Air-Conditioning and Mechanical Ventilation (ACMV) Systems

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Preface

For a tropical climate like Singapore, air-conditioning and mechanical ventilation (ACMV) system is required to operate throughout the year to maintain thermal comfort inside the buildings. ACMV system consumes about 60% of the total energy consumption of a typical commercial building. Energy efficient design and optimisation of the ACMV system contribute significantly in reducing energy consumption and achieving cost savings. Designing and operating the central air-conditioning systems in an energy efficient manner while maintaining the recommended Indoor Air Quality (IAQ) remains a challenge for design engineers and operators. More industrial facility and building owners and professionals are interested to understand the energy efficient design and optimisation opportunities for the ACMV systems to trim their overall operating costs.

Electricity is the main form of energy used to operate the ACMV systems. Non-renewable fossil fuels such as oil, gas and coal are generally used to generate the electricity. These fossil fuels during combustion emit carbon dioxide, which contributes to global warming. Since the ACMV systems of commercial buildings and industrial plants consume a significant amount of electrical energy, government agencies in many countries including Singapore’s National Environment Agency (NEA) and Building and Construction Authority (BCA) are also promoting proper energy management of the ACMV systems as a means to trim carbon dioxide emissions.

This reference manual (monograph) on ACMV systems is developed mainly for the professionals who are interested to broaden their knowledge in energy efficient design, optimisation and operation of the ACMV systems to reduce the energy and operating costs as well as the students who are specializing in energy systems. The monograph is written based on the author’s experience in research and energy saving projects for over 10 years. Many actual case studies and examples with technical analysis presented in this document have been successfully implemented. The readers can confidently apply the concepts on their individual needs. The monograph contains five self-contained chapters on the different aspects and systems relating to the energy management of the ACMV systems.

Chapter 1 provides an introduction to the air-conditioning chilled water system, vapour-compression refrigeration cycles, types of air-conditioning systems, components and layouts of central chilled water systems, measurement and calculation of chilled water system performance, optimisation strategies of chilled water system based on design and operational aspects and the thermal energy storage systems.

Chapter 2 is devoted to different types of pumping systems used in chilled water systems, interaction of pump and system characteristic curves, pump sizing, applications of affinity laws, constant and variable flow systems and optimisation strategies for pumping systems.

Chapter 3 deals with different types of air handling systems used in ACMV systems, types of fans and their performances, interaction of fan and system characteristic curves, selection of fan, applications of affinity laws, constant and variable air volume AHU systems, energy management strategies for air handling systems and modified air handling systems.

Chapter 4 covers the Psychrometrics of air-conditioning processes, properties of moist air, determination of moist air properties, types of heat gain in spaces, processes involve in AHU and spaces which include mixing of different air streams, cooling and dehumidification of air as well as ventilation effects.

Chapter 5 discusses the different configurations of cooling tower systems, heat transfer processes in cooling towers, selection and optimisation strategies for cooling towers, installation and maintenance of cooling towers.

Finally, a case study on retrofitting of central chilled water system is presented in Chapter 6.

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I am extremely grateful to my past and present colleagues, especially Professor N.E. Wijeysundera and Dr. Lal Jayamaha, without whom I would not have been able to successfully complete so many research projects, energy audits and energy saving projects. Finally, I would like to express my sincere thanks to my wife Minni, son Shadman and daughter Suhi for their continued patience, understanding and support throughout the preparation of this monograph, which involved many long hours when my face was glued to my computer screen.

Dr. Md Raisul Islam

Chapter-1: Air-conditioning Chilled Water System

1. Air-conditioning Chilled Water System

The purpose of an Air-Conditioning and Mechanical Ventilation system is to maintain comfort condition in the air-conditioned space irrespective of the outdoor ambient condition. Comfort condition usually refers to a specific range of temperature, relative humidity, cleanliness and distribution of air to meet the comfort requirements of the occupants in air-conditioned spaces. For a tropical country like Singapore, outdoor ambient temperature is generally higher than the comfort temperature of conditioned spaces. Therefore, air-conditioning chilled water systems are required to operate throughout the year to maintain the comfort condition in the commercial buildings, hotels, hospitals, industrial buildings etc. Central air-conditioning plant is the single largest energy consumer that consumes about 60% of the total energy consumption of a typical commercial building. Wise selection of the components of central air-conditioning plant and energy smart control and operation strategies may contribute to the significant energy and cost savings. In this section of the reference manual, the working principles of central air-conditioning chilled water systems will be discussed. Methodologies for the evaluation of the energy performance and relative advantages and disadvantages of different types of chilled water systems will be presented. Finally, the opportunities for the energy savings and relevant control strategies will be elaborated.

Learning Outcomes

Participants will be able to:

i. Analyse the pressure-enthalpy diagram and identify the influences of operating variables on chiller performance.
ii. Evaluate the performance of the chilled water systems.
iii. Select appropriate chilled water systems based on cooling load behavior of the buildings and other considerations.
iv. Calculate potential energy and cost savings due to the retrofitting of chilled water systems.
v. Develop energy smart operational and control strategies.

1.1 Vapour-Compression Refrigeration Cycles

We all know from experience that heat flows naturally from high-temperature to low-temperature spaces without requiring any devices. However, the reverse process of transferring heat from low-temperature conditioned spaces to the high-temperature outdoor ambient air requires a special device called Air-Conditioning and Mechanical Ventilation system. The air-conditioning machines, commonly known as chillers, are cyclic devices. The working fluid used in the air-conditioning cycle is called refrigerant. R123 and R134a are two commonly used refrigerants in air-conditioning systems. Ozone depletion potentials (ODP) of R123 and R134a are 0.02 and 0, respectively. However, Global worming potential (GWP) of R123 and R134a are 77 and 1430, respectively. Based on Singapore Green Mark criteria, ODP of refrigerant should be 0 or GWP should be less than 100. Both R123 and R134a meet Singapore Green Mark criteria. CFC refrigerants such as R12 and R22 are not environment friendly. Therefore, they are phased out and should not be used in air-conditioning systems. Manufacturers of the air-conditioning systems also carefully consider other properties of refrigerants such as efficiency, toxicity, atmospheric life, operating pressure, cost etc. The most frequently used air-conditioning cycle is the vapour-compression air-conditioning cycle, which consists of four main components: (a) Compressor, (b) Condenser, (c) Expansion valve and (d) Evaporator.

The arrangement of the components and the variation of pressure and specific enthalpy of the refrigerant at different stages of the air-conditioning cycle are shown in Figure-1.1.

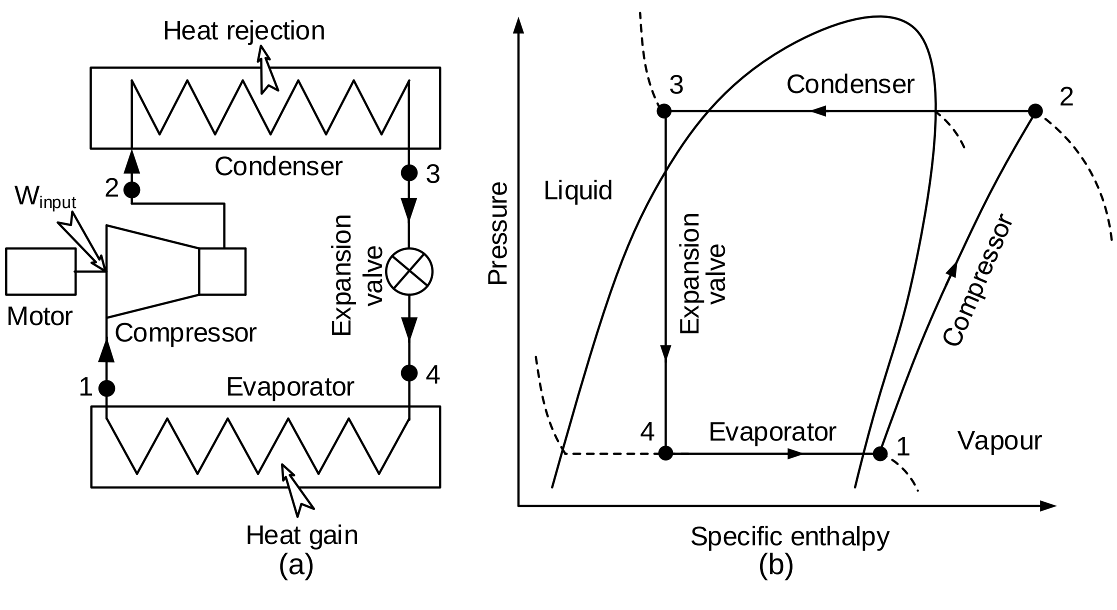


Figure 1.1 (a) Main components of air-conditioning system, (b) Pressure vs. specific enthalpy diagram

1.2 Function of Main Components

1) **Compressor:** The function of the compressor is to increase the pressure of refrigerant vapour from evaporator pressure (point-1) to the condenser pressure (point-2). The compressor is driven by an electric motor using an appropriately designed gear box or transmission system. Compressor receives refrigerant as a saturated or slightly superheated vapour at low temperature and pressure (point-1) and discharges as superheated vapour at relatively high pressure and temperature (point-2).

2) **Condenser:** The condenser is a heat exchanger. The function of the condenser is to reject heat from the refrigerant vapour to the surrounding ambient air or the cooling water known as condenser water. Superheated refrigerant vapour of relatively high pressure and temperature (point-2) enters the condenser, rejects latent heat of condensation and leaves as saturated or slightly subcooled liquid refrigerant of high pressure and temperature (point-3).

3) **Expansion valve:** The function of the expansion valve is to reduce the pressure of liquid refrigerant from the condenser pressure (point-3) to the evaporator pressure (point-4). Same as any other liquid, boiling temperature of liquid refrigerant drops due to the reduction of refrigerant pressure after the expansion valve (point-4). Low pressure liquid refrigerant partly evaporates (change phase from liquid to vapour) by absorbing latent heat of vapourisation from the liquid refrigerant itself and enters the evaporator as a liquid vapour mixture (point-4) of low pressure and temperature.

4) **Evaporator:** Same as condenser, the evaporator is also a heat exchanger. In the evaporator, the liquid refrigerant absorbs heat from the surrounding air or circulating water known as chilled water. Partially evaporated liquid vapour mixture of refrigerant of low pressure and temperature (point-4) enters the evaporator, absorbs latent heat of vapourisation from the surrounding air or chilled water and then leaves as saturated or slightly superheated refrigerant vapour of low pressure

and temperature (point-1). The cycle is completed as the refrigerant leaves the evaporator and again enters the compressor.

1.3 Types of Air-Conditioning Systems

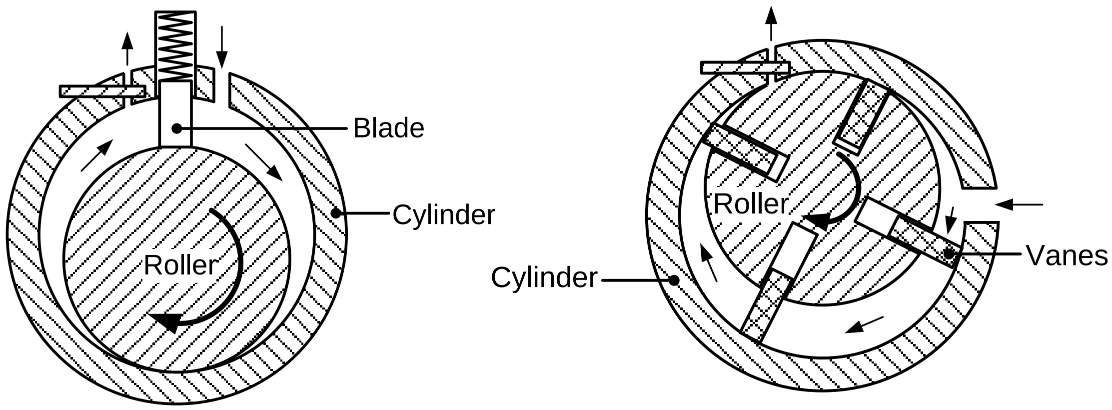
Air-conditioning systems are broadly classified as: (a) Vapour compression type and (b) Vapour absorption type.

Vapour compression type air-conditioning systems: The compressor is the biggest energy consuming component of vapour compression type air-conditioning systems. Vapour compression type systems are further classified according to the types of compressors used for the compression of the refrigerant. The common types of compressors used in the chillers are:

i. Rotary compressor.
ii. Reciprocating compressor.
iii. Scroll compressor.
iv. Screw compressor.
v. Centrifugal compressor.

Rotary compressor: Rotary compressors belong to the positive displacement type. There are mainly two designs for the rotary compressors which are: (a) Rolling piston type and (b) Rotating vane type as shown in Figure-1.2. In the rolling piston type compressor, a roller is mounted on an eccentric shaft. One spring loaded blade always remains in contact with the roller and acts as a moving separation wall between the high and low pressure refrigerant. Because of the eccentric motion of the roller, refrigerant enters the cylinder through the inlet port and is trapped by the roller. As the roller moves eccentrically, the space between the cylinder and the roller reduces. This results in the compression of the refrigerant.

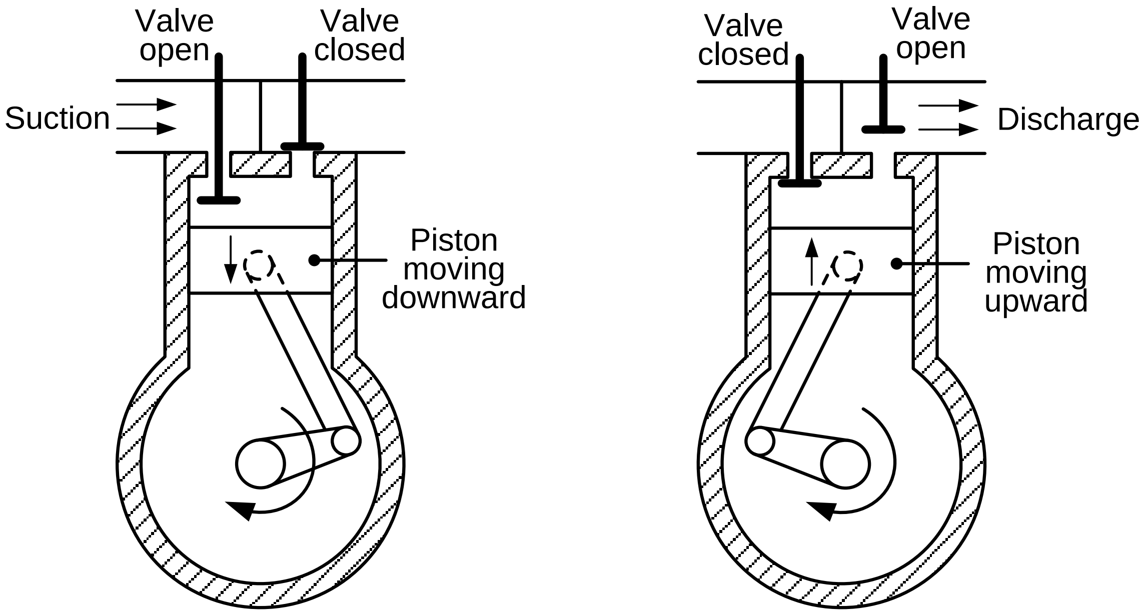
In the rotating vane type compressor, the assembly of the rotor is off-center with respect to the cylinder. The rotor is concentric with the shaft. More than one vanes slide within the rotor and keep in contact with the cylinder. As the rotor rotates, refrigerant enters the cylinder through the inlet port. The refrigerant is trapped between the sliding vanes. Because of the off-center assembly of the rotor and cylinder, the trapped refrigerant is squeezed to high pressure and finally discharged through the outlet port. Rotary compressors have high volumetric efficiency due to negligible clearance. They are generally used for low cooling capacity applications.



(a) Rolling piston type compressor (b) Rotating vane type compressor

Figure 1.2 Rotary compressor

Reciprocating compressor: Main components of reciprocating compressors are cylinder, piston, connecting rod, crank shaft, motor and valves. The crank shaft is driven by a motor. Rotational motion of crank shaft is converted to reciprocating motion of the piston using a connecting rod. Suction and discharge valves sequentially open and close the inlet and outlet ports of the cylinder while the piston reciprocates inside the cylinder resulting in the suction and compression of the refrigerant. The hermetically sealed compressors have their motor enclosed along with the cylinder and crank shaft inside a dome. The motor of the hermetically sealed compressor is cooled by the incoming suction vapour of refrigerant. They have the advantages of less noise and no refrigerant leakage. In open type compressors, on the other hand, the motor is mounted externally and cooled by the surrounding air. Reciprocating compressors also belong to the positive displacement type. Reciprocating compressors are typically used for low-capacity applications up to about 100 RT (350 kW). Multiple compressors are used for high-capacity application. Figure-1.3 shows the suction and discharge of a single compressor. Multiple compressors are usually installed in parallel to increase the refrigerant flow rate and the cooling capacity of the air-conditioning machine.



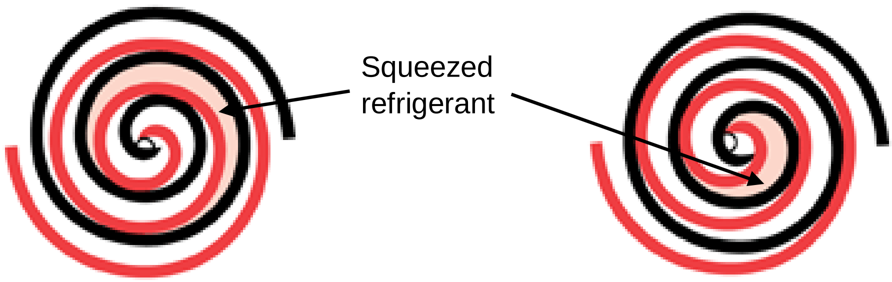
(a) Suction of refrigerant (b) Discharge of refrigerant

Figure 1.3 Reciprocating compressor

Scroll compressor: Scroll compressors contain two spiral-shaped scroll members which are interfitted as shown in Figure-1.4. One scroll remains stationary while the

a) Refrigerant enters the outer openings

b) Open passages are sealed off as refrigerant is drawn in.



c) Refrigerant is compressed into two increasingly smaller pockets

d) High pressure refrigerant is discharged at the center port

Figure 1.4 Compression process of scroll compressor other rotates eccentrically. Due to the eccentric motion and profile of the scrolls, refrigerant drawn in through the inlet opening is compressed between the scrolls during the rotation as shown in Figure-1.4 and then discharged at the outlet port. Scroll compressors also belong to the positive displacement type. Scroll compressors are typically used for low-capacity applications up to about 20 RT (70 kW).

Screw compressor: Screw compressors contain a set of male and female helically grooved rotors as shown in Figure-1.5. The male rotor is generally connected with the motor and functions as the driving rotor. The other female rotor generally functions as the driven rotor. Refrigerant enters into the grooves of the rotor through the inlet port. Due to the rotation of the rotors, compression of refrigerant is achieved by direct volume reduction of the grooves. Finally, the compressed refrigerant is discharged through the outlet port. Screw compressors also belong to the positive displacement type. Screw compressors are typically used for medium-capacity applications up to about 600 RT (2110 kW).

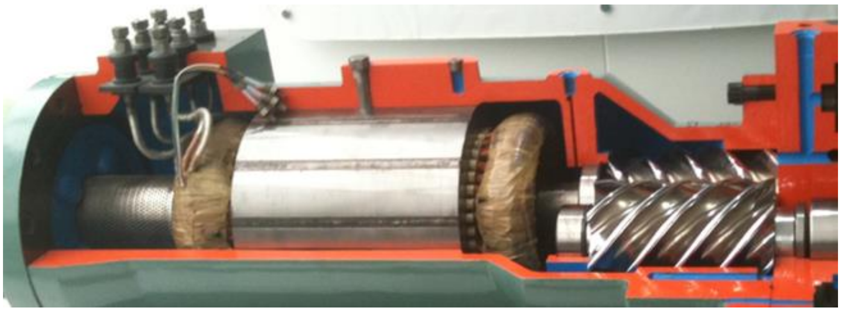
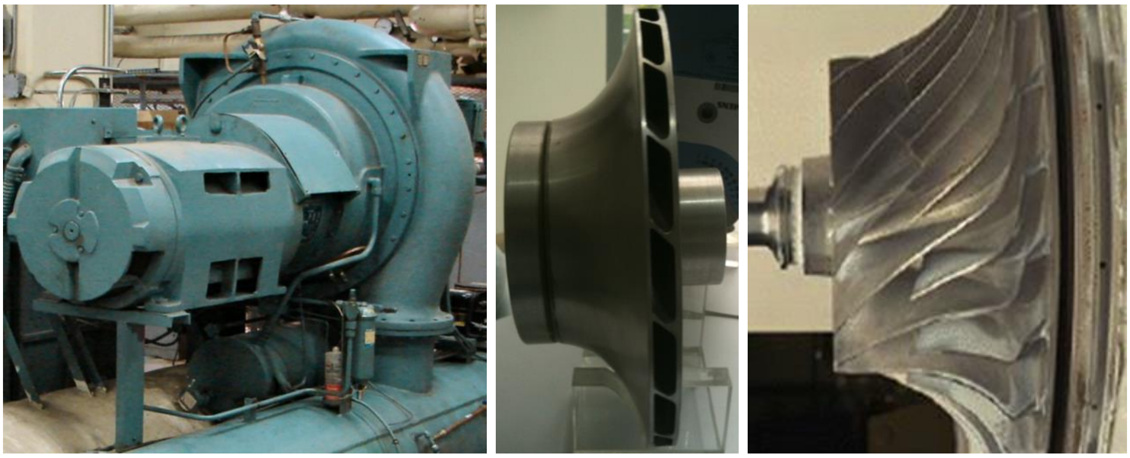


Figure 1.5 Sectional view of screw compressor

Centrifugal compressor: Centrifugal compressors consist of impeller housed inside a volute casing. The impeller is mounted on a shaft and rotates at high speed. Refrigerant enters the impeller in the axial direction and is discharged radially at high velocity. The kinetic energy of the refrigerant is then converted to static pressure in the diffuser. Photograph of the centrifugal compressor is shown in Figure-1.6. Multiple impellers are used with intercooler for multistage compression. Centrifugal compressors are used for high-capacity applications, usually above 300 RT (1055 kW).



(a) Assembly of centrifugal compressor (b) Two different designs of impeller

Figure 1.6 Centrifugal compressor

Vapour absorption type air-conditioning systems: A great majority of air-conditioning plants operate on vapour-compression refrigeration systems that use electricity as the primary input energy. Vapour absorption chilled water system is a viable alternative to vapour compression refrigeration machines. Vapour absorption systems use heat energy as the main input energy form. This makes it possible to use energy forms such as waste heat, solar energy and others of temperature about 80 to 100^{o}C to operate vapour absorption chilled water systems. Moreover, vapour absorption systems could be conveniently incorporated in Combined Heat and Power (CHP) plants that produce both electricity and low grade heat.

Figure-1.7 (a) shows a schematic diagram of a vapour compression system. The low-pressure refrigerant evaporates in the evaporator to produce the cooling effect. The resulting refrigerant vapour is compressed to the condenser pressure by a compressor that usually consumes electrical energy. In the condenser, the high-pressure refrigerant is condensed and the resulting heat is released to the ambient. The condensed refrigerant is finally expanded to the evaporator pressure through an expansion valve to continue the cycle.

