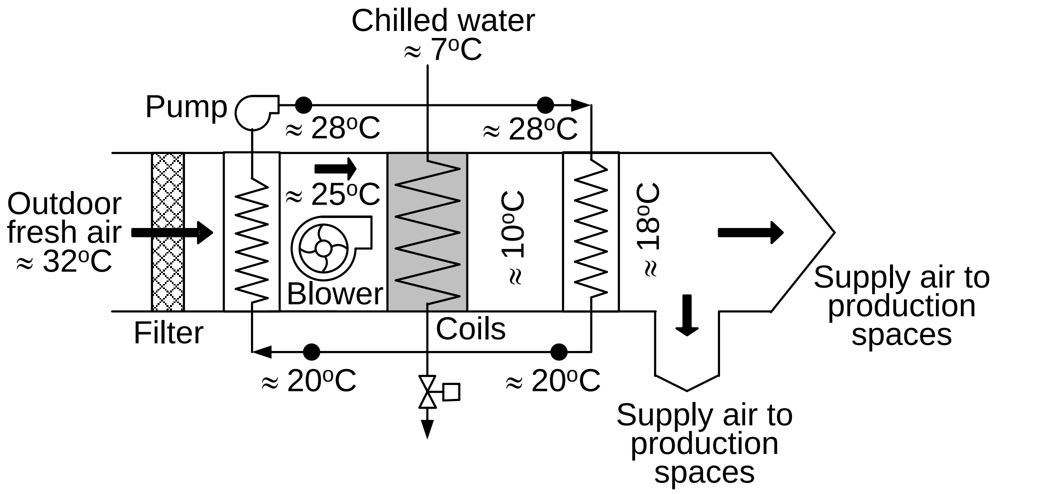


Figure 3.39 Example of temperature distribution in run-around coil systems

3.15.4 AHU with Direct Expansion Systems

i. In the refrigerant direct expansion systems, refrigerant is evaporated inside a finned evaporator coil installed inside the duct of the AHU and the supply air flows directly over the evaporator coil as shown in Figure-3.40.

ii. Air flow velocity and the depth of evaporator coil are designed in such a manner that the flowing air reaches the required dew point while flowing over the evaporator coil, resulting in sufficient dehumidification.

iii. Refrigerant direct expansion is not an energy efficient process. This system can be used to maintain relatively low RH in small spaces but not large or the entire space of a building.

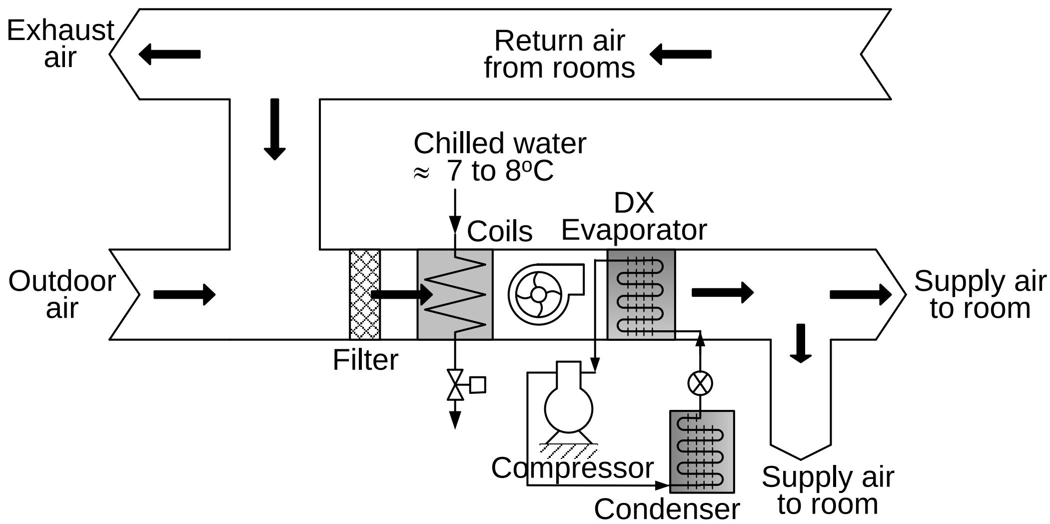


Figure 3.40 Schematic diagram of AHU with direct expansion refrigeration system

iv. Direct expansion coils are usually installed in series with the chilled water coil of AHUs that serve spaces with low RH requirement. Central chilled water systems can be operated at a relatively higher chilled water supply temperature, resulting in improvement of the chilled water system energy efficiency.

3.15.5 AHU with Solid Desiccant Systems

i. Commonly used solid desiccant systems consist of a porous rotor of solid desiccant with two isolated air paths through the rotor as shown in Figure-3.41. Recirculating air flows through one part of the rotor while hot regenerative air flows through the other. As the rotor rotates, solid desiccant adsorbs moisture from the stream of recirculating air in one path and then moves to the other path where adsorbed moisture is evaporated from the solid desiccant by the hot regenerative air.

ii. Hot regenerated solid desiccant then moves to the path of recirculating air again and transfers the absorbed heat while dehumidifying the recirculating air. Dry recirculating air usually leaves the rotor at a higher enthalpy than it entered. The dehumidified circulating air is cooled before supplying to the space resulting in almost no reduction of cooling load for the chilled water system.

iii. As the AHUs need to provide only sensible cooling, chilled water supply temperature of about 8 to 10^{o}C (depending on the design of the cooling coil) can be used in the AHU system. This results in significant improvement of chiller efficiency.

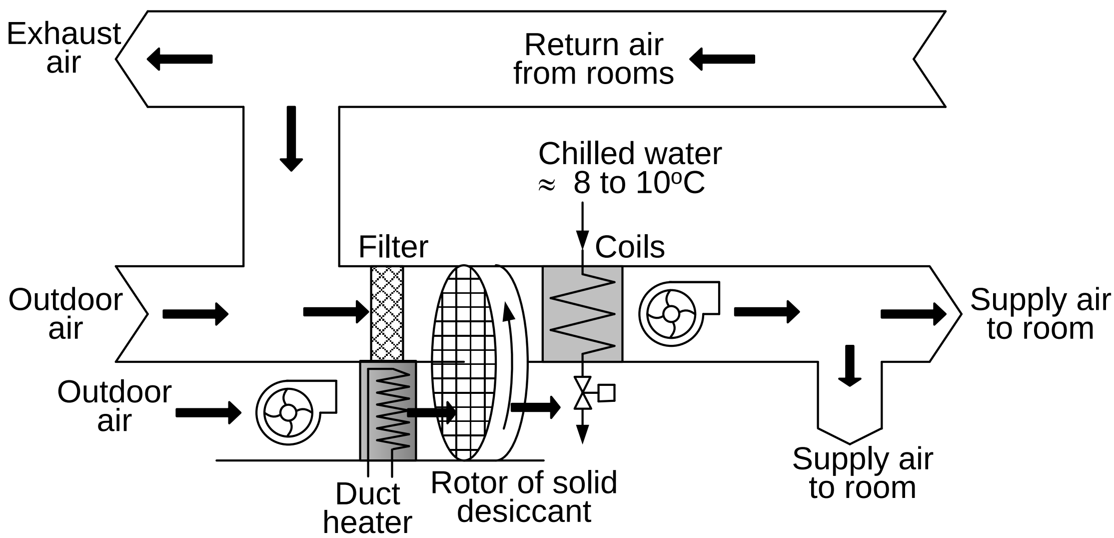


Figure 3.41 Schematic diagram of AHU with solid desiccant systems

3.15.6 AHU with Energy Recovery Systems

Based on the type of application and activities involved in the air-conditioning spaces, a certain percentage of the return air from the room is exhausted to the atmosphere and an equal amount of outdoor fresh air is supplied with the return air to the AHUs. As the exhaust air is cooler than the outdoor fresh air, energy recovery wheel and heat pipe systems can be used to precool the outdoor fresh air using the exhaust air.

Energy recovery wheel: Energy recovery wheel is a porous disk made of fairly high heat capacity materials. The energy recovery wheel is installed between the side-by-side exhaust and fresh air ducts as shown in Figure-3.42 and is rotated slowly. Sensible heat is transferred from the material of the wheel to the exhaust cold air. As the wheel rotates, the cold section of the wheel enters the fresh air duct and heat is transferred from the fresh air to the cold material of wheel resulting in the precooling of fresh intake air. The flow passages of the energy recovery wheel can be coated with hygroscopic materials (such as desiccant) to partially absorb the moisture of the fresh intake air. Hence, the fresh intake air is precooled and dehumidified. This process is known as total energy recovery process. As the wheel rotates, wet hygroscopic materials enter the exhaust air duct where the moisture of the hygroscopic materials is removed by the exhaust cold air of low relative humidity.

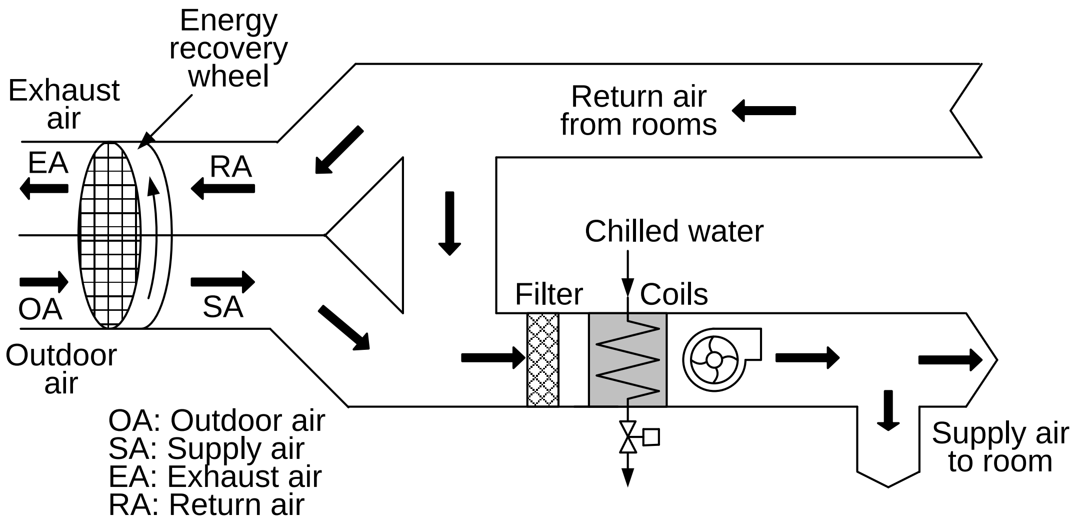


Figure 3.42 Energy recovery wheel installed in AHU duct

Efficiency of sensible energy recovery process is expressed as:

$$\eta_{\text{sensible}} = \left(\frac{T_{OA} - T_{SA}}{T_{OA} - T_{RA}} \right) x100 \quad (3.33)$$

where

$\eta_{sensible}$ = efficiency of sensible energy recovery process, %

T_{OA} = temperature of outdoor air, ^oC

T_{SA} = temperature of supply air, ^oC

T_{RA} = temperature of return air, ^oC

Sensible energy recovery rate can be calculated as:

$$Q_{\text{sensible}} = V\rho C_p \left(T_{OA} - T_{SA} \right) \quad (3.34)$$

where

$Q_{sensible}$ = sensible energy recovery rate, kW

V = volume flow rate of outdoor fresh air, m^3/s

ρ = density of outdoor fresh air, kg/m^3

C_p = specific heat capacity of air, kJ/(kg K)

T_{OA} = temperature of outdoor air, ^oC

T_{SA} = temperature of supply air, ^oC

Efficiency of total energy recovery process is expressed as:

$$\eta_{\text{total}} = \left(\frac{h_{OA} - h_{SA}}{h_{OA} - h_{RA}} \right) x100 \quad (3.35)$$

where

η_{total} = efficiency of total energy recovery process, %

h_{OA} = enthalpy of outdoor air, kJ/kg dry air

h_{SA} = enthalpy of supply air, kJ/kg dry air

h_{RA} = enthalpy of return air, kJ/kg dry air

Total energy recovery rate can be calculated as:

$$Q_{\text{total}} = m_{dry\,air} \left(h_{OA} - h_{SA} \right) \quad (3.36)$$

where

Q_{total} = total energy recovery rate, kW

$m_{dry\ air}$ = mass flow rate of outdoor fresh air, kg dry air/s

h_{OA} = enthalpy of outdoor air, kJ/kg dry air

h_{SA} = enthalpy of supply air, kJ/kg dry air

Heat pipe: A typical heat pipe is a closed evaporator-condenser system consisting of a sealed tube made up of high thermal conductivity material (such as copper, aluminium etc.) whose inside wall is lined with a capillary structure or wick as shown in Figure-3.43. Based on the operating temperatures of the hot and cold streams, the tube is partially filled with a working fluid (such as water, ammonia, R134a, etc.), vacuumed to a particular pressure and then sealed. Fins can be installed on the evaporator and condenser sections to enhance the heat transfer performance. The evaporator and condenser of the heat pipe are installed at the hot and cold streams respectively. The liquid working fluid turns into vapour by absorbing the latent heat of vapourisation at the evaporator of the heat pipe. The vapour then travels along the heat pipe to the condenser and condenses back into liquid by rejecting the latent heat. The liquid then returns to the evaporator due to the capillary action exerted by the wick structure and the cycle repeats.

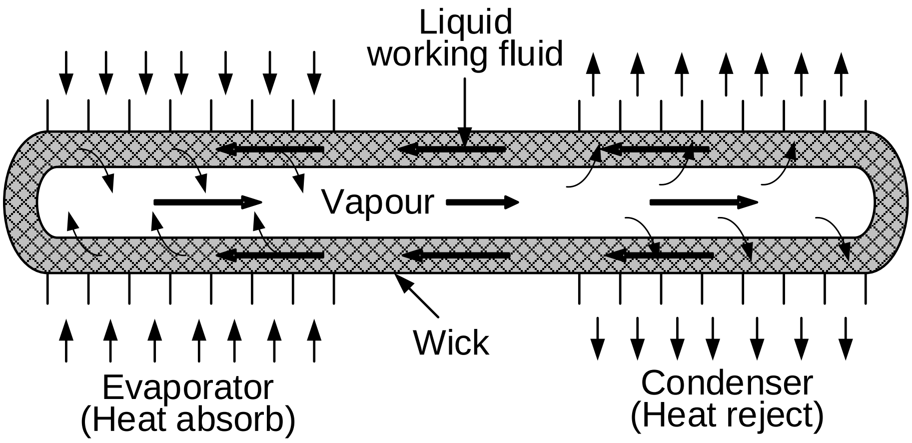


Figure 3.43 Typical heat pipe

Similar to the energy recovery wheel, a number of heat pipes are installed between the side-by-side exhaust and fresh air ducts as shown in Figure-3.44. The evaporator and condenser of the heat pipes are mounted in the fresh air duct and return air duct respectively. Liquid working fluid of the heat pipes absorbs the latent heat of vapourisation from the fresh air resulting in precooling of the fresh air. The vapour then travels to the condenser and rejects the latent heat to the exhaust air. The liquid working fluid then moves to the evaporator by capillary action and repeats the cycle. As there is no moving part, operation and maintenance costs for heat pipe energy recovery systems are low.

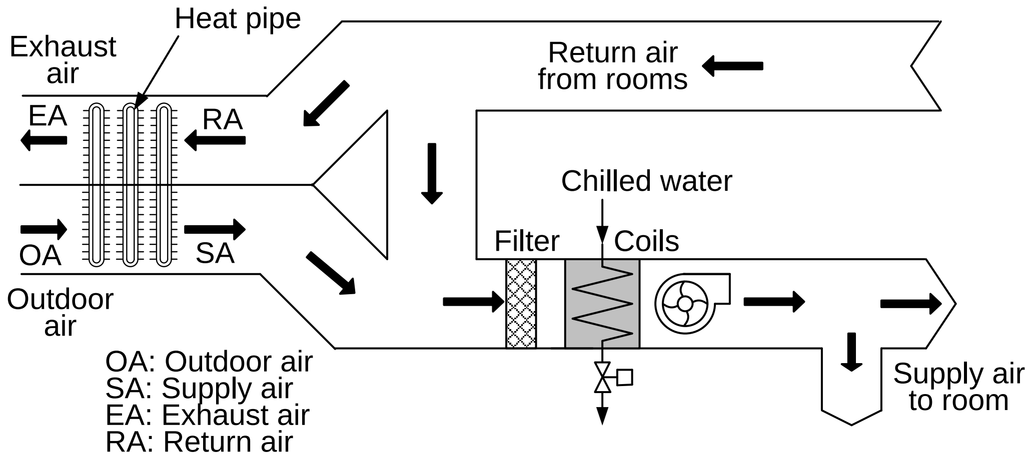


Figure 3.44 Heat pipe system installed in AHU duct.

3.15.7 Displacement Ventilation System

In conventional overhead ventilation systems, cool and dehumidified air is supplied to the air-conditioned spaces at relatively high velocity from or near the ceiling. High flow velocity of supply air causes induction and mixing of room air with the supply air. As the cold supply air mixes with the warm air in the room before reaching the level of the occupants, supply air is required to be cooled to a relatively low temperature. Moreover, supply clean air and contaminated room air mix resulting in relatively poor indoor air quality.

A displacement ventilation system, on the other hand, uses natural forces to distribute the air in the air-conditioned spaces. The system uses an innovative technology based on two major principles namely buoyancy and stratification. Conditioned cold air is supplied at low velocity through air diffusers located near the floor. As the density of cold air is high, the supply cold air spreads as a thin layer over the floor by displacing the warmer contaminated room air. Due to the absorption of heat from the heat sources, such as occupants and appliances, the cold air becomes warmer and less dense. The warm low density air creates upward convection flow know as thermal plumes. Finally, the warm air is extracted at ceiling height above the occupied zone. Displacement ventilation systems are quieter than conventional overhead systems and provide a desirable acoustic environment. Supply air temperature could also be increased slightly in comparison to the conventional ventilation system. Due to the minimum mixing of the supply air and the contaminated room air, indoor air quality could be improved. However, the system may create sensations of cold at the feet region while warm sensations at the head resulting in discomfort. Displacement ventilation systems usually provide acceptable comfort if the cooling load is relatively low.

3.15.8 Heat Pump System

Heat pumps and air-conditioning machines operate on the same cycle. Similar to the air-conditioning machines, main components of heat pumps are compressor, condenser, expansion valve and evaporator. The objective of the air-conditioning machines is to cool down spaces. Discharging of heat from the condenser to the ambient air is a necessary part of the operation of the air-conditioning machines. In contrast to air-conditioning machines, the objective of heat pumps is to heat up a medium or space. Cooling the ambient air by absorbing the heat in the evaporator is a necessary part of operation of the heat pumps. A typical heat pump system is shown in Figure-3.45.

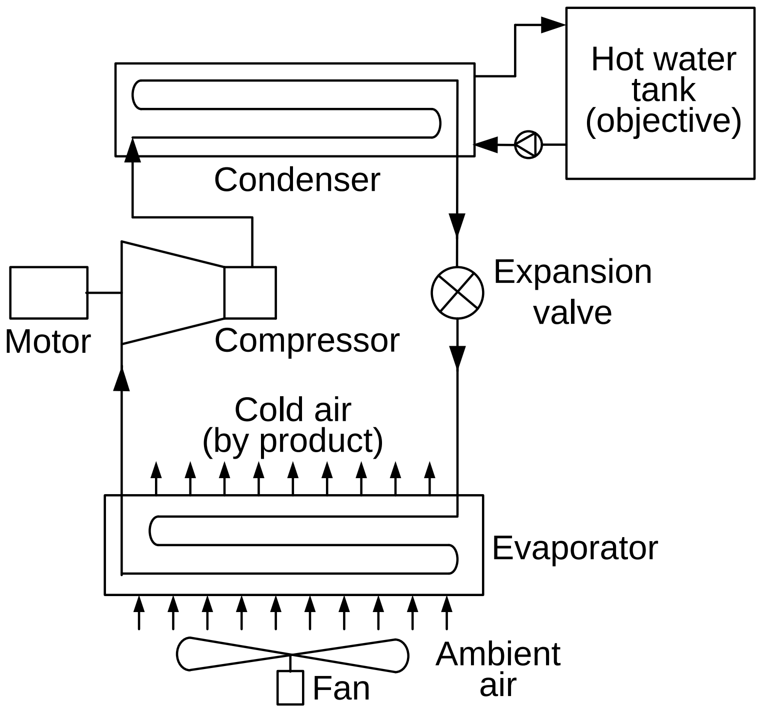


Figure 3.45 Typical heat pump system

The coefficient of performance (COP) of heat pumps is defined as the ratio of the desired heating effect to the electrical power input to the motor of the compressor:

$$COP_{HP} = \frac{\text{Desired heating effect in condenser, } Q_{cond} \text{ (kW)}}{\text{Input power to motor of compressor, } W_{input} \text{ (kW)}} \tag{3.37}$$

Therefore,

$$W_{input}\,(kW) = \frac{Q_{cond}\,(kW)}{COP_{HP}} \tag{3.38}$$

The COP of air-conditioning machines, on the other hand, is defined as the ratio of the desired cooling effect to the electrical power input to the motor of the compressor:

$$COP_{A/C} = \frac{\textit{Desired cooling effect in evaporator, } Q_{eva}\,(kW)}{\textit{Input power to motor of compressor, } W_{input}\,(kW)} \tag{3.39}$$

For the hermetic compressor, energy balance of the refrigeration cycle gives:

$$Q_{cond} = Q_{eva} + W_{input} \tag{3.40}$$

where,

Q_{cond} = heat rejection rate of condenser, kW

Q_{eva} = heat absorption rate of evaporator, kW

W_{input} = input electrical power to the compressor, kW

Combining Eqs. (3.37), (3.39) and (3.40) gives:

$$COP_{HP} = \frac{Q_{eva} + W_{input}}{W_{input}}$$

$$COP_{HP} = \frac{Q_{eva}}{W_{input}} + 1$$

$$COP_{HP} = COP_{A/C} + 1 \tag{3.40}$$

Heat pumps are used in hotel, kitchen, swimming pool and industrial processes for generating hot water of about 50 to 90^{o}C in an energy efficient manner. For instance, the required heat energy for a hot water system is Q kW. If electric heater is used to heat up the water, input electrical energy to the electric heater will be Q kW. However, if the COP of a heat pump is 4 and is used to heat up the water, the input electrical energy to the compressor of the heat pump will be Q/4 kW which is only one-fourth of the input energy of electric heater (Eq. 3.38). The cold air generated by the evaporator of the heat pumps is usually rejected to the atmosphere. The generated cold air can be supplied to the non-critical areas such as common corridor of hotel, hospital or industrial buildings to provide the cooling or the cold air can be supplied to the PAU to reduce its cooling load. If the cooling effect of the heat pump is used for space cooling or precooling the fresh air, the overall performance $COP_{overall}$ of the heat pump becomes:

$$COP_{overall} = \frac{Heating\ effect, Q_{cond}\ (kW) + Cooling\ effect, Q_{eva}\ (kW)}{Input\ power\ to\ motor\ of\ compressor, W_{input}\ (kW)} \qquad (3.41)$$

$$COP_{overall} = COP_{HP} + COP_{A/C} \qquad (3.42)$$

3.16 Fan Performance and Operational Requirements Based on Singapore Standards:

Fan Power Limitation: Based on Singapore Standard SS553, allowable specific power consumption for the fans of air-conditioning systems is summarised in Table 3.6.