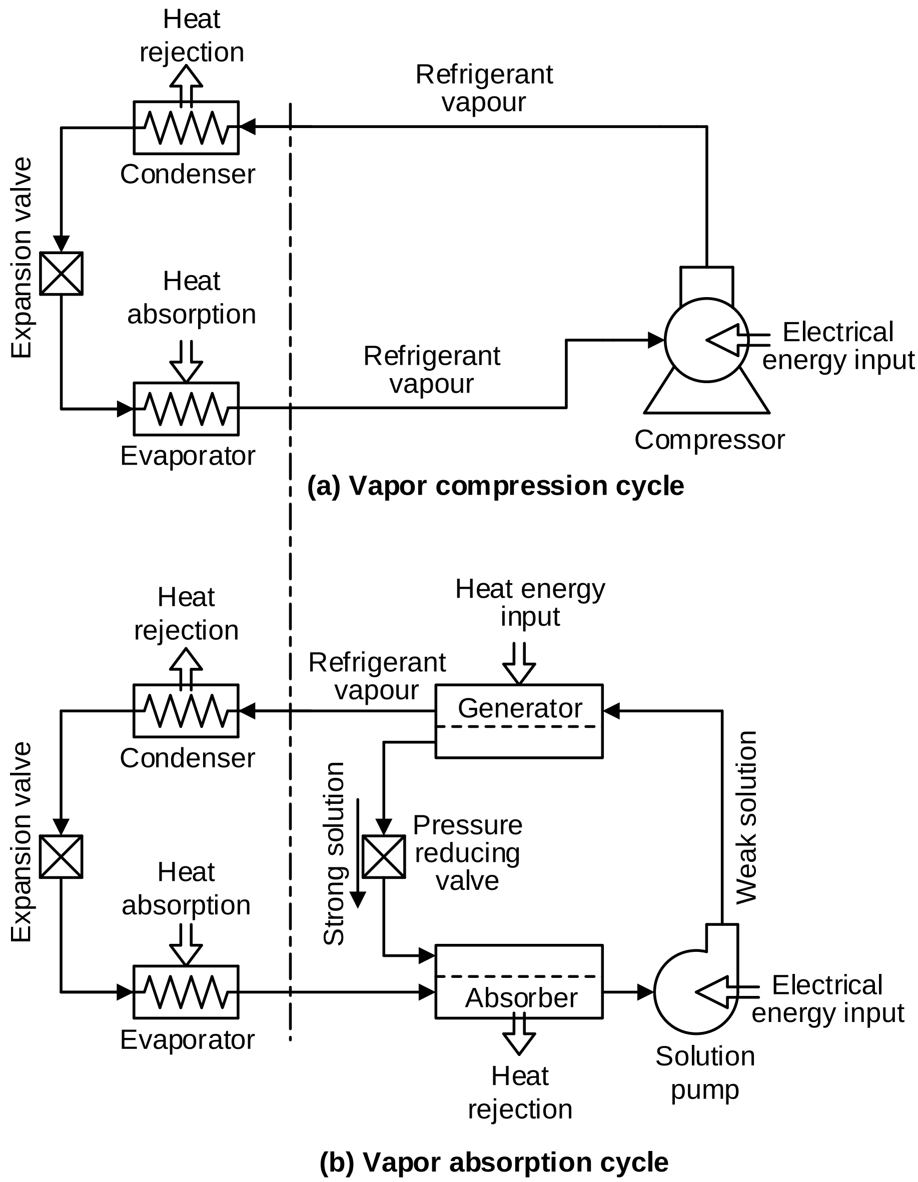


Figure 1.7 Comparison of vapour compression and vapour absorption cycles

In the vapour absorption system, shown schematically in Figure-1.7 (b), the compressor is replaced by a combination of an absorber and a generator. In order to accomplish the function of the compressor, the absorber-generator units include a secondary fluid called the absorbent. The refrigerant vapour from the evaporator is absorbed in the absorber unit, which operates at a low pressure close to that of the evaporator. As the absorption process evolves the heat of absorption, it is necessary to cool the absorber using an external cooling system.

The dilute solution in the absorber is pumped to the generator with a solution pump that requires some electrical energy input. This energy input, however, is much smaller than that required in the compressor of a vapour compression system. The solution in the generator is heated using an external energy source, to evolve the refrigerant out of the solution. The concentrated solution then flows back to the absorber through a pressure reducing valve. The refrigerant continues its passage through the rest of the cycle in a manner similar to that of a vapour compression system.

The vast majority of commercially available vapour absorption chilled water systems use lithium bromide as the absorbent and water as the refrigerant. A detailed schematic diagram of such a system is shown in Figure-1.8. The main components of the system are condenser, refrigerant expansion valve, evaporator, absorber, generator, solution heat exchanger and solution pressure reducing valve. The absorber and the evaporator are housed in the same sealed enclosure, as they operate at about the same low pressure. Similarly, the generator and the condenser are housed in an enclosure at a higher pressure. The water to be chilled flows through the tube bundle that is located in the evaporator. The heat removal from the absorber and the condenser is done by circulating water between these units and the cooling tower, which eventually rejects the heat to the atmosphere. The solution in the generator is heated by passing steam or a hot fluid through the tube bundle in the generator. This constitutes the main energy input to the system.

The concentrated solution from the generator flows through a heat exchanger where it heats the weak solution on its way to the generator. This regenerative process helps to reduce the heat input required in the generator. The condensed refrigerant is expanded through an expansion valve and sprayed over evaporator tubes to produce the cooling effect by evaporation. In the absorber, the water vapour is absorbed in the film of solution as it flows over the cooled absorber surfaces. A simplified schematic diagram of the vapour absorption chilled water system with typical temperatures of the main components is shown in Figure-1.9.

Lithium Bromide-water is widely used as an absorbent and refrigerant pair in vapour absorption chilled water systems due to their many desirable properties. Lithium bromide is a hygroscopic salt. The boiling point of lithium bromide is much higher than that of water. Therefore, the refrigerant vapour is boiled off from the solution as pure water vapour. This is a major advantage because there is no absorbent carry over to the other sections of the refrigeration cycle, such as the evaporator. However, there is a possibility of crystallisation of lithium bromide from the concentrated solution in the colder sections of the systems. The concentrated solution at the exit of the heat exchanger is particularly vulnerable due to heat removal from the solution.

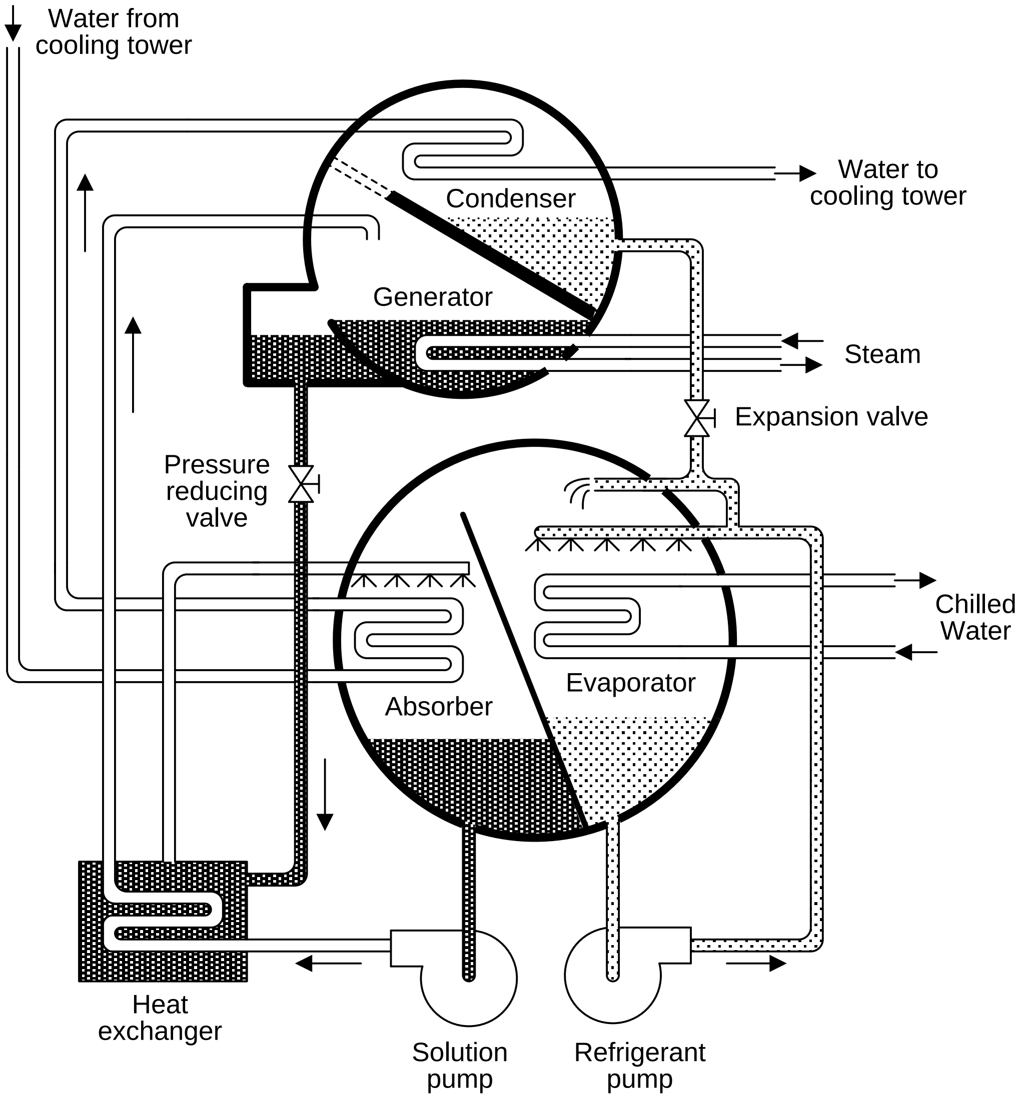


Figure 1.8 Detailed schematic diagram of vapour absorption chilled water system

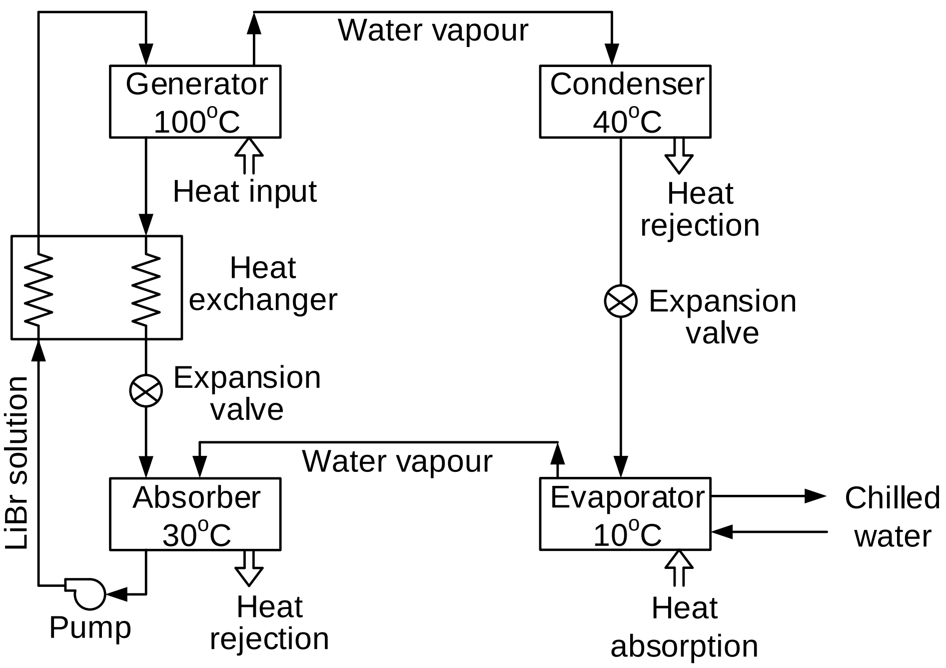


Figure 1.9 Simplified schematic diagram of vapour absorption chilled water system

As lithium bromide is corrosive in the presence of air, inhibitors like lithium nitrate and lithium chromate are added to absorption systems to retard corrosion. The efficiency of the vapour absorption chiller is quite low in comparison to the vapour compression chilled water system. However, vapour absorption chilled water system still could be an attractive alternative as waste heat of industrial plants is generally used to operate the system.

1.4 Components and Layouts of Central Chilled Water System

Based on cooling load of the air-conditioning spaces, different types of air-conditioning systems such as central air-conditioning chilled water systems, water-cooled package units, variable refrigerant volume (VRV) systems and stand-alone package units are commonly used in commercial buildings and industries. Central air-conditioning chilled water systems are commonly selected to support relatively large cooling demand. A typical water-cooled central air-conditioning chilled water system is shown in Figure-1.10. Main components of central air-conditioning systems are:

i. Chiller
ii. Chilled water pump (CHWP)
iii. Condenser water pump (CDWP)
iv. Cooling tower and
v. Air handling unit (AHU)

As shown in Figure-1.10, chilled water pump (CHWP) circulates chilled water through the evaporator, air handling units (AHUs) and fan coil units (FCUs) located at different parts of the building. Refrigerant evaporates in the evaporator by absorbing latent heat of vapourisation from the circulating water. The chilled water then flows from the evaporator to the cooling coil of AHUs and FCUs. Blowers of the AHUs and FCUs blow air over the cooling coils to transfer heat from the circulating air to the chilled water. The treated cold air is then circulated to the air-conditioned spaces to maintain the comfort conditions. Typical arrangement of an air handling unit is shown in Figure-

1.11. After absorbing heat from the cooling coils of AHUs or FCUs, the chilled water is pumped back again to the evaporator of the chiller to cool down and repeat the cycle.

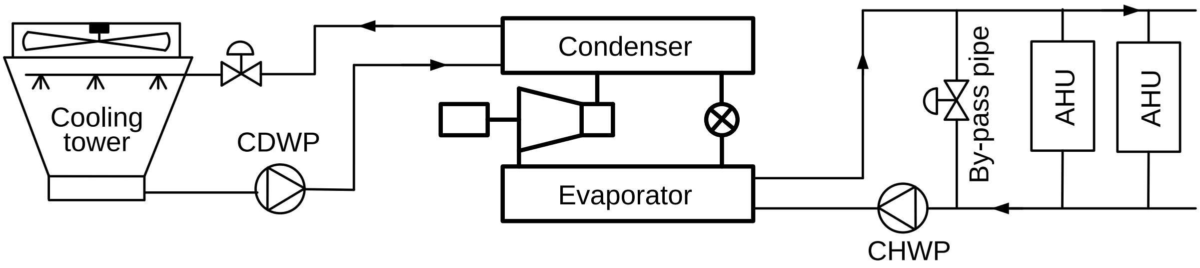


Figure 1.10 Main components of water-cooled central air-conditioning system

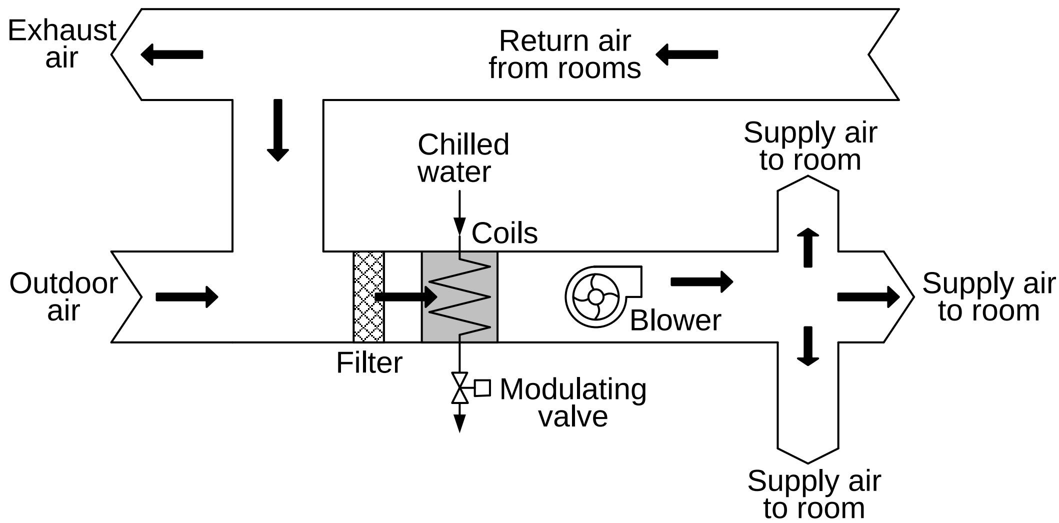


Figure 1.11 Main components of a typical air handling unit (AHU)

On the other hand, condenser water pump (CDWP) circulates water through the condenser of the chiller and the cooling tower as shown in Figure-1.10. The heat absorbed by the refrigerant in the evaporator and the heat added to the refrigerant by the compressor during the compression process of refrigerant are rejected to the circulating condenser water as latent heat of condensation. The warm condenser water then flows to the cooling tower (Figure-1.10) and rejects heat to the ambient air. Relatively cool condenser water accumulates in the basin of the cooling tower. Finally, the condenser water is pumped back again to the condenser of the chiller and continue the cycle. In air-cooled chillers, ambient air is blown using fans over the condenser coil to release condenser heat directly to the ambient air.

Usually more than one chiller with associated pumps and cooling towers are installed in the plant to support the cooling load of the spaces. Commonly designed chilled water pumping systems are: (a) primary chilled water pumping system and (b) primary-secondary chilled water pumping system. Figure-1.12 shows a central air-conditioning plant containing three sets of chillers with primary chilled water pumping system. Three sets of chilled water pumps (CHWP), condenser water pumps (CDWP) and cooling towers are connected with the chillers using common header pipes. The primary chilled water pumps can be operated at constant speed irrespective to the cooling load of the spaces. The speed of the primary chilled water pumps can also be modulated using Variable Speed Drive (VSD) based on real time demand of chilled water.

If the primary CHWPs are operated at constant speed, the flow modulating valve installed on the by-pass pipe is partially opened to by-pass excess chilled water when chilled water demand for the AHUs drops due to low cooling load of the spaces. On the other hand, if the speed of the primary CHWPs is modulated based on real time demand of chilled water, the flow modulating valve of the by-pass pipe remains close until chilled water demand for AHUs drops below the minimum chilled water flow requirements for the chillers. If chilled water demand for the AHUs drops below the minimum chilled water flow requirements for the chillers, the flow modulating valve of the by-pass pipe is opened partially to make up the shortfall of the chilled water for the chillers. Due to the reduction of the speed of CHWPs, power consumption of the CHWPs drops remarkably during part load operation of the chillers.

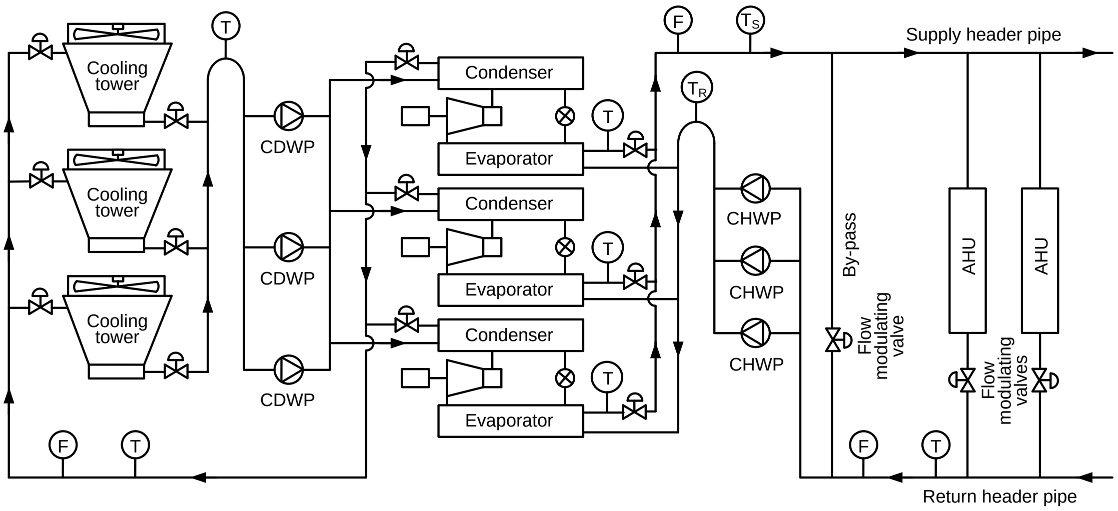


Figure 1.12 Central air-conditioning plant with primary chilled water pumping system. CHWP: Chilled water pump; CDWP: Condenser water pump; T: Temperature sensor; F: Flow meter.

Figure-1.13 shows a central air-conditioning plant containing three sets of chillers with primary-secondary chilled water pumping system. The primary chilled water pumps (PCHWPs) are usually small and operated at constant speed irrespective of the chiller cooling load. PCHWPs are designed to circulate chilled water through the chillers and the small decoupler piping loop (Figure-1.13) generally located in the chilled water system plant room. The secondary chilled water pumps (SCHWPs) are big and designed to supply chilled water to all the AHUs of the building. The speed of the SCHWPs are modulated using VSD based on real time demand of chilled water resulting in significant SCHWPs energy savings at low part load conditions. If the flow of the PCHWPs is higher than the flow of the SCHWPs, chilled water flows from supply header to return header through the decoupler pipe. One chiller with associated pumps and cooling tower can be turned off if the flow through the decoupler pipe is greater than or equals to the flow of one chiller. On the other hand, if the flow through the SCHWPs is higher than the flow of the PCHWPs, chilled water will flow from the return header to the supply header through decoupler pipe. Under this scenario, another chiller with associated pumps and cooling tower are turned on.

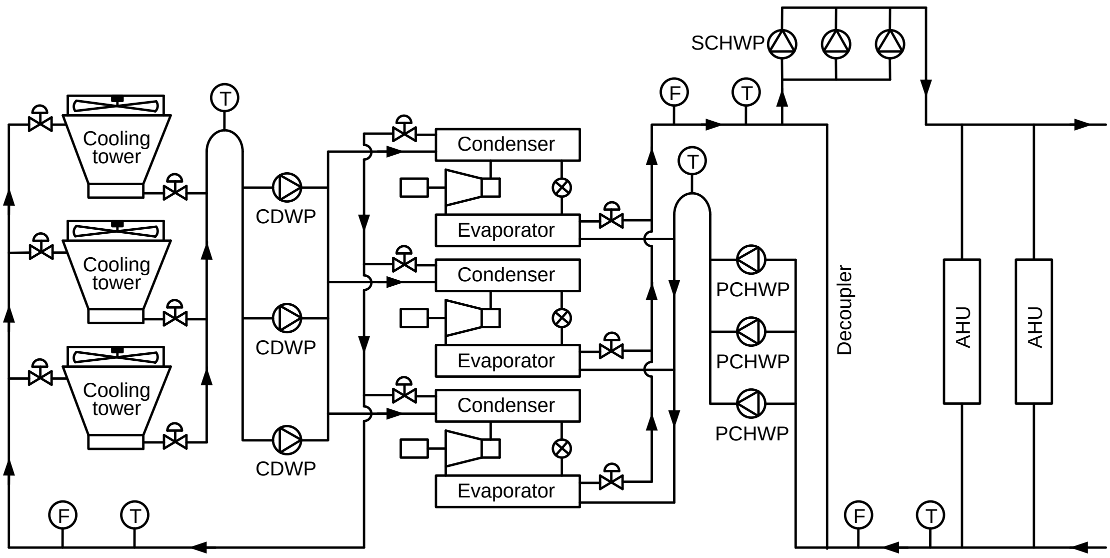


Figure 1.13 Central air-conditioning plant with primary and secondary chilled water pumping system. PCHWP: Primary chilled water pump; SCHWP: Secondary chilled water pump; CDWP: Condenser water pump; T: Temperature sensor; F: Flow meter.

1.5 Measurement and Calculation of Chilled Water System Performance

Calculation of Chiller Cooling Load: The cooling load of a chiller can be calculated using mass flow rate of refrigerant at point-1 and specific enthalpy of refrigerant at point-1 and 4 (Figure-1.14) using the following equation:

$$Q_{ev} = m_{r,1}\left(h_{r,1} - h_{r,4}\right) \tag{1.1}$$

where,

Q_{ev} = Cooling load for chiller or rate of heat gain in evaporator, kW

$m_{r,1}$ = Mass flow rate of refrigerant at point-1, kg/s

$h_{r,1}$ = Specific enthalpy of refrigerant at point-1, kJ/kg

$h_{r,4}$ = Specific enthalpy of refrigerant at point-4, kJ/kg

However, necessary flow meter, temperature and pressure sensors are usually not installed by the chiller manufacturers in the appropriate locations. Therefore, the cooling load of a chiller is usually measured by installing the flow meter and temperature sensors at the chilled water supply and return pipes as shown in Figure-1.14. Chilled water flow meter can be installed at the chilled water supply or return pipe. As chilled water return temperature is higher than the supply temperature, chilled water flow meter should be installed at the chilled water return pipe to minimise moisture condensation rate.