Table 3.6 Fan specific power consumption limitations for air-conditioning systems

Constant volume flow systems	Variable volume flow systems
1.7 kW/(m^3/s)	2.4 kW/(m^3/s)
0.472 W/CMH	0.67 W/CMH

(Source: Reproduced from Singapore Standard SS553 with permission from SPRING Singapore. Please refer SS553 for details. Website: www.singaporestandardseshop.sg)

Table 3.6 is applicable to air-conditioning systems having a total fan power exceeding 4 kW.

Air-Conditioning Space Requirements: Based on Singapore Standard SS553, air-conditioning space requirements are:

i. Normal design dry-bulb temperature for comfort air-conditioning can vary from 23°C to 25°C.
ii. When air-conditioning systems are in operation, the operative temperature should be maintained between 24°C and 26°C.
iii. Air movement in the air-conditioned spaces should not exceed 0.30 m/s measured at the occupants' level, which is 1500 mm from the floor.

iv. Average RH in the air-conditioned spaces should not exceed 65% for new buildings and 70% for existing buildings.

v. Cool air leaving supply diffuser should be designed at a temperature less than 2^{o}C below room dew point to prevent moisture condensing on the diffuser surface.

Outdoor Air Supply Requirement: Based on Singapore Standard SS553, outdoor air supply requirements for comfort air-conditioning spaces are summarised in Table 3.7.

Table 3.7 Outdoor air supply requirements for comfort air-conditioning spaces

Type of building/Occupancy	Minimum outdoor air supply	
	L/s per m^2 floor area	L/s per person
Restaurants	3.4	5.1
Dance halls	7.0	10.5
Offices	0.6	5.5
Shops, supermarkets and department stores	1.1	5.5
Theatres and cinemas seating area	2.0	3.0
Lobbies and corridors	0.3	3.3
Concourses	1.1	3.3
Hotel guest rooms	15.0 L/s per room	5.5

(Source: Reproduced from Singapore Standard SS553 with permission from SPRING Singapore. Please refer SS553 for details. Website: www.singaporestandardseshop.sg)

Chapter-4: Psychrometrics of Air-conditioning Processes

4. Psychrometrics of Air-conditioning Processes

In buildings with central air-conditioning system, moist air is treated in the air handling units (AHUs) and then supplied to the spaces to maintain pre-set space temperature and relative humidity (RH). The treatment processes of moist air in the AHUs involve heat and mass transfer between the moist air and the cooling or heating coils of AHUs. Common heat and mass transfer processes of AHUs include sensible cooling and heating processes, dehumidification by partial condensation of moisture and humidification by adding water vapour in the AHUs and air-conditioned spaces. The properties of moist air at different stages of AHUs and air-conditioned spaces can conveniently be determined using Psychrometric chart. In this section of the reference manual, relevant properties of the moist air are defined and examples on calculation of properties using the Psychrometric chart are presented. Finally, the Psychrometric principles are used to calculate the heat load, moisture transfer rate and required air flow rate for the sizing and optimisation of the AHU systems. Cooling load and associated energy saving opportunities for the AHUs by modulating the fresh air flow rate are discussed with examples.

Learning Outcomes

Participants will be able to:

i. Determine properties of moist air using Psychrometric chart.
ii. Draw the heat and mass transfer processes involved in the AHUs and air-conditioned spaces on the Psychrometric chart.
iii. Calculate the heat and mass transfer rate of the AHUs and air-conditioned spaces.
iv. Analyse the effects of outdoor fresh air flow rate on the cooling load of AHU coil.
v. Size the cooling and heating coils and calculate the required air flow rate for the AHU.

4.1 Properties of Moist Air

A typical AHU with air distribution system is shown schematically in Figure-4.1. The main components of an AHU are filters, cooling and heating coils, dampers and fans. Sensible and latent heat loads generated inside the air-conditioned spaces by occupants, lights, appliances, electric machines, building façade etc. are absorbed by the treated supply air in order to maintain pre-set conditions in the spaces. The relatively warm and humid air is then returned from the conditioned spaces to the AHUs. To maintain good space Indoor Air Quality (IAQ), certain percentage of the return air is exhausted to the atmosphere and the required amount of outdoor fresh air is mixed with the return air as shown in Figure 4.1. The quantity of outdoor fresh air required for an AHU mainly depends on ventilation requirements of the occupants and the necessary pressurisation to prevent infiltration of air into the spaces. The mixture of the return and outdoor fresh air is filtered and then treated (cooling and dehumidification or heated) by the coils. Finally, the fan transports the treated air to the spaces through the supply ducting system.

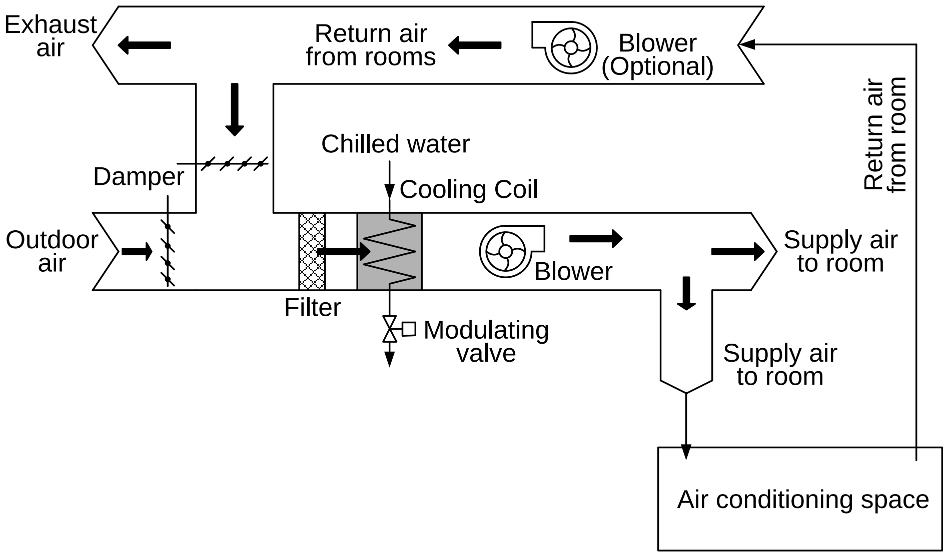


Figure 4.1 Typical air handling unit (AHU) and ducting system

Heat and moisture transfer processes involved in the AHU and air-conditioned spaces can conveniently be analysed using Psychrometric chart. A number of parameters of moist air such as dry-bulb temperature, wet-bulb temperature, dew-point temperature, specific volume, humidity ratio, relative humidity and enthalpy are appropriately presented together on the Psychrometric chart as shown in Figure-4.2. Before analysing the heat and moisture transfer processes using the Psychrometric chart, a brief description of the parameters are presented in the following section:

Dry-Bulb Temperature, T_{db} (Unit: ^{o}C): The temperature of air indicated by an accurate thermometer shielded from the effect of radiation is known as the dry-bulb temperature.

Wet-Bulb Temperature, T_{wb} (Unit: ^{o}C): The temperature at which water by evaporation into air can bring the air to saturation adiabatically is known as the wet-bulb temperature. If the bulb of a thermometer is covered with wet cotton and air is blown at a velocity of about 2 to 4 m/s over the wet cotton, the temperature indicated by the thermometer is wet-bulb temperature. As air is blown over the wet cotton, water is evaporated by absorbing latent heat of vapourisation from the wet cotton. As a result, wet cotton temperature reaches the wet-bulb temperature. Water evaporation rate and the resulting wet-bulb temperature depend on the relative humidity of surrounding air. If relative humidity of surrounding air is 100% (saturated air), water evaporation rate will drop to zero and wet-bulb temperature will be equal to the dry-bulb temperature.

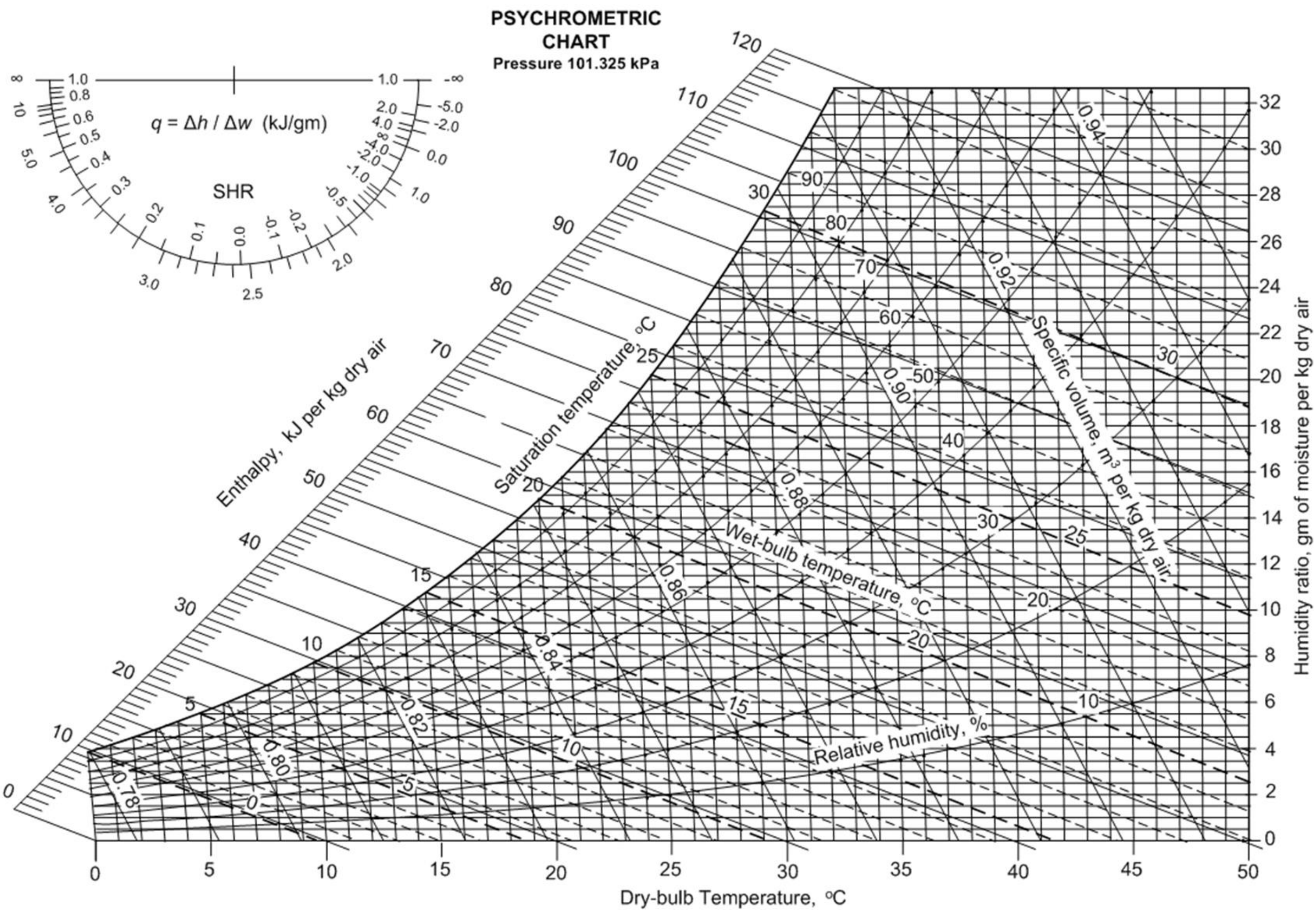


Figure 4.2 Psychrometric chart

Dew-point Temperature, T_{dp} (Unit: ^{o}C): If moist air (air vapour mixture) is cooled at constant pressure, the temperature at which condensation of moisture starts is known as the dew-point temperature.

Specific Volume, v (m^3/kg dry air): The volume of air-vapour mixture that contains unit mass (e.g. 1 kg) of dry air is called specific volume. Let us consider a container of volume 2 m^3 that contains a mixture of 2.35 kg dry air and 0.02 kg moisture as shown in Figure 4.3. The specific volume of air-vapour mixture = 2 m^3 / 2.35 kg of dry air = 0.85 m^3/kg dry air.

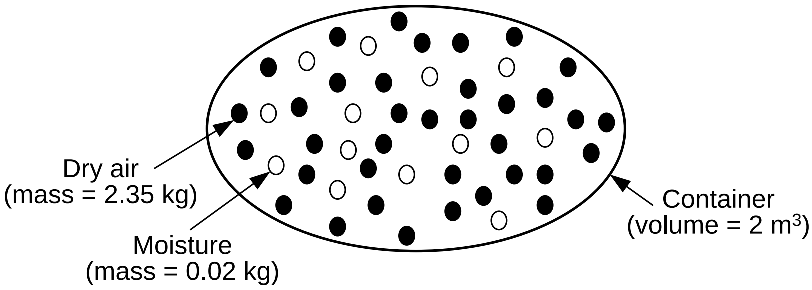


Figure 4.3 Air vapour mixture in a control volume

Humidity Ratio, ω (Unit: kg moisture / kg dry air): Mass of water vapour present in a unit mass of dry air is called Humidity ratio. It is also known as absolute humidity or specific humidity.

$$Humidity\ Ratio = \frac{Mass\ of\ moisture\ in\ a\ certain\ volume\ of\ moist\ air}{Mass\ of\ air\ in\ the\ same\ volume\ of\ moist\ air} \qquad (4.1)$$

For the control volume shown in Figure 4.3:

Humidity ratio = 0.02 kg moisture / 2.35 kg dry air

= 0.0085 kg moisture / kg dry air

= 8.5 g moisture / kg dry air

As humidity ratio represents the absolute or actual moisture content of moist air, absolute humidity does not change with the change of temperature unless addition of water vapour or condensation of moisture takes place.

Relative Humidity, ϕ (Unit: %): Relative humidity is the ratio of actual moisture content in moist air of specific volume at a temperature T and pressure P to the maximum quantity of moisture the same moist air can hold at the same temperature T and pressure P.

$$Relative\ Humidity = \frac{Actual\ amount\ of\ moisture\ in\ a\ specific\ volume\ of\ moist\ air\ at\ T\ \&\ P}{Maximum\ amount\ of\ moisture\ the\ moist\ air\ can\ hold\ at\ the\ same\ T\ \&\ P} \qquad (4.2)$$

For the control volume shown in Figure 4.3, the moisture content in the moist air is 0.02 kg. Suppose the temperature and pressure of the moist air are 23^{o}C and one atmospheric pressure respectively. If the same moist air can hold maximum 0.04 kg of moisture (to be saturated) at the temperature of 23^{o}C and one atmospheric pressure:

Relative humidity of the moist air = 0.02 / 0.04 = 50%.

Moisture holding capacity of moist air increases with the increase in temperature. Suppose that the control volume shown in Figure-4.3 is heated to the temperature of 30^{o}C and moisture-holding capacity of the air is increased to 0.064 kg at the temperature of 30^{o}C:

Relative humidity of the moist air = 0.02 / 0.064 = 31.3%.

Note that the actual or absolute moisture content of the moist air is not changed (remains at 0.02 kg) due to the increase of temperature from 23^{o}C to 30^{o}C. However, relative humidity of the moist air drops from 50% to 31.3% due to the increase in moisture holding capacity of the air at a higher temperature. Similarly, the same air may reach 100% relative humidity without changing the actual moisture content if the temperature of the air is decreased, which reduces the maximum moisture holding capacity.

Enthalpy, h (Unit: kJ/kg of dry air): Enthalpy is the energy content of moist air. Enthalpy is expressed per kg of dry air basis. Suppose that the energy content of the moist air (mixture of 2.35 kg dry air and 0.02 kg moisture) shown in Figure-4.3 is 105 kJ.

Enthalpy = 105 kJ / 2.35 kg dry air = 44.7 kJ/kg dry air.

4.2 Determination of Moist Air Properties Using Psychrometric Chart

All the above-mentioned parameters of the moist air are properly presented together on the Psychrometric chart. If any two parameters of the moist air are known, it is possible to identify the intersection point of the parameters and all the remaining parameters can conveniently be obtained corresponding to the intersection point from the Psychrometric chart as shown in Figure 4.4.

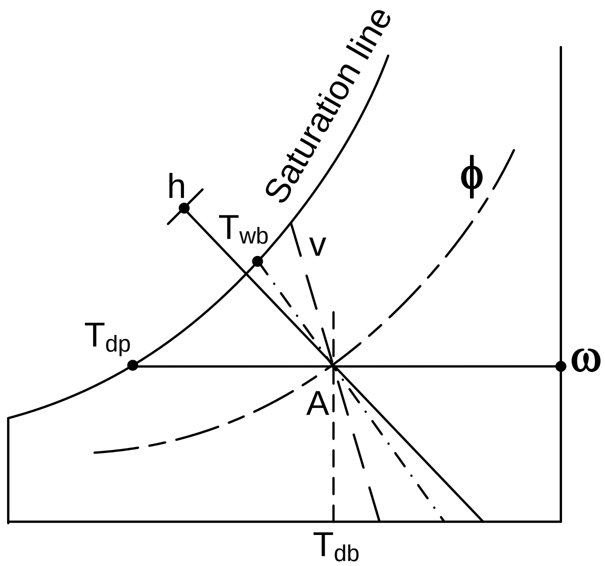


Figure 4.4 Psychrometric chart showing different property lines

Suppose that A is the intersection point of dry-bulb temperature T_{db} and relative humidity ϕ. All remaining properties of the moist air can conveniently be determined from the Psychrometric chart by following the corresponding lines from the intersection point A as shown in Figure 4.4.

Example 4.1

A room contains air at 1 atm, 25^{o}C and 60% relative humidity. Using Psychrometric chart, determine (a) Wet-bulb temperature, (b) Absolute humidity, (c) Dew-point temperature, (d) Specific volume and (e) Enthalpy of air.

Solution

Identify the intersection point of 25^{o}C dry-bulb temperature and 60% relative humidity on the Psychrometric chart. Following the corresponding lines on Psychrometric chart:

(a) Wet bulb temperature = 19.5^{o}C

(b) Absolute humidity = 12 g moisture/kg dry air

(c) Dew-point temperature = 16.8^{o}C

(d) Specific volume of air = 0.861 m^{3} /kg dry air

(e) Enthalpy of air = 56 kJ/kg dry air

4.3 Types of Heat Gain in Spaces

Heat gain in the air-conditioned spaces represents the rate at which heat energy is transferred to or generated within the spaces. Conversely, the cooling load is defined as the rate at which the heat energy must be removed at any instant by the conditioned air passing through the spaces to maintain the comfort temperature and relative humidity. Ideally, the heat gain and the cooling load should be the same. However, they generally differ due to the thermal inertia (storage effect) of the building structure. The heat gain or heat load has two components:

i. Sensible heat: It causes the increase of space temperature.
ii. Latent heat: It causes the increase of space moisture content. Moisture contains the latent heat of vapourisation which is removed by condensing moisture in the cooling coil of AHU.

Sensible and latent heat are generated inside the air-conditioning spaces mainly by (1) occupants, (2) lights, (3) appliances, (4) electric machines, (5) presence of hot pipes and ducts in the conditioned spaces and (6) miscellaneous loads. Sensible and latent heat are also transferred to the conditioning spaces by (1) direct solar radiation through windows, (2) convection and conduction heat transfer through building envelope which includes walls, windows, doors, roofs, floors etc., (3) direct solar radiation absorbed by the façade of the building, (4) moisture migrating into the air-conditioning spaces due to the effect of external vapour pressure, (5) infiltration of air into the building and (6) outdoor air induced for ventilation purposes. Air-conditioning system is designed to provide the conditioned air at a flow rate that can match the sensible and latent cooling loads of the spaces.

4.4 Processes Involved in AHU and Spaces

4.4.1 Sensible Heating

Consider air is flowing over a heating coil from state-1 to state-2 as shown in Figure 4.5.

Main observations:

i. No change of absolute moisture content ω of air as no moisture is added or removed during sensible heating process

ii. RH decreases due to the increase of moisture holding capacity of air at higher temperature

iii. Sensible heating process is horizontal line on Psychrometric chart

iv. At any position on the saturated line, $T_{db} = T_{wb}$

v. Heat transfer rate to the air can be calculated as $Q = \dot{m}_{air}(h_2 - h_1)$

where

$\dot{m}_{air}$ = Mass flow rate of dry air, kg dry air / s

h = Enthalpy of moist air, kJ/kg dry air

Q = Heat transfer rate to the moist air, kW

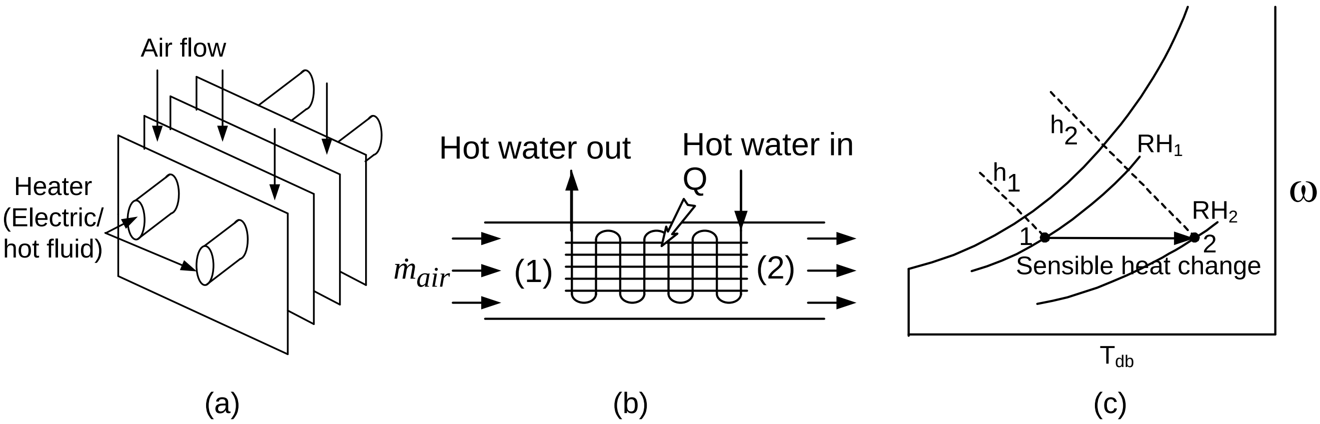


Figure 4.5 Sensible heating process: (a) Typical finned heating coil, (b) States of air in heating process and (c) States of air on Psychrometric chart

4.4.2 Sensible Cooling

Consider air is flowing over a cooling coil from state-1 to state-2 as shown in Figure 4.6. As cold water temperature is higher than the dew-point temperature of flowing air, no condensation of moisture occurs.