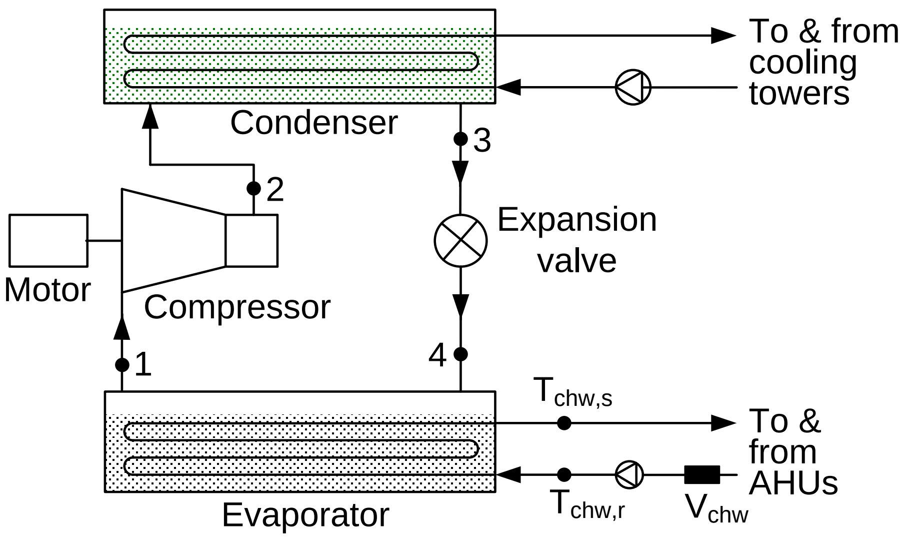


Figure 1.14 Location of sensors for measuring chiller cooling load

Cooling load of chiller or rate of heat gain in the evaporator can be calculated as:

$$Q_{ev} = V_{chw}\rho_{chw}C_p\left(T_{chw,r} - T_{chw,s}\right) \tag{1.2}$$

where

V_{chw} = Chilled water flow rate, m^3/s

ρ_{chw} = Chilled water density, kg/m^3

C_p = Specific heat of chilled water, kJ/kg. K

$T_{chw,r}$ = Chilled water return temp, ^oC

$T_{chw,s}$ = Chilled water supply temp, ^oC

Q_{ev} = Cooling load for chiller or the rate of heat gain in evaporator, kW

The cooling effect of the chiller is commonly expressed in refrigeration ton RT. One RT of cooling effect is defined as the rate of heat input required to melt one ton of ice in 24-hour period, which is equal to about 3.517 kW.

Therefore, cooling load of chiller or the rate of heat gain in evaporator in RT = (Rate of heat gain in evaporator in kW) / 3.517.

If the flow rate of the chilled water is measured in GPM, the following expression can be used to calculate the cooling load of the chiller in RT:

$$Q_{ev} = V_{chw}\left(T_{chw,r} - T_{chw,s}\right)/13.3 \quad (1.3)$$

where

V_{chw} = Chilled water flow rate, GPM

$T_{chw,r}$ = Chilled water return temp, ^oC

$T_{chw,s}$ = Chilled water supply temp, ^oC

Q_{ev} = Cooling load for chiller or the rate of heat gain in evaporator, RT

Cooling load of chiller or the rate of heat gain in evaporator in kW = (Rate of heat gain in evaporator in RT) x 3.517.

Conversion Unit: 1 GPM = 0.0631 L/s = (0.0631×10^{-3}) m^3/s

Calculation of Heat Rejection Rate by Condenser: The heat rejection rate from the condenser of a chiller can be calculated using mass flow rate of refrigerant at point-3 and specific enthalpy of refrigerant at point-2 and 3 (Figure-1.15) using the following equation:

$$Q_{cd} = m_{r,3}\left(h_{r,2} - h_{r,3}\right) \quad (1.4)$$

where,

Q_{cd} = Heat rejection rate from condenser, kW

$m_{r,3}$ = Mass flow rate of refrigerant at point-3, kg/s

$h_{r,2}$ = Specific enthalpy of refrigerant at point-2, kJ/kg

$h_{r,3}$ = Specific enthalpy of refrigerant at point-3, kJ/kg

However, as stated earlier, the necessary flow meter, temperature and pressure sensors are usually not installed by the chiller manufacturers in the appropriate locations. Therefore, the heat rejection rate from the condenser is usually measured by installing the flow meter and temperature sensors at the condenser water supply and return pipes as shown in Figure-1.15. Condenser water flow meter can be installed at the condenser water supply or return pipe.

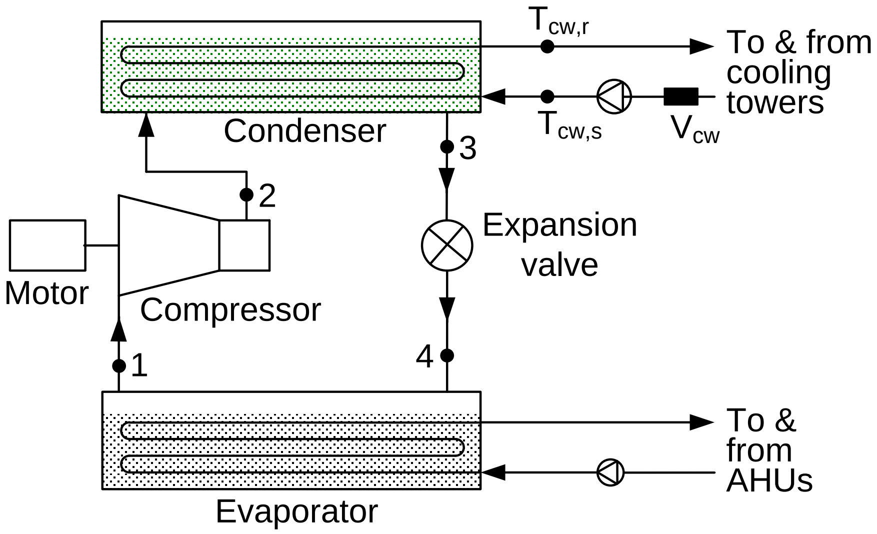


Figure 1.15 Location of sensors for measuring heat rejection rate of condenser

Heat rejection rate from the evaporator can be calculated as:

$$Q_{cd} = V_{cw}\rho_{cw}C_p\left(T_{cw,r} - T_{cw,s}\right) \tag{1.5}$$

where

V_{cw} = Condenser water flow rate, m^3/s

ρ_{cw} = Condenser water density, kg/m^3

C_p = Specific heat of condenser water, kJ/kg. K

$T_{cw,r}$ = Condenser water return temp, ^oC

$T_{cw,s}$ = Condenser water supply temp, ^oC

Q_{cd} = Heat rejection rate from condenser, kW

Heat rejection rate from condenser in RT = (Heat rejection rate from condenser in kW) / 3.517.

If the flow rate of the condenser water is measured in GPM, the following expression can be used to calculate the heat rejection rate from the condenser in RT:

$$Q_{cd} = V_{cw}\left(T_{cw,r} - T_{cw,s}\right) / 13.3 \tag{1.6}$$

where

V_{cw} = Condenser water flow rate, GPM

$T_{cw,r}$ = Condenser water return temp, °C

$T_{cw,s}$ = Condenser water supply temp, °C

Q_{cd} = Heat rejection rate from condenser, RT

Heat rejection rate from condenser in kW = (Heat rejection rate from condenser in RT) x 3.517.

Energy Balance Equation for Chiller: Heat loss or gain by radiation, convection, bearing friction, oil coolers, etc. are relatively small and usually neglected in overall energy balance for chiller. Omitting the effect of the above heat losses and gains, the general overall heat balance equation (based on energy conservation law: input energy is equal to the output energy) for a chiller as shown in Figure-1.16 can be written as:

$$Q_{ev} + W_{input} = Q_{cd} \tag{1.7}$$

where

Q_{ev} = Heat gain in evaporator, kW

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

Q_{cd} = Heat rejection rate by the condenser, kW

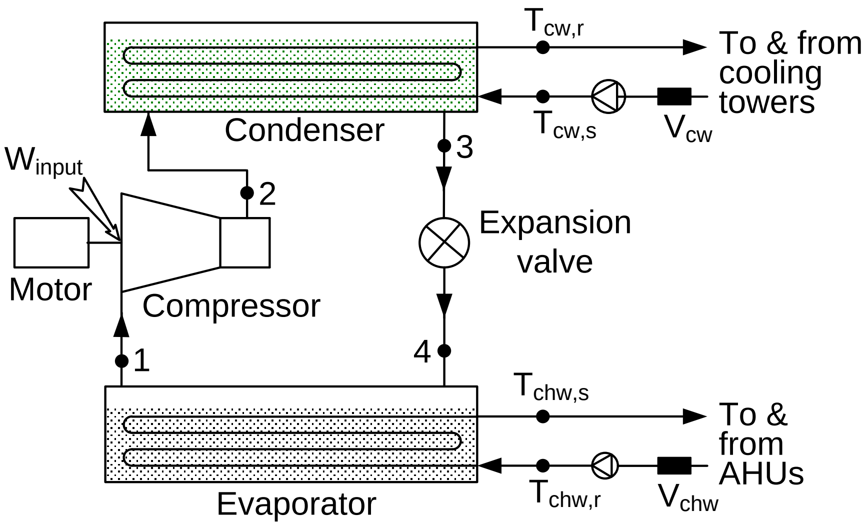


Figure 1.16 Location of sensors for measuring the chiller cooling load and heat rejection rate of condenser

Power input to the compressor W_{input} can be calculated using mass flow rate of refrigerant at point-1 or 2 and specific enthalpy of refrigerant at point-1 and 2 (Figure-1.16) using the following equation:

$$W_{input} = m_{r,1}\left(h_{r,2} - h_{r,1}\right) \tag{1.8}$$

where,

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

$m_{r,1}$ = Mass flow rate of refrigerant at point-1, kg/s

$h_{r,1}$ = Specific enthalpy of refrigerant at point-1, kJ/kg

$h_{r,2}$ = Specific enthalpy of refrigerant at point-2, kJ/kg

In practice, usually input power to the compressor W_{input} is calculated by measuring the input power to the motor of the compressor using power meter. Motor and compressor could be coupled or connected using different configurations which should be analysed carefully to determine the actual input power to the compressor.

Motor and Compressor Coupling: The motor and the compressor of chiller could be connected using the following coupling systems:

Type-1: Open-Type Compressor with Prime Mover and External Gear Drive: In open-type compressors, the motor is mounted externally and cooled by the surrounding air. Photograph of an open-type compressor system is shown in Figure1.17.

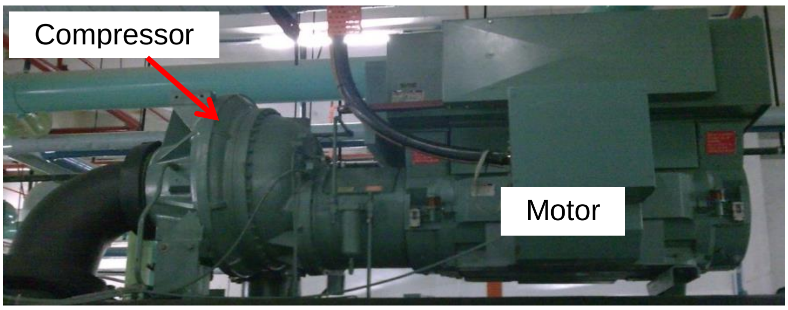


Figure 1.17 Photograph of an open-type compressor system

Schematic diagram of an open-type compressor system indicating energy values is shown in Figure-1.18.

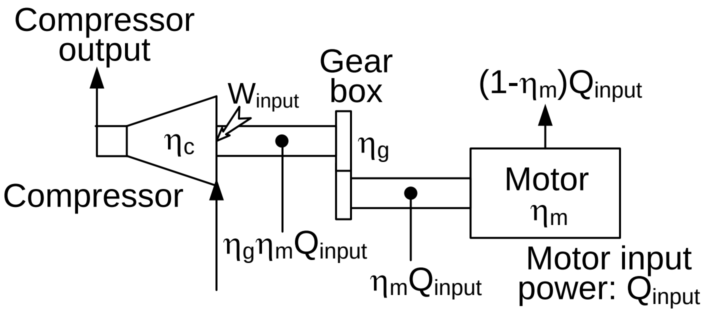


Figure 1.18 Schematic diagram of an open-type compressor system with gear drive

Energy analysis of an open-type compressor is presented below:

Let electric power input (Figure-1.18) to the motor (also known as prime mover) = Q_{input}

If the efficiency of the motor is η_m, output power from the prime mover can be expressed as

$$Q_{prime\,mover} = \eta_m Q_{input} \tag{1.9}$$

If the efficiency of the gear box is η_g, output power of the gear box will be equal to the input power to the compressor:

$$W_{input} = \eta_g \eta_m Q_{input} \tag{1.10}$$

If the friction loss of the gear box is Q_{gear}, the input power to the compressor can also be expressed as:

$$W_{input} = Q_{prime\,mover} - Q_{gear} \tag{1.11}$$

If the efficiency of the compressor is η_c, output power of the compressor can be calculated as:

$$W_{output} = \eta_c \eta_g \eta_m Q_{input} \tag{1.12}$$

Input power to the motor of the compressor is measured using the power meter. For open-type compressor with prime mover and external gear drive, input power to the compressor (W_{input}) expressed in Equation (1.10) or (1.11) should be used in the energy balance Equation (1.7).

Type-2: Open-Type Compressor with Direct Drive: If the motor is directly coupled with the compressor as shown in Figure-1.19, the efficiency of the direct drive η_t is usually considered as 100% (no loss).

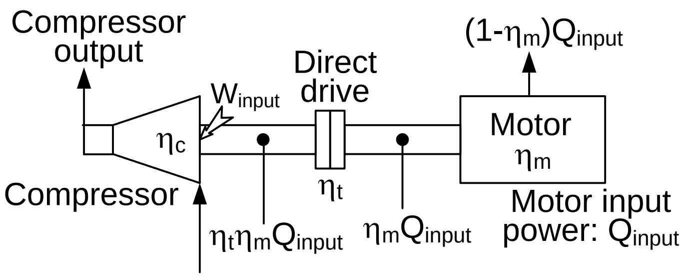


Figure 1.19 Schematic diagram of an open-type compressor system with direct drive

If the efficiency of the direct drive is η_t , output power of the direct drive will be equal to the input power to the compressor:

$$W_{input}=\eta_t\eta_m Q_{input} \tag{1.13}$$

If the friction loss of the direct drive is $Q_{direct\ drive}$, the input power to the compressor can also be expressed as:

$$W_{input} = Q_{prime\ mover} - Q_{direct\ drive} \tag{1.14}$$

If the efficiency of the compressor is η_c, output power of the compressor can be calculated as:

$$W_{output}=\eta_c\eta_t\eta_m Q_{input} \tag{1.15}$$

Input power to the motor of the compressor is measured using the power meter. For open-type compressor with prime mover and external direct drive, input power to the compressor (W_{input}) expressed in Equation (1.13) or (1.14) should be used in the energy balance Equation (1.7).

Type-3: Hermetically Sealed Compressor Systems: Hermetically sealed compressors have their motor enclosed along with compressor inside a dome. The motor of the hermetically sealed compressor is cooled by the incoming suction vapour of refrigerant. They have the advantages of less noise and no refrigerant leakage.

For the hermetically sealed compressors, input energy to the motor of the compressor is finally transported to the condenser by the refrigerant. Therefore:

Compressor input power W_{input} = Electrical power input to the motor of compressor, kW

$$W_{input} = Q_{input} \tag{1.16}$$

For the hermetically sealed compressor systems, input power to the compressor (W_{input}) expressed in Equation (1.16) should be used in the energy balance Equation (1.7). Input power to the motor of the compressor is measured using the power meter.

Chiller Overall Heat Balance Calculation for Measured Data Verification: To verify the accuracy of the measured data, an energy balance is carried out for the chiller using the following equation:

$$Error = \frac{(Q_{ev} + W_{input}) - Q_{cd}}{Q_{cd}} x100 \tag{1.17}$$

Equation (1.17) represents the percentage unbalanced energy which depends on the measurement accuracy of the chilled and condenser water supply and return temperatures, chilled and condenser water flow rates and input power to the motor of the compressor. Based on SS 591 criteria, calculated unbalanced energy should be less than 5% for minimum 80% of the measured data.

Calculation of Chiller Efficiency:

i. Chiller efficiency is defined as the magnitude of electric power consumed by the motor of compressor or chiller power consumption to produce one unit of cooling effect.
ii. Chiller efficiency in kW/RT = Electric power consumed by motor of compressor in kW / Cooling effect produced in RT

$$\eta_{chiller} = \frac{Chiller\ power\ consumption, kW}{Chiller\ cooling\ load, RT} \tag{1.18}$$

iii. Chiller efficiency is also expressed as Coefficient of Performance (COP). Coefficient of performance is defined as:

$$COP = \frac{Chiller\ cooling\ load, kW}{Chiller\ power\ consumption, kW} \tag{1.19}$$

COP is a dimensionless quantity. COP of chiller can also be calculated using the specific enthalpies of refrigerant at different points of the refrigeration cycle illustrated in Figure-1.1. Eqs. (1.1) and (1.8) represent the cooling load and power consumption of the chiller, which can be substituted in Eq. (1.19) to calculate the COP of the chiller.

iv. Input electrical power to the motor of compressor can be measured using power meter.

v. Cooling produced can be calculated using the measured chilled water flow rate, supply and return temperatures and Equations (1.1), (1.2) or (1.3).

Calculation of Water-Cooled Central Chilled Water System Energy Efficiency: Water-cooled central chilled water system energy efficiency is defined as the ratio of the instantaneous total power consumption by the chillers, chilled water pumps (such as primary, secondary, tertiary etc.), condenser water pumps and cooling towers to the instantaneous total cooling load produced by the central chilled water system.

$$Cooling\ load\ of\ central\ chilled\ water\ system, RT = \sum Individual\ chiller\ cooling\ load, RT \quad (1.20)$$

Water-cooled central chilled water system energy efficiency, kW/RT:

$$\eta_{system} = \frac{\sum Individual\ Chiller, CHWPP, CHWSP, CDWP\ \&\ CT\ electric\ power\ consumption, kW}{Cooling\ load\ of\ central\ chilled\ water\ system, RT} \quad (1.21)$$

where

CHWPP: Chilled Water Primary Pump

CHWSP: Chilled Water Secondary Pump

CT: Cooling Tower

CDWP: Condenser Water Pump

i. If chilled water tertiary pump is used, power consumption of tertiary pump is to be included in the calculation of overall energy efficiency of water-cooled central chilled water system.

ii. AHU fan power is not included in the calculation of overall energy efficiency of water-cooled central chilled water system.

1.6 Measurement and Instrumentation Requirements

Based on the guidelines published in Singapore Standard SS591 - Code of practice for long term measurement of central chilled water system energy efficiency, the overall measurement system shall be capable of calculating resultant central chilled water system energy efficiency with the uncertainty within ±5% for in-situ measurements. The following instrumentation and installation are required:

i. The maximum allowable uncertainty of the temperature measurement system shall be ±0.05ºC over the working range of 0ºC to 40ºC. The uncertainty of the temperature measurement shall include the uncertainty of the entire chain of the temperature measurement i.e. the temperature sensor, transmitter, wiring, data acquisition system etc. Temperature sensors shall be insertion type, installed in thermowells for both chilled and condenser water temperatures measurements. The temperature sensors shall be in direct contact with the water. Test-plugs or additional thermowells before and after each temperature sensor shall be installed for verification purposes.

ii. If the pipe diameter allows so and it is possible to shut down the plant, full-bore magnetic in-line flow meters shall be installed. The maximum allowable uncertainty of the flow measurement system shall be ±1%. The uncertainty of the flow measurement shall include the uncertainty of the entire chain of the flow measurement i.e. the flow meter, transmitter, wiring, data acquisition system etc. In the event where clamp-on and insertion-type flow meters are installed due to circumstances where for example, the shut-down of the central chilled water system is not possible, freezing of pipes in not feasible or the pipe diameter does not allow so, the maximum allowable uncertainty of the flow meter shall be ±2%. Flow meters shall be installed away from distortion sources (such as elbows, expanders, valves etc.) which will cause rotational distortion in the velocity flow profile. For magnetic in-line flow meter, the sensors shall be installed with minimum clearance of 5 pipe diameters upstream and 3 pipe diameters downstream.

iii. The power measurement device shall yield true RMS power based on the measure current, voltage and power factor. The maximum allowable uncertainty of the power measurement system shall be ±2%. The uncertainty of the power measurement shall include the uncertainty of the entire chain of the power measurement i.e. the digital power meter, any associated voltage and current transformers, wiring, data acquisition system etc.

iv. The data acquisition system shall be able to record and store values up to at least 3 decimal places and capable of collecting data at all points at a minimum sampling interval of one minute without measurably affecting control performance. The memory space of the data acquisition system shall be sized for at least 36 months of storage capacity.

v. For weather station, temperature measurement shall have an uncertainty within ±0.2ºC over the working range of 15ºC to 40ºC and relative humidity measurement shall have an uncertainty within ±2% over the working rage of 50% to 90%.

Calculation of Overall Uncertainty of Measurement: Overall uncertainty of measurement of the central chilled water system energy efficiency can be calculated as follows:

Maximum allowable uncertainty of temperature measurement system is ±0.05ºC over the working range of 0ºC to 40ºC.

Maximum allowable uncertainty of the temperature measurement system used to measure a temperature difference of 5.5ºC between the supply and return of the chilled or condenser water is $100\text{x}\sqrt{\left(0.05^2+0.05^2\right)}/5.5=\pm1.3\%$

Maximum allowable uncertainty for the chilled water or condenser water flow measurement system is ±1%[1]

Maximum allowable uncertainty for the power measurement system is ±2%

Therefore, the overall root mean square uncertainty of measurement of the central chilled water system energy efficiency:

$$Error_{rms}=\sqrt{\left(\sum\left(U_n\right)^2\right)} \tag{1.22}$$

$$Error_{rms}=\sqrt{\left(1.3^2+1^2+2^2\right)}=\pm2.6\% \tag{1.23}$$

where

U_n = uncertainty of individual variable *n*, %

n = measured variables such as mass flow rate, electrical input power or temperature difference

rms = root mean square

Therefore, the overall uncertainty for the calculated central chilled water system energy efficiency is 2.6% which falls within ±5% of the true value.

[1] In the event where clamp-on and insertion-type flow meters are installed due to circumstances where for example, the shut-down of the central chilled water system is not possible, freezing of pipes is not feasible or the pipe diameter does not allow so, the maximum allowable uncertainty of the flow meter shall be ±2%.

1.7 Chiller Performance Based on IPLV, NPLV and Singapore Standards

Integrated Part Load Value (IPLV): As chillers do not always operate at design load and constant operating conditions, integrated part load value (IPLV) predicts chiller efficiency at Air-Conditioning, Heating and Refrigeration Institute (AHRI) standard rating point using the following equation:

$$IPLV = \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}} \tag{1.24}$$

where

A = kW/RT at 100% load (*ECW* 29.4 ^oC / *EDB* 35 ^oC)

B = kW/RT at 75% load (*ECW* 23.9 ^oC / *EDB* 26.7 ^oC)

C = kW/RT at 50% load (*ECW* 18.3 ^oC / *EDB* 18.3 ^oC)

D = kW/RT at 25% load (*ECW* 18.3 ^oC / *EDB* 12.8 ^oC)

ECW = Entering condenser water temperature for the water-cooled chiller

EDB = Entering dry bulb temperature of the ambient air to the condenser coil of air-cooled chiller

For a tropical country like Singapore, it is not possible to cool down the condenser water using the cooling tower to the temperature of 23.9 ^oC or below. Similarly, the temperature of the ambient air does not drop to 26.7 ^oC or below. Therefore, calculated IPLV for the chiller will not represent the actual performance of the chillers in Singapore climate condition.