Main observations:

i. No change of absolute moisture content ω of air as the temperatures of cold water and the flowing air are higher than the dew-point temperature of flowing air. No condensation of moisture takes place during the cooling process.

ii. RH increases due to the decrease of moisture holding capacity of air at lower temperature

iii. Sensible cooling process is horizontal line on Psychrometric chart

iv. At any position on the saturated line, $T_{db} = T_{wb}$

v. Sensible cooling process line is simply the reverse of sensible heating process.

vi. Heat transfer rate from the air can be calculated as $Q = \dot{m}_{air}(h_1 - h_2)$

where

$\dot{m}_{air}$ = Mass flow rate of dry air, kg dry air / s

h = Enthalpy of moist air, kJ/kg dry air

Q = Heat transfer rate from the moist air, kW

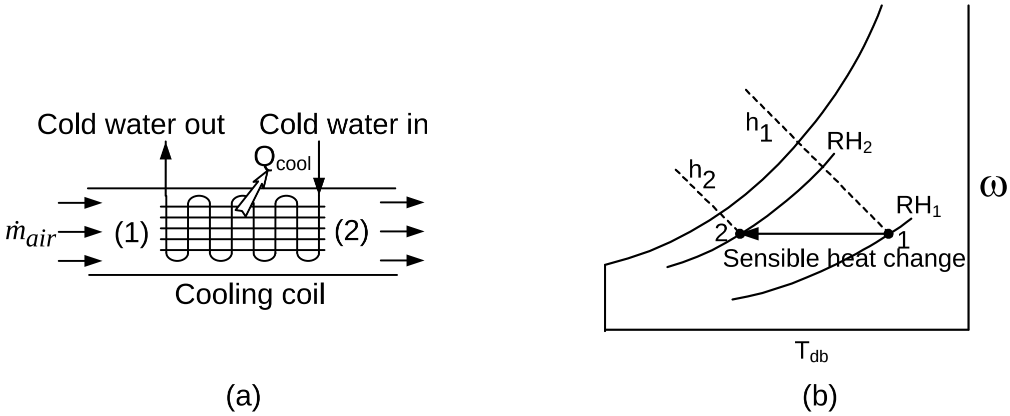


Figure 4.6 Sensible cooling process: (a) States of air in cooling process and (b) States of air on Psychrometric chart

4.4.3 Mixing of Two Air Streams

Mixing of different air streams is quite common in AHU systems. Consider stream-1 and stream-2 of moist air are mixing in a mixing chamber and then flowing out of the mixing chamber as shown in Figure 4.7.

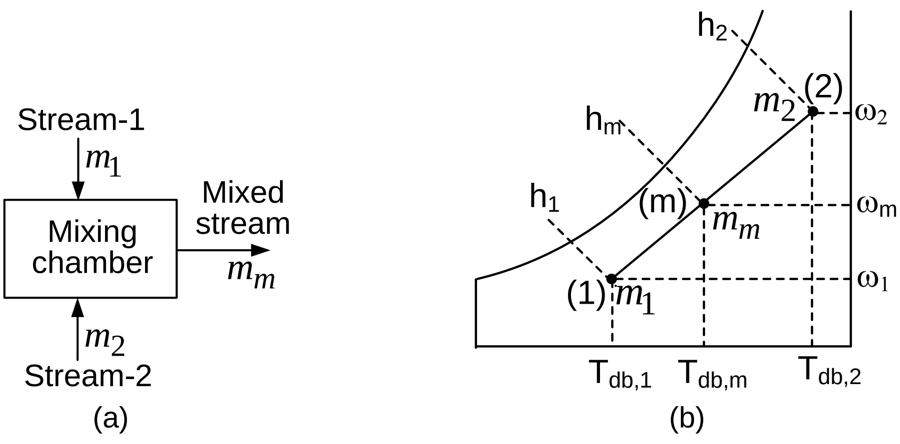


Figure 4.7 Mixing process of air streams: (a) Mixing of two air steams and (b) States of air streams on Psychrometric chart

Mass balance of dry air flow for the mixing chamber gives:

$$m_1 + m_2 = m_m \tag{4.3}$$

where

m_1 = mass flow rate of dry air for stream-1, kg dry air / s

m_2 = mass flow rate of dry air for stream-2, kg dry air / s

m_m = mass flow rate of dry air for mixed stream, kg dry air / s

Mass balance of moisture flow for the mixing chamber gives (McQuiston et al., 2005):

$$m_1\omega_1 + m_2\omega_2 = m_m\omega_m$$

$$(m_m - m_2)\omega_1 + m_2\omega_2 = m_m\omega_m$$

$$m_m\omega_1 - m_2\omega_1 + m_2\omega_2 = m_m\omega_m$$

$$\omega_m = \omega_1 + \frac{m_2}{m_m}(\omega_2 - \omega_1) \tag{4.4}$$

where

ω_1 = absolute humidity of air stream-1, kg moisture / kg dry air

ω_2 = absolute humidity of air stream-2, kg moisture / kg dry air

ω_m = absolute humidity of mixed air stream, kg moisture / kg dry air

Similarly, energy balance for the air streams gives:

$$m_1h_1 + m_2h_2 = m_mh_m$$

$$h_m = h_1 + \frac{m_2}{m_m}(h_2 - h_1) \tag{4.5}$$

where

h_1 = enthalpy of air stream-1, kJ / kg dry air

h_2 = enthalpy of air stream-2, kJ / kg dry air

h_m = enthalpy of mixed air stream, kJ / kg dry air

Energy balance for the air streams also gives:

$$T_{db,m} = T_{db,1} + \frac{m_2}{m_m}(T_{db,2} - T_{db,1}) \tag{4.6}$$

where

$T_{db,1}$ = dry-bulb temperature of air stream-1, ^oC

$T_{db,2}$ = dry-bulb temperature of air stream-2, ^oC

$T_{db,m}$ = dry-bulb temperature of mixed air stream, ^oC

If two properties for both stream-1 and 2 are known, point-1 and 2 can be located on Psychrometric chart. Absolute humidity, enthalpy and dry-bulb temperature of the mixed stream can be calculated using Eqs. (4.4), (4.5) and (4.6) respectively. Properties of mixed stream can also be determined from the intersection point of the straight line between point-1 and 2 (Figure 4.7 b) and any of the calculated property lines (absolute humidity or enthalpy or dry-bulb temperature) of the mixed stream.

Example 4.2

Figure 4.8 shows the mixing process of the return and outdoor fresh air of a typical AHU system. The measured values of the dry-bulb temperature, RH and flow rate of the return air (point-1) and the outdoor fresh air (point-2) are presented in Table 4.1. Calculate dry-bulb temperature, RH, absolute humidity, specific volume and enthalpy of the mixed air stream (point-3).

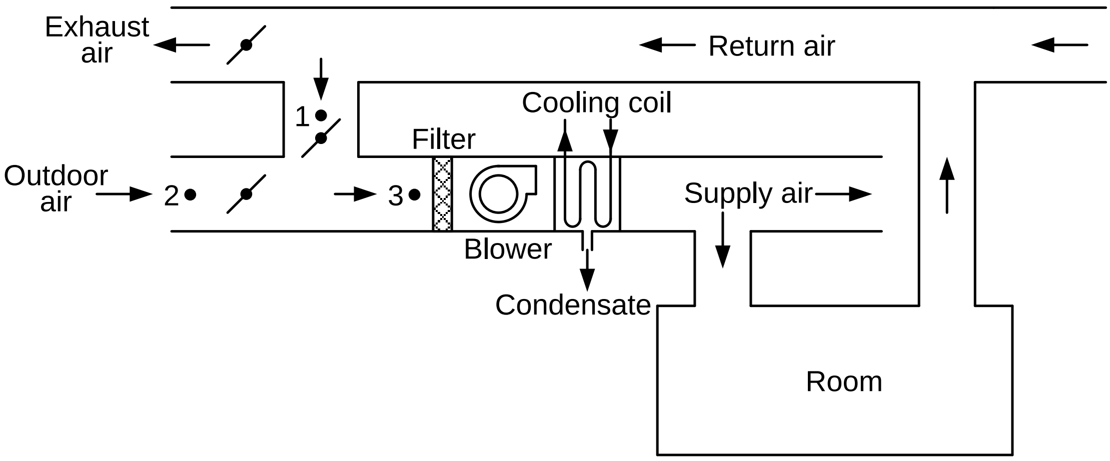


Figure 4.8 Mixing process of return and outdoor fresh air streams

Table 4.1 Measured parameters of the return and outdoor fresh air streams

Description	Point-1	Point-2
Flow, CMH	10,000	2,000
Dry-bulb Temperature, ^oC	23	32
RH, %	65	80

Solution

Return air flow rate Q_1 = 10,000CMH = 10,000/3600 = 2.78 m^3/s

Similarly, outdoor fresh air flow rate Q_2 = 2,000CMH = 2,000/3600 = 0.56 m^3/s

For dry-bulb temperature of 23°C and RH of 65%, point-1 can be located on Psychrometric chart as shown in Figure 4.9.

Using Psychrometric chart for point-1:

Specific volume v_1 = 0.854 m^3 /kg dry air

Absolute humidity ω_1 = 11.5 g moisture/kg dry air

Enthalpy h_1 = 52.5 kJ/kg dry air

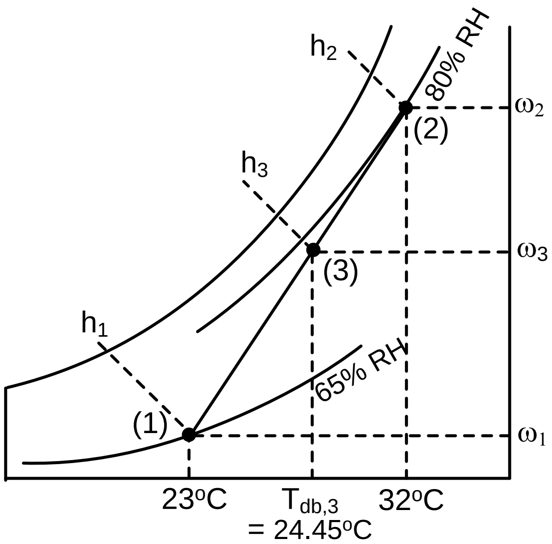


Figure 4.9 Mixing process of return and outdoor fresh air on Psychrometric chart

For dry-bulb temperature of 32°C and RH of 80%, point-2 can be located on Psychrometric chart as shown in Figure 4.9.

Using Psychrometric chart for point-2:

Specific volume v_2 = 0.898 m^3 /kg dry air

Absolute humidity ω_2 = 24.5 g moisture/kg dry air

Enthalpy h_2 = 95 kJ/kg dry air

Mass flow rate of dry air with return air m_1= Q_1/v_1 = 2.78/0.854 = 3.255 kg dry air/s

Mass flow rate of dry air with outdoor air m_2 = Q_2/v_2 = 0.56/0.898 = 0.624 kg dry air/s

Mass flow rate of dry air with mixed air m_3 = m_1+m_2 = 3.879 kg dry air/s

Mixed air properties:

Dry-bulb temperature $T_{db,3} = T_{db,1} + \frac{m_2}{m_3}\left(T_{db,2} - T_{db,1}\right)$

$T_{db,3}$ = 23+(0.624/3.879)(32-23) = 24.45^{o}C

Absolute humidity $\omega_3 = \omega_1 + \frac{m_2}{m_m}\left(\omega_2 - \omega_1\right)$

ω_3 = 11.5+(0.624/3.879)(24.5-11.5)

=13.59 g moisture/kg dry air

Enthalpy $h_3 = h_1 + \frac{m_2}{m_m}\left(h_2 - h_1\right)$ = 52.5+(0.624/3.879)(95-52.5) = 59.3 kJ/kg dry air

Using Psychrometric chart:

For $T_{db,3}$ = 24.45^{o}C and ω_3 = 13.59 g moisture/kg dry air

Relative humidity of mixed air RH_3 = 70%

4.4.4 Cooling and Dehumidification of Air

To maintain pre-set temperature and relative humidity inside the air-conditioned spaces, mixed air is cooled and dehumidified in the cooling coil by using chilled water of temperature lower than the dew point temperature of mixed air. The cooling and dehumidification process of moist air is shown in Figure 4.10.

Sensible cooling of moist air takes place in the entry region (point-1 to 1') of cooling coil where moist air is cooled to the dew point temperature. As a result, moisture does not condense in the entry region from point-1 to 1'of the coil. As moist air further flows across the cooling coil, moist air is cooled below the dew point temperature following saturation line from point-1 to 1' of the Psychrometric chart, resulting in condensation of moisture. A small fraction of the moist air flows between the gap of fins and cooling coils without contacting the surface of fins and cooling coils. This fraction of moist air is called by-pass air. Quantity of by-pass air depends on the compactness of cooling coil. It is assumed that by-pass air exits the cooling coil at the inlet condition (point-1). Major fraction of the moist air will get in contact with the surface of fins and cooling coils and exit the cooling coil at a temperature close to the fins and cooling coils surface temperature known as apparatus dew point (ADP) temperature. After leaving the cooling coil, the by-pass air will mix with the major stream of air and form a homogenous mixture of state-3 as shown in Figure 4.10 (c). As a result, real condition of air leaving the cooling coil (condition-3) is not saturated air at the ADP.

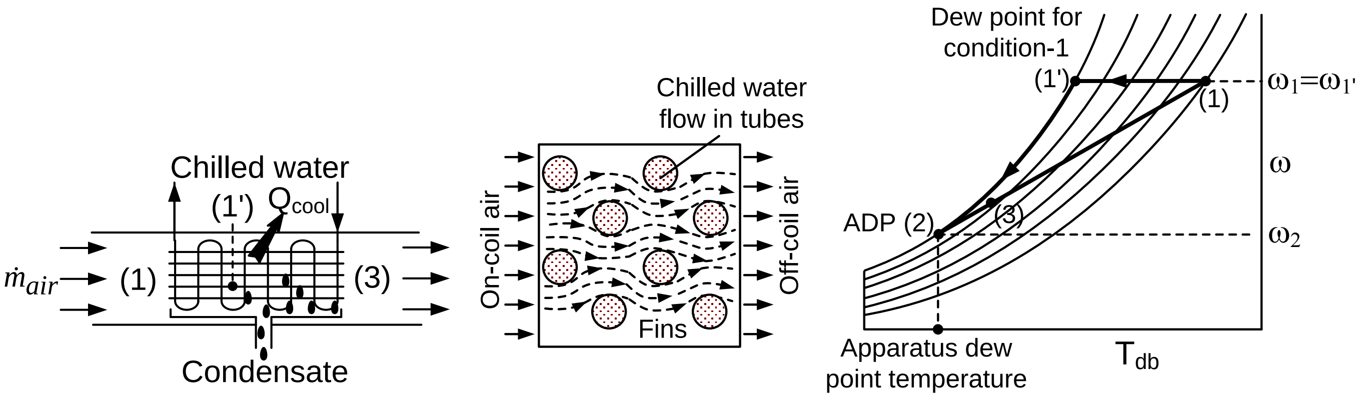


Figure 4.10. Cooling of moist air below dew point: (a) Process of cooling coil, (b) Flow of air inside cooling coil and (c) States of air on Psychrometric chart

4.4.5 Heat Loads for AHU

In a typical AHU system, the mixture of return and outdoor fresh air is filtered and then cooled and dehumidified by the cooling coil. The cold and dehumidified air is transported by a fan to the air-conditioned spaces through the supply ducting system as shown in Figure 4.11. The treated supply air absorbs sensible and latent heat of the spaces. Finally, the relatively warm and humid air is returned through the return duct to the AHU. To maintain space IAQ, certain percentage of the return air is exhausted to the atmosphere and the required amount of outdoor fresh air is mixed with the return air and the cooling cycle is repeated. Figure 4.12 shows the AHU and space conditioning processes on the Psychrometric chart.

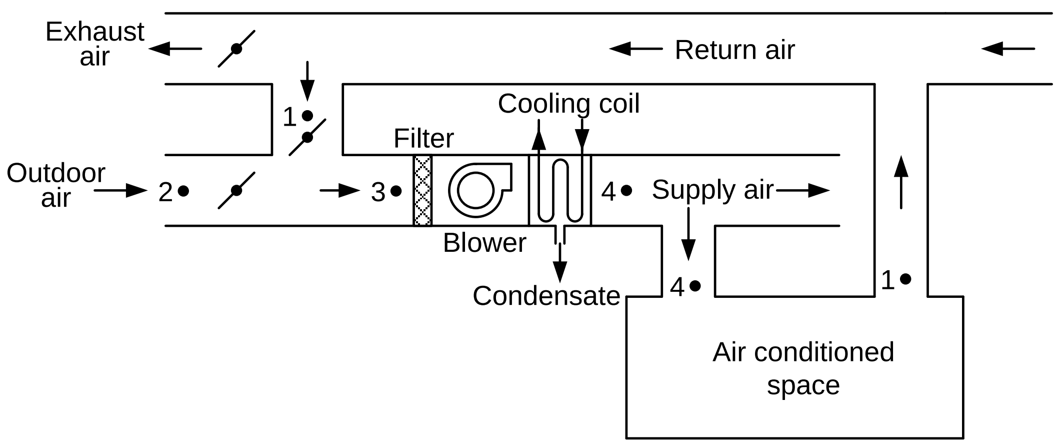


Figure 4.11 Typical AHU and ducting system

Space total heat load = $m_4(h_1\text{-}h_4)$

Space sensible heat load = $m_4(h_5\text{-}h_4)$

Space latent heat load = $m_4(h_1\text{-}h_5)$

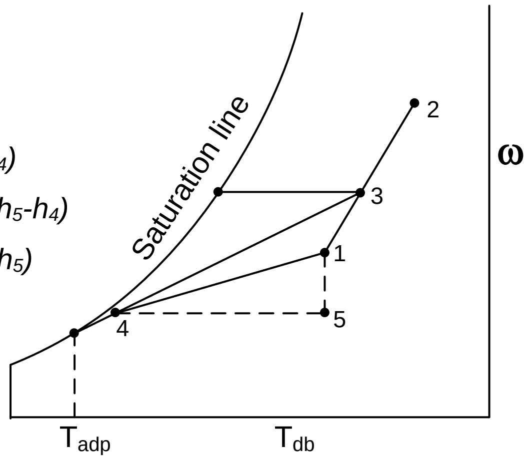


Figure 4.12 Typical AHU and space conditioning processes on the Psychrometric chart

4.4.6 Ventilation Effects

To maintain the concentration of CO_2 and other contaminants below the maximum acceptable values, certain percentage of the return air is exhausted to the atmosphere and a specific amount of outdoor fresh air is introduced and mixed with the return air. The amount of outdoor fresh air that is introduced intentionally to the air-conditioning space is known as the ventilation air. For a tropical country like Singapore, the outdoor fresh air is hot and humid. Air-conditioning systems take significant amount of energy to condition the ventilation air. To minimise the energy consumption of the air-conditioning systems, the quantity of the ventilation air should be maintained at the minimum possible amount to meet the indoor air quality requirements. The required quantity of ventilation air depends on the number of occupants and the type of space usage. ASHRAE Standard 62 is generally used to determine the quantity of ventilation air required for the spaces. Usually, the amount of ventilation air works out to be about 10 to 15 percent of the total circulating air.

Example 4.3

For the Example 4.2, consider dry-bulb temperature and relative humidity of the supply air after the cooling coil as 14^{o}C and 95% respectively.

Calculate:

a) Heat removal rate by the chilled water. Assume condensate temperature is 14^{o}C
b) Space total heat load
c) Space sensible heat load

d) Space latent heat load

e) Heat removal rate by chilled water if 0% outdoor air is used

f) Heat removal rate by chilled water if 100% outdoor air is used

Solution

Figure 4.15 shows the AHU system and the space conditioning processes on the Psychrometric chart.

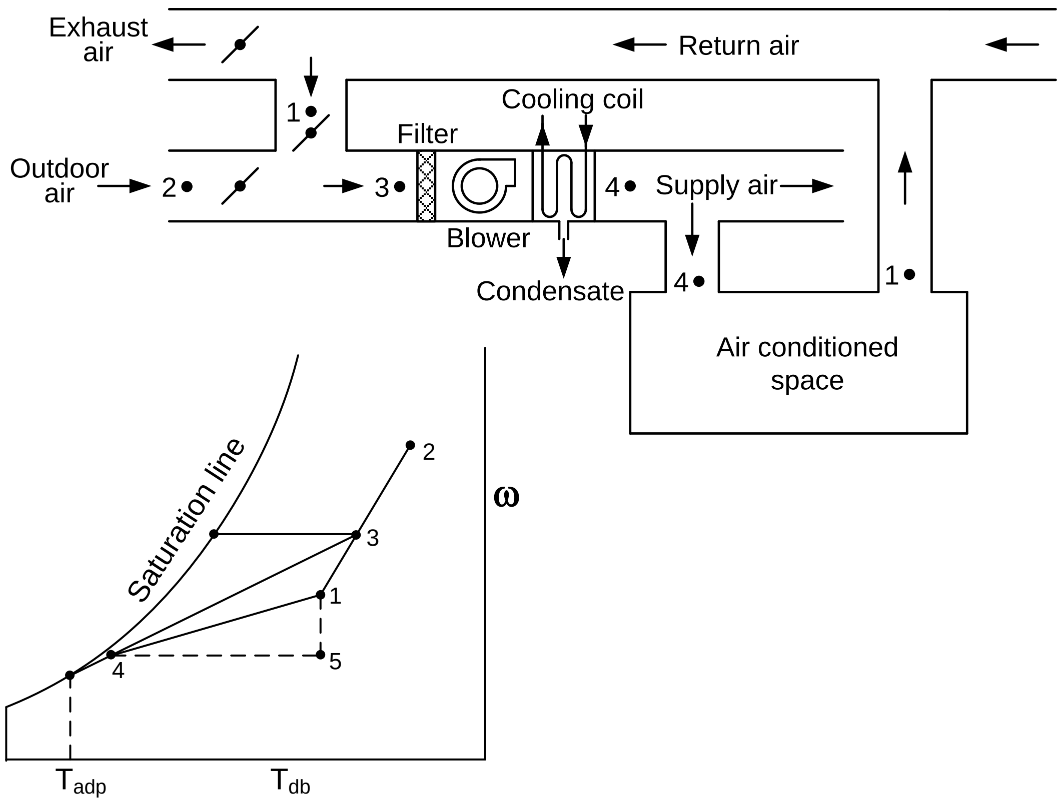


Figure 4.15 AHU system and the space conditioning processes

From Example 4.2:

$T_{db,3}$ = 24.45°C, ω_3 = 13.59 g moisture/kg dry air, RH_3 = 70%

h_3=59.3 kJ/kg dry air and m_3 = 3.879 kg dry air/s

Using Psychrometric chart for $T_{db,4}$ = 14°C and RH_4 = 95%:

h_4 = 38 kJ/kg dry air, ω_4 = 9.5 g moisture/kg dry air

Dry air flow rate at point-3 and 4 will remain the same, i.e. m_3 = m_4 = 3.879 kg dry air/s

From steam table, enthalpy of condensate at 14°C = 58.8 kJ/kg

Moisture condensation rate, m_c = $m_3(\omega_3 - \omega_4)/1000$

= 3.879(13.59-9.5)/1000 = 0.0159 kg/s

Heat transfer processes involved in AHU are shown in Figure 4.16.

Heat removal rate by chilled water = $m_w h_{w,out} - m_w h_{w,in}$

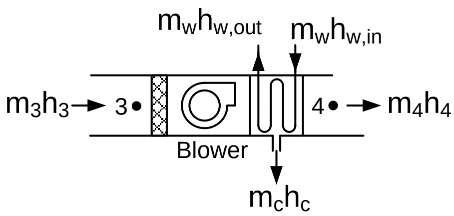


Figure 4.16 Heat transfer processes involve in AHU

a) Energy balance for the AHU gives:

$$m_w h_{w,out} + m_4h_4 + m_ch_c = m_w h_{w,in} + m_3h_3$$

$$m_w h_{w,out} - m_w h_{w,in} = m_3(h_3-h_4) - m_ch_c = 3.879(59.3-38)-0.0159\text{x}58.8$$

$$m_w h_{w,out} - m_w h_{w,in} = 81.7 \text{ kW}$$

$$m_w h_{w,out} - m_w h_{w,in} = 81.7/3.517 = 23.23 \text{ RT}$$

b) Space total heat load = $m_4(h_1-h_4)$

= 3.879(52.5 – 38) = 56.25 kW

= 56.25/3.517 = 16 RT

c) Vertical line from point-1 and horizontal line from point-4 intersect at point-5 (see Psychrometric chart).

Using Psychrometric chart: h_5 = 47.5 kJ/kg dry air

Space sensible heat load = $m_4(h_5-h_4)$

= 3.879(47.5 - 38) = 36.85 kW

= 36.85/3.517 = 10.48 RT

d) Space latent heat load = $m_4(h_1-h_5)$

= 3.879(52.5-47.5) = 19.4 kW

= 19.4/3.517 = 5.5 RT

e) Heat removal rate by chilled water if 0% outdoor air is used:

Moisture condensation rate, m_c = $m_3(\omega_1 - \omega_4)/1000$

$$=3.879(11.5-9.5)/1000 = 0.0077 \text{ kg/s}$$

Energy balance gives:

$$m_w h_{w,out} - m_w h_{w,in} = m_3(h_1 - h_4) - m_c h_c$$

$$= 3.879(52.5-38)-0.0077\text{x}58.8 = 55.8 \text{ kW} = 55.8/3.517 = 15.86 \text{ RT}$$

f) Heat removal rate by chilled water if 100% outdoor air is used:

Moisture condensation rate, $m_c = m_3(\omega_2 - \omega_4)/1000$

$$= 3.879(24.5 - 9.5)/1000 = 0.058 \text{ kg/s}$$

Energy balance gives:

$$m_w h_{w,out} - m_w h_{w,in} = m_3(h_2 - h_4) - m_c h_c$$

$$= 3.879(95-38)-0.058\text{x}58.8 = 217.7 \text{ kW} = 217.7/3.517 = 61.9 \text{ RT}$$

Note: Cooling load for AHU coil is increased from 15.86 RT to 61.9 RT (about 4 times) due to the increase of fresh air flow rate from 0% to 100%. Fresh air flow rate should be modulated based on demand of the air-conditioned spaces to optimise the cooling load and energy consumption of AHUs.

4.4.7 Handling of Sensible and Latent Cooling Load of Spaces

The amount of sensible and latent heat generated in an air-conditioning space depends on a number of factors such as number of occupants, type of space usage etc. To maintain the pre-set conditions in the air-conditioning spaces, the cooling coil of the AHU should be designed to remove the sensible and latent heat generated within the spaces. The sensible and latent heat removal capacity of the cooling coil of an AHU could be different even though the total cooling capacity is the same as shown in Figure-4.17. For the processes AB and AC, the total cooling load for the AHU is the same; however the sensible and latent heat removal rates are different. As a result, the AHU system could maintain the pre-set space temperature, but not the desired relative humidity. For proper handling of the sensible and latent heat loads of the spaces, it is necessary to design the cooling coil of the AHU based on sensible and latent heat loads generated within the spaces and operate the AHU system at the design conditions such as air flow rate and chilled water temperature and flow rate.

The ratio of the sensible heat load to the total heat load (sensible and latent) is known as sensible heat factor (SHF). Room or space sensible heat factor (RSHF) and cooling coil sensible heat factor could be different if outdoor fresh air is mixed with the return air. Sensible heat factors for room and cooling coil are considered in the design of the cooling coil of AHU for proper handling of the sensible and latent heat load of the spaces.

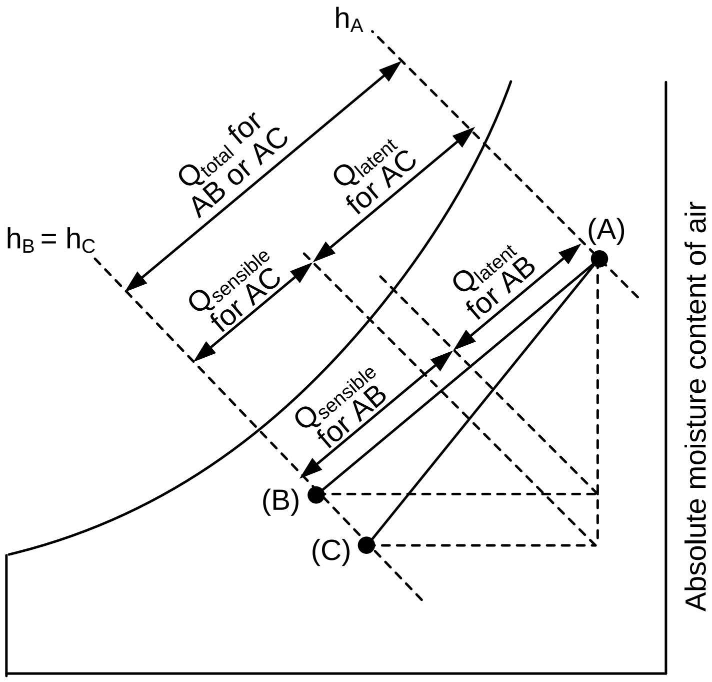


Figure 4.17 Sensible, latent and total heat removal capacity of the cooling coil of AHU

Example 4.4

An air-conditioned space has a sensible heat gain of 32 kW and a latent heat gain of 8 kW, and is maintained at a temperature of 25^oC_{db}. The supply air enters the space at 20^oC_{db}, 50% relative humidity. Outside conditions are 38^oC_{db} and 26^oC_{wb} and outside air and return air are mixed in the proportions of 1 kg outside air to 3 kg of return air.

The AHU consists of a cooling and heating coil in series with air entering the cooling coil first. The apparatus dew point of the cooling coil is 7^oC, and atmospheric pressure is 1.01325 bar.

a) Sketch an outline diagram of the AHU system and sketch the process lines on Psychrometric chart.
b) Identify enthalpies, specific volumes, and any other parameters used in your calculations.
c) Determine required supply air flow rate.
d) Determine the bypass factor for the cooling coil.

e) Determine the heat transfer rate required from the heating and the cooling coils, in kW.

Solution

(a) Outline diagram of the AHU system and process lines on Psychrometric chart are shown in Figure 4.18 and 4.19 respectively.

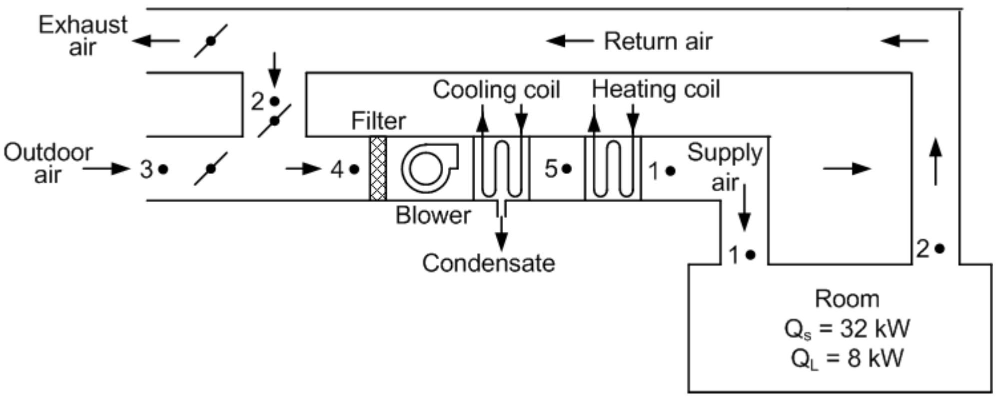


Figure 4.18 Outline diagram of the AHU system

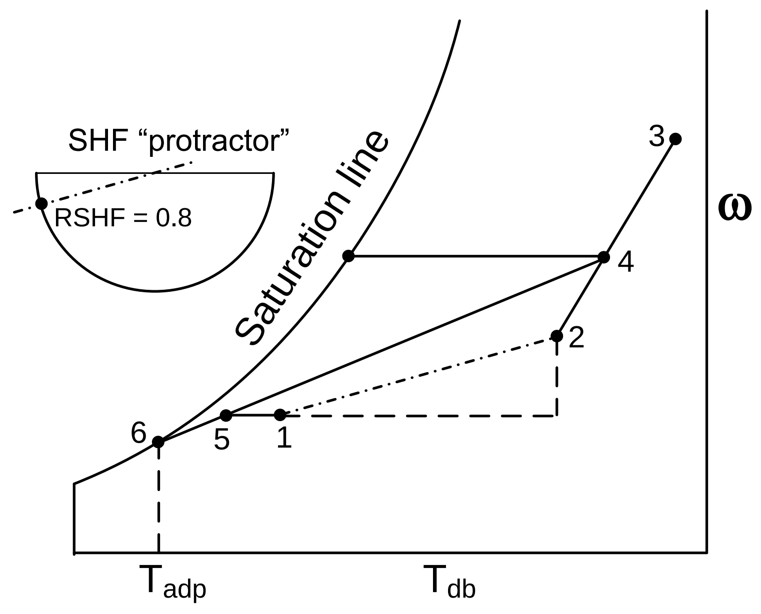


Figure 4.19 Process lines on Psychrometric chart

(b) Determination of properties:

Room sensible heat gain Q_s = 32 kW

Room latent heat gain Q_L = 8 kW

Room sensible heat ratio (RSHF) = Q_s / (Q_s+ Q_L) = 32/(32+8) = 0.8

RSHF line on the SHF "protractor" of Psychrometric chart is shown in Figure-4.19.

h_1 = 39 kJ/kg dry air (using 20°C_{db}, 50% RH)

h_2 = 45 kJ/kg dry air (using line 1-2 parallel to RSHF = 0.8 and 25°C_{db})

h_3 = 79.8 kJ/kg dry air (using 38°C_{db}, 26°C_{wb})

$$T_{db,4} = T_{db,2} + \frac{m_3}{(m_2 + m_3)}\left(T_{db,3} - T_{db,2}\right)$$

$T_{db,4}$ = 25+1x(38-25)/(3+1) = 28.25°C_{db}

Using Psychrometric chart:

h_4 = 54 kJ/kg dry air (using 28.25°C_{db} and line 2-3)

h_5 = 32.5 kJ/kg dry air (using intersection of horizontal line 1-5 and line 4-6)

v_1 = 0.84 m^3/kg dry air

(c) Determination of required supply air flow rate:

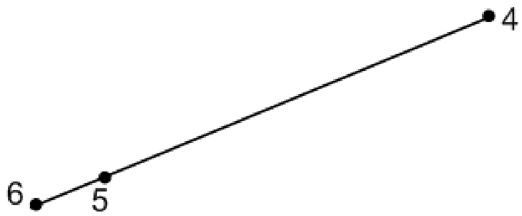
Total room load = 40 kW = M_{sa} ($h_2 - h_1$)

M_{sa} = 40/(45-39) = 6.67 kg dry air/s

V_{sa} = M_{sa} x v_1 = 6.67 x 0.84 = 5.6 m^3/s

(d) Determination of bypass factor for the cooling coil:

Cooling coil process line 4-5-6 as shown in the "Process lines on Psychrometric chart (Figure 4.19)"



T_6 = 7°C_{db}, T_4 = 28.25°C_{db},

Using Psychrometirc chart T_5 = 13.2°C_{db}

Bypass factor for the cooling coil

= (T_5 - T_6)/(T_4 - T_6) = (13.2-7)/(28.25-7) = 0.29

(e) Determination of heat transfer rate required from the heating and the cooling coils:

Heat transfer rate from cooling coil = M_{sa} ($h_4 - h_5$) = 6.67(54-32.5) = 143.4 kW

Heat transfer rate from heating coil = M_{sa} ($h_1 - h_5$) = 6.67(39-32.5) = 43.3 kW

4.5 Air-conditioning Space Requirements Based on Singapore Standards

Air-conditioning space requirements based on Singapore Standard SS553:

i. Normal design dry-bulb temperature for comfort air-conditioning can vary from 23ºC to 25ºC
ii. When air-conditioning systems are in operation, the operative temperature should be maintained with 24ºC to 26ºC
iii. Air movement should not exceed 0.30 m/s, measured at the occupants' level 1500 mm from the floor
iv. Average RH should not exceed 65% for new buildings and 70% for existing buildings.

Chapter-5: Cooling Tower Systems

5. Cooling Tower Systems

Cooling tower systems are used to reject heat from the water-cooled central air-conditioning systems, water-cooled package units and process cooling systems to the atmosphere. Based on the water and air flow configurations, cooling towers are broadly classified as forced-draft cross flow, induced-draft cross flow, forced-draft counter flow and induced-draft counter flow types. Cooling towers of induced-draft cross flow configuration are most commonly used. Main components of cooling towers are water spray systems, packing materials (known as “fill”) and fans. Warm water is sprayed from the top of the cooling tower. The warm water flows as a thin film over the packing materials. Ambient air is induced or forced through the cooling tower by the fans. Heat is transferred from the warm water to the flowing air as sensible and latent heat. Finally, the cold water is accumulated at the basin of the cooling tower. Performance of the cooling towers depends on a number of factors such as the operation of the water spray system, the fill, the air flow rate and the ambient air conditions. This chapter deals mainly with the heat transfer mechanisms, selection, energy optimisation strategies, installation and maintenance of the cooling towers.

Learning Outcomes

Participants will be able to:

i. Determine the influence of different parameters on the performance of the cooling towers.
ii. Size the cooling power for a specific application.
iii. Apply affinity laws to calculate fan power consumption at part load operations.
iv. Develop control strategies and calculate potential energy savings.
v. Understand the installation and maintenance requirements.

5.1 Configuration of Cooling Tower Systems

Cooling towers are mainly used to reject heat from the water-cooled central air-conditioning systems, water-cooled air-conditioning packaged units and industrial processes. Simplified configurations of the cooling tower for central air-conditioning and industrial process cooling applications are shown in Figures 5.1 and 5.2

respectively. Different components and flow configurations water and air are presented in Figure-5.3.

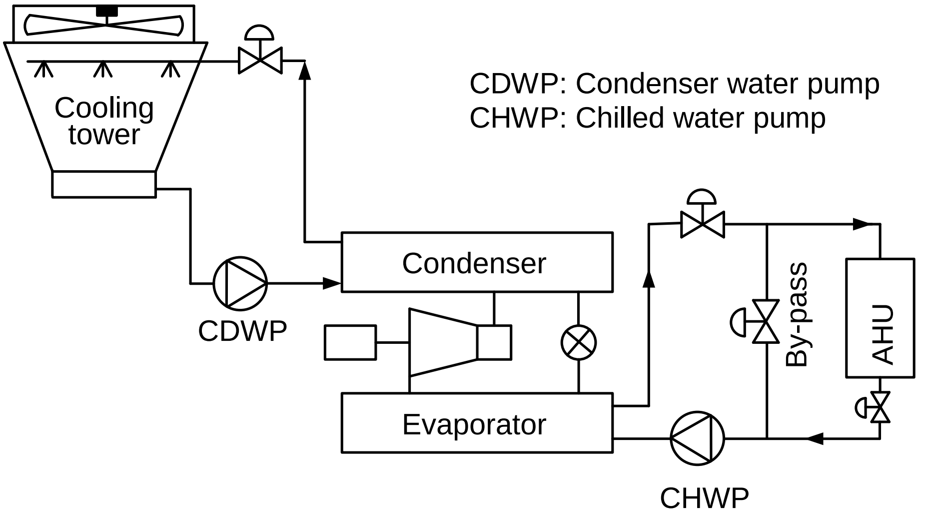


Figure 5.1 Central air-conditioning chilled water system with cooling tower.

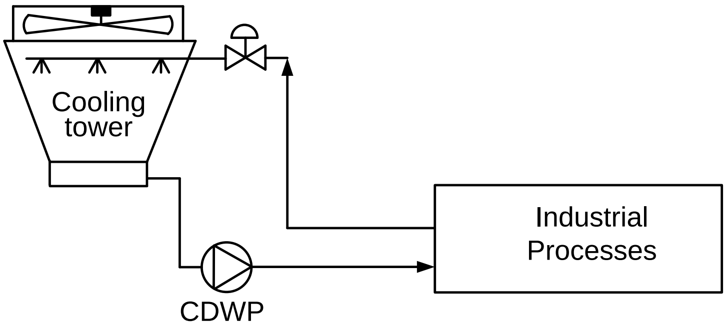


Figure 5.2 Cooling tower for cooling of industrial processes.

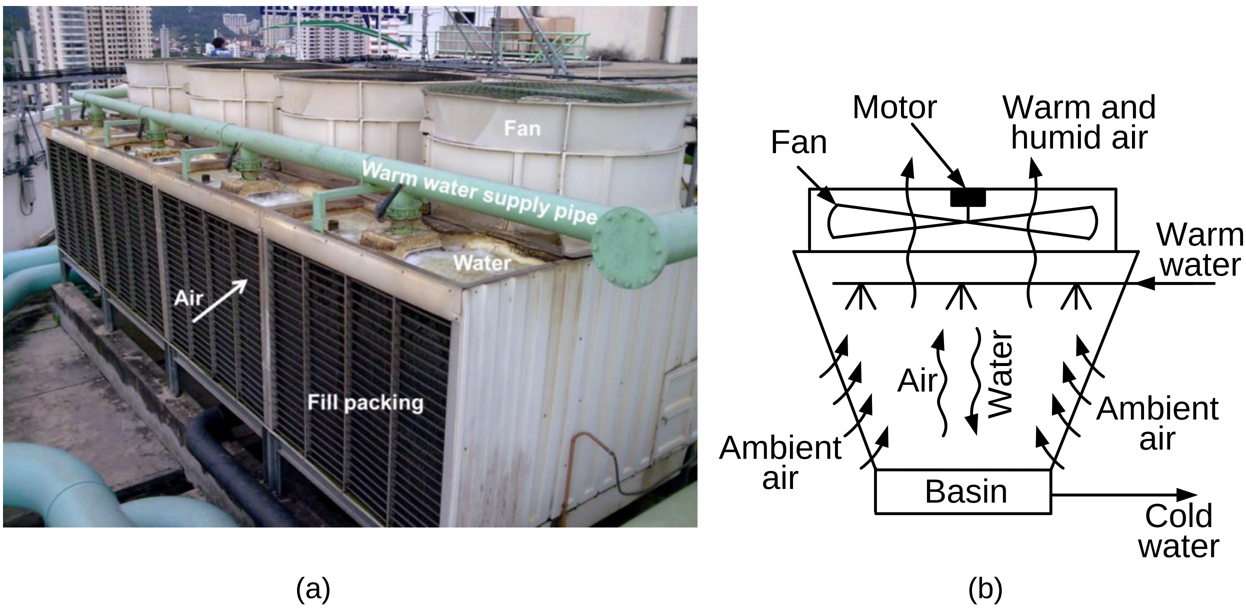


Figure 5.3 (a) Components of cooling tower and (b) flow configuration of water and air

5.2 Heat Transfer Processes in Cooling Towers

Warm water is sprayed over the fill packing materials of the cooling towers. The warm water flows downward as a thin film as shown in Figure-5.4. Heat is transferred from the exposed surface of the water film to the flowing air as sensible and latent heat.

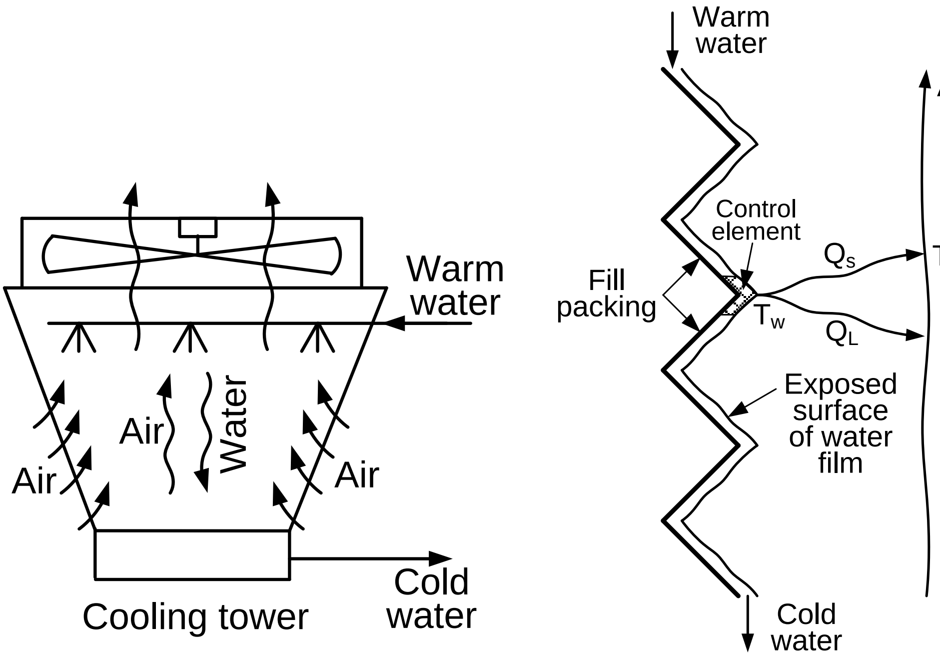


Figure 5.4 Heat transfer processes of cooling towers

The rate of sensible heat transfer from a small control element of the water film can be determined as:

$$Q_S = Ah_C(T_W - T_a) \tag{5.1}$$

where,

Q_s = rate of sensible heat transfer from the control element, W

A = exposed surface area of small control element of water film, m^2

h_c = convective heat transfer coefficient of air, W/m^2 K

T_w = temperature of water film of control element, ^{o}C

T_a = temperature of air, ^{o}C

Convective heat transfer coefficient of air depends on a number of factors such as air flow velocity, flow geometry, temperature and thermophysical properties of air. Eq. (5.1) shows that the sensible heat transfer rate depends on the local temperature of water film and air stream. If the temperature of the surrounding air is higher than the water film, water film will gain heat from instead of reject heat to the flowing air.

Therefore, sensible heat transfer rate depends on the surrounding ambient air temperature.