Non-Standard Part Load Value (NPLV): Non-standard part load value (NPLV) for the chiller efficiency is also calculated using the same Equation (1.24). However, NPLV predicts chiller efficiency at specific rating conditions other than Air-Conditioning, Heating and Refrigeration Institute (AHRI) standard rating points. For example, entering condenser water (ECW) and entering dry bulb (EDB) temperatures can be defined based on the actual variation of the ambient condition of a specific country.

Singapore Standards SS530: Based on Singapore Standard SS530: 2014, minimum efficiency of few selected vapour compression and vapour absorption chillers are presented in Tables 1.1 and 1.2 respectively.

Table 1.1 Minimum efficiency of vapour compression chillers based on Singapore Standard SS530: 2014

Equipment type	Size category	Minimum efficiency, COP (W/W)	
		Path A	Path B
Water-cooled, electrically operated, positive displacement (rotary screw and scroll)	≥ 1055 kW and < 2100 kW	5.771 FL	5.633 FL
		6.770 IPLV	8.586 IPLV
Water-cooled, electrically operated, centrifugal	≥ 1055 kW and < 1407 kW	6.286 FL	5.917 FL
		6.770 IPLV	9.027 IPLV

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

Note 1: One can elect to comply with either Path A or Path B but both FL and IPLV of the selected path shall be complied with.

Note 2: IPLV = 0.01A + 0.42B + 0.45C + 0.12D

Where A = COP at 100% (30ºC) per AHRI STD 551/591 (SI) for water-cooled chillers
B = COP at 75% (24.5ºC)
C = COP at 50% (19ºC)
D = COP at 25% (19ºC)
Where A = COP at 100% (35ºC) per AHRI STD 551/591 (SI) for air-cooled chillers
B = COP at 75% (27ºC)
C = COP at 50% (19ºC)
D = COP at 25% (13ºC)

Table 1.2 Minimum efficiency of vapour absorption chillers based on Singapore Standard SS530: 2014

Equipment type	Size category	Minimum efficiency	
		COP	kW/RT
Water-cooled absorption single effect	All capacities	0.700 FL	5.024
Absorption double effect, indirect-fired	All capacities	1.000 FL	3.517
		1.050 IPLV	3.349

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

As Singapore is located in the tropics and the weather condition remains relatively constant throughout the year, IPLV for the chiller does not represent the actual performance of the chillers in Singapore climate condition. The following operating variables (Table 1.3) of chillers based on Air-Conditioning, Heating and Refrigeration Institute (AHRI) are commonly used to compare the performance of different vapour compression chillers.

Table 1.3 Operating variables generally used to evaluate the performance of chillers

Operating Variables	Values
Chilled water supply temperature	6.7 ^{o}C (44 ^{o}F)
Condenser water supply temperature	29.4 ^{o}C (85 ^{o}F)
Chilled water ΔT at maximum load	5.6 ^{o}C (10 ^{o}F)
Condenser water ΔT at maximum load	5.6 ^{o}C (10 ^{o}F)
Chilled water flow rate	2.4 usgpm/RT (0.15 l/s per RT)
Condenser water flow rate	3 usgpm/RT (0.19 l/s per RT)

The minimum efficiencies of the chilled water and condenser water pumps are presented in Singapore Standard SS553 and cooling tower performance requirements are presented in Singapore Standard SS530: 2014. These will be discussed in Chapters 2 and 5 respectively. Water-cooled central chilled water system energy efficiency is defined as the ratio of the total power consumption by the chillers, chilled water pumps (such as primary, secondary, tertiary etc.), condenser water pumps and cooling towers to the total cooling load of the chilled water system (refer to Eq. (1.21)). Based on Singapore Standards, breakdown of the minimum efficiency of the chilled water systems of capacity ≥1055 kW and < 1407 kW (≥ 300 RT and < 400 RT) are summarised in Table 1.4.

Table 1.4 Breakdown of minimum efficiency of water cooled chilled water system of capacity ≥ 1055 kW and < 1407 kW (≥ 300 RT and < 400 RT)

Description	Singapore Standards	kW/RT
Water cooled centrifugal chiller (≥ 300 RT and < 400 RT)	SS530: 2014	0.559
Chilled water pumps	SS553	0.053
Condenser water pumps	SS553	0.057
Cooling towers	SS530: 2014	0.054

Water-cooled central chilled water system energy efficiency	0.723

Due to the advancement in chiller technology, new and existing water-cooled central chilled water systems that have undergone major retrofit are conveniently achieving energy efficiency of 0.65 kW/RT (COP 5.4). Several projects have achieved energy efficiency of 0.6 kW/RT (COP 5.8) or better. Steps that significantly contribute to the improvement of central chilled water systems energy efficiency are:

i. Selection of high efficiency chillers of suitable capacities to match the cooling load profile of the facility in an energy efficient manner
ii. Selection of high efficiency pumps and cooling towers and modulation of their capacities based on instantaneous cooling load
iii. Selection of fittings and appropriate design of chilled water plant piping layout and
iv. Implementation of energy smart control strategies

The contributions of the above steps are elaborated upon in the following sections.

1.8 Optimisation of Chilled Water System

Operating efficiency of chiller depends on a number of factors such as:

i. Type of chiller
ii. Rated efficiency of chiller
iii. Part-load performance of chiller
iv. Matching of chiller capacity to the cooling load of the building
v. Chilled water and condenser water set point temperatures
vi. Subcooling of condenser
vii. Condition of condenser tubes

The influences of the above factors on the performance of the chilled water systems and the corresponding energy saving opportunities are presented in the following sections.

1.8.1 Air-Cooled to Water-Cooled Chiller

Based on the cooling systems adopted for the condenser coil, chillers can be divided into:

i. Air-Cooled Chiller (ACC): Condenser coil is cooled by blowing ambient air.

ii. Water-cooled Chiller (WCC): Condenser coil is cooled by condenser water which is cooled using the cooling tower.

The relative advantages and disadvantages of the ACC and WCC are summarised below:

i. The convective heat transfer coefficient for water is higher than air. As a result, the condensing temperature and pressure of the refrigerant in the condenser is lower for WCC, which helps to reduce the compressor lift and the compressor power consumption.
ii. Typical efficiency of (a) ACC ranges from 1.0 to 1.5 kW/RT and (b) WCC ranges from 0.5 to 0.6 kW/RT.
iii. First cost of WCC systems is higher than ACC systems as WCC requires plant room, condenser water pumps, cooling towers and condenser water piping systems. Extra initial cost for the WCC can be paid back in few years due to the higher energy efficiency of WCC.

iv. ACC have many specific applications such as ACC can be used to support small area (Example: Data center area, small occupied area) when WCC is turned off.

Example 1.1

Air-conditioning of a building is provided using a 350 RT Air-Cooled Chiller (ACC). The building cooling load from 8am to 10pm and the energy consumption of the ACC are shown in Table 1.5.

Table 1.5 Energy consumption of the existing ACC

Operating hours	Number of hours	Cooling load, RT	Existing ACC efficiency, kW/RT	Present energy consumption, kWh
	A	B	C	A x B x C
0800 to 0900	1	250	1.30	325
0900 t0 1000	1	250	1.30	325
1000 to 1100	1	275	1.28	352
1100 to 1200	1	275	1.28	352
1200 to 1300	1	300	1.25	375
1300 to 1400	1	350	1.20	420
1400 to 1500	1	300	1.25	375
1500 to 1600	1	300	1.25	375
1600 to 1700	1	300	1.25	375

1700 to 1800	1	250	1.30	325
1800 to 1900	1	250	1.30	325
1900 to 2000	1	250	1.30	325
2000 to 2100	1	200	1.40	280
2100 to 2200	1	200	1.40	280
			Total	**4809**

If the ACC is replaced with a WCC having the part load efficiency shown in Table 1.6, compute the achievable energy and cost savings for this replacement. Make the necessary assumptions.

Table 1.6 Part load efficiency of the proposed WCC

Load (RT)	Efficiency (kW/RT)
200	0.62
250	0.60
275	0.59
300	0.58
350	0.57

Solution

Using the part load efficiency of the proposed WCC, the estimated energy consumption of the WCC is presented in Table 1.7.

Table 1.7 Estimated energy consumption of the proposed WCC

Operating hours	Number of hours	Cooling load, RT	Existing ACC efficiency, kW/RT	Present energy consumption, kWh	Proposed WCC efficiency, kW/RT	Proposed energy consumption, kWh
	A	B	C	A x B x C	D	A x B x D
0800 to 0900	1	250	1.30	325	0.60	150
0900 t0 1000	1	250	1.30	325	0.60	150
1000 to 1100	1	275	1.28	352	0.59	162.25
1100 to 1200	1	275	1.28	352	0.59	162.25
1200 to 1300	1	300	1.25	375	0.58	174
1300 to 1400	1	**350**	**1.20**	420	**0.57**	199.50
1400 to 1500	1	300	1.25	375	0.58	174
1500 to 1600	1	300	1.25	375	0.58	174
1600 to 1700	1	300	1.25	375	0.58	174

1700 to 1800	1	250	1.30	325	0.60	150
1800 to 1900	1	250	1.30	325	0.60	150
1900 to 2000	1	250	1.30	325	0.60	150
2000 to 2100	1	200	1.40	280	0.62	124
2100 to 2200	1	200	1.40	280	0.62	124
			Total	**4809**		**2218**

Therefore, achievable energy savings = 4809 – 2218 = 2591 kWh/day

Extra operating cost involved due to the use of WCC are:

a) Operating condenser water pumps
b) Operating cooling tower fans
c) Use of make-up water for cooling tower

Following assumptions can be made:

i. Total power consumption of the condenser water pumps and cooling towers usually ranges from 0.08 to 0.15 kW/RT.
ii. Assuming that the total power consumption for the condenser water pumps and the cooling towers = 0.12 kW/RT
iii. Make-up water consumption of the cooling tower is about 1 to 1.5% of circulating flow rate, or approximately 1 x 10^{-2} m^3/hr per RT.
iv. Assuming that the make-up water consumption of cooling towers = 1 x 10^{-2} m^3/hr per RT

Based on the above assumptions, estimated electrical energy and water consumption of the condenser water pumps and cooling towers are presented in Table 1.8.

Table 1.8 Calculated electrical energy and water consumption for the condenser water pump and cooling tower

Operating hours	Number of hours	Cooling load, RT	Energy consumption for condenser water pump and cooling tower, kWh	Make-up water consumption for cooling tower, m^3
	A	B	A x B x 0.12 kW/RT	B x 1 x 10^{-2}
0800 to 0900	1	250	30	2.50
0900 t0 1000	1	250	30	2.50
1000 to 1100	1	275	33	2.75
1100 to 1200	1	275	33	2.75
1200 to 1300	1	300	36	3.00
1300 to 1400	1	**350**	**42**	3.50
1400 to 1500	1	300	36	3.00
1500 to 1600	1	300	36	3.00
1600 to 1700	1	300	36	3.00
1700 to 1800	1	250	30	2.50

1800 to 1900	1	250	30	2.50
1900 to 2000	1	250	30	2.50
2000 to 2100	1	200	24	2.00
2100 to 2200	1	200	24	2.00
		Total	**450**	**37.50**

Therefore, extra electrical energy consumption by condenser water pumps and cooling towers = 450 kWh/day

Make-up water consumption of cooling towers = 37.5 m^3/day

Therefore, the net electrical energy savings = 2591 – 450 = 2141 kWh/day

Peak electrical demand savings:

= Peak cooling load (RT) x difference in ACC and WCC efficiencies (kW/RT) – Peak cooling load (RT) x Combined efficiency of condenser water pumps and cooling towers, kW/RT

= 350 x (1.2 – 0.57) - 350 x 0.12

= 178.5 kW

To calculate the cost savings, the following utility tariffs can be assumed:

Electricity tariff = $0.20 / kWh

Electricity demand charge = $8 / kW per month

Water usage = $1 / m^3

Net annual cost savings (based on operating 365 days a year)

= [(2141 kWh/day x Electricity tariff) - (37.5 m^3/day x water tariff)] x days/year + [178.5 kW x monthly electricity demand charges x 12 months/year]

= [(2141 x 0.20) – (37.5 x 1)] x 365 + [178.5 x 8 x 12]

= $159,742 per year

1.8.2 Chiller Efficiency and Life Cycle Costing

i. Rated efficiency (at 100% loading) of chillers ranges from 0.5 to 0.65 kW/RT.
ii. Components and relative magnitudes of chiller life cycle costs are shown in Figure-1.20.

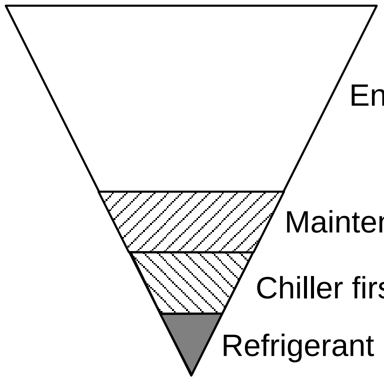


Figure 1.20 Components and relative magnitudes of chiller life cycle costs

iii. Energy cost exceeds the first cost by 10 to 15 times over the operating life as shown in Figure 1.20.

iv. High efficiency chillers are expensive but incur low energy cost. As a result, the additional upfront cost for these high efficiency chillers can be paid back in about 3 to 5 years.

Example 1.2

Consider three 600RT capacity chiller options to support the constant process cooling load of 600 RT. The efficiencies and first costs for the three 600 RT chiller options are presented in Table 1.9.

Table 1.9 First costs and rated efficiencies of three 600 RT chiller options

Chiller option	Chiller first cost, $	Chiller efficiency, kW/RT
1	420,000	0.50
2	408,000	0.55
3	390,000	0.65

Compute the life cycle cost for a 10-year period based on the following assumptions:

Chiller is to operate 10 hrs/day and 300 days a year

Electricity tariff = $0.20 / kWh

Electricity cost escalation is 2% per year

Note:

Chiller energy consumption, kWh = Hours of operation (Hrs) x Cooling Load (RT) x Chiller efficiency (kW/RT)

Energy cost = Chiller energy consumption (kWh) x Electricity tariff ($/kWh)

Energy cost = Hours of operation (Hrs) x Cooling Load (RT) x Chiller efficiency (kW/RT) x Electricity tariff ($/kWh)

Solution

For the chiller of efficiency 0.5 kW/RT, the electrical energy cost for year-1
= 10 hours/day x 300 days/year x 600RT x 0.5 kW/RT x $0.20/kWh
= $180,000

As the escalation of electricity tariff is 2% per year, the energy cost for year-2 will be
= $180,000 x (1+0.02) = $183,600

Similarly, energy cost for year-3 and the following years can be calculated. Life cycle cost comparison for the three chiller options is given in Table 1.10. All costs presented in Table 1.10 are energy costs except year-0 capital cost or chiller initial cost. Maintenance costs for the three chiller options are assumed to be the same and not included in the comparison of the chiller options.

Table 1.10 Life cycle cost for the chiller options

	Option-1, 0.5 kW/RT	Option-2, 0.55 kW/RT	Option-3, 0.65 kW/RT
	$	$	$
Year-0	**420,000**	**408,000**	**390,000**
Year-1	180,000	198,000	234,000
Year-2	183,600	201,960	238,680
Year-3	187,272	205,999	243,454
Year-4	191,017	210,119	248,323
Year-5	194,838	214,322	253,289
Year-6	198,735	218,608	258,355
Year-7	202,709	222,980	263,522
Year-8	206,763	227,440	268,792
Year-9	210,899	231,989	274,168
Year-10	215,117	236,628	279,652
Total	**2,390,950**	**2,576,045**	**2,952,235**

Notes:

i. Electricity cost incurred to operate option-1 is the lowest from year-1 onwards.
ii. Capital cost of option-1 is the highest. However, life cycle cost (which consists of capital and operating costs) for implementing option-1 is the lowest.
iii. Hence, option-1 should be selected.

Example 1.3

Consider two 600RT capacity chiller options to support the variable cooling load of a commercial building as shown in Table 1.11. Part load efficiencies for the chiller options are presented in Table 1.12.

Table 1.11 Variation of building cooling load with time

Operating hours	Number of hours	Cooling load, RT
0800 to 0900	1	360
0900 t0 1000	1	450
1000 to 1100	1	480
1100 to 1200	1	540
1200 to 1300	1	600
1300 to 1400	1	600
1400 to 1500	1	540
1500 to 1600	1	480
1600 to 1700	1	420
1700 to 1800	1	360

Table 1.12 Part load efficiency for the chiller options

Chiller load, RT	Loading, %	Chiller-A efficiency, kW/RT	Chiller-B efficiency, kW/RT
600	100	0.50	0.55
540	90	0.52	0.57
480	80	0.55	0.60
420	70	0.59	0.64
360	60	0.60	0.66
300	50	0.67	0.73

The capital costs of chiller-A and B are $420,000 and $408,000 respectively. Compute the life cycle cost for a 10-year period based on the following assumptions:

Chiller is to operate 10 hrs/day and 250 days a year

Electricity tariff = $0.20 / kWh

Electricity cost escalation is 2% per year

Solution

Daily energy consumption of chiller-A, with rated efficiency of 0.5 kW/RT, can be calculated as shown in Table 1.13.

Table 1.13 Daily energy consumption for chiller-A of rated efficiency 0.5 kW/RT

Operating hours	Number of hours	Cooling load, RT	Operating chiller capacity, RT	Loading, %	Chiller efficiency, kW/RT	Energy consumption, kWh
	A	B	C	D =100 (B / C)	E	A x B x E
0800 to 0900	1	360	600	60	0.6	216.0
0900 t0 1000	1	450	600	75	0.57	256.5
1000 to 1100	1	480	600	80	0.55	264.0
1100 to 1200	1	540	600	90	0.52	280.8
1200 to 1300	1	600	600	100	0.5	300.0
1300 to 1400	1	600	600	100	0.50	300.0
1400 to 1500	1	540	600	90	0.52	280.8
1500 to 1600	1	480	600	80	0.55	264.0
1600 to 1700	1	420	600	70	0.59	247.8
1700 to 1800	1	360	600	60	0.60	216.0
Total (kWh/day)						2625.9

Annual energy consumption = 2625.9 x 250 = 656,475 kWh/year

Annual electricity cost = 656,475 x 0.20 = $ 131,295 /year

Similarly, daily energy consumption for chiller-B of rated efficiency 0.55 kW/RT can be calculated as shown in Table 1.14.

Table 1.14 Daily energy consumption for chiller-B of rated efficiency 0.55 kW/RT

Operating hours	Number of hours	Cooling load, RT	Operating chiller capacity, RT	Loading, %	Chiller efficiency, kW/RT	Energy consumption, kWh
	A	B	C	D =100 (B / C)	E	A x B x E
0800 to 0900	1	360	600	60	0.66	237.6
0900 t0 1000	1	450	600	75	0.62	279.0
1000 to 1100	1	480	600	80	0.60	288.0
1100 to 1200	1	540	600	90	0.57	307.8
1200 to 1300	1	600	600	100	0.55	330.0
1300 to 1400	1	600	600	100	0.55	330.0
1400 to 1500	1	540	600	90	0.57	307.8

1500 to 1600	1	480	600	80	0.60	288.0
1600 to 1700	1	420	600	70	0.64	268.8
1700 to 1800	1	360	600	60	0.66	237.6
					Total (kWh/day)	2874.6

Annual energy consumption = 2874.6 x 250 = 718,650 kWh/year

Annual electricity cost = 718,650 x 0.20 = $ 143,730 /year

Assuming the escalation of the electricity tariff of 2% per year, the life cycle cost for the two chiller options are summarised in Table 1.15.

Table 1.15 Summary of life cycle cost for the two chiller options

	Option-1, 0.5 kW/RT	Option-2, 0.55 kW/RT
	$	$
Year-0	**420,000**	**408,000**
Year-1	131,295	143,730
Year-2	133,921	146,605
Year-3	136,599	149,537
Year-4	139,331	152,527
Year-5	142,118	155,578
Year-6	144,960	158,690
Year-7	147,859	161,863
Year-8	150,817	165,101
Year-9	153,833	168,403
Year-10	156,910	171,771
Total	**1,857,644**	**1,981,803**

Notes:

i. First row of Table-1.15 (year-0) represents the capital cost of chillers

ii. Second (year-1) to eleventh (year-10) rows of Table-1.15 show the operating electricity cost of chillers. It is assumed that the electricity tariff will be escalated by 2% every year.

iii. Electrical energy cost incurred to implement option-1 is the lowest from year-1 onwards.

iv. Capital cost of option-1 is higher. However, life cycle cost (which consists of capital and operating costs) for implementing option-1 is lower.

v. Hence, option-1 should be selected.

1.8.3 Chiller Sizing and Configuration

Operating efficiency of chillers depends on the chiller loading. An example of the variation of chiller efficiency with loading for different brands of chillers using different types of compressor and refrigerant is shown in Figure-1.21

Important points for the sizing and configuration of chillers:

i. To operate a chilled water system in an energy efficient manner, building cooling load profile needs to be matched with the energy efficient loadings of the chillers.

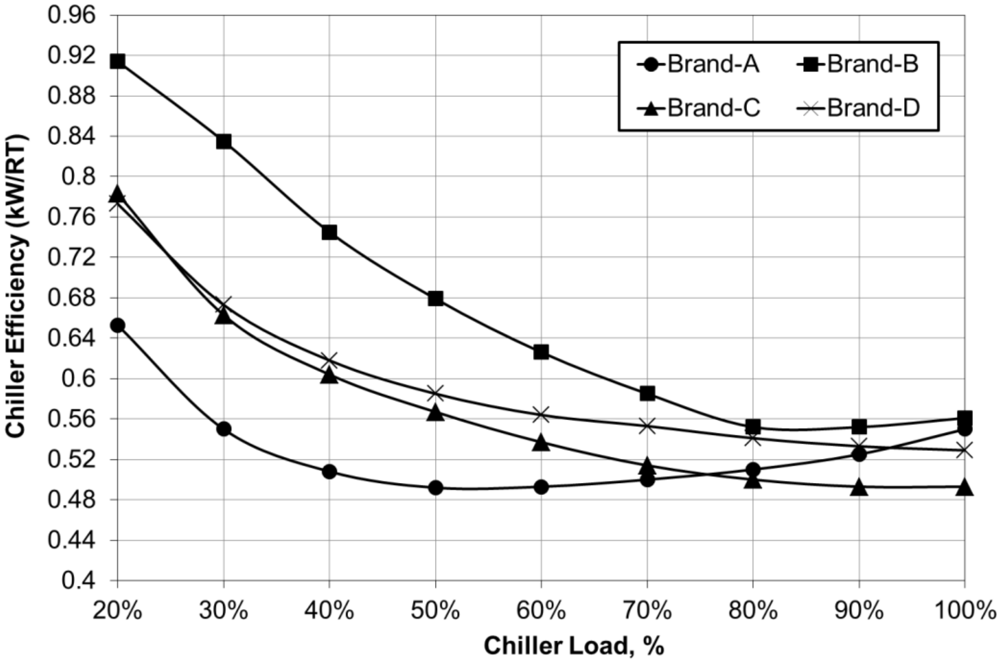


Figure 1.21 Variation of chiller efficiency with loading for chillers of different brands using different types of compressor and refrigerant

ii. For existing industrial facilities and buildings, cooling load profile of the facility or building should be measured and new chillers should be selected accordingly during retrofitting of the chilled water system.

iii. For new industrial facilities and buildings, cooling load profile of the facility and building should be determined using energy simulation software and chillers should be selected according to the simulated cooling load. The capacity of the selected chillers is often oversized due to the assumptions, such as type of space usage, number of occupants, temperature, relative humidity and ventilation requirements of spaces, made in cooling load simulation and the selection of chiller based on the expected peak cooling load.

iv. In many cases, this over-sizing of the chillers is repeated when the chillers are replaced many years later.