Latent heat transfer rate from the water film depends on the evaporation rate of water. During evaporation, water absorbs latent heat of vapourisation from the water film and surrounding air. As a result, the temperature of the water film drops. The latent heat transfer rate from a small control element of the water film can be evaluated as:

$$Q_L = \dot{m}_v A h_{fg} \tag{5.2}$$

$$Q_L = k_m (m_{v,f} - m_{v,a}) A h_{fg} \tag{5.3}$$

where,

Q_L = rate of latent heat transfer from the control element, W

$\dot{m}_v$ = mass flux of water vapour, kg/m^2 s

A = exposed surface area of small control element of water film, m^2

h_{fg} = latent heat of vapourisation at T_w, J/kg

k_m = mass transfer coefficient, kg/m^2 s

$m_{v,f}$ = mass fraction of water vapour at exposed surface of water film, dimensionless

$m_{v,a}$ = mass fraction of water vapour of air, dimensionless

Mass fraction of water vapour at the exposed surface of the water film can be calculated as:

$$m_{v,f} = \frac{\rho_{v,f}}{\rho_f} \tag{5.4}$$

where

$$\rho_{v,f} = \frac{(P_{sat\,at\,T_w})M_v}{RuT_w} \tag{5.5}$$

$$\rho_f = \frac{(P_{sat\,at\,T_w})M_v}{RuT_w} + \frac{P_{a,f}M_a}{RuT_w} \tag{5.6}$$

Similarly, mass fraction of water vapour in the air stream can be evaluated as:

$$m_{v,a} = \frac{\rho_{v,a}}{\rho_a} \tag{5.7}$$

where

$$\rho_{v,a} = \frac{(RH)(P_{sat\,at\,T_a})M_v}{RuT_a} \tag{5.8}$$

$$\rho_a = \frac{(RH)(P_{sat\ at\ T_a})M_v}{RuT_a} + \frac{P_a M_a}{RuT_a} \tag{5.9}$$

where,

M_a = molecular weight of air, 29 kg/kmol

M_v = molecular weight of vapour, 18 kg/kmol

P_a = pressure of air vapour mixture of air stream, Pa

$P_{a,f}$ = pressure of air vapour mixture at exposed surface of water film, Pa

P_{sat} = saturated vapour pressure, Pa

RH = relative humidity of air, fraction

R_u = universal gas constant, 8.3143 kJ/kmol K

T_a = temperature of air, K

T_w = temperature of water, K

$\rho_{v,f}$ = partial density of vapour at exposed surface of water film, kg/m^3

$\rho_{v,a}$ = partial density of vapour at air stream, kg/m^3

ρ_f = density of air vapour mixture at exposed surface of water film, kg/m^3

ρ_a = density of air vapour mixture at air stream, kg/m^3

Eqs. (5.3) to (5.9) show that the evaporation rate of water and the resulting latent heat transfer rate from the water film depend on the relative humidity of the surrounding air. The latent heat transfer rate decreases with the increase in the relative humidity of the air. If the relative humidity of the surrounding air is increased to 100 percent (saturation condition), the water evaporation rate and the corresponding latent heat transfer rate will drop to zero. Total sensible and latent heat transfer rate from the cooling tower can be evaluated by summing up the sensible and latent heat transfer rate from each control element. Therefore, total heat transfer rate from the cooling towers depends on both the temperature and relative humidity of the surrounding air.

Problem 5.1

Process water enters a cooling tower at 40^{o}C and leaves at 30^{o}C. Ambient air flows in cross flow configuration through the packing materials of the cooling tower at an average temperature of 32^{o}C and relative humidity of 70%. Saturated vapour pressure and latent heat of vapourisation of water at different temperatures are shown in Table 5.1 below. Assume that the convective heat and mass transfer coefficients of circulating air are 5 W/m^2 K and 0.007 kg/m^2 s respectively. For the average temperature of the cooling water, calculate the sensible, latent and total heat flux from water to the circulating air.

Table 5.1 Saturated vapour pressure and latent heat of vapourisation of water

Temperature, ^oC	Saturated vapour pressure, kPa	Latent heat of vaporisation (h_{fg}), kJ/kg
30	4.246	2430.5
35	5.628	2418.6
40	7.384	2406.7
45	9.593	2394.8

Solution:

Average temperature of water = (40+30)/2 = 35^{o}C

Saturated vapour pressure at exposed surface of water film (using interpolation):

$$P_{sat\,at\,T=35^oC} = 5.628x10^3\,Pa$$

Partial density of vapour at exposed surface of water film:

$$\rho_{v,f} = \frac{RH(P_{sat\,at\,T=35^oC})M_v}{RuT_w} = \frac{1x5.628x10^3x18}{8314.0(273+35)} = 0.0396 \text{ kg/m}^3$$

Density of air vapour mixture at exposed surface of water film :

$$\rho_f = \frac{RH(P_{sat\,at\,T=35^oC})M_v}{RuT_w} + \frac{P_{a,f}M_a}{RuT_w}$$

$$= \frac{1x5.628x10^3x18}{8314.0(273+35)} + \frac{(100-1x5.628)x10^3x29}{8314.0(273+35)} = 0.0396+1.069 = 1.1086 \text{ kg/m}^3$$

Mass fraction of water vapour at exposed surface of water film:

$$m_{v,f} = \frac{\rho_{v,f}}{\rho_f} = \frac{0.0396}{1.1086} = 0.0357$$

Ambient air temperature = 32^{o}C and RH = 70%.

Saturated vapour pressure at 32^{o}C, $P_{sat\,at\,T=32^oC} = 4.799x10^3\,Pa$

Partial density of vapour at air stream:

$$\rho_{v,a} = \frac{(RH)(P_{sat\,at\,T_a})M_v}{RuT_a} = \frac{0.7x4.799x10^3x18}{8314.0(273+32)} = 0.0238 \text{ kg/m}^3$$

Density of air vapour mixture at air stream:

$$\rho_a = \frac{(RH)(P_{sat\,at\,T_a})M_v}{RuT_a} + \frac{P_{a,bulk}M_a}{RuT_a}$$

$$= \frac{0.7x4.799x10^3x18}{8314.0(273+32)} + \frac{(100-0.7x4.799)x10^3x29}{8314.0(273+32)}$$ = 0.0238 +1.1052 = 1.129 kg/m^3

Mass fraction of water vapour in bulk air:

$$m_{v,a} = \frac{\rho_{v,a}}{\rho_a} = \frac{0.0238}{1.129} = 0.0211$$

Sensible heat flux (per unit area) from water film to bulk air:

$$\dot{Q}_S = Ah_c(T_w - T_a)$$

$$\dot{Q}_S = 1x5(35-32) = 15\,W/m^2$$

Latent heat flux (per unit area) from water film to bulk air:

$$\dot{Q}_L = k_m(m_{v,f} - m_{v,a})Ah_{fg}$$

$$\dot{Q}_L = 0.007(0.0357-0.0211)x1x2418.6 = 0.247\,\mathrm{kW/m^2}$$

$$\dot{Q}_L = 0.247\,\mathrm{kW/m^2} = 247\,\mathrm{W/m^2}$$

Total heat flux (per unit area) from water film to bulk air:

$$\dot{Q}_{Total} = \dot{Q}_S + \dot{Q}_L = 15 + 247\,\mathrm{W} = 262\,\mathrm{W/m^2}$$

Notes:

i. Sensible heat transfer rate (15 W/m^2) is quite small in comparison to the latent heat transfer rate (247 W/m^2).

ii. Total heat transfer rate and water leaving temperature from the cooling tower can be calculated by numerical integration over the entire surface of fill packing.

iii. Sensible heat transfer rate depends on flowing air temperature.

iv. Latent heat transfer rate depends on flowing air relative humidity (RH).

v. Unfortunately, both temperature (average daytime temperature is about 32^{o}C) and RH (about 75%) of ambient air in Singapore are relatively high, resulting in poor heat transfer performance of cooling towers in Singapore.

vi. Water leaving temperature for an ideal cooling tower could be reduced to the wet bulb temperature of the air, but the cooling tower will have to be tremendous in size.

vii. Cooling tower Approach Temperature = (Temperature of leaving water) – (Wet bulb temperature of the air).

viii. Approach Temperature for commonly designed cooling tower is 5°F or 2.8°C.

5.3 Selection of Cooling Towers

The efficiency of water-cooled chillers depends on condenser water supply temperature. If the cooling towers are undersized, condenser water supply temperature from the cooling towers will rise and the efficiency of the chillers will drop. Similarly, if the cooling towers are oversized, condenser water supply temperature can be lowered and the efficiency of the chillers can be improved. Cooling towers need to reject the heat of the air-conditioning spaces as well as the heat added by the compressor to the refrigerant during the compression process known as the heat of compression. The heat of compression depends on the efficiency of the compressor, and the prime mover systems (such as open type or hermetically shield). Usually, the heat of compression is about 25 percent of the chiller cooling capacity. Therefore, the size of the cooling towers should be about 125 percent of the chiller cooling capacity. As a rule of thumb, cooling tower rated capacity of 1.5 times the chiller cooling capacity is used. If oversized cooling tower is used, the heat transfer surface area of the cooling tower is increased. As a result, the air flow rate can be reduced to transfer the same amount of heat, leading to the reduction of cooling tower fan power consumption. Moreover, oversized cooling tower may help to improve the efficiency of the chillers due to the supply of lower condenser water temperature. However, first cost of the oversized cooling tower will be higher. Therefore, sizing of the cooling towers is a compromise between the first cost of the cooling tower and the operating cost savings of the cooling towers and the chillers.

Cooling towers are selected based on:

i. Required water flow rate to the cooling tower
ii. Required entering & leaving temperatures of water
iii. Wet and dry bulb temperatures of surrounding air
iv. Approach temperature, which is the difference between the temperature of water leaving the cooling towers and the wet bulb temperature of the surrounding air. Usually, cooling towers are designed for the approach temperature of about 5°F or 2.8°C.

5.4 Optimisation Strategies of Cooling Towers

The cooling towers are generally selected based on the cooling capacity of the chillers and peak heat load of industrial processes. The operation of the cooling towers is usually interlocked with systems such as the chillers and industrial process. Often the heat load of the systems varies during the actual operation (such as when the cooling load of the chillers is low or few industrial processes are turned off), but the capacity of the cooling towers is maintained at the fixed rated value. Power consumption of the cooling towers can be optimised by varying the capacity of the cooling towers in relation to the actual changes in load. The capacity of the cooling towers is a function of the air flow rate through them. The capacity of the cooling towers can be modulated by varying the air flow rate by: (a) Fan staging and (b) Variable speed fans.

Fan Staging: In the fan staging systems, few fans of the cooling towers are turned on or off based on the actual changes in heat rejection load to maintain the pre-set temperature of the condenser water leaving the cooling towers. However, frequent on and off may cause premature wear and tear of the motor drive systems of cooling tower fans.

Variable Speed Fans: In the variable speed fan systems, the speed of the cooling tower fans is modulated using the variable speed drives (VSDs) to maintain the pre-set temperature of the condenser water leaving the cooling towers as shown in Figure-5.5. Although the heat and mass transfer coefficients generally vary nonlinearly with the velocity of air, it can be assumed with reasonable accuracy that the speed of the cooling tower fans can be reduced proportionally with the reduction of the heat rejection load of cooling towers. Based on affinity laws, the power consumption of the fan is proportional to the cube of the fan speed. If the heat rejection load of a cooling tower is dropped to 90 percent, the power consumption of the fan will drop to about 73 percent ($0.9^3 = 0.729$) due to the proportional reduction of the fan speed. Therefore, the variable speed fan systems contribute significant savings to cooling tower fan power consumption under part load operation. Moreover, variable speed fan systems reduce the wear and tear of the motor drive systems and water drift losses due to the reduction of fan speed and corresponding air flow velocity.

The latent heat transfer performance of cooling towers depends on the relative humidity of the surrounding air. Relative humidity of air is a function of the wet bulb temperature. If the wet bulb temperature of the surrounding air drops below the design value, the cooling towers are able to cool the water economically at a lower temperature while maintaining the design approach temperature. As the efficiency of chillers improves with the reduction of the condenser water temperature, this will lead to the improvement of chiller efficiency. Therefore, condenser water approach temperature can be used as the input parameter for modulating the speed of the fans of cooling towers as shown in Figure-5.6 for further enhancement of energy performance of the cooling towers and the chillers.

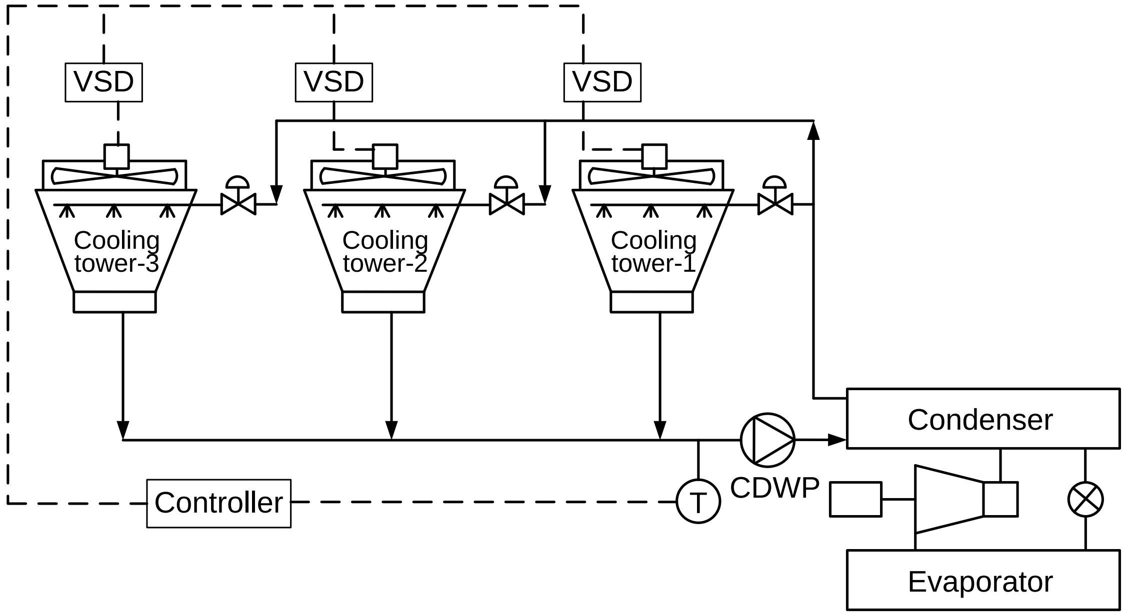


Figure 5.5 Control of cooling tower capacity based on water leaving temperature from the cooling tower

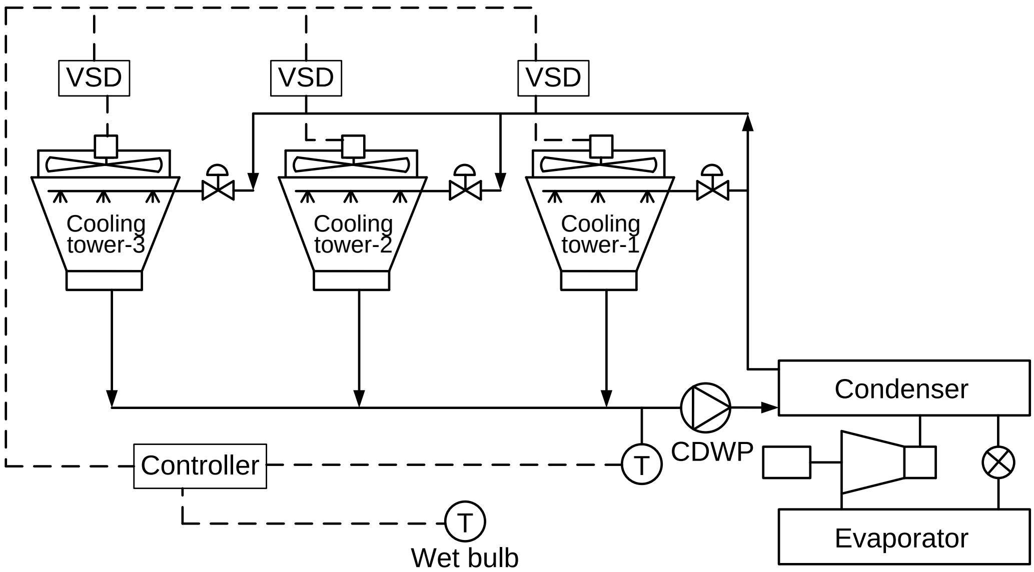


Figure 5.6 Control of cooling tower capacity based on approach temperature

A comparison of the specific power consumption of cooling tower fans (power consumption of fans divided by the heat load of chillers) for constant speed, fan staging and variable speed fan systems is shown in Figure-5.7. For constant speed operating strategy, specific power consumption of fans increases with the decrease of heat rejection load of the cooling towers. The specific power consumption profile for fan staging strategy follows the same profile of constant speed operating strategy. Power consumption for fan staging strategy drops once a certain number of fans are turned off. The specific power consumption decreases with the decrease of heat rejection load of the cooling towers and reaches the minimum value for the variable speed fan operating strategy.

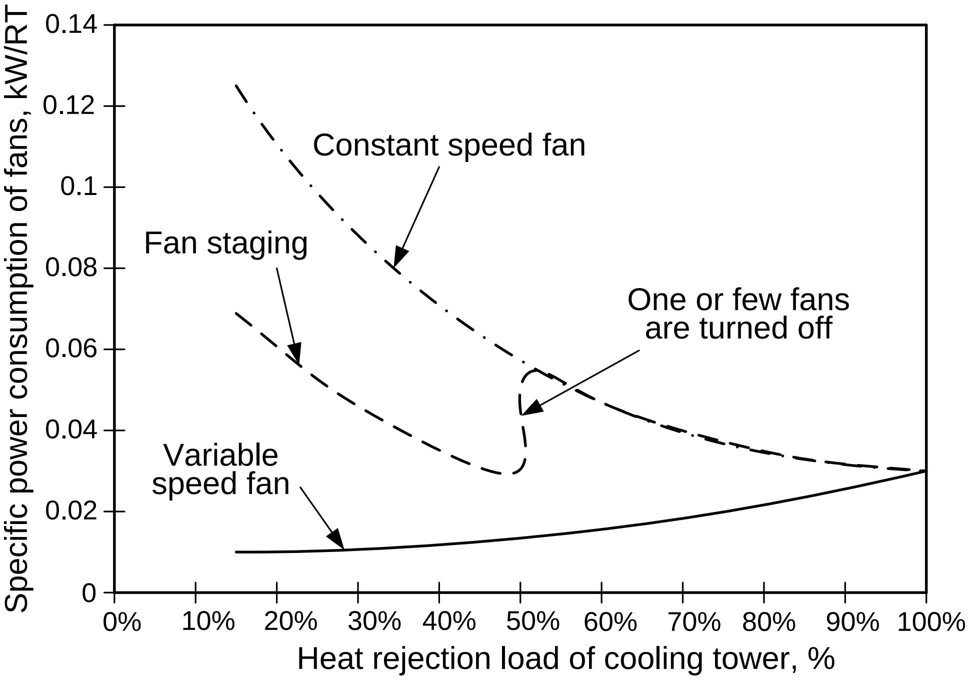


Figure 5.7 Specific power consumption of cooling tower fans for different operating strategies

Problem 5.2

A chilled water system has 4 nos. of cooling towers. Power consumption of fan for each cooling tower at full speed is 20 kW. Presently 3 nos. of cooling towers are operated at full speed and the other cooling tower acts as a stand-by unit. If all 4 nos. cooling towers are operated at reduced capacity by reducing the speed of the fans using VSD, calculate the potential power savings.

Solution:

If all 4 nos. of cooling towers are operated instead of 3 nos., the capacity of each of the 4 nos. of cooling towers is required to reduce to 3/4 of the rated capacity.

Assuming the capacity of the cooling tower is proportional to the air flow rate, required air flow rate through each cooling tower will be 3/4 of the rated flow rate.

If rated speed (or full speed) of the fan and corresponding air flow rate for each cooling tower are N_{full} and Q_{full} respectively, the required air flow rate for each of the 4 nos. of cooling towers is $Q_{required} = (3/4)\,Q_{full}$.

Fans of each of the 4 nos. of cooling towers are required to operate at speed $N_{required}$, which can be calculated using affinity law as:

$$\frac{N_{required}}{N_{full}} = \frac{Q_{required}}{Q_{full}}$$

$$\frac{N_{required}}{N_{full}} = \frac{(3/4)Q_{full}}{Q_{full}}$$

$$\frac{N_{required}}{N_{full}} = \frac{3}{4}$$

Given that the rated power consumption of the fan of each cooling tower at full speed is W_{full} = 20 kW. The power consumption of the fan of each cooling tower $W_{required}$ at the reduced speed can be calculated using the affinity law as:

$$W_{required} = W_{full}\left(\frac{N_{required}}{N_{full}}\right)^3$$

$$W_{required} = 20\left(\frac{3}{4}\right)^3 = 8.4 \text{ kW}$$

Total power consumption of the fans of 4 nos. of cooling towers = 8.4 x 4 = 33.6 kW

Previous power consumption of the fans of 3 nos. of cooling towers = 20 x 3 = 60 kW

Power saving = 60 – 33.6 = 26.4 kW

Percentage saving of power = 100 x (26.4 / 60) = 44%.

5.5 Installation of Cooling Towers

The heat transfer performance of the cooling towers depends on the rate of air flow through them. Required air flow rate depends on the capacity of the cooling towers. Sufficient space between the cooling towers and the surrounding obstructions (such as wall) should be provided as shown in Figure-5.7 to ensure the flow of air freely into the cooling towers. The minimum distance between the cooling towers and the surrounding obstructions to be maintained depends on the capacity of the cooling towers and is usually specified by the manufacturer of the cooling towers.

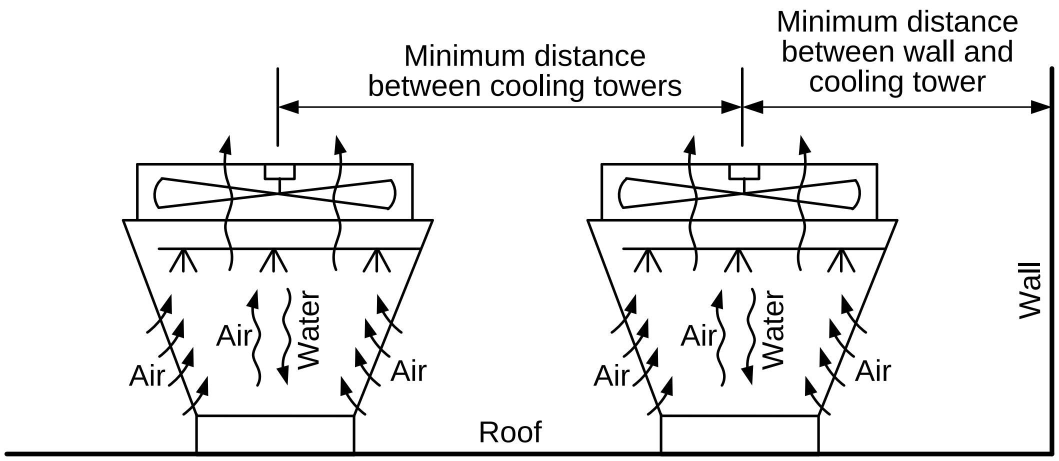


Figure 5.7 Minimum distance between the cooling towers and the surrounding obstructions

Moreover, if the height of the surrounding obstructions or walls is higher than the cooling towers, warm and humid air discharged from the cooling towers may recirculate back into the air intakes as shown in Figure-5.8, which will affect the heat transfer performance of the cooling towers. Extension duct can be installed at the discharge of the fans (Figure-5.8) to direct the warm and humid air flow away from the air intakes. However, the extension duct will create additional pressure head for the fans.