Example 1.4

Cooling load profile of a building and the frequencies of cooling load occurrences are shown in Figures-1.22 and 1.23 respectively.

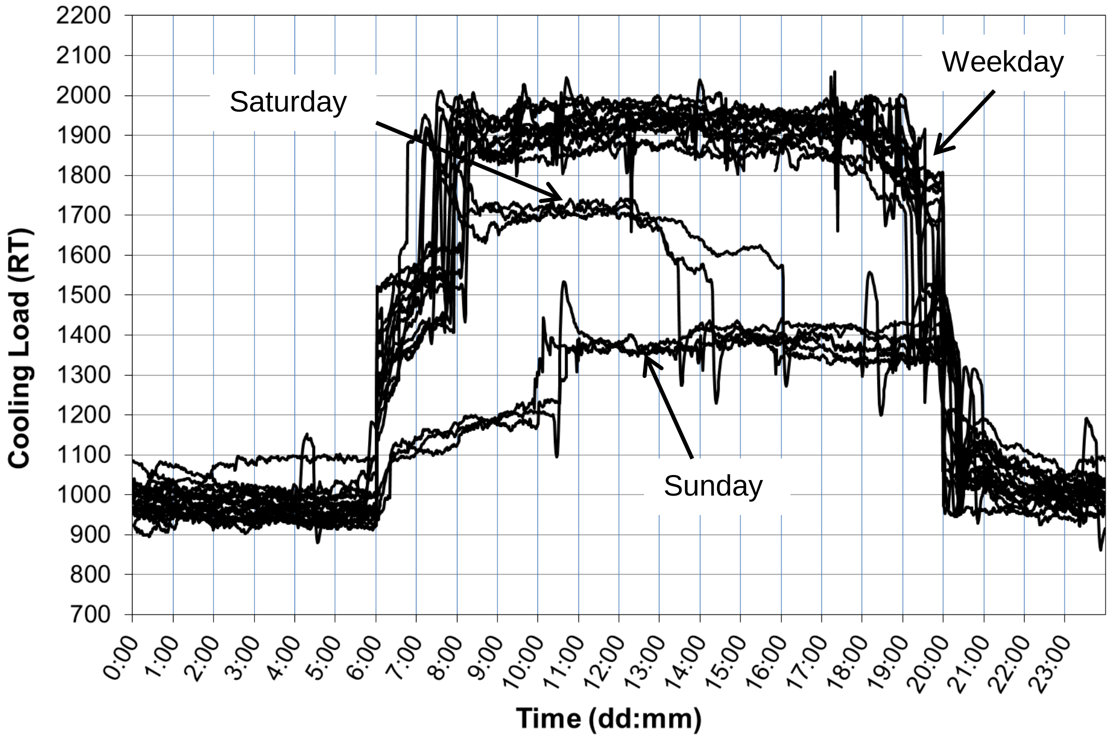


Figure 1.22 Cooling load profile of a building

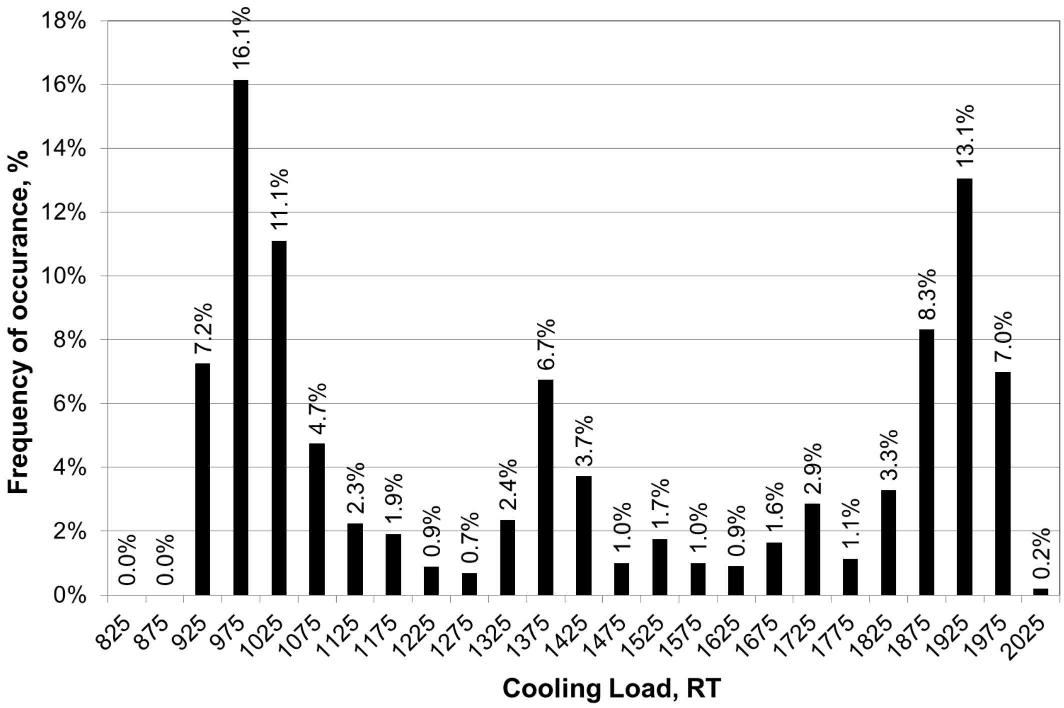


Figure 1.23 Frequencies of cooling load occurrences

To support the building cooling load, the following three options of chiller combinations are considered.

Option-1: 1 no. of 2050 RT chiller
Option-2: 2 nos. of 1100 RT chillers
Option-3: 1 no. of 500 RT and 2 nos. of 800 RT chillers

Which chiller combination should be selected? Justify your selection. Part load efficiencies for the chillers are shown in Table 1.16.

Table 1.16 Part load efficiencies of chillers

Chiller loading, %	2050 RT	1100 RT	800 RT	500 RT
100	0.528	0.525	0.532	0.56
90	0.521	0.524	0.531	0.54
80	0.522	0.524	0.533	0.52
70	0.529	0.523	0.534	0.51
60	0.542	0.545	0.538	0.5
50	0.566	0.558	0.563	0.51
40	0.600	0.586	0.590	0.52
30	0.653	0.639	0.641	0.57
20	0.745	0.755	0.754	0.68

Solution:

Daily electrical energy consumption for Option-1 (1 no. of 2050 RT chiller) can be calculated as shown in Table 1.17.

Table 1.17 Daily electrical energy consumption for Option-1 (1 no. of 2050 RT chiller)

Av. RT	Occurrence	Operating hours/day	Operating chiller	Part Load	Chiller Eff, kW/RT	Energy consumption, kWh
A	B	C = B x 24	D	E = A / D	F	A x C x F
925	0.072	1.74	1x2050	45.1	0.583	938.1
975	0.161	3.87	1x2050	47.6	0.574	2167.7
1025	0.111	2.66	1x2050	50.0	0.566	1545.0
1075	0.047	1.14	1x2050	52.4	0.560	684.8
1125	0.023	0.54	1x2050	54.9	0.554	336.7
1175	0.019	0.46	1x2050	57.3	0.548	294.7
1225	0.009	0.22	1x2050	59.8	0.542	143.2
1275	0.007	0.17	1x2050	62.2	0.539	114.6
1325	0.024	0.57	1x2050	64.6	0.536	402.3
1375	0.067	1.62	1x2050	67.1	0.533	1187.1
1425	0.037	0.89	1x2050	69.5	0.530	675.0
1475	0.010	0.24	1x2050	72.0	0.528	187.1
1525	0.017	0.42	1x2050	74.4	0.526	335.9
1575	0.010	0.24	1x2050	76.8	0.524	196.7
1625	0.009	0.22	1x2050	79.3	0.522	185.3
1675	0.016	0.39	1x2050	81.7	0.522	343.2
1725	0.029	0.69	1x2050	84.1	0.522	617.9
1775	0.011	0.27	1x2050	86.6	0.521	249.0
1825	0.033	0.79	1x2050	89.0	0.521	749.0
1875	0.083	2.00	1x2050	91.5	0.522	1953.6
1925	0.131	3.13	1x2050	93.9	0.524	3161.8
1975	0.070	1.68	1x2050	96.3	0.525	1739.7
2025	0.003	0.07	1x2050	98.8	0.527	70.6
Total	**1.000**	**24.00**				**18,279**

Average daily energy consumption for option-1 = 18,279 kWh / day

Assuming that the chilled water system is operated for 365 days a year, the annual energy consumption = 18,279 x 365 = 6,671,835 kWh / year

Similarly, daily electrical energy consumption for option-2 (2 nos. of 1100 RT chillers) can be calculated as shown in Table 1.18.

Table 1.18 Daily electrical energy consumption for Option-2 (2 nos. of 1100 RT chillers)

Av RT	Occurrence	Operating hours/day	Operating chiller	Part Load	Chiller Eff, kW/RT	Energy consumption, kWh
A	B	C = B x 24	D	E = A / D	F	A x C x F
925	0.072	1.74	1x1100	84.1	0.524	843.1
975	0.161	3.87	1x1100	88.6	0.524	1978.8
1025	0.111	2.66	1x1100	93.2	0.524	1430.4
1075	0.047	1.14	1x1100	97.7	0.525	642.0
1125	0.023	0.54	2x1100	51.1	0.557	338.5
1175	0.019	0.46	2x1100	53.4	0.554	298.0
1225	0.009	0.22	2x1100	55.7	0.551	145.6
1275	0.007	0.17	2x1100	58.0	0.548	116.5
1325	0.024	0.57	2x1100	60.2	0.545	409.1
1375	0.067	1.62	2x1100	62.5	0.540	1202.7
1425	0.037	0.89	2x1100	64.8	0.534	680.1
1475	0.010	0.24	2x1100	67.0	0.530	187.8
1525	0.017	0.42	2x1100	69.3	0.525	335.3
1575	0.010	0.24	2x1100	71.6	0.523	196.4
1625	0.009	0.22	2x1100	73.9	0.523	185.7
1675	0.016	0.39	2x1100	76.1	0.524	344.5
1725	0.029	0.69	2x1100	78.4	0.524	620.2
1775	0.011	0.27	2x1100	80.7	0.524	250.4
1825	0.033	0.79	2x1100	83.0	0.524	753.3
1875	0.083	2.00	2x1100	85.2	0.524	1961.1
1925	0.131	3.13	2x1100	87.5	0.524	3161.8
1975	0.070	1.68	2x1100	89.8	0.524	1736.4
2025	0.003	0.07	2x1100	92.0	0.524	70.2
Total	**1.000**	**24.00**				**17,888**

Average daily energy consumption for option-2 = 17,888 kWh / day

Annual electrical energy consumption = 17,888 x 365 = 6,529,120 kWh / year

Similarly, daily electrical energy consumption for option-3 (1 no. of 500 RT and 2 nos. of 800 RT chillers) can be calculated as shown in Table 1.19.

Table 1.19 Daily electrical energy consumption for option-3 (1 no. of 500 RT and 2 nos. of 800 RT chillers)

Av RT	Occurrence	Operating hours/day	Operating chiller	Part Load	Chiller Eff, kW/RT	Energy consumption, kWh
A	B	C = B x 24	D	E = A / D	F	A x C x F
925	0.072	1.74	1x800+1x500	71.2	0.525	844.7
975	0.161	3.87	1x800+1x500	75.0	0.526	1986.4
1025	0.111	2.66	1x800+1x500	78.8	0.528	1441.3
1075	0.047	1.14	1x800+1x500	82.7	0.530	648.1
1125	0.023	0.54	1x800+1x500	86.5	0.532	323.3
1175	0.019	0.46	1x800+1x500	90.4	0.535	287.8

1225	0.009	0.22	1x800+1x500	94.2	0.538	142.2
1275	0.007	0.17	1x800+1x500	98.1	0.541	115.0
1325	0.024	0.57	2x800	82.8	0.532	399.3
1375	0.067	1.62	2x800	85.9	0.532	1184.9
1425	0.037	0.89	2x800	89.1	0.531	676.3
1475	0.010	0.24	2x800	92.2	0.531	188.1
1525	0.017	0.42	2x800	95.3	0.532	339.8
1575	0.010	0.24	2x800	98.4	0.532	199.8
1625	0.009	0.22	2x800+1x500	77.4	0.529	187.8
1675	0.016	0.39	2x800+1x500	79.8	0.530	348.4
1725	0.029	0.69	2x800+1x500	82.1	0.531	628.5
1775	0.011	0.27	2x800+1x500	84.5	0.531	253.7
1825	0.033	0.79	2x800+1x500	86.9	0.532	764.8
1875	0.083	2.00	2x800+1x500	89.3	0.533	1994.7
1925	0.131	3.13	2x800+1x500	91.7	0.534	3222.1
1975	0.070	1.68	2x800+1x500	94.0	0.535	1772.8
2025	0.003	0.07	2x800+1x500	96.4	0.537	72.0
Total	**1.000**	**24.00**				**18,022**

Average daily energy consumption for option-3 = 18,022 kWh / day

Annual electrical energy consumption = 18,022 x 365 = 6,578,030 kWh / year

Comparison of energy consumption for the three different options of chiller combinations is summarised in Table 1.20.

Table 1.20 Comparison of energy consumption for three different chiller combinations

Chiller combinations	Daily average energy consumption, kWh/day	Annual energy consumption, kWh/year
1 x 2050 RT	18,279	6,671,835
2 x 1100 RT	**17,888**	**6,529,120**
2 x 800 + 1 x 500 RT	18,022	6,578,030

Notes:

i. Among the three options of chiller combinations, energy consumption for option-2 (2 nos. of 1100 RT chillers) is the lowest. Therefore, option-2 could be selected for implementation.

ii. Table 1.19 shows that the loading of the chillers for option-3 (1 no. of 500 RT and 2 nos. of 800 RT chillers) is slightly better than option-2 (2 nos. of 1100 RT chillers) (Table 1.18). However, energy consumption for option-2 is slightly lower due to the use of bigger chillers of capacity 1100 RT. The energy efficiency of bigger chillers is usually better.

iii. One standby chiller is usually installed in the plant room to support the cooling load during the period of maintenance or breakdown of the chiller. For option-2, the capacity of the standby chiller should be 1100 RT. Therefore, total capacity of the installed chillers will be 3x1100 = 3300 RT. However, for option-3, the capacity of the standby chiller should be 800 RT. Therefore, total capacity of the installed chillers will be 3x800+1x500 = 2900 RT. As a result, installation cost for option-3 will be lower than option-2. The difference of the annual energy savings for option-2 and 3 is only 6,578,030 - 6,529,120 = 48,910 kWh. Therefore, option-2 could also be considered for implementation. However, option-3 will require more plant room space due to the installation of 4 nos. of chillers in comparison to option-2 which requires installing 3 nos. of chillers.

iv. As the capacity of each chiller for option-2 is the same, i.e. 1100 RT, capacity of the associated chilled water pumps, condenser water pumps and cooling towers will be the same. As a result, control, operation and maintenance will be simpler for option-2.

v. Three different chiller combinations are tested in this problem. Other combinations of chillers of different capacities could be tested to find the optimum combination that consumes minimum electrical energy and satisfies other constraints of the project.

1.8.4 Peak and Off-Peak Operation

i. The off-peak period cooling load is usually quite small in comparison to the peak period.

ii. Example: Night time (off-peak period) cooling load could be very small in comparison to the day time (peak period) cooling load of an office building. In some cases, only the cooling load of the data center may need to be supported during the off-peak period. However, the cooling load of the office area as well as the data center area needs to be supported during the peak period.

iii. If peak period chillers of higher capacity are operated to support the off-peak period cooling load, the chillers will be running at low part load with poor energy efficiency during the off-peak period.

iv. Dedicated small “Off-peak chillers” should be installed and operated to maintain good energy efficiency during the off-peak period.

Example 1.5

Peak period (8am to 10pm) cooling load of an office building varies between 1500 to 2000 RT. However, off-peak period (10pm to 8am) cooling load varies between 200 to 250 RT to support the data center area. Presently, a chiller of capacity 600 RT is used to provide cooling during off-peak period and the chiller operating efficiency varies between 1 to 1.2 kW/RT as shown in Table 1.21. Calculate the achievable savings if a new chiller of capacity 300 RT of efficiency 0.56 and 0.58 kW/RT for the loading of 250 and 200 RT respectively is used to provide cooling during the off-peak period.

Table 1.21 Present energy consumption during the off-peak period

Time	Hrs/day	Cooling load (RT)	Present chiller efficiency (kW/RT)	Present chiller consumption (kWh/day)
	A	B	C	A x B X C
2200 to 2300	1	250	1.0	250
2300 to 2400	1	250	1.0	250
0000 to 0100	1	250	1.0	250
0100 to 0200	1	250	1.0	250
0200 to 0300	1	200	1.2	240
0300 to 0400	1	200	1.2	240
0400 to 0500	1	200	1.2	240
0500 to 0600	1	200	1.2	240
0600 to 0700	1	250	1.0	250
0700 to 0800	1	250	1.0	250
Total	10			**2460**

Solution:

The achievable energy saving due to the operation of the new chiller of capacity 300 RT is calculated and summarised in Table 1.22.

Table 1.22 Achievable energy saving due to the operation of the new 300 RT chiller

Time	Hrs/day	Cooling load (RT)	Present chiller efficiency (kW/RT)	Present chiller consumption (kWh/day)	Proposed chiller efficiency (kW/RT)	Proposed chiller consumption (kWh/day)
	A	B	C	A x B X C	D	A x B X D

2200 to 2300	1	250	1	250	0.56	140
2300 to 2400	1	250	1	250	0.56	140
0000 to 0100	1	250	1	250	0.56	140
0100 to 0200	1	250	1	250	0.56	140
0200 to 0300	1	200	1.2	240	0.58	116
0300 to 0400	1	200	1.2	240	0.58	116
0400 to 0500	1	200	1.2	240	0.58	116
0500 to 0600	1	200	1.2	240	0.58	116
0600 to 0700	1	250	1	250	0.56	140
0700 to 0800	1	250	1	250	0.56	140
Total	10			**2460**		**1304**

Energy savings = 2460 – 1304 = 1,156 kWh/day

1.8.5 Consolidation of Chilled Water Systems

i. Chilled water system's energy consumption can be reduced or the operating performance of the chillers can be improved through the consolidation of the chilled water systems.

ii. Possible opportunities are:
   - Different buildings within one facility (Example: Different blocks of university) having different chilled water systems.
   - Tower block (office area) and podium block (retail area) are served by two separate chilled water systems.
   - Different standalone systems serving specific areas.

iii. Larger scale application could be the District Cooling Systems.

Example 1.6

Cooling loads of Building-A and Building-B are supported by two separate chilled water systems.

1. Chilled water system serving Building-A has 3 x 500 RT chillers and operates from 8am to 9pm. Cooling load varies from 150 to 650 RT as shown in Table 1.23.

2. Chilled water system serving Building-B has 2 x 400 RT chillers and operates from 9am to 9pm. Cooling load varies from 200 to 325 RT as shown in Table 1.24.

Assuming that the energy consumption of the pumps and cooling towers remains the same, estimate the achievable energy savings by consolidating the two chilled water systems.

Table 1.23 Present energy consumption of chilled water system of Building-A

Operating hours	Number of hours	Cooling load, RT	Operating chiller capacity, RT	Loading, %	Chiller efficiency, kW/RT	Energy consumption, kWh
	A	B			C	A x B x C
0800 to 0900	1	500	1 x 500	100.0	0.57	285
0900 t0 1000	1	550	2 x 500	55.0	0.74	407
1000 to 1100	1	600	2 x 500	60.0	0.70	420
1100 to 1200	1	650	2 x 500	65.0	0.72	468
1200 to 1300	1	600	2 x 500	60.0	0.70	420
1300 to 1400	1	600	2 x 500	60.0	0.70	420
1400 to 1500	1	550	2 x 500	55.0	0.74	407
1500 to 1600	1	500	1 x 500	100.0	0.57	285
1600 to 1700	1	500	1 x 500	100.0	0.57	285
1700 to 1800	1	350	1 x 500	70.0	0.68	238
1800 to 1900	1	150	1 x 500	30.0	0.95	143
1900 to 2000	1	150	1 x 500	30.0	0.95	143
2000 to 2100	1	150	1 x 500	30.0	0.95	143
					Total	4,063

Chiller energy consumption serving Building-A = 4,063 kWh/day

Table 1.24 Present energy consumption of chilled water system of Building-B

Operating hours	Number of hours	Cooling load, RT	Operating chiller capacity, RT	Loading, %	Chiller efficiency, kW/RT	Energy consumption, kWh
	A	B			C	A x B x C
0900 t0 1000	1	300	1 x 400	75.0	0.71	213
1000 to 1100	1	325	1 x 400	81.3	0.71	231
1100 to 1200	1	350	1 x 400	87.5	0.68	238
1200 to 1300	1	350	1 x 400	87.5	0.68	238
1300 to 1400	1	325	1 x 400	81.3	0.71	231
1400 to 1500	1	325	1 x 400	81.3	0.71	231
1500 to 1600	1	325	1 x 400	81.3	0.71	231
1600 to 1700	1	300	1 x 400	75.0	0.71	213
1700 to 1800	1	300	1 x 400	75.0	0.71	213

1800 to 1900	1	300	1 x 400	75.0	0.71	213
1900 to 2000	1	200	1 x 400	50.0	0.98	196
2000 to 2100	1	200	1 x 400	50.0	0.98	196
					Total	2,643

Chiller energy consumption serving Library = 2,643 kWh/day

Solution

Consolidated chilled water system could be:

i. A new chilled water system with more efficient chillers, pumps and cooling towers and improved piping layout.

ii. One of the existing chilled water systems with or without additional chillers.

iii. Need to ensure new system static pressure does not exceed equipment design pressure (may need to use the heat exchangers to divide the static pressure head).

iv. In this example, the peak combined cooling load for Building-A and Building-B is 1000 RT as shown in Table 1.25. The logical choice is to use the Building-A chilled water system to serve as the consolidated plant. 2 nos. of existing 500 RT chillers can be operated to support the combined cooling load of the buildings. The other 500 RT chiller will be used as the standby chiller.

Table 1.25 Energy consumption after chilled water system consolidation

Operating hours	Number of hours	Combined Cooling load, RT	Operating chiller capacity, RT	Loading, %	Chiller efficiency, kW/RT	Energy consumption, kWh
	A	B			C	A x B x C
0800 to 0900	1	500	1 x 500	100.0	0.57	285
0900 t0 1000	1	850	2 x 500	85.0	0.55	468
1000 to 1100	1	925	2 x 500	92.5	0.56	518
1100 to 1200	1	1000	2 x 500	100.0	0.57	570
1200 to 1300	1	950	2 x 500	95.0	0.57	542
1300 to 1400	1	925	2 x 500	92.5	0.56	518
1400 to 1500	1	875	2 x 500	87.5	0.55	481
1500 to 1600	1	825	2 x 500	82.5	0.55	454
1600 to 1700	1	800	2 x 500	80.0	0.57	456
1700 to 1800	1	650	2 x 500	65.0	0.72	468
1800 to 1900	1	450	1 x 500	90.0	0.56	252

1900 to 2000	1	350	1 x 500	70.0	0.68	238
2000 to 2100	1	350	1 x 500	70.0	0.68	238
					Total	5,487

Energy consumption after chilled water system consolidation = 5,487 kWh/day

Chilled water system energy savings due to consolidation

= (4,063 + 2,643) - 5,487

= 1,219 kWh/day

= 1,219 x 365

= 444,750 kWh/year

If electricity tariff is $0.20 /kWh, the annual cost saving due to the consolidation of the chilled water systems = (444,750 kWh/year) x ($0.20 /kWh)

= $ 88,950 per year

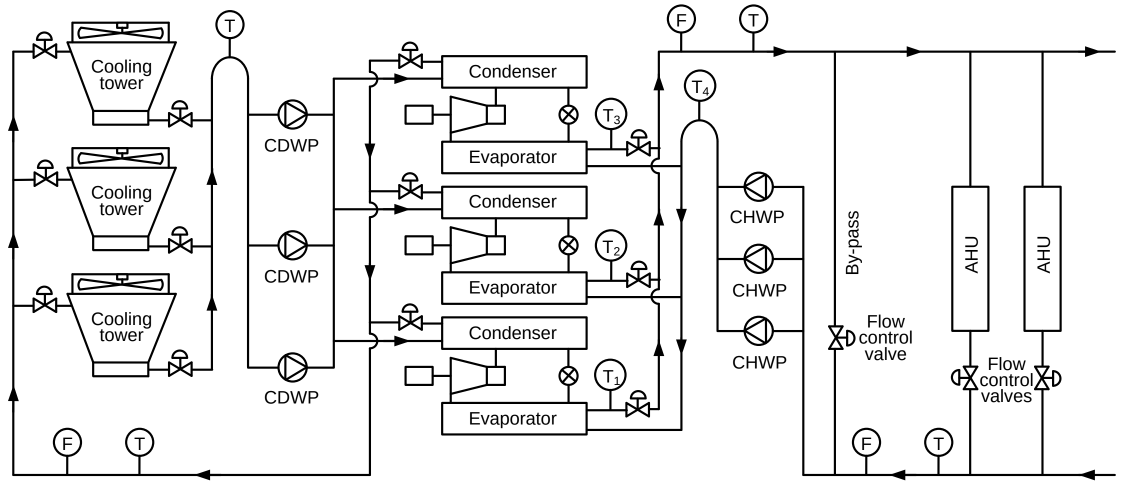
1.8.6 Chiller Sequencing Strategies

Main objective of chiller sequencing is to select which chiller or set of chillers to be operated to support building load in an energy efficient manner. During sequencing of the chillers, the selected chillers need to satisfy the following:

i. Building cooling load.
ii. Chilled water flow requirements: To ensure enough chilled water will flow to each AHU
iii. Chilled water set-point requirements: To ensure AHUs can provide the required sensible and latent cooling

Since chiller capacity and compressor power consumption change with the change of chilled water and condenser water supply temperature, compressor power (kW) should also be considered for sequencing to protect the compressor. Few chilled water system sequencing strategies are presented below:

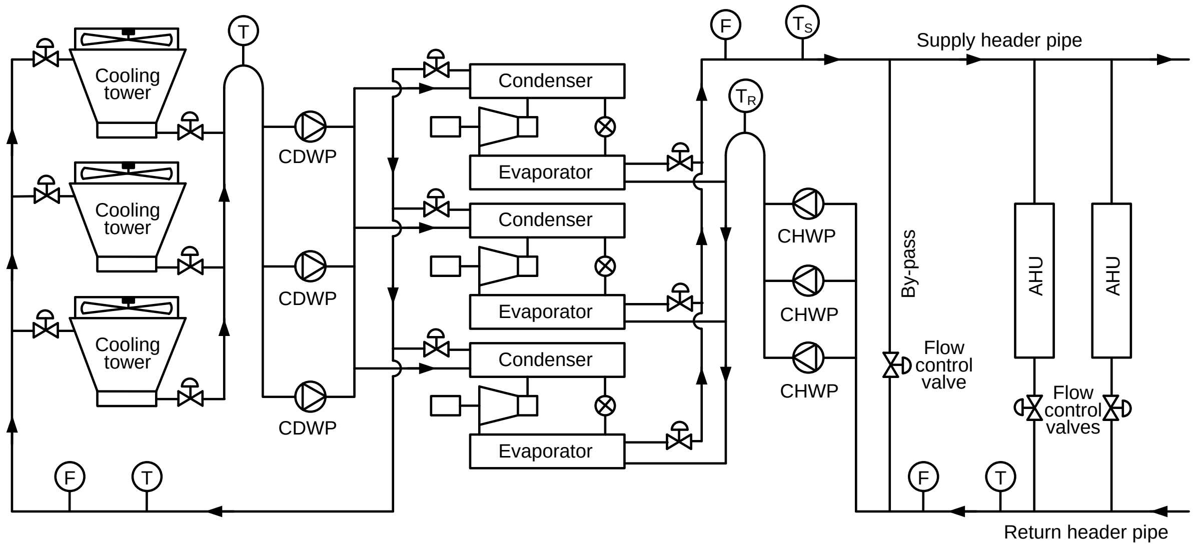
Chiller Sequencing Strategy-1:



Start another one chiller if:
T_1, T_2 or T_3 is greater than chilled water pre-set supply temperature.

Stop one chiller if:
ΔT (difference between chilled water return T_4 and supply T_1, T_2 or T_3 temperature) is less than a fixed value (0.5 for equal capacity chillers).

Chiller Sequencing Strategy-2:



Start another one chiller if:
T_s is greater than chilled water pre-set supply temperature

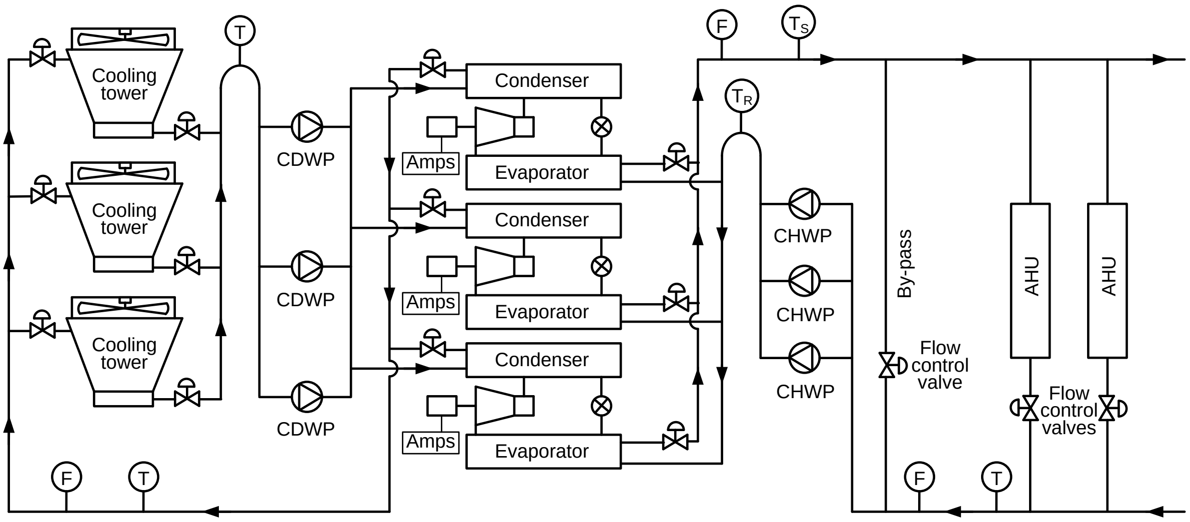
Or

Cooling load = Chiller capacity

Stop one chiller if:

- Cooling load can be supported by operating 1 no. less chiller
  - Monitor to ensure T_s is not greater than chilled water set point temperature

Recommended Strategy for Primary Chilled Water Pumping System

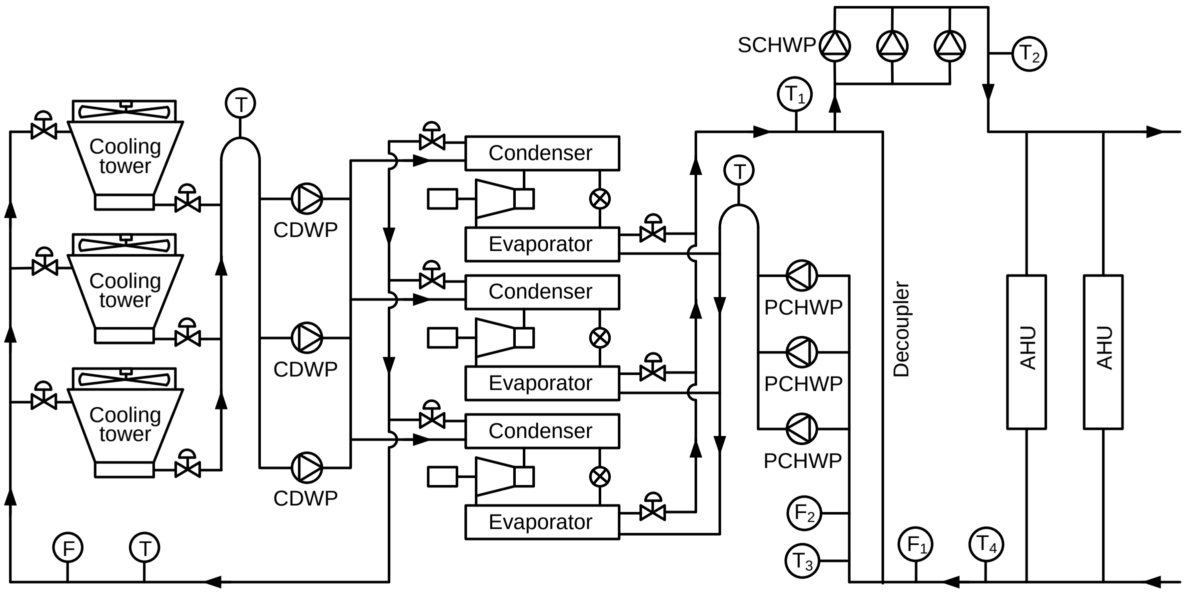


Start another one chiller if:
T_s is greater than chilled water pre-set supply temperature
Or
{Cooling load = Chiller capacity
And
Compressor current ≥ Full load current}

Stop one chiller if:

- Cooling load can be supported by operating 1 no. less chiller
  - Monitor to ensure T_s is not greater than chilled water set point temperature

Recommended Strategy for Primary-Secondary Chilled Water Pumping System



Start another one chiller if:
T_1 is greater than chilled water pre-set supply temperature
Or
Secondary loop chilled water flow F_1 > Primary loop chilled water flow F_2

Stop one chiller if:

- Cooling load can be supported by operating 1 no. less chiller
- Flow in decoupler pipe is about 110% of design flow of one operating chiller

Recommended Chiller Sequencing and Control Strategy: Chilled water system energy efficiency can be optimised further by:

i. Sequencing chillers based on optimum part load efficiency.
ii. Analysing performance characteristics of chillers under variable chilled and condenser water flow rate and temperature.
iii. Analysing performance characteristics of pumps and using appropriate operating and control strategies.
iv. Analysing performance characteristics of cooling towers and fans and using appropriate operating and control strategies.
v. Analysing performance characteristics of AHUs, chilled water flow and temperature requirements for AHUs, air-conditioning space temperature, relative humidity (RH) and fresh air flow requirements.

1.8.7 Chilled and condenser water temperature reset

The refrigeration cycle of vapour compression air-conditioning systems is shown in Figure-1.24. The difference of the enthalpies of refrigerant at the inlet and outlet of the evaporator ($h_1 - h_4$) represents the cooling effect produced per unit mass of the flowing refrigerant. The function of the compressor is to increase the pressure of the flowing refrigerant from the evaporator pressure P_1 to the condenser pressure P_2. The power consumption of the compressor depends on the difference of the condenser and evaporator pressures ($P_2 - P_1$) (also known as compressor lift) and the mass flow rate of refrigerant through the compressor. If the difference of enthalpies ($h_1 - h_4$) is increased, the required mass flow rate of refrigerant through the cycle to get the same cooling effect will be decreased. As a result, the power consumption of the compressor to produce the same cooling effect will decrease. Similarly, if the evaporator pressure P_1 is increased and condenser pressure P_2 is decreased, the difference of ($P_2 - P_1$) will decrease. Consequently, the power consumption of the compressor will decrease. Chiller efficiency (kW/RT) is defined as the ratio of compressor power consumption to produce unit cooling effect. Therefore, if the power consumption of the compressor is reduced by reducing the value of ($P_2 - P_1$) or required mass flow rate of the refrigerant is reduced by increasing the cooling effect ($h_1 - h_4$), the efficiency of the chiller will be increased. The steps for changing the above parameters by regulating different operating variables are discussed in the following sections.

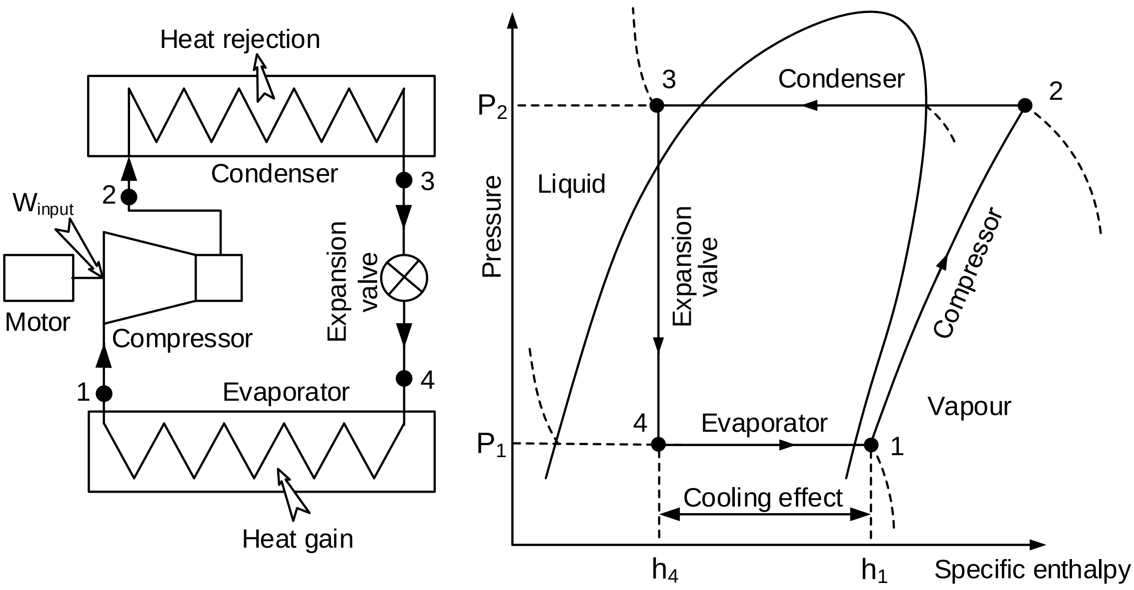


Figure 1.24 Refrigeration cycle of vapour compression air-conditioning systems

Effect of Chilled Water Temperature Reset: The effect of the reset of chilled water supply temperature is shown in Figure-1.25. As saturation pressure of refrigerant depends on its corresponding saturation temperature, the pressure of refrigerant inside the evaporator increases with the increase of chilled water supply temperature. 1-2-3-4 and 1'-2'-3-4' represent the refrigeration cycles for low and increased chilled water supply temperatures, respectively. It is obvious from Figure-1.25 that due to the increase of chilled water supply temperature, the difference of the condenser and evaporator pressures will decrease from (P_2 – P_1) to ($P_{2'}$ – $P_{1'}$) and the cooling effect produced per unit mass of the flowing refrigerant will increase from (h_1 – h_4) to ($h_{1'}$ – $h_{4'}$). Consequently, refrigerant flow rate through the cycle will decrease; compressor power consumption will decrease and the energy efficiency of the chiller will improve. However, due to the increase of chilled water supply temperature, the air handling unit (AHU) may not be able to provide required latent and sensible cooling. Surface area of the cooling coil of AHU should be large enough and designed carefully to provide the required cooling with the higher chilled water supply temperature.

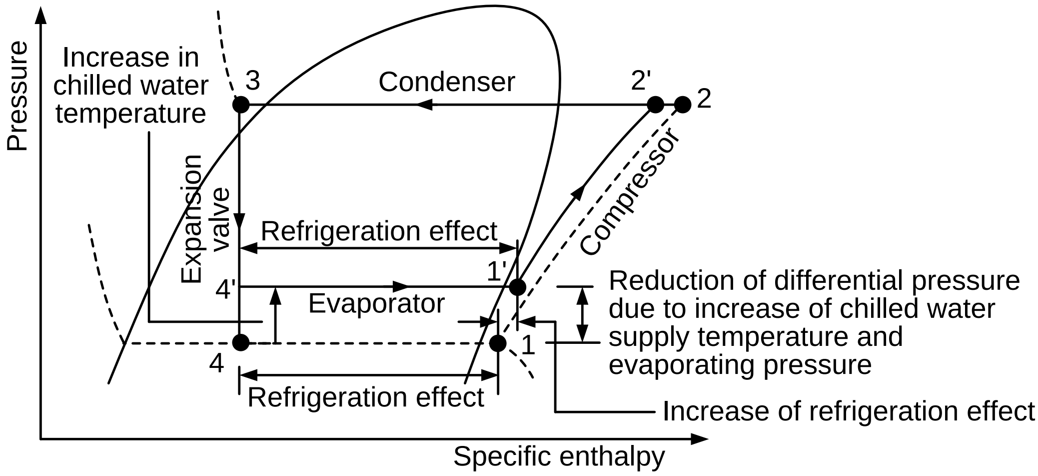


Figure 1.25 Effect of chilled water temperature reset

In general, an improvement in chiller efficiency of about 1 to 2 percent can be achieved by increasing the chilled water supply temperature by 0.6 ^oC (1 ^oF) due to the reduction of compressor lift and the increase of cooling effect [Jayamaha, 2006]. The typical variation of chiller efficiency with the change of chilled water supply temperature is shown in Figure-1.26. Usually, chilled water systems are designed for chilled water supply temperature of 6.7 ^oC to support the peak cooling load. However, peak cooling load generally occurs only for short period of time. Most of the time chillers are operated at part load. Under such operating conditions, it is not necessary to supply chilled water at 6.7 ^oC to the cooling coil of AHUs and the chilled water supply temperature can be reset to a higher value.

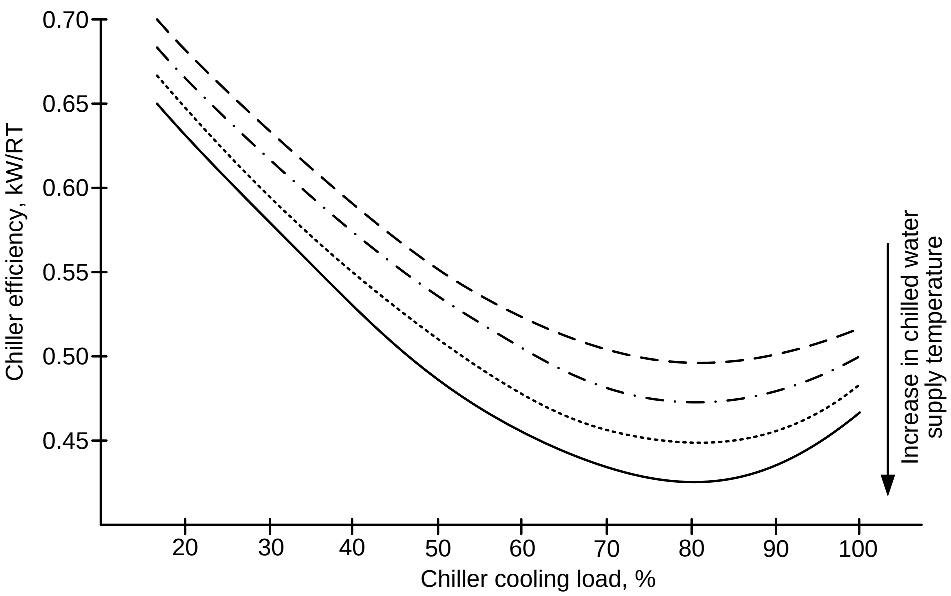


Figure 1.26 Typical variation of chiller efficiency with chilled water supply temperature

Method of Chilled Water Temperature Reset: During the period of low cooling load of the spaces, the chilled water supply temperature can be reset based on:

i. Position of the chilled water flow modulating valve of the cooling coils of AHUs. Chilled water supply temperature can be increased if the modulating valves close more than the pre-set value during the period of low cooling loads of spaces and vice versa.
ii. Pre-set value of chilled water return temperature. For constant flow rate of chilled water, during the period of low cooling load of the spaces, the chilled water return temperature will be lower than the pre-set value. Under such operating conditions, chilled water supply temperature can be increased.
iii. Measured actual cooling load of spaces or outdoor temperature. Heat transfer rate through the building envelope to the air-conditioned spaces decreases with the decrease of outdoor temperature. Chilled water supply temperature can be increased during the period of low outdoor temperature or space cooling load.

For constant air volume (CAV) air handling units (AHUs), air flow rate across the cooling coils of the AHUs and the spaces is kept constant irrespective of the cooling load of the spaces. The temperature of the supply air is modulated based on the space cooling load. If the temperature of the chilled water is increased at low load conditions, volume flow rate of chilled water to the cooling coils of AHUs is required to increase to get the same cooling effect. Consequently, power consumption of the variable flow chilled water pumps is increased due to the need for more chilled water flow. Typical variation of specific power consumption of the compressor of chiller and chilled water pump to provide unit cooling effect or efficiency (kW/RT) is shown in Figure-1.27. As expected chiller efficiency improves but chilled water pumping efficiency drops with the increase of chilled water supply temperature. The point of minimum overall specific power consumption of the chiller and the chilled water pump represents the optimum chilled water supply temperature. This optimum temperature depends on the performance characteristics of the chiller and pump.

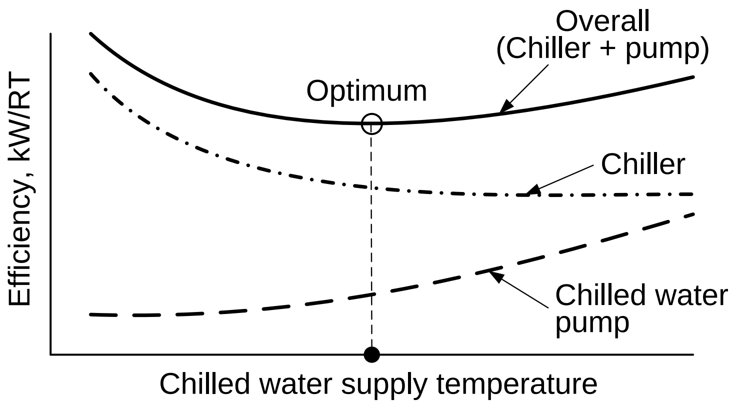


Figure 1.27 Variation of chiller and chilled water pump efficiency with chilled water supply temperature

Moreover, cooling coil of the AHU provides not only sensible cooling but also latent cooling by condensing moisture of the circulating air. Moisture removal ability of the AHU depends on the chilled water supply temperature to the cooling coil. Due to the increase of chilled water supply temperature, the circulating air flowing across the cooling coil of AHU may not reach to the required dew point temperature. Consequently, moisture condensation rate in AHU may drop resulting in higher relative humidity in the air-conditioned spaces. On the other hand, for variable air volume (VAV) systems, increase of the chilled air supply temperature may lead to the increase of the supply air temperature from the AHUs. As a result, more air may need to be supplied to the spaces to satisfy the same cooling load. This results in higher power consumption of the AHU fans.

Example 1.7

The average cooling load of a chiller is 1000 RT. Present chilled water supply temperature is 7 ^{0}C and the corresponding chiller efficiency is 0.51 kW/RT. If the chilled water supply temperature is increased to 8.5 ^{0}C, calculate annual energy savings. The chiller is operated 10 hours per day and 350 days of a year. Assume that chiller efficiency is improved by 2% for every 1 ^{o}C increase of chilled water supply temperature.