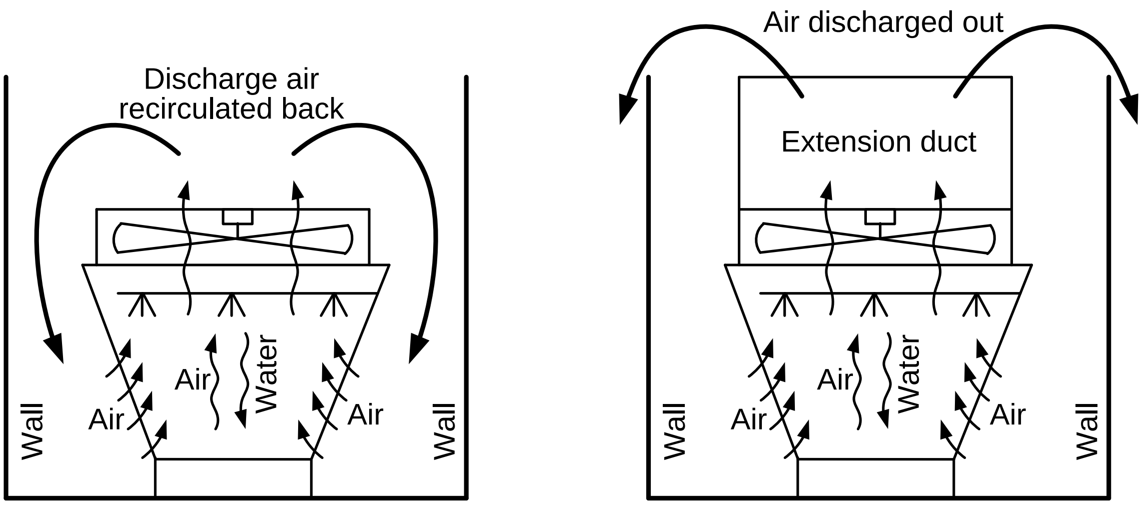


Figure 5.8 Extension duct to direct the warm and humid discharged air flow away from the air intakes

5.6 Maintenance of Cooling Towers

The performance of cooling towers depends on the flow of air through the cooling tower and the condition of spray system and the fill. Fan should be maintained regularly and operated at the required speed to ensure that enough air flows uniformly through the fill without any obstruction. The tension of the belts used in the belt drive systems of the fans should be checked regularly to prevent belt slippage. The spray system and the fill should also be maintained regularly to ensure that they are in good condition. If the spray system is defective, water will not be sprayed over the entire fill packing materials. Similarly, if the fill is damaged, sprayed water will flow directly into the basin rather than flowing uniformly as a thin water film over the entire fill packing materials. Hence, the exposed surface area and contact time (between water and air) of the flowing water will reduce, leading to the drop of heat transfer rate from the warm water to the flowing air.

Moreover, a cooling tower is an open system. The flowing water is exposed to the ambient air. Hence, part of the flowing water is evaporated and dust of the ambient air is captured by the flowing water. Makeup water is required to be supplied continuously to compensate for the water evaporated. Certain fraction of the water is also regularly drained from the bottom of the cooling tower basin and topped up with the clean water to reduce the concentration of solids in the water.

Water treatment is also important for the cooling towers to prevent scaling, corrosion and fouling on the heat exchanger surfaces of the industrial processes or the condenser tubes of the chillers. Water treatment usually consists of chemical or nonchemical treatment. The pH value of cooling tower water needs to be carefully controlled. If the pH value of the water is too high, the tendency for scaling increases. On the other hand, if the pH value is too low, the tendency for corrosion increases. In the chemical treatment processes, the pH value of the water is generally controlled to 6.5 to 8.5. This is more for controlling scaling than completely eliminating it.

Problem 5.3

Cooling water leaves the condenser of a chilled water system and enters a cooling tower at 35°C at a rate of 0.1 m^3/s. The water is cooled to 29°C in the cooling tower by air which enters the tower at 1 atm, 28°C, 70% RH and leaves at 32°C, 95% RH as shown in Figure-5.9. Neglecting the power input to the fan, determine (a) the required volume flow rate of air into the cooling tower and (b) the required mass flow rate of the makeup water. The enthalpies of water at 35°C and 29°C are 146.68 kJ/kg and 121.61 kJ/kg respectively.

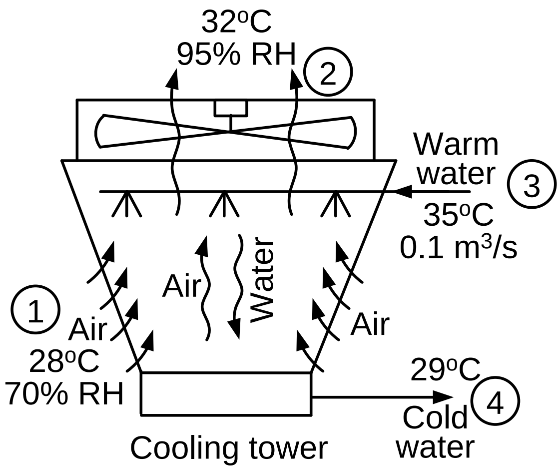


Figure 5.9 Operating parameters of cooling tower

Solution:

Applying the mass conservation law for the dry air at point-1 and 2 of the cooling tower (Figure-5.9) gives:

$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

where, $\dot{m}_a$ is the mass flow rate of dry air through the cooling tower.

Applying the mass conservation law for the water flow rate of the cooling tower gives:

$$\dot{m}_{w3} + \dot{m}_{a1}\omega_1 = \dot{m}_{w4} + \dot{m}_{a2}\omega_2$$

$$\dot{m}_{w3} - \dot{m}_{w4} = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{makeup}$$

where, $\dot{m}_w$ is the mass flow rate of cooling water and ω is the absolute humidity of the flowing air through the cooling tower.

Similarly, applying the energy conservation law for the cooling tower gives:

$$\dot{m}_{a1}h_{a1} + \dot{m}_{w3}h_{w3} = \dot{m}_{a2}h_{a2} + \dot{m}_{w4}h_{w4}$$

$$\dot{m}_a(h_{a2} - h_{a1}) = \dot{m}_{w3}h_{w3} - m_{w4}h_{w4}$$

$$\dot{m}_a(h_{a2} - h_{a1}) = \dot{m}_{w3}h_{w3} - [\dot{m}_{w3} - \dot{m}_a(\omega_2 - \omega_1)]h_{w4}$$

$$\dot{m}_a = \frac{\dot{m}_{w3}(h_{w3} - h_{w4})}{(h_{a2} - h_{a1}) - (\omega_2 - \omega_1)h_{w4}}$$

where, h_a and h_w are the enthalpies of the air and the water respectively.

Using the temperatures and relative humidity of air at points 1 and 2 of the cooling tower, the following parameters can be determined using Psychrometric chart:

Enthalpy of air at point-1 h_{a1} = 71 kJ/kg

Absolute humidity at point-1 ω_{a1} = 16.75x10^{-3} kg H_2O/kg dry air

Specific volume of air at point-1 v_1 = 0.877 m^3/kg dry air

Enthalpy of air at point-2 h_{a2} = 106.5 kJ/kg

Absolute humidity at point-2 ω_{a2} = 29x10^{-3} kg H_2O/kg dry air

For the given temperatures of water at point-3 and 4:

Enthalpy of water at point-3 $h_{w3} = h_{w@35C}$ = 146.68 kJ/kg H_2O

Enthalpy of water at point-4 $h_{w4} = h_{w@29C}$ = 121.61 kJ/kg H_2O

Mass flow rate of dry air:

$$\dot{m}_a = \frac{(0.1x1000)(146.68 - 121.61)}{(106.5 - 71) - \left(29x10^{-3} - 16.75x10^{-3}\right)121.61} = 73.7 kg/s$$

Therefore, required volume flow rate of air = $\dot{m}_a v_1 = 64.6\, m^3/s$

Required mass flow rate of the makeup water:

$$\dot{m}_{makeup} = \dot{m}_a(\omega_2 - \omega_1) = 73.7(29 - 16.75)x10^{-3} = 0.9\ kg/s$$

Note:

Volume flow rate of water = 0.1 m^3/s

Mass flow rate of water = Volume flow rate of water x water density

= 0.1 x 1000 = 100 kg/s

Water evaporation rate = makeup water flow rate = 0.9 kg/s

Water evaporation rate is 100 x (0.9/100) = 0.9%.

Water evaporation rate is approximately 1% of the circulating water.

5.7 Cooling tower performance based on Singapore Standards

Based on Singapore Standard SS530: 2014, the heat transfer performance requirements for the cooling towers are summarised in Table 5.2.

Table 5.2 Performance requirements for heat rejection equipment

Equipment type	Rating condition	Performance required
Propeller or axial fan cooling towers	Entering water: 35.0°C Leaving water: 29.4°C Outdoor air: 23.4°C WB	≥ 3.23 (L/s)/kW
Centrifugal fan cooling towers	Entering water: 35.0°C Leaving water: 29.4°C Outdoor air: 23.4°C WB	≥ 1.7 (L/s)/kW

(Source: Reproduced from Singapore Standard SS530: 2014 with permission from SPRING Singapore. Please refer SS530: 2014 for details. Website: www.singaporestandardseshop.sg)

Table 5.2 demonstrates that propeller or axial fan cooling towers should be able to cool 3.23 L/s of water from 35°C to 29.4°C by consuming fan power of 1 kW.

The quantity of rejected heat due to the cooling of 3.23 L/s of water from 35°C to 29.4°C can be calculated as:

$$Q_{reject} = V\rho C_p \Delta T \quad (5.10)$$

$$Q_{reject} = 3.23\frac{L}{s}\, x\, \frac{1m^3}{1000L}\, x\, \frac{1000kg}{m^3}\, x4.2\frac{kJ}{kgK}\, x(35-29.4)K = 75.97\, kW \quad (5.11)$$

where,

Q_{reject} = rate of heat rejection from the cooling tower, kW

V = volume flow rate of water, m^3/s

ρ = density of water, kg/m^3

C_p = specific heat of water, 4.2 kJ/(kg K)

ΔT = difference of entering and leaving temperatures of water, °C

For a chiller of capacity Q_c RT and efficiency of η kW/RT, the heat rejection load for the cooling tower can be determined as

$$Q_{reject} = 3.517Q_c + \eta Q_c \tag{5.12}$$

Comparing Eqs. (5.11) and (5.12) gives:

$$Q_c = \frac{75.97}{3.517 + \eta} \tag{5.13}$$

Based on Singapore Standard SS530: 2014, COP of water-cooled centrifugal chillers of size category (≥ 1055 kW and < 1407 kW) is 6.286 at full load. Putting the chiller efficiency of 3.517/6.286 = 0.559 kW/RT in Eq. (5.13) gives:

$$Q_c = \frac{75.97}{3.517 + 0.559} = 18.6RT \tag{5.14}$$

Therefore, a cooling tower should be able to reject the total heat of a chiller of capacity 18.6 RT by consuming fan power of 1 kW. Hence, the specific power consumption of the cooling tower based on SS530: 2014 is 1/18.6 = 0.054 kW/RT.

6. Case Study on Retrofitting of Chilled Water System

An energy audit was carried out in a commercial building. Air-conditioning for the building was provided using water-cooled central chilled water system, which consisted of 3 nos. of chillers of capacity 500 RT each, 3 nos. of chilled water pumps, 3 nos. of condenser water pumps and 3 nos. of cooling towers. 2 nos. of chillers were operated to support the building cooling load while the other chiller was used as a standby unit. Pumps and cooling towers were operated at constant speed (no VSD) and connected to the chillers using common header pipes as shown in Figure 6.1. Total 9 nos. of triple duty valves (which combine the features of a check valve, a balancing valve and a shut-off valve) were installed to modulate the flow of water in the chillers and cooling towers. Based on the findings of the energy audit, it was recommended to retrofit the chilled water system with high efficiency chillers, pumps, cooling towers and to modify the piping system.

Because of budget and space constraints, 2 nos. of existing chillers of capacity 500 RT each were replaced with 2 nos. of high efficiency chillers of capacity 400 RT each. The other existing chiller was kept as the standby unit. The existing 3 nos. of chilled water pumps, 3 nos. of condenser water pumps and 3 nos. of cooling towers were also replaced with appropriately sized high efficiency pumps and cooling towers. VSDs were installed to regulate the capacity of the pumps and cooling towers based on real-time cooling load. Due to budget constraints, minor changes were made to the existing piping system. As the pressure loss of the triple duty valve is quite high, 9 nos. of existing triple duty valves were removed to reduce the unnecessary pressure loss. 4 nos. of butterfly valves were installed on the chilled and condenser water common header pipes. The butterfly valves were normally closed to operate the pumps and chillers as one-to-one system as shown in Figure 6.2. Chilled and condenser water flow rates to each chiller can be modulated conveniently by regulating the speed of the corresponding pumps using VSDs. Moreover, the modified piping system provides flexibility of circulating chilled or condenser water to a specific chiller by operating the corresponding pumps of the other chillers. For instance, if chiller-1 and chilled water pump CHWP-2 fail (Figure 6.2), butterfly valve BF-1 can be opened and chilled water pump CHWP-1 can be operated to circulate the chilled water through chiller-2.

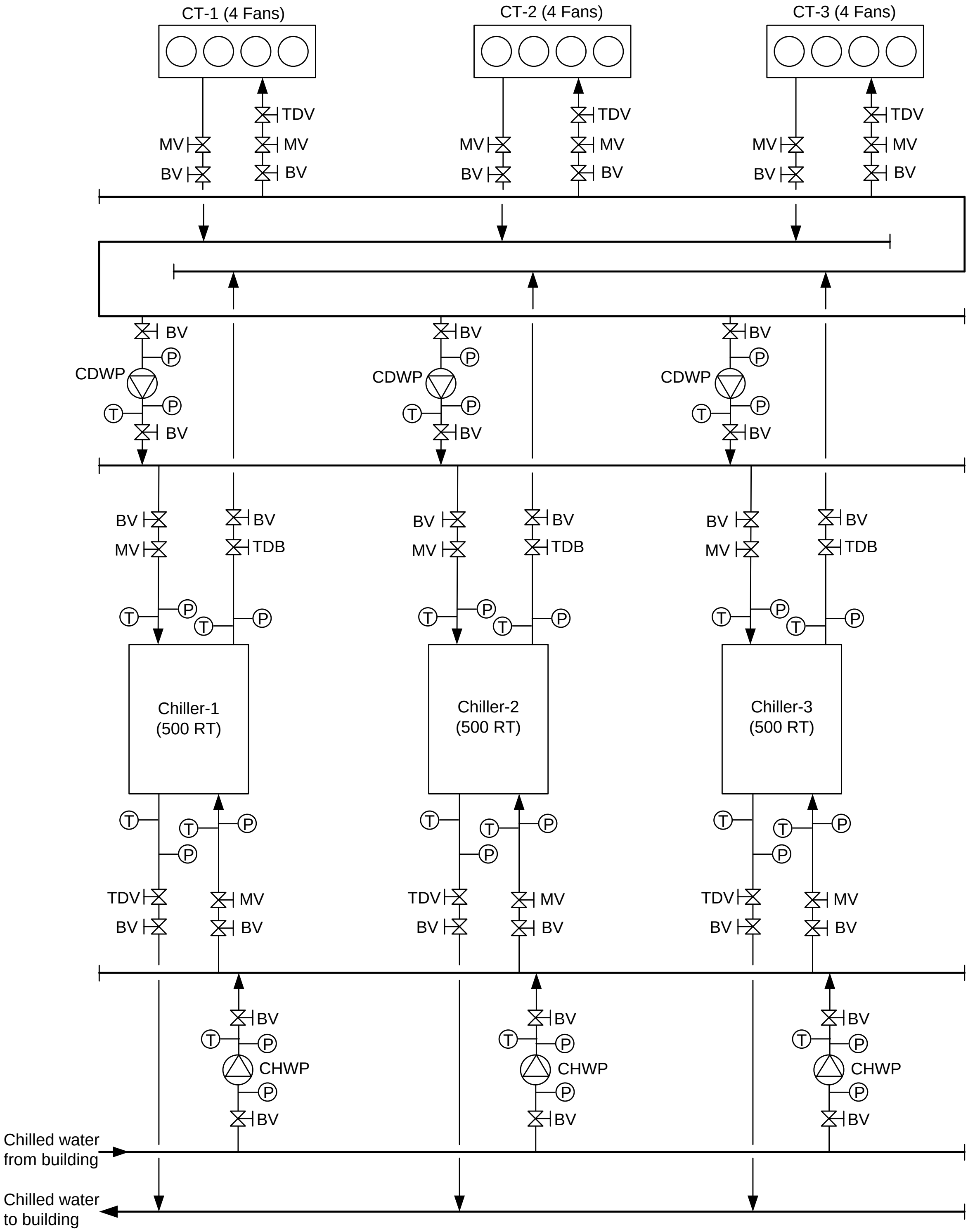


Figure 6.1 Water-cooled central chilled water system before retrofitting work (CHWP: Chilled water pump, CDWP: Condenser water pump, CT: Cooling tower, BV: Butterfly valve, MV: Motorized valve, TDV: Triple duty valve)

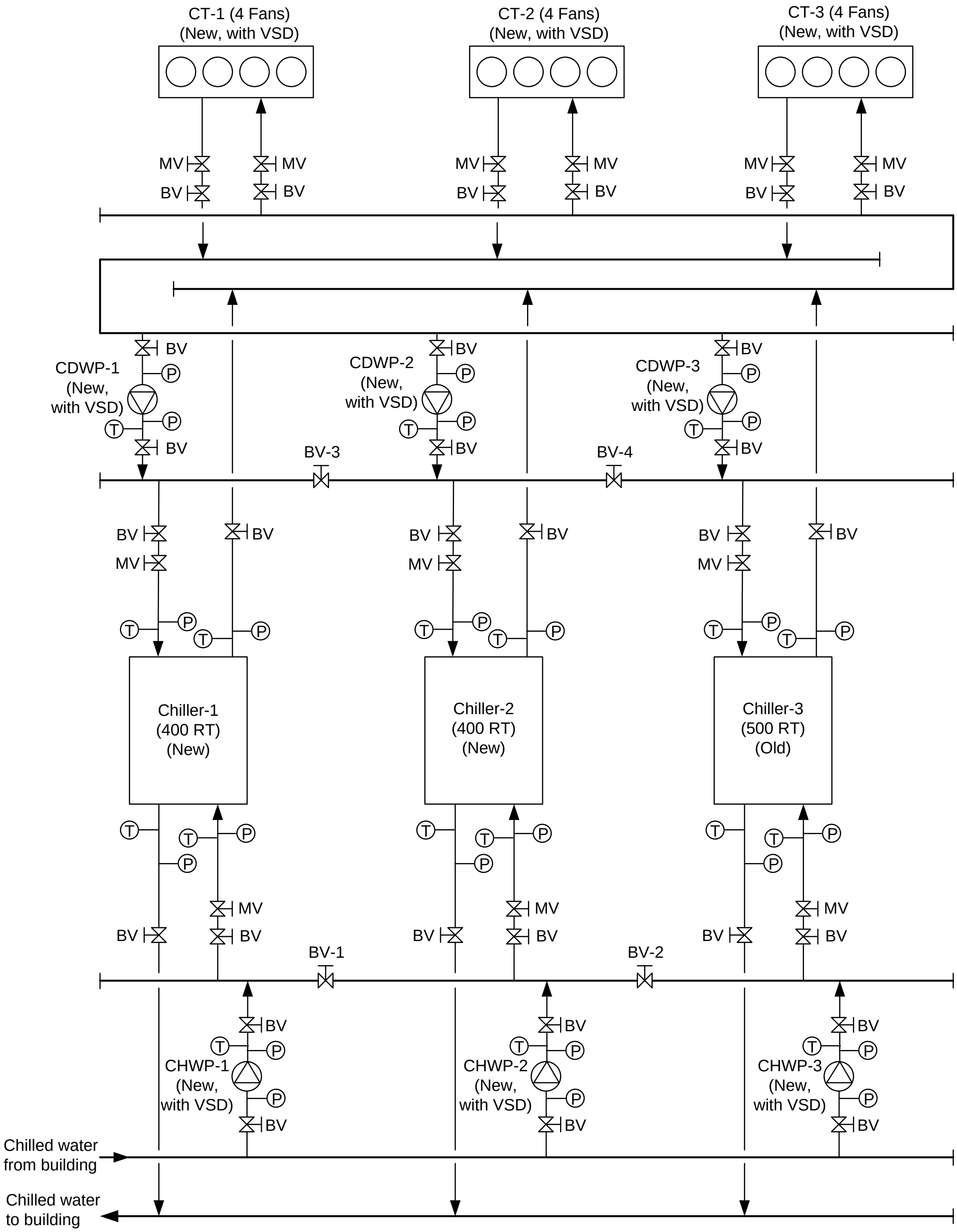


Figure 6.2 Water-cooled central chilled water system after retrofitting work (CHWP: Chilled water pump, CDWP: Condenser water pump, CT: Cooling tower, BV: Butterfly valve, MV: Motorized valve)

The performances of the old and retrofitted chilled water systems were measured. The main findings are presented below:

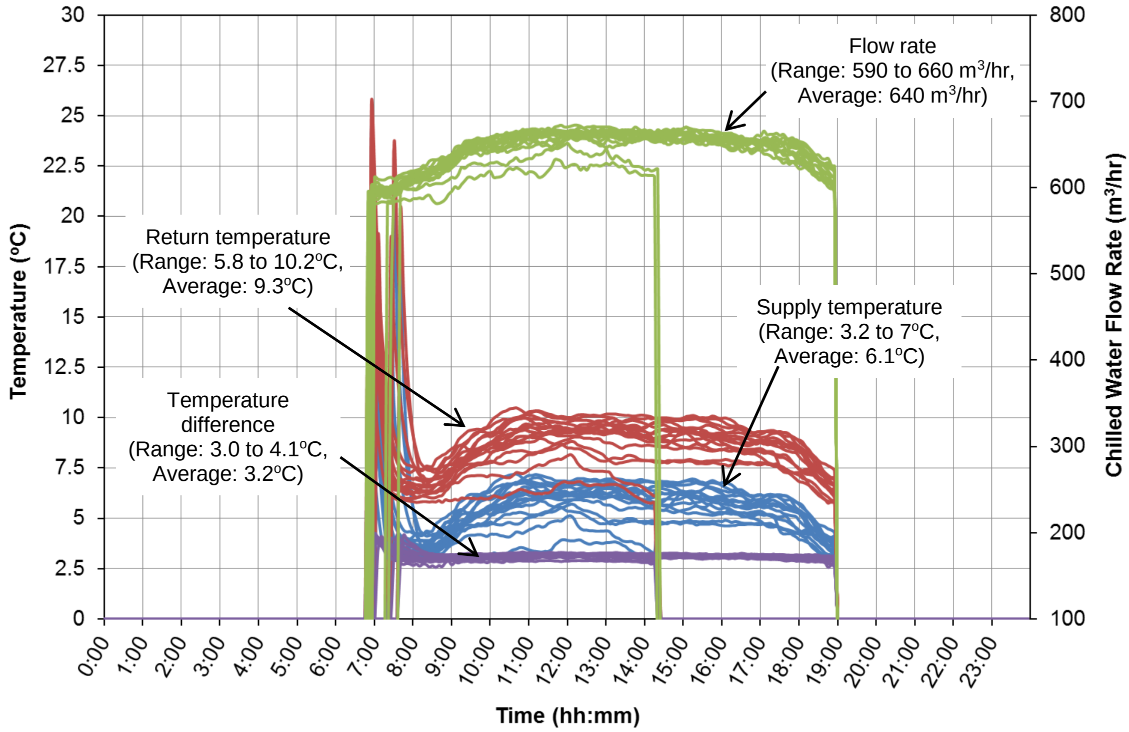


Figure 6.3 Total flow rate and temperature profiles of chilled water for 2 nos. of old chillers

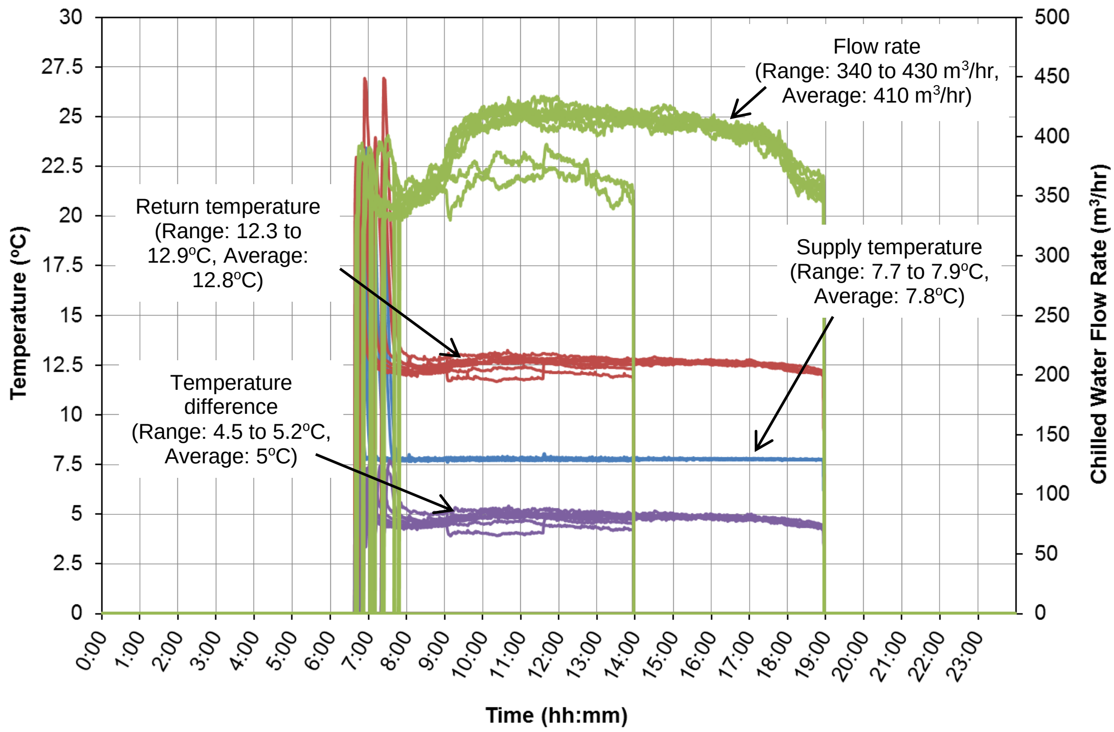


Figure 6.4 Total flow rate and temperature profiles of chilled water for 2 nos. of retrofitted chillers

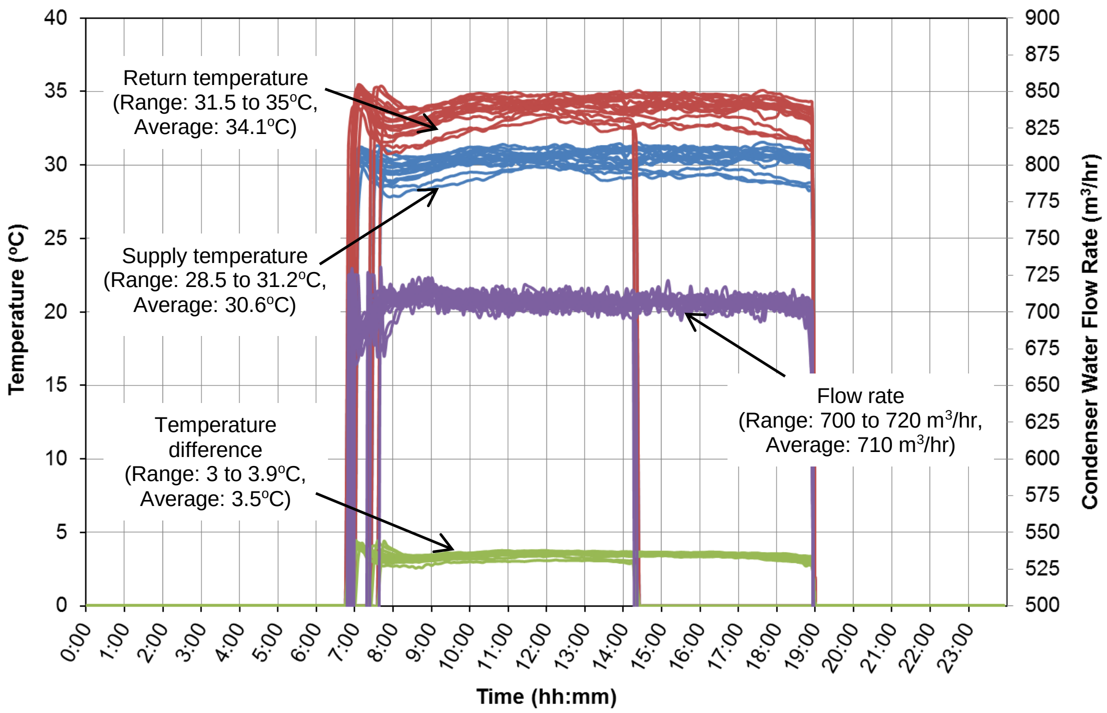


Figure 6.5 Total flow rate and temperature profiles of condenser water for 2 nos. of old chillers

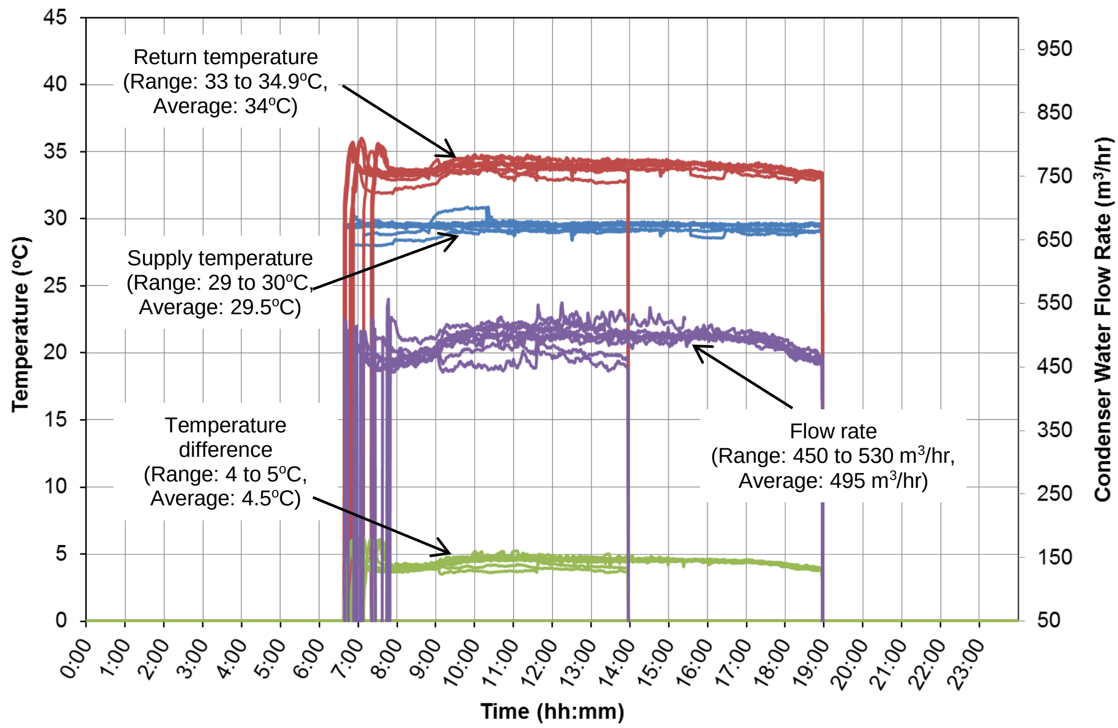


Figure 6.6 Total flow rate and temperature profiles of condenser water for 2 nos. of retrofitted chillers

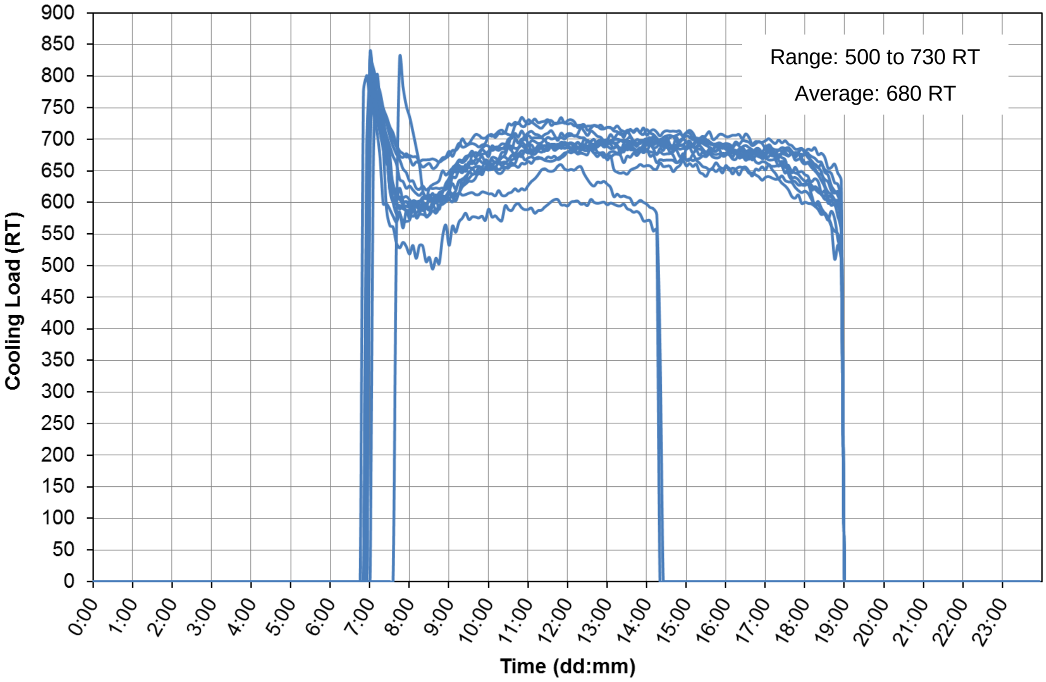


Figure 6.7 Total cooling load profile for 2 nos. of old chillers

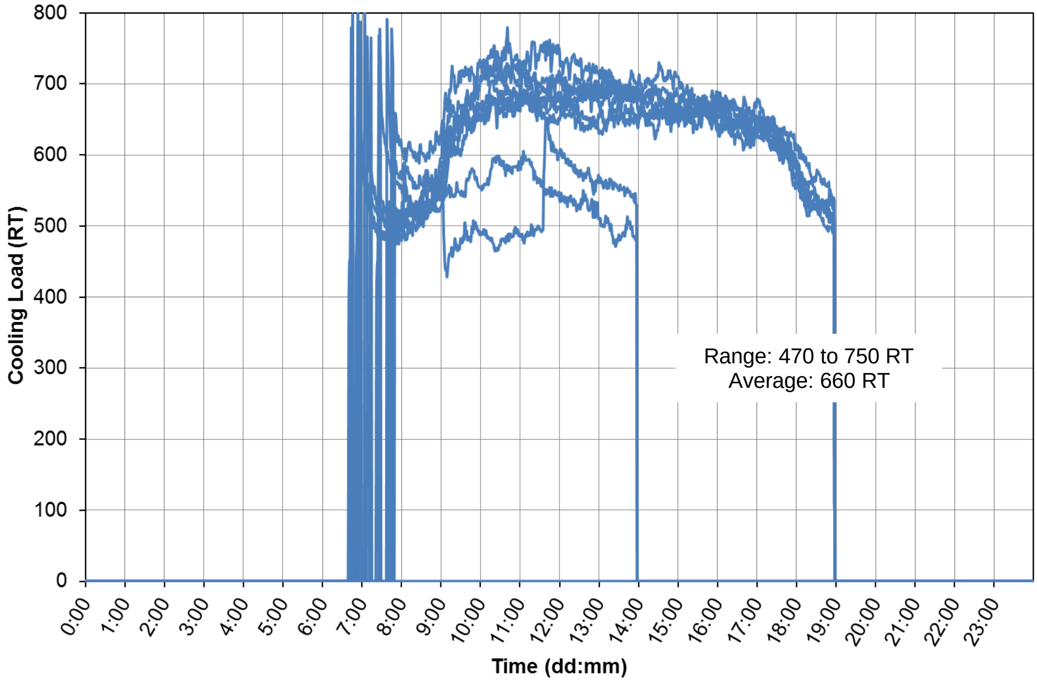


Figure 6.8 Total cooling load profile for 2 nos. of retrofitted chillers

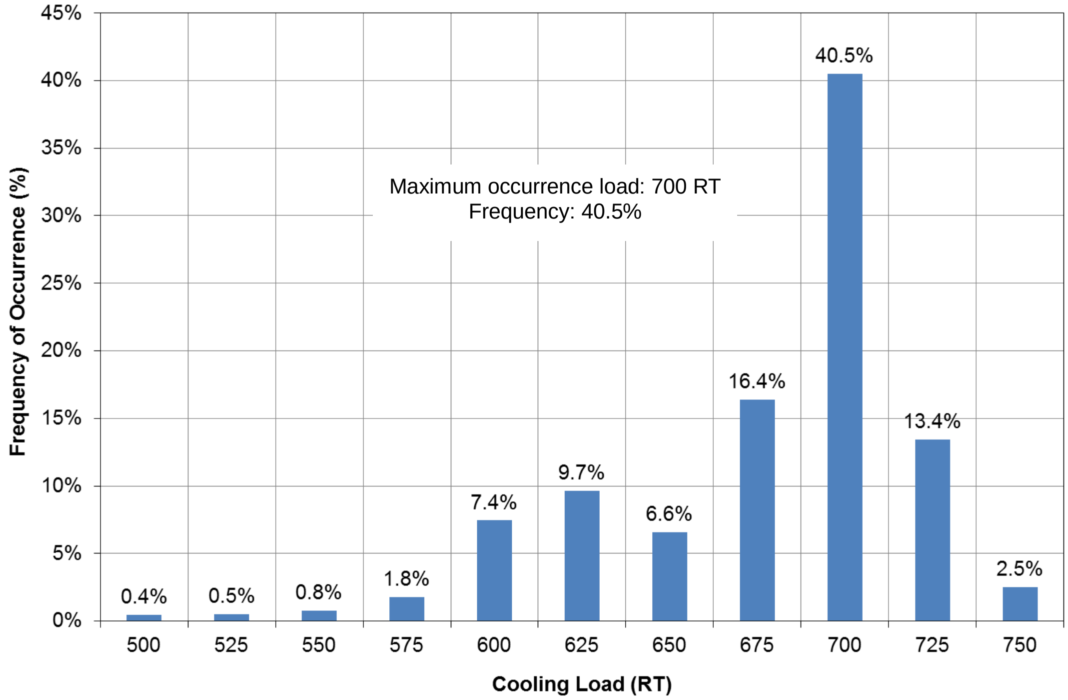


Figure 6.9 Frequency of cooling load occurrence for old chilled water system

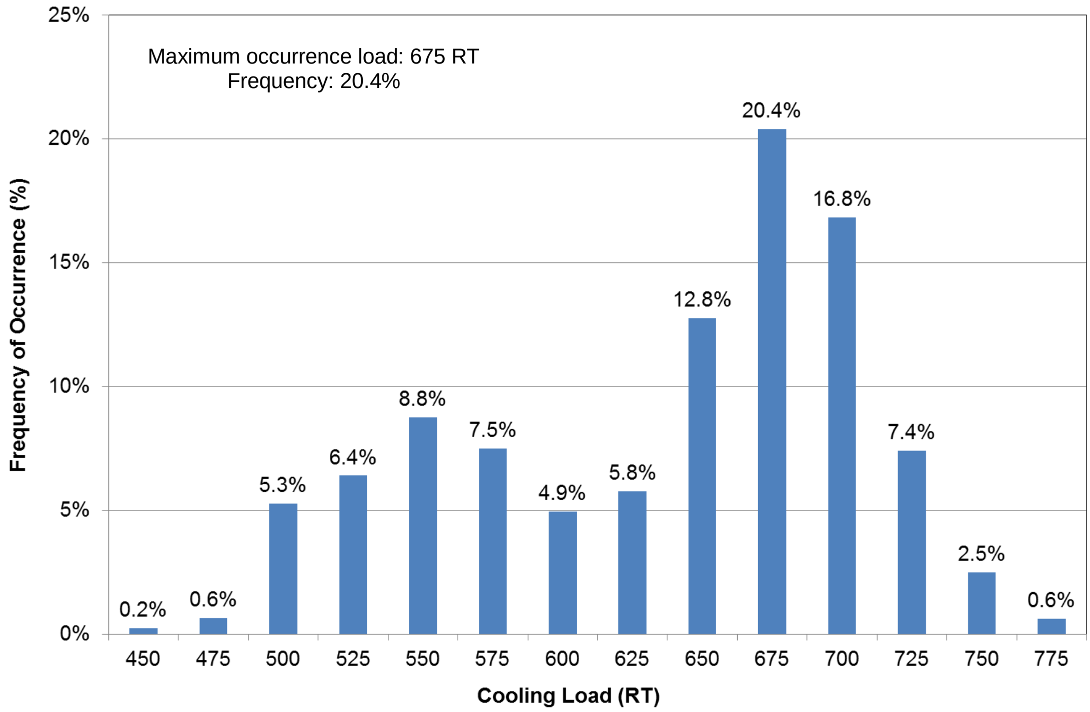


Figure 6.10 Frequency of cooling load occurrence for retrofitted chilled water system