Solution

Chiller cooling load = 1000 RT

Present chiller efficiency = 0.51 kW/RT

Present chilled water supply temperature = 7 ^{0}C

Proposed chilled water supply temperature = 8.5 ^{0}C

Proposed increment of chilled water supply temperature = 8.5 – 7 = 1.5 ^{o}C

Given that chiller efficiency is improved by 2% for every 1 ^{o}C increase of chilled water supply temperature

Improvement of chiller efficiency due to the increase of chilled water supply temperature by 1.5 ^{o}C = 1.5 x 2 = 3%

Chiller new efficiency = 0.51 kW/RT x (1 – 0.03) = 0.4947 kW/RT

Chiller is operated for 10 hrs/day and 350 days/year

Annual energy savings = (0.51 - 0.4947) x 1000 x 10 x 350 = 53,550 kWh/year

Effect of Condenser Water Temperature Reset: The effect of the reset of condenser water supply temperature is shown in Figure-1.28. The pressure of refrigerant inside the condenser decreases with the decrease of condenser water supply temperature. 1-2-3-4 and 1-2'-3'-4' represent the refrigeration cycles for high and low supply temperatures of condenser water, respectively. It is obvious from Figure-1.28 that due to the decrease of condenser water supply temperature, the difference of the condenser and evaporator pressures will decrease from (P_2 – P_1) to ($P_{2'}$ – P_1) and the cooling effect produced per unit mass of the flowing refrigerant will increase from (h_1 – h_4) to (h_1 – $h_{4'}$). Consequently, refrigerant flow rate through the cycle will decrease to produce the same cooling effect; compressor power consumption will decrease and the energy efficiency of the chiller will improve. However, due to the decrease of condenser water supply temperature, the power consumption of the cooling tower will increase. The increase of cooling tower power consumption and the reduction of chiller compressor power consumption need to be analysed carefully to determine the net energy saving.

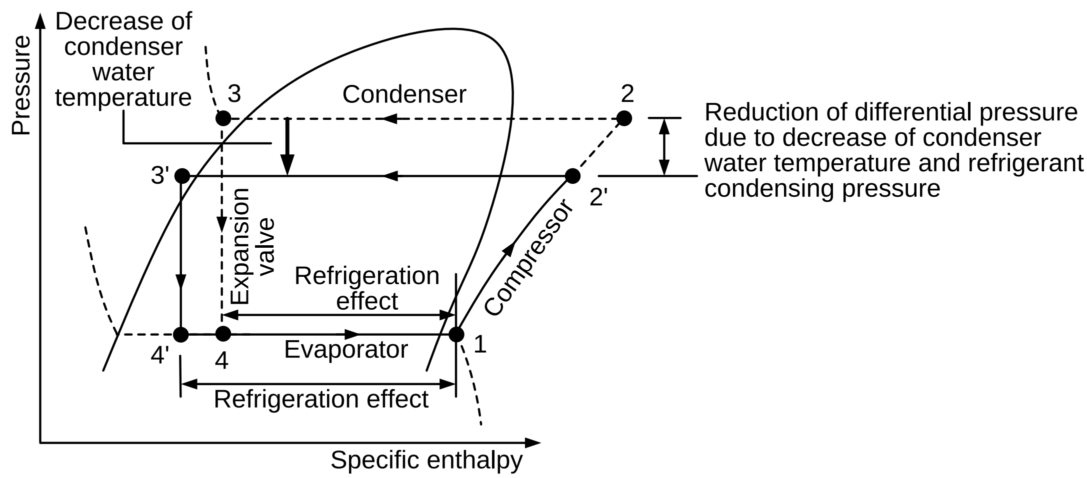


Figure 1.28 Effect of condenser water temperature reset

Same as the chilled water supply temperature, usually an improvement in chiller efficiency of about 1 to 2 percent can be achieved by reducing the condenser water supply temperature by 0.6 ^{o}C (1 ^{o}F) due to the reduction of compressor lift and the increase of cooling effect [Jayamaha, 2006]. However, to reduce the temperature of condenser water, cooling tower fans are required to operate at higher speed which results in the increase of the cooling tower fan power consumption. Typical variation of specific power consumption of the chiller and cooling tower to provide unit cooling effect or efficiency (kW/RT) is shown in Figure-1.29. As expected, chiller efficiency improves but cooling tower efficiency drops with the decrease of condenser water supply temperature. The point of minimum overall specific power consumption of the chiller and the cooling tower represents the optimum condenser water supply temperature which depends on the performance characteristics of the chiller and the cooling tower.

Wet-bulb temperature represents the minimum achievable condenser water temperature. The size of the cooling tower will be extremely large, construction and operating cost will be high and the system will not be economically feasible if condenser water is targeted to cool to the wet-bulb temperature. As the condenser water temperature reaches closer to the wet-bulb temperature, heat transfer rate from the condenser water to the air reduces and hence the power consumption of the cooling tower increases to transfer the same amount of heat. Usually cooling towers are designed to cool the condenser water to about 2 to 3 ^{o}C higher than the wet-bulb temperature. If the wet-bulb temperature of the ambient air drops, cooling towers can cool the condenser water to a lower temperature which helps to improve the efficiency of the chiller.

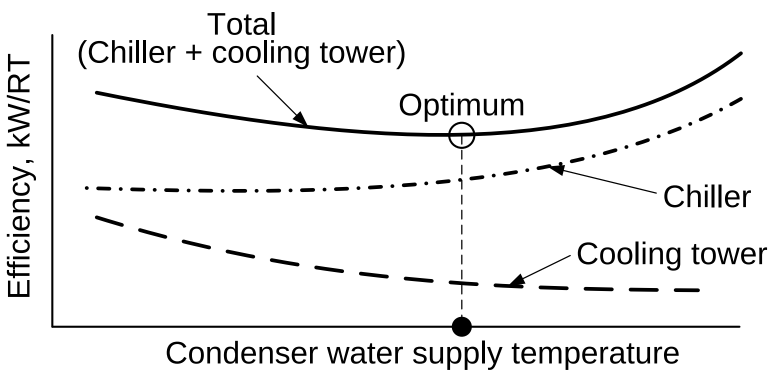


Figure 1.29 Variation of chiller and cooling tower efficiency with condenser water supply temperature

1.8.8 Condenser Tube Cleaning and Water Treatment

Refrigerant rejects heat to the cooling water through the condenser tubes. If scale or fouling develops on the tubes surface, the resistance to heat transfer increases. As a result, temperature and corresponding pressure of refrigerant in the condenser increases to enhance the driving force of heat transfer, which lowers the efficiency of the chiller. During the heat transfer process in the cooling tower, the condenser water is exposed to the ambient air. As a result, dust of ambient air is easily captured by the condenser water. Captured dust gradually deposits on the surface of the tubes causing scaling or fouling. Research has shown that scale of 0.6 mm thickness will increase the chiller compressor power consumption by about 20 percent [Jayamaha, 2006]. Therefore, it is important to take necessary steps to prevent scaling or fouling and keep the condenser tubes clean. Generally, scale and fouling on condenser tubes is controlled by:

i. Water treatment system: The acidity of water is increased using chemical treatment processes to control the alkalinity, which increases the tendency of scaling.
ii. Water blow down: Certain percent of condenser water is continuously blown down and clean water is added to maintain the solid content of water below certain level.
iii. Tube cleaning: Condenser tubes are cleaned periodically by brushing. The efficiency of chillers returns to the rated value once the tubes are cleaned. Again the efficiency drops gradually as the scale and fouling gradually builds up on the tube surfaces. As a result, chiller efficiency profile takes the shape of "saw tooth" as shown in Figure-1.30. The height of the tooth represents the maximum drop of chiller efficiency before the cleanings.

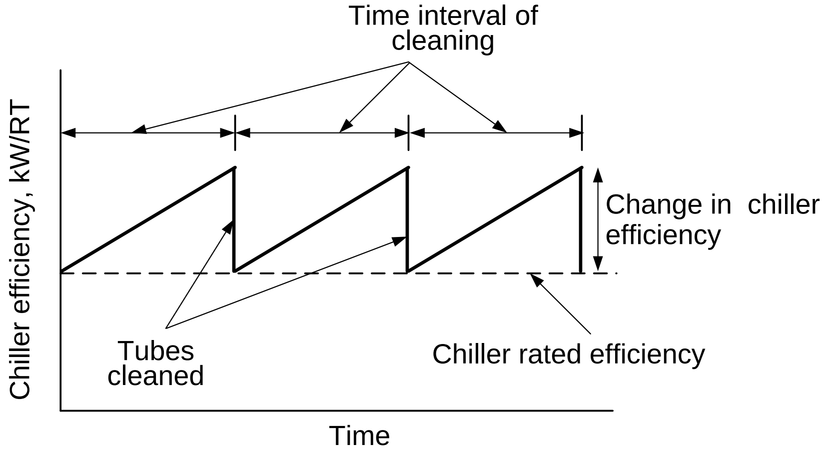


Figure 1.30 Variation of chiller efficiency due to periodic cleaning condenser tubes

To continuously maintain the chiller at the rated efficiency, automatic tube cleaning systems (with sponge balls or brushes) as shown in Figure-1.31 is used. The diameter of the balls is slightly bigger than the internal diameter of the condenser tubes. The balls from the ball station are injected by the pump at regular intervals and circulated through the tubes. As the balls travel through the condenser tubes, deposited scale and fouling on the tube surfaces are removed. Finally the balls are collected by the ball catcher and returned to the ball station.

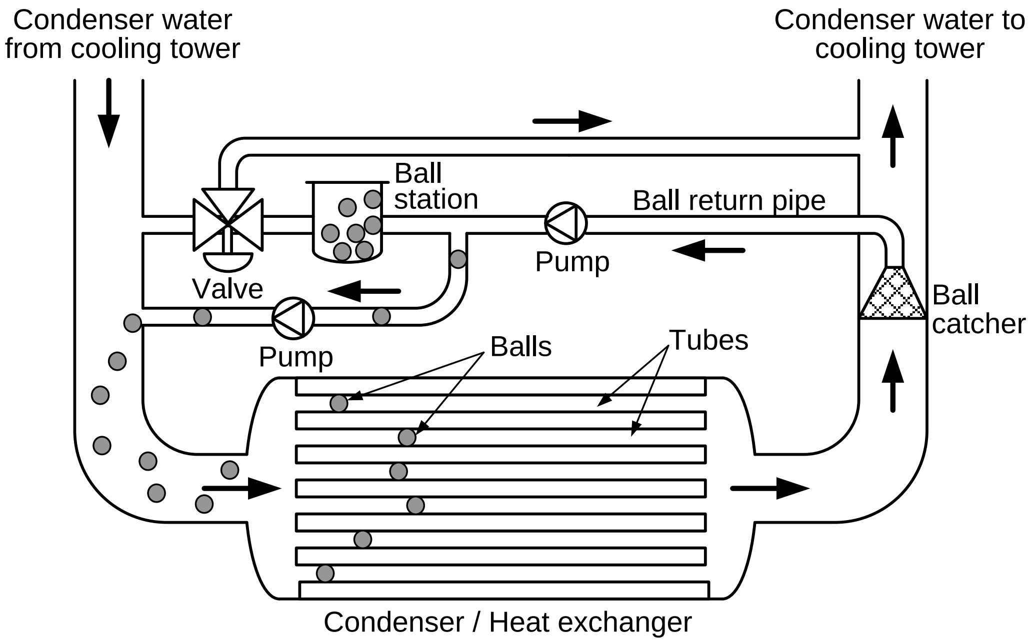


Figure 1.31 Automatic condenser tube cleaning system

1.9 Thermal Energy Storage Systems

Thermal energy storage systems used in air-conditioning systems store cold energy in storage tanks for use at a later period. The capacity of the storage tank is expressed in RT-hour. A thermal storage tank of capacity 500 RT-hour can support space cooling load of 100 RT for a period 5 hours. A typical thermal energy storage with chilled water system is shown in Figure-1.32. By regulating the opening of the valves (Figure-1.32), the system can store the cold energy in the chilled water storage tank during the period of no or part cooling load of the building and support space cooling load of the building by operating the chiller or using the stored cold energy of chilled water storage tank. Three common types of storage systems are:

i. Chilled water storage system: Chilled water of about 4 to 5 ^oC is stored in the thermal energy storage tank. Specific heat capacity of chilled water (about 4.2 kJ/kg K) is used to store the cold energy. For storage of 100 RT-hour cold energy, the required volume of chilled water storage system is about 55 m^3. Standard chillers can be used in chilled water storage system. As the chiller operates at almost normal operating condition to produce chilled water of about 4 to 5 ^oC, there is no remarkable drop in chiller efficiency.

ii. Ice storage system: Encapsulated ice, ice-on-coil and slurry ice are commonly stored in the thermal energy storage tank. Latent heat of fusion of water (about 333 kJ/kg) is used to store the cold energy. For storage of 100 RT-hour cold energy, required volume of chilled water storage system is about 8 m^3, which is much lesser in comparison to the chilled water storage system. Special chillers that are designed to operate at low temperature (such as -7^oC) are used in ice storage systems. As the chiller operates at low temperature, the drop in efficiency of the chiller can be about 50%. Operating efficiency of ice-making chiller usually varies from about 0.8 to 1.0 kW/RT.

iii. Eutectic salts system: An Eutectic salts system uses a mixture of inorganic salts, water and other elements that freezes at a temperature above 0^oC. The system uses the advantage of high latent heat of fusion to store the cold energy. However, the efficiency of the chiller does not drop significantly like ice-making chillers as the chilled water system is operated above 0^oC. Standard chillers can be used in an eutectic salts system. Operating efficiency of the chiller depends on type of eutectic salts mixture used and the corresponding freezing point.

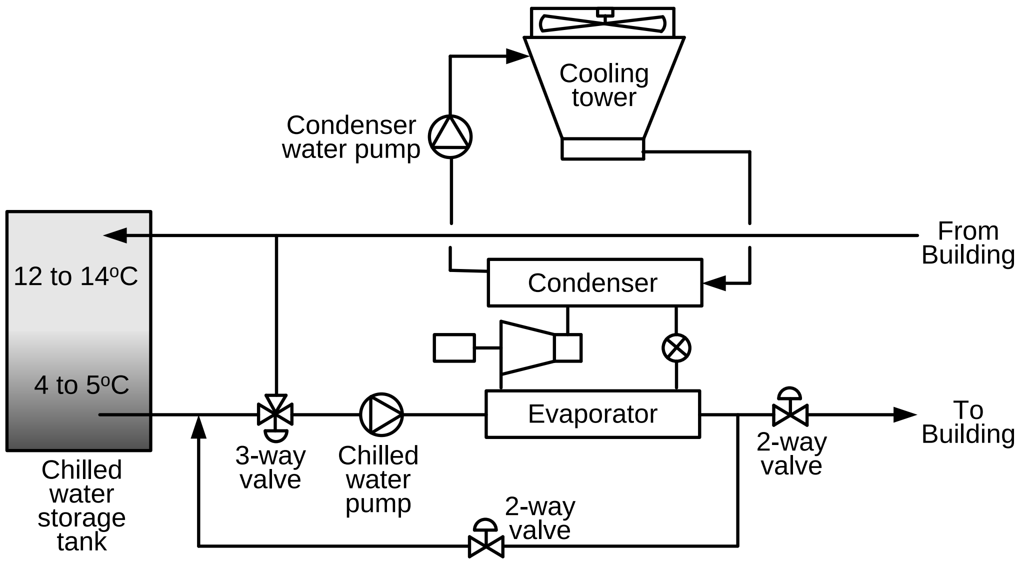


Figure 1.32 Typical thermal energy storage with chilled water system

Feasibility of thermal energy storage systems depends on a number of factors such as:

i. Cooling load profile: If the cooling load demand for different spaces or processes occurs for short period of time and infrequently, cooling load demand of the spaces or processes can be supported using the thermal energy storage systems without frequently turning on and off the chiller or operating the chiller at low part load with poor efficiency.

ii. Electricity tariffs / availability: If the electricity tariff is time dependent (peak and off-peak tariff), thermal energy storage tank could be charged by operating the chiller during the period of off-peak tariff and use the stored cold energy to support the cooling demand of the spaces or processes of peak tariff period. High electricity demand charge can also be saved by charging the thermal energy storage tank during the period of low electricity demand. Similarly, electricity supply could be limited during the period of high cooling demand. Thermal energy storage tank could be charged when electricity is available and the stored cold energy could be used to support the cooling demand when the supply of electricity is limited.

iii. Others: Because of the nature of operation or associated huge losses, interruption of cooling is not tolerated for some industrial production processes, labs and critical operational spaces. For such applications, one additional chiller is operated to avoid the interruption of cooling because of the risk of tripping of one running chiller. Consequently, the chilled water system operates at low part load with poor efficiency. For those applications, the additional chiller can be turned off. The chilled water system will be running at better loading and efficiency. Thermal energy storage system will take over the cooling load if any chiller trips. Similarly, if a chilled water system does not have spare peak capacity but has excess off-peak capacity, the energy storage tank could be charged during the off-peak period to support the cooling load during peak period. This eliminates the need to install another chiller.

Two common storage strategies are used in thermal energy storage systems, and they are:

i. Full storage strategy: The cooling load during the entire peak period is met by the thermal energy storage system as shown in Figure-1.33. Chilled water systems are operated during the off-peak period to charge the energy storage tanks. During peak period, the chilled water system is turned off and the cooling load is supported only by the thermal energy storage tank. This strategy is particularly attractive if the difference between the peak and off-peak electricity tariff is big and the demand charge of electricity is high.

ii. Partial storage strategy: In the partial storage strategy, part of the peak period cooling demand is supported by operating the chiller and the remaining cooling demand is met by the thermal energy storage tank as shown in Figure-1.34. Chilled water systems are operated during the off-peak period as well to charge the energy storage tanks. This strategy is attractive if the peak cooling demand is much higher than the average cooling demand. Other than installing chillers of high capacity to meet the maximum cooling demand, appropriately sized small capacity chillers with a thermal energy storage system can be installed to meet the cooling load. This helps to reduce the capital cost required for the chillers and the associated systems.

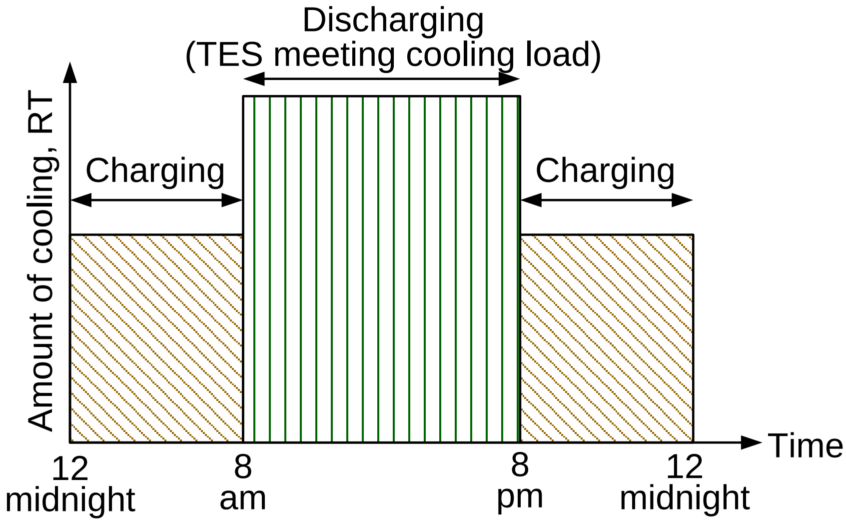


Figure 1.33 Full storage strategy

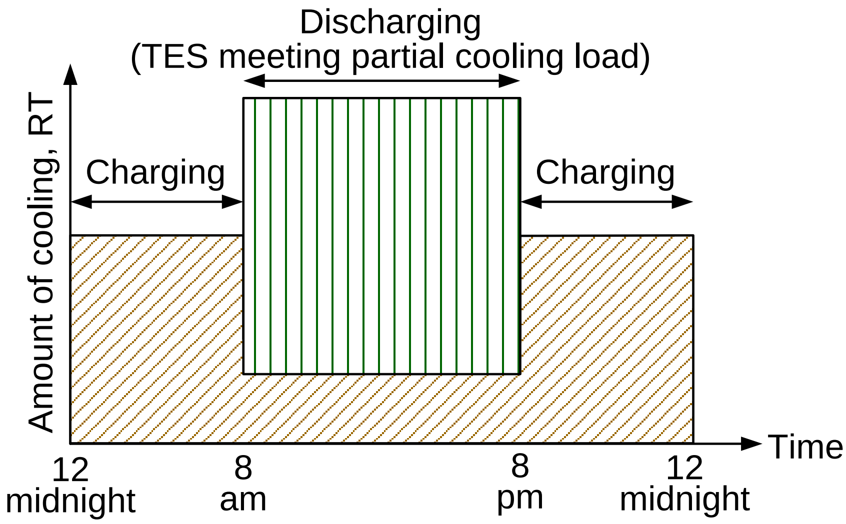


Figure 1.34 Partial storage strategy

1.10 District Cooling Systems

A larger scale application of the thermal energy storage system is known as district cooling system. A number of high capacity chillers and thermal energy storage tanks (chilled water and / or ice systems) are installed in the district cooling systems. Chilled water is distributed from the thermal energy storage tanks to the nearby buildings using distribution pumps as shown in Figure-1.35. Dedicated heat exchangers are installed in each building to transfer the heat load of the buildings to the chilled water. Chilled water flow meter and temperature sensors are usually installed at the inlet and outlet of the heat exchangers as shown in Figure-1.35 to determine the heat load transferred (in kWh or RTh) by each building.

Heat load transferred by a building can be calculated as:

$$Q_{building} = \sum \frac{V_{chw}\rho_{chw}C_p\left(T_{chw,r} - T_{chw,s}\right)\Delta t}{3.517} \qquad (1.25)$$

where

V_{chw} = Chilled water flow rate, m^3/s

ρ_{chw} = Chilled water density, kg/m^3

C_p = Specific heat of chilled water, kJ/kg. K

$T_{chw,r}$ = Chilled water return temp, ^oC

$T_{chw,s}$ = Chilled water supply temp, ^oC

ΔT = time interval of data recording, hr

$Q_{building}$ = Heat load transferred by the building, RTh

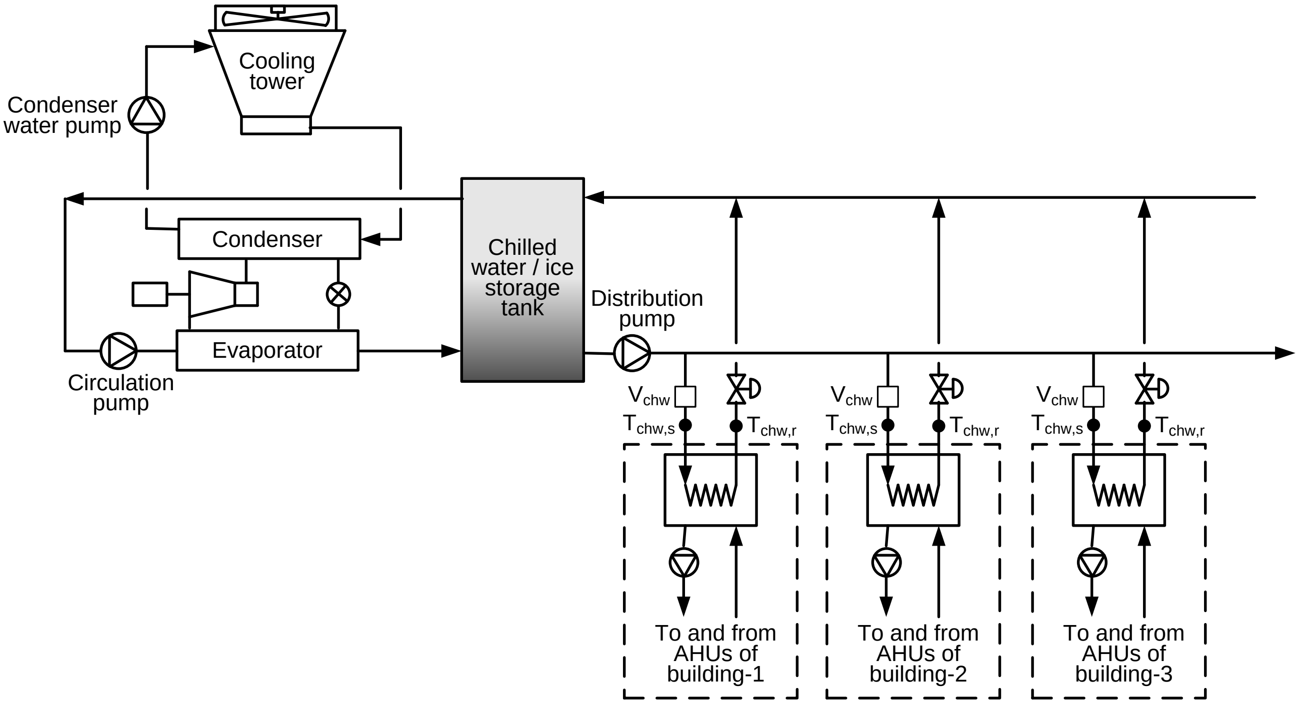


Figure 1.35 Typical district cooling system

Chapter-2: Pumping Systems

2. Pumping Systems

Chilled water and condenser water pumping systems are used in water-cooled central air-conditioning chilled water systems. The function of the chilled water pumps is to provide the primary force to overcome the pressure losses caused by different components of the chilled water loops and circulate the required amount of chilled water through the evaporator of the chillers and cooling coils of the AHUs. Similarly, condenser water pumps overcome the associated pressure losses and circulate the required amount of condenser water through the condenser of the chillers and cooling towers. Centrifugal pumps are commonly used in which an impeller is rotated inside a volute or diffuser casing by a motor to generate the required static pressure and kinetic energy for the flow of water. Based on the piping layout of the chilled water systems, the pumps could be horizontally mounted end-suction, vertically mounted in-line or horizontally and vertically split case types. This chapter deals mainly with the interaction of the pump and system characteristic curves, selection of pumps, configuration of pumping systems, control strategies and energy saving opportunities in pumping systems.