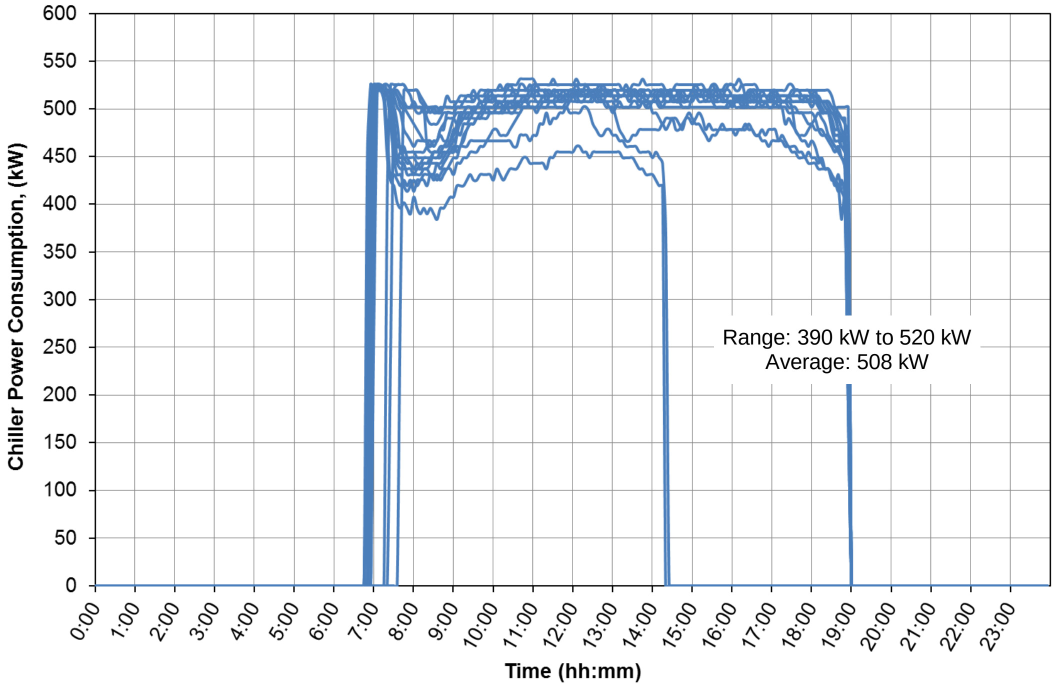


Figure 6.11 Total power consumption profile for 2 nos. of old chillers

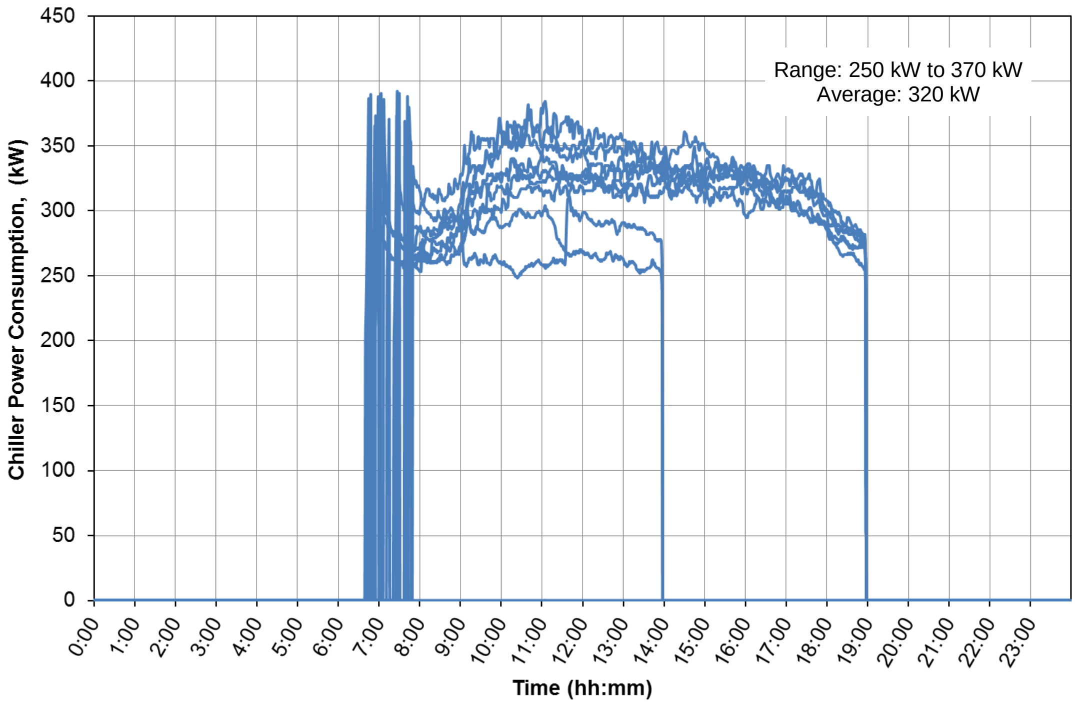


Figure 6.12 Total power consumption profile for 2 nos. of retrofitted chillers

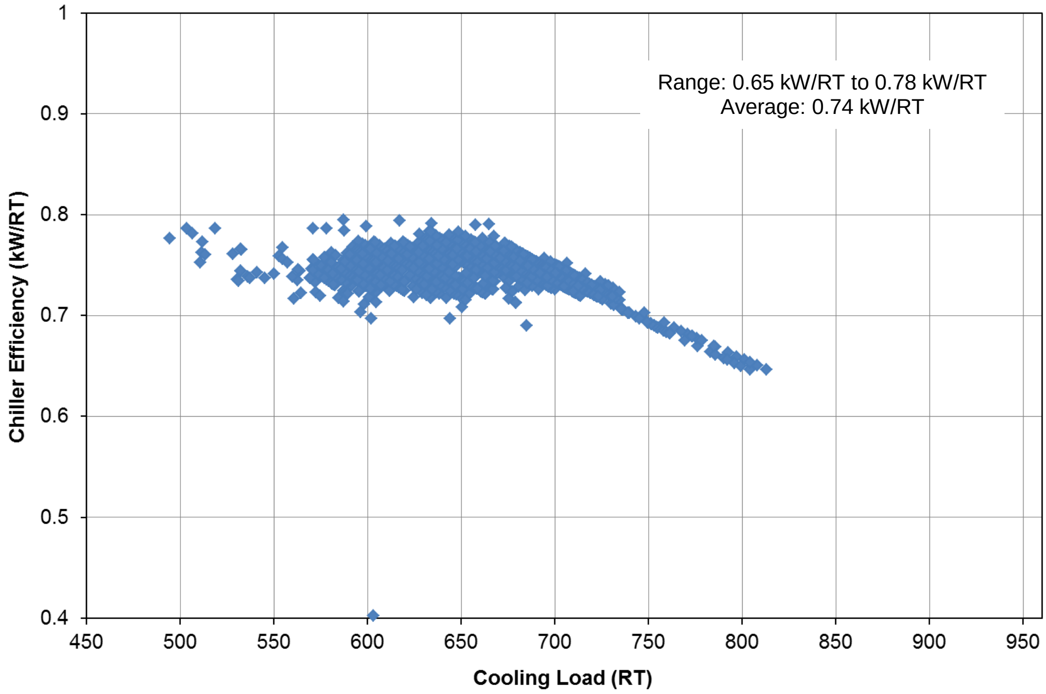


Figure 6.13 Efficiency profile for 2 nos. of old chillers

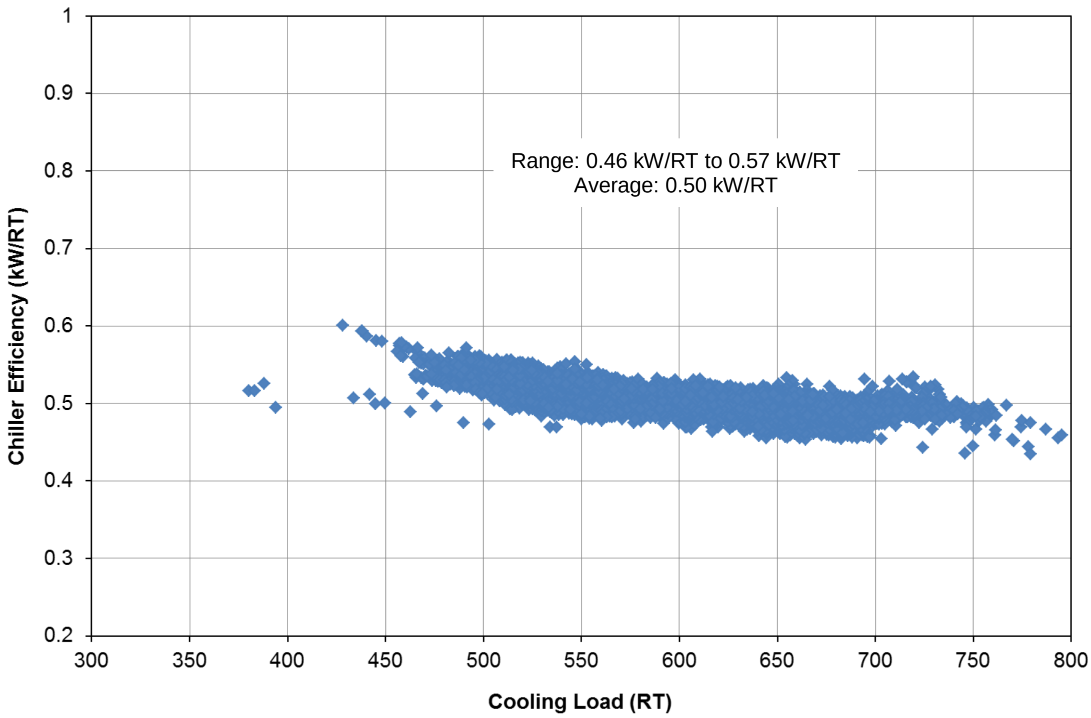


Figure 6.14 Efficiency profile for 2 nos. of retrofitted chillers

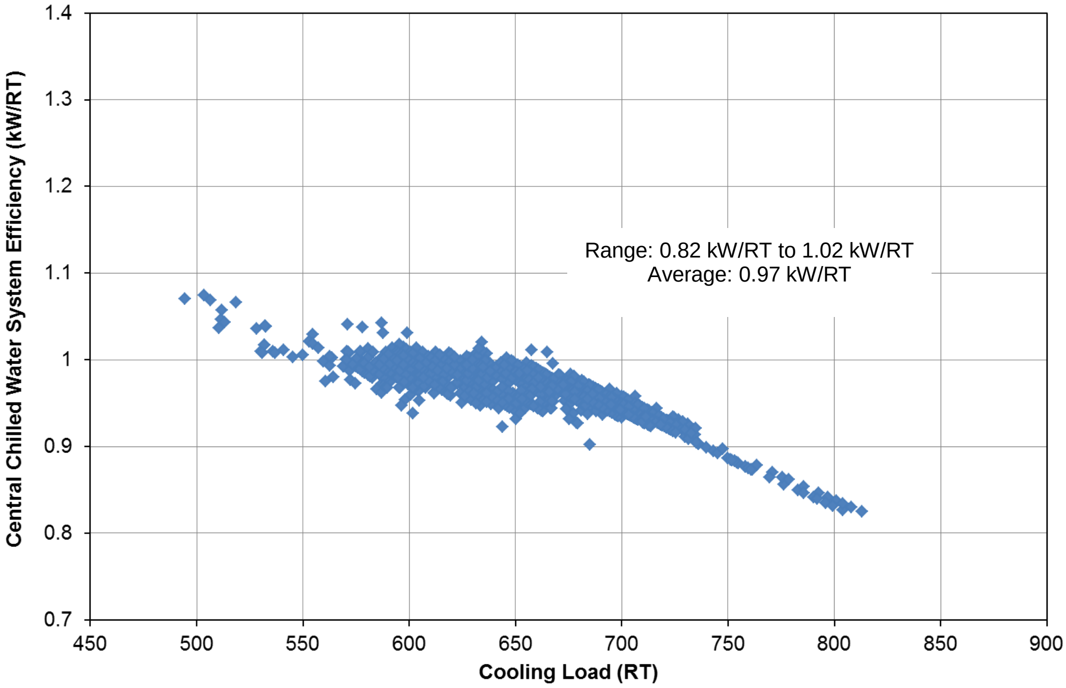


Figure 6.15 Old central chilled water system efficiency profile

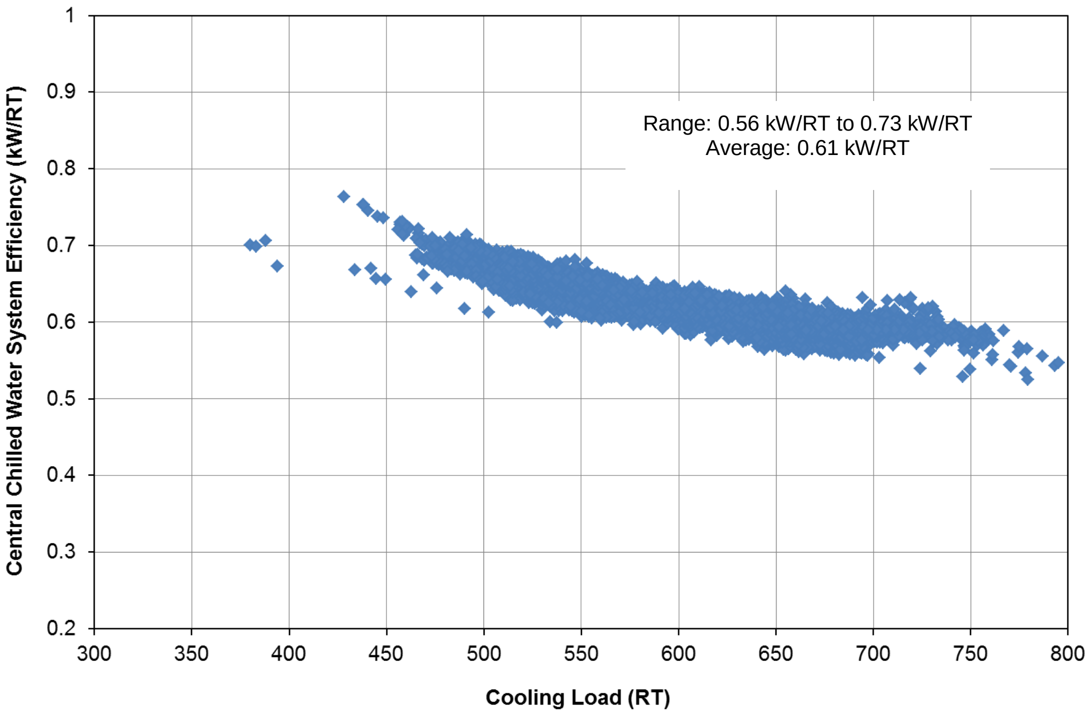


Figure 6.16 New central chilled water system efficiency profile

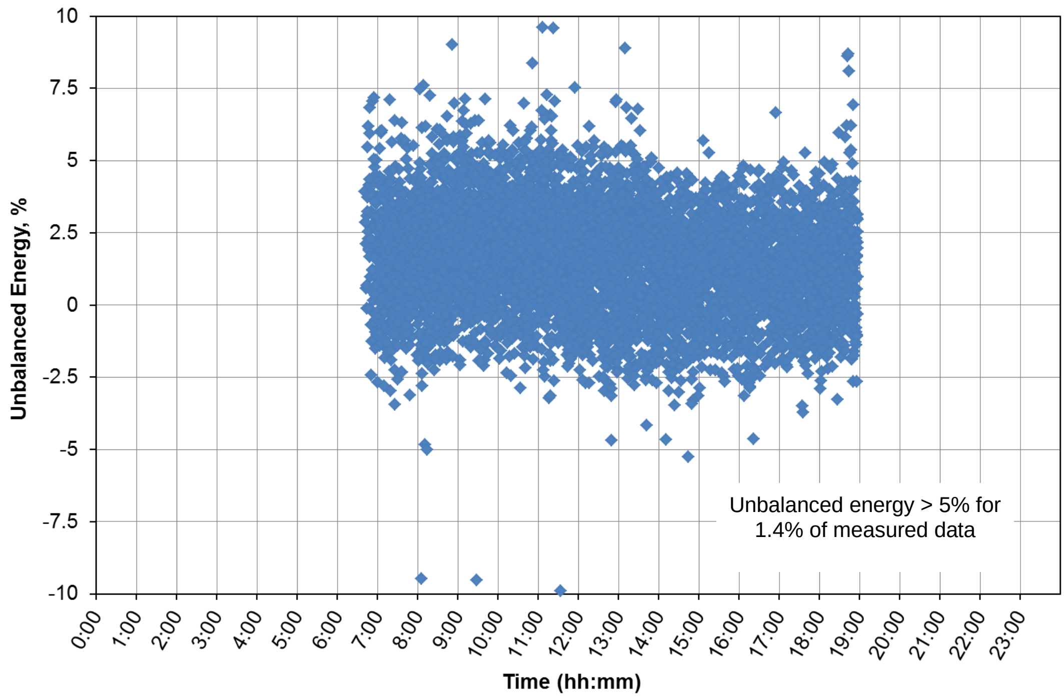


Figure 6.17 Percentage of unbalanced energy for old chilled water system

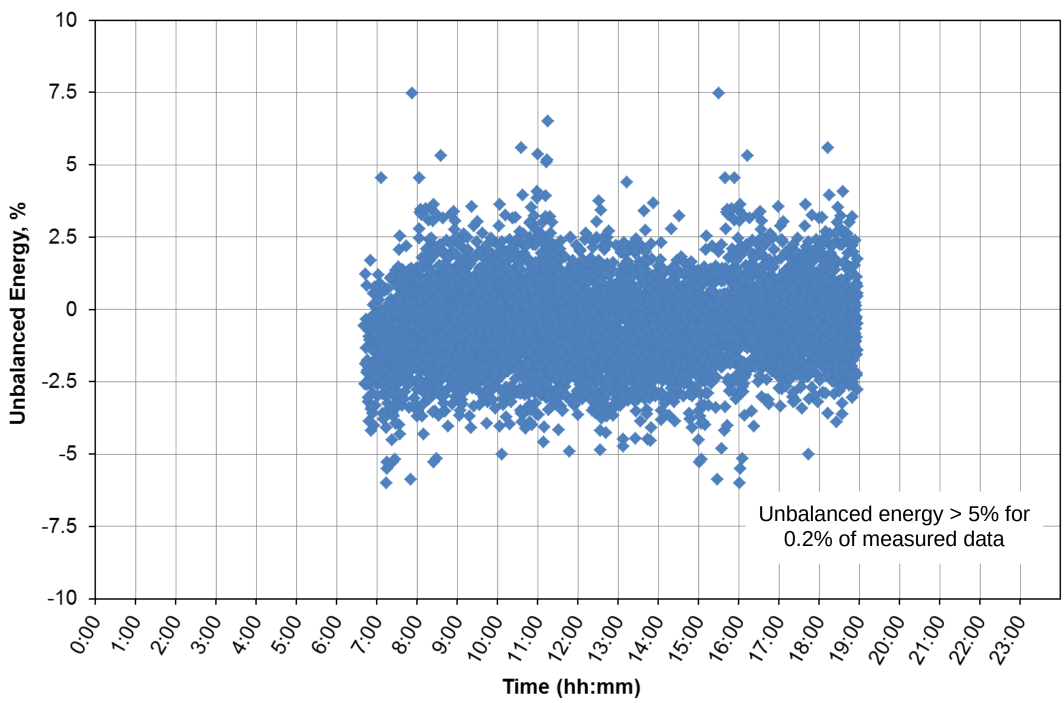


Figure 6.18 Percentage of unbalanced energy for retrofitted chilled water system

Based on the above data plots, the performance of the old and retrofitted chilled water systems are summarised in Table 6.1 below:

Table 6.1 Performance of the old and retrofitted chilled water systems

Parameters	Old chilled water system (2 nos. of operating chillers)		Retrofitted chilled water system (2 nos. of operating chillers)	
	Design	Measured	Design	Measured
Chilled water flow rate, m^3/hr	545.2	590 to 660 (Av. 640)	436.1	340 to 430 (Av. 410)
Chilled water supply temperature, ^oC	6.7	3.2 to 7.0 (Av. 6.1)	6.7	7.7 to 7.9 (Av. 7.8)
Chilled water return temperature, ^oC	12.2	5.8 to 10.2 (Av. 9.3)	12.2	12.3 to 12.9 (Av. 12.8)
Chilled water temperature difference, ^oC	5.5	3.0 to 4.1 (Av. 3.2)	5.5	4.5 to 5.2 (Av. 5.0)
Condenser water flow rate, m^3/hr	681.5	700 to 720 (Av. 710)	545.2	450 to 530 (Av. 495)
Condenser water supply temperature, ^oC	29.5	28.5 to 31.2 (Av. 30.6)	29.5	29.0 to 30.0 (Av. 29.5)
Condenser water return temperature, ^oC	35.0	31.5 to 35.0 (Av. 34.1)	35.0	33 to 34.9 (Av. 34.0)
Condenser water temperature difference, ^oC	5.5	3.0 to 3.9 (Av. 3.5)	5.5	4.0 to 5.0 (Av. 4.5)
Cooling load, RT	2x500	500 to 730 (Av. 680)	2x400	470 to 750 (Av. 660)
Maximum occurrence of cooling load, RT	---	700 (Frequency: 40.5%)	---	675 (Frequency: 20.4%)
Chiller power consumption, kW	---	390 to 520 (Av. 508)	360 (at 100% loading)	250 to 370 (Av. 320)
Chiller efficiency, kW/RT	---	0.65 to 0.78 (Av. 0.74)	0.45 (at 100% loading)	0.46 to 0.57 (Av. 0.50)
Central chilled water system efficiency, kW/RT	---	0.82 to 1.02 (Av. 0.97)	---	0.56 to 0.73 (Av. 0.61)

The main observations for the old and retrofitted chilled water systems are summarised below:

i. Chilled and condenser water flow rates for the old chilled water system were higher than the design flow rates.

ii. Chilled and condenser water flow rates for the retrofitted chilled water system were modulated using variable speed pumps based on real-time cooling load.

iii. Chilled water supply temperature for the old chilled water system was varying from 3.2^{o}C to 7.0^{o}C with an average value of 6.1^{o}C.

iv. To enhance the energy performance of the retrofitted chilled water system, chilled water supply temperature was increased to 7.8^{o}C. AHUs were able to maintain preset space temperature of 24^{o}C and relative humidity of 65% using the raised chilled water supply temperature of 7.8^{o}C.

v. Average percentage of loading for the old chillers was about 100x680/(2x500) = 68%. As the percentage of loading and chilled water supply temperature for the old chillers were low, the operating efficiency of the chillers was poor (average: 0.74 kW/RT).

vi. For the old chilled water system, the average efficiencies of the chillers and the overall central chilled water system were 0.74 kW/RT and 0.97 kW/RT respectively. Therefore, the combined energy efficiency of the chilled water pumps, condenser water pumps and cooling towers was about (0.97 – 0.74) = 0.23 kW/RT.

vii. To increase the percentage of loading, the capacity of the retrofitted chillers was reduced to 400 RT. Hence, the average percentage of loading during the period of commissioning was improved to about 100x660/(2x400) = 82.5%.

viii. Due to the increase of chilled water supply temperature and chiller loading, the average operating efficiency of the retrofitted chiller was improved to 0.50 kW/RT.

ix. For the retrofitted chilled water system, 9 nos. of triple duty valves were removed which helped to reduce the pressure loss of the piping system significantly. The capacity of the cooling towers was also modulated based on real-time cooling load by changing the speed of the fans using VSDs.

x. As the speed of the pumps and cooling towers were modulated based on real-time cooling load and unnecessary pressure losses of the piping system were eliminated, the energy performance of the pumps and cooling towers improved remarkably.

xi. For the retrofitted chilled water system, the average efficiencies of the chillers and the overall central chilled water system were 0.50 kW/RT and 0.61 kW/RT respectively. Therefore, the combined energy efficiency of the retrofitted chilled water pumps, condenser water pumps and cooling towers was (0.61 – 0.50) = 0.11 kW/RT.

xii. Based on SS 591 criteria, calculated unbalanced energy for central chilled water systems should be less than 5% for minimum 80% of the measured data. Figures 6.17 and 6.18 show that SS 591 criteria are fulfilled for the old and retrofitted central chilled water systems.

Calculation of Savings: Based on the frequency of cooling load occurrence during the period of commissioning of the retrofitted chilled water system (Figure 6.10), the energy and operating cost savings are presented below:

Cooling Load, RT (A)	Occurrence (B)	Operating hours/day (C=B x Operating hours)	Old chilled water system efficiency, kW/RT (D)	Retrofitted chilled water system efficiency, kW/RT (E)	Energy consumption of old chilled water system, kWh (A)x(C)x(D)	Energy consumption of retrofitted chilled water system, kWh (A)x(C)x(E)
450	0.002	0.024	1.05	0.73	11.34	7.88
475	0.006	0.072	1.05	0.68	35.91	23.26
500	0.053	0.636	1.05	0.67	333.90	213.06
525	0.064	0.768	1.02	0.66	411.26	266.11
550	0.088	1.056	1.01	0.64	586.61	371.71
575	0.075	0.900	1.00	0.63	517.50	326.03
600	0.049	0.588	0.99	0.62	349.27	218.74
625	0.058	0.696	0.98	0.60	426.30	261.00
650	0.128	1.536	0.97	0.59	968.45	589.06
675	0.204	2.448	0.96	0.58	1586.30	958.39
700	0.168	2.016	0.94	0.58	1326.53	818.50
725	0.074	0.888	0.93	0.59	598.73	379.84
750	0.025	0.300	0.89	0.58	200.25	130.50
775	0.006	0.072	0.86	0.56	47.99	31.25
Operating hours/day =		**12**	**Energy consumption, kWh/day =**		**7,400.34**	**4,595.33**

Daily electrical energy consumption of old chilled water system to support the present cooling load = 7,400.34 kWh/day

Daily electrical energy consumption of retrofitted chilled water system to support the present cooling load = 4,595.33 kWh/day

Number of days the chilled water plant is operated = 280 days per year

Annual electrical energy consumption of old chilled water system to support the present cooling load = 7,400.34 x 280 = 2,072,095 kWh/year

Annual electrical energy consumption of retrofitted chilled water system to support the present cooling load = 4,595.33 x 280 = 1,286,692 kWh/day

Annual electrical energy savings = 2,072,095 – 1,286,692 = **785,403 kWh / year**

Percentage of chilled water plant energy savings = 100x (785,403 / 2,072,095) = **37.9%**

Electricity tariff = $0.20 / kWh

Annual operating cost savings = 785,403 x 0.20 = **$157,081 per year**

Cost for the retrofitting of chilled water plant = $820,000

Simple payback period = 820,000 / 157,081 = **5.2 years**

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