Learning Outcomes

Participants will be able to:

i. Determine the piping system characteristic curves and analyse the interaction of the pump and system characteristic curves under different operating conditions.
ii. Size the pumps and calculate the energy consumption of different components of pumping systems.
iii. Apply affinity laws to evaluate the performance of pumping systems.
iv. Develop control strategies and calculate potential energy savings.

2.1 Types of Pumping Systems

Pumping systems used in water-cooled central air-conditioning chilled water systems are:

i. Closed loop system: Chilled water loops of the central air-conditioning chilled water systems are closed loop systems as shown in Figure-2.1. In closed loop systems, water is enclosed within the piping systems and does not come into contact with air. Pumps of closed loop systems need to overcome the frictional losses caused by different components within the closed loops such as pipes, bends, valves, strainers, heat exchangers and other fittings.

ii. Open loop system: Condenser water loops of the central air-conditioning chilled water systems are open loop systems. In open loop systems, water flows through an open section of the piping systems instead of flowing entirely inside the pipes. Water is sprayed in the open cooling tower (Figure-2.1) which is the open section of the condenser water piping system. Pumps need to develop an additional static pressure head equivalent to the vertical height of the open sections to circulate the water. In addition to the static pressure head, pumps of open loop systems also need to overcome the frictional losses caused by different components of the open loops such as pipes, bends, valves, strainers, heat exchangers and other fittings.

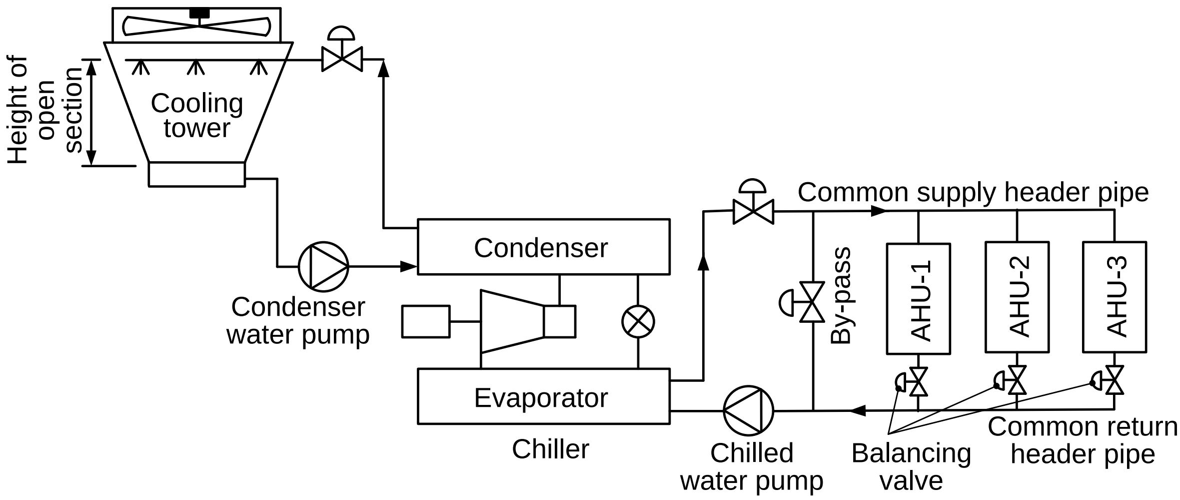


Figure-2.1 Closed and open loop piping of central air-conditioning chilled water systems

Closed and open loop piping systems used in central air-conditioning chilled water systems are designed as:

i. Direct return system: Chilled water loop shown in Figure-2.1 is an example of the direct return system. Chilled water is supplied to the individual AHU from the common supply header pipe. After flowing through the cooling coils of the AHUs, chilled water directly returns back through the common return header pipe to the chilled water pump. It is obvious from Figure-2.1 that the total travelling lengths of chilled water through the pipes for AHU-1 and AHU-3 become the shortest and the longest respectively. Consequently, more water will flow through AHU-1 in comparison to AHU-3 if cooling coils of the same pressure drop are selected for the AHUs. Balancing valves are used to balance the flow of water through the AHUs.

ii. Reverse return system: Schematic diagram of a reverse return chilled water loop is shown in Figure-2.2. Chilled water flows through a reverse piping system of common return header pipe before it returns to the chilled water pump. As a result, the total travelling length of chilled water through the pipes for all AHUs becomes equal. If cooling coils of the same pressure drop are selected for all the AHUs (which is generally not possible for practical AHUs of different cooling capacities), the system will be self-balanced and will eliminate the requirement for the balancing valves. However, total pipe length of the reverse return systems is longer than the direct return systems.

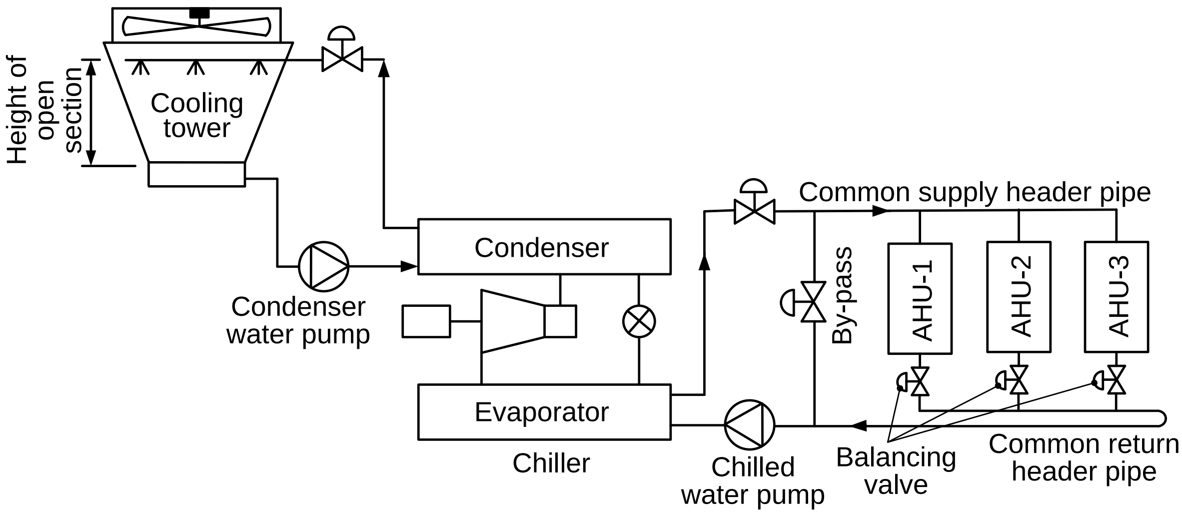


Figure-2.2 Reverse return chilled water piping system of a chilled water system

2.2 System Characteristic Curves

For incompressible flow in pipes, the pressure drop due to the friction loss in the pipe can be expressed using Darcy-Weisbach equation as:

$$\Delta P_{friction} = f\,\frac{L}{D}\frac{V^2}{2g} \tag{2.1}$$

where

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

f = Moody friction factor, dimensionless

L = length of the pipe, m

D = inner diameter of the pipe, m

V = average velocity of fluid, m/s

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

The friction factor *f* depends on piping material and the roughness of internal surface of the pipes. Average flow velocity of fluid in a pipe can be calculated as:

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

$$V = \frac{4Q}{\pi D^2} \tag{2.2}$$

where

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

A= cross sectional area of pipe, m^2

Combining Eqs. (2.1) and (2.2) gives:

$$\Delta P_{friction} = f \frac{L}{D^5} \frac{8Q^2}{g\pi^2} \tag{2.3}$$

Pressure losses for the fittings (also known as dynamic losses) such as elbow, tee-joint, valve, strainer etc. of the piping systems can be calculated as:

$$\Delta P_{fitting} = C_o \frac{V^2}{2g} \tag{2.4}$$

$$\Delta P_{fitting} = C_o \frac{8Q^2}{g\pi^2 D^4} \tag{2.5}$$

where

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

C_o = loss coefficient of the fitting, dimensionless

V = average velocity of fluid, m/s

Q = volume flow rate of fluid, 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 pressure losses of the pipes and the fittings represent the total pressure drop of a piping system. If a pump is installed in the piping system, the pump must produce pressure head equal to the total pressure drop of the piping system to maintain the flow. Total pressure drop can be expressed in Pa as:

$$\Delta P_{total} = \left(\Delta P_{friction} + \Delta P_{fitting}\right)\rho g \tag{2.6}$$

where

ΔP_{total} = total pressure drop, Pa

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

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

ρ = density of water, ≈1000 kg/m^3

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

Based on Eqs. (2.3), (2.5) and (2.6), the friction, fitting and total pressure drop of a piping system are proportional to the square of the water flow rate. Therefore, the plot of the total pressure drop of a piping system versus flow rate, known as system curve, is approximately parabolic as shown in Figure-2.3. Thus, the equation of the system curve of a piping system can be expressed in general form as:

$$\Delta P \propto Q^2 \tag{2.7}$$

$$\Delta P = CQ^2 \tag{2.8}$$

where

C = constant of proportionality

Each piping system has its own system curve due to its own unique pipe sizing, piping geometry and fittings. The system curve of a piping system will change, if any of these components, such as percentage of valve opening, is changed. Figure-2.3(a) and (b) represent the system curves for the closed and open loop piping systems. Pressure drop for the closed loop piping systems drops to zero once the flow is reduced to zero. However, pumps need to generate static pressure head equivalent to the height of the open component (such as the height of cooling tower) to start the flow in open loop piping systems. Note that the static pressure head is the same regardless of the flow rate. If the vertical height of the open section of a cooling tower is H_{CT}, static pressure head can be calculated as

$$\Delta P_{static} = H_{CT}\rho g \tag{2.9}$$

where

ΔP_{static} = static pressure head, Pa

H_{CT} = vertical height of cooling tower, m

ρ = density of water, ≈1000 kg/m^3

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

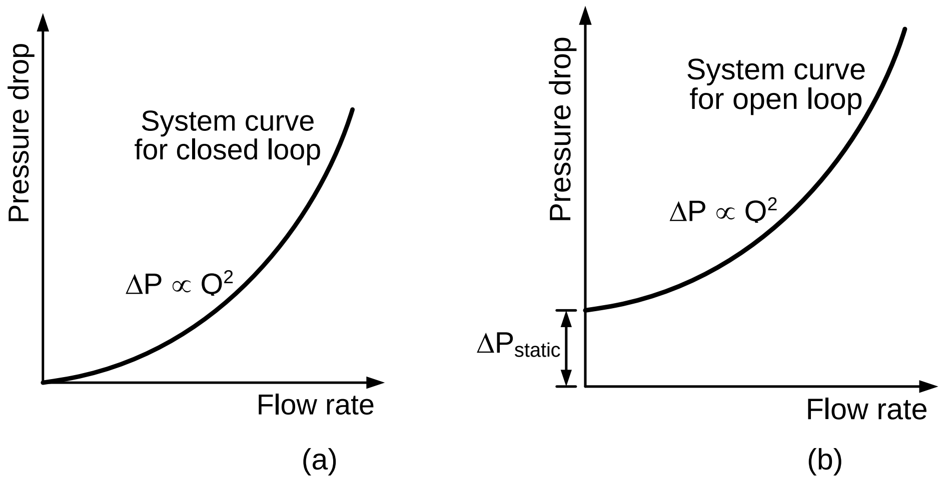


Figure-2.3 System curves. (a) closed loop system and (b) open loop system

Steps for developing the system curve of closed loop systems: To develop the system curve of an existing closed loop system (such as the chilled water loop shown in Figure-2.4), pressures at the inlet $P_{in,m}$ and exit $P_{out,m}$ of the chilled water pump and chilled water flow rate Q_m should be measured. As the total pressure drop of the piping system is equal to the pressure head generated by the pump, total pressure drop of the chilled water piping system is:

$$\Delta P_m = P_{out,m} - P_{in,m} \tag{2.10}$$

where

ΔP_m = measured pressure drop of piping system, Pa

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

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

Putting the value of the measured total pressure drop ΔP_m and water flow rate Q_m in Eq. (2.8) gives:

$$\Delta P_m = CQ_m^2$$

$$C = \frac{\Delta P_m}{Q_m^2} \tag{2.11}$$

where

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

Combining Eqs. (2.8) and (2.11) gives the equation of the system curve as:

$$\Delta P = \frac{\Delta P_m}{Q_m^2} Q^2 \tag{2.12}$$

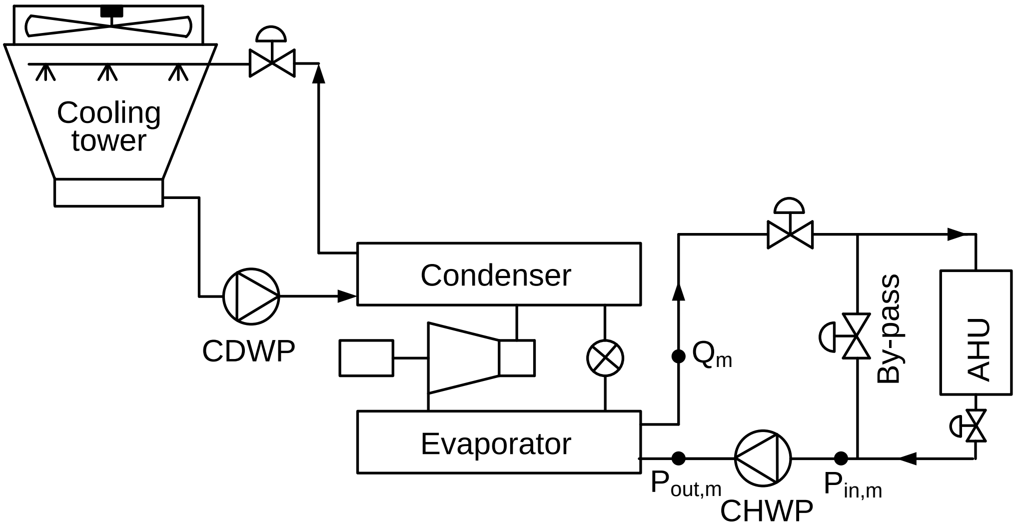


Figure-2.4 Measurement of pressure and water flow rate in closed loop

Eq. (2.12) represents the system curve for the closed loop chilled water piping system. For a new piping system, total pressure drop for the piping system needs to be calculated corresponding to the design flow rate of water and then the system curve can be determined following the same procedure presented above. If none of the piping system parameters such as pipe size, piping geometry, fittings, opening of the valves etc. are changed; the equation and profile of the system curve will remain unchanged. The total pressure drop of the piping system will change from ΔP_m to ΔP_1 if the water flow rate through the piping system is changed from Q_m to Q_1. The system pressure drop ΔP_1 for the water flow rate of Q_1 can be calculated using Eq. (2.12) as:

$$\Delta P_1 = \frac{\Delta P_m}{Q_m^2} Q_1^2 \tag{2.13}$$

If any of the piping system parameters such as the opening of the flow modulating valve are changed, the system curve will move either to the left or right as shown in Figure-2.5 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 modulating valve. Similarly, the system curve moves to the right if the system pressure is decreased due to the opening of the modulating valve. The changes in the system pressure drop with the flow rate of water and the movement of system curve with the opening of the flow modulating valve are represented in Figure-2.5.

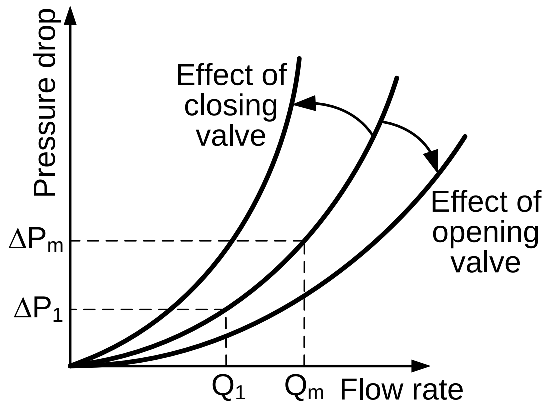


Figure 2.5 Changes in the system pressure drop with the flow rate of water and the movement of system curve with the opening of the flow modulating valve

Steps for developing the system curve of open loop systems: Same as the closed loop piping systems, to develop the system curve of an existing open loop system (such as condenser water loop shown in Figure-2.6), pressures at the inlet $P_{in,m}$ and exit $P_{out,m}$ of the condenser water pump, condenser water flow rate Q_m and height of open section of the cooling tower $H_{CT,m}$ should be measured. Total pressure drop of the condenser water piping system including the static pressure head of the cooling tower is:

$$\Delta P_m = P_{out,m} - P_{in,m} \tag{2.14}$$

where

ΔP_m = measured pressure drop of piping system, Pa

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

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

The static pressure head across the cooling tower can be calculated using Eq. (2.9) as:

$$\Delta P_{static,m} = H_{CT,m} \rho g \tag{2.15}$$

where

$\Delta P_{static,m}$ = measured static pressure head, Pa

$H_{CT,m}$ = measured vertical height of open section of the cooling tower, m

ρ = density of water, ≈1000 kg/m^3

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

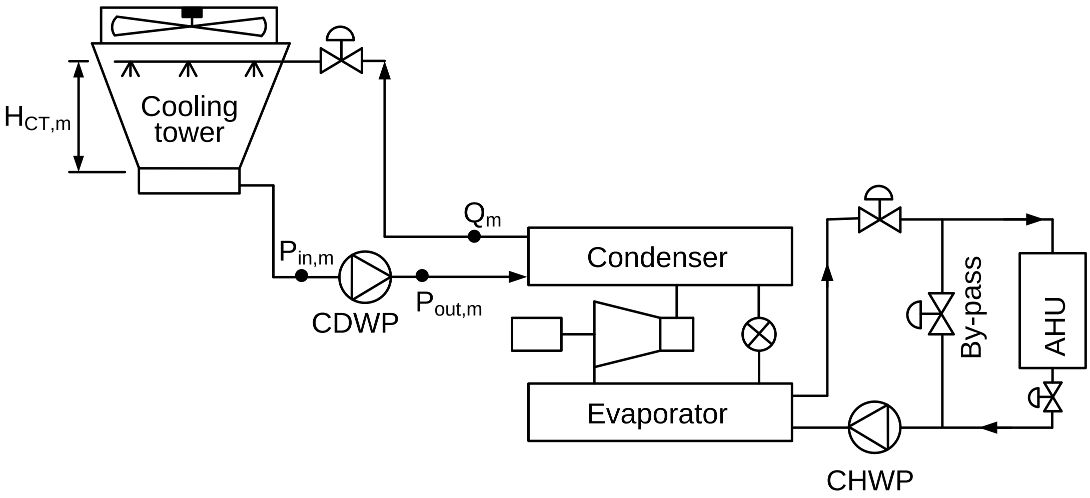


Figure-2.6 Measurement of pressure, water flow rate and height of open section of the cooling tower of open loop piping system

Total pressure drop of the piping system and the static pressure head are shown in the open loop system curve (Figure-2.7).

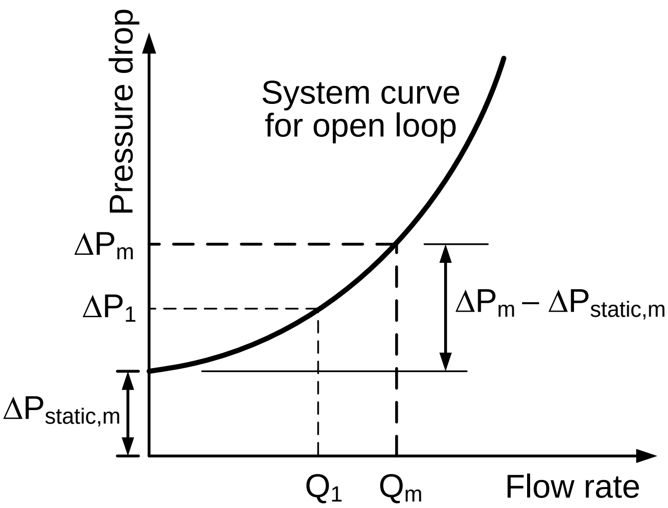


Figure 2.7 Total pressure drop and static pressure head of open loop piping system

In Figure-2.7, (ΔP_m - $\Delta P_{static,m}$) represents the total loss of pressure head due to the friction of the pipe and different fittings such as elbow, tee-joint, valve, strainer etc. of the piping systems. Therefore, based on Eq. (2.8):

$$\Delta P_m - \Delta P_{static,m} = CQ_m^2 \tag{2.16}$$

$$C = \frac{\Delta P_m - \Delta P_{static,m}}{Q_m^2} \tag{2.17}$$

where

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

Thus, the equation of the system curve for open loop piping system becomes:

$$\Delta P = \Delta P_{static,m} + \frac{\Delta P_m - \Delta P_{static,m}}{Q_m^2} Q^2 \tag{2.18}$$

Eq. (2.18) represents the system curve for the open loop condenser water piping system. For a new piping system, static pressure head due to the height of open section of the cooling tower and the total pressure drop for the piping system corresponding to the design flow rate of water need to be calculated and then the system curve can be determined following the same procedure elaborated above. Same as the closed loop piping systems, if none of the piping system parameters such as pipe size, piping geometry, fittings, opening of the valves, height of cooling tower etc. are changed; the equation and profile of the system curve will remain unchanged. The total pressure drop of the piping system will change from ΔP_m to ΔP_1 if the water flow rate through the piping system is changed from Q_m to Q_1. The system pressure drop ΔP_1 for the water flow rate of Q_1 can be calculated using Eq. (2.18) as:

$$\Delta P_1 = \Delta P_{static,m} + \frac{\Delta P_m - \Delta P_{static,m}}{Q_m^2} Q_1^2 \tag{2.19}$$

If any of the piping system parameters such as the opening of the flow modulating valve is changed, the system curve will move either to the left or right as shown in Figure-2.8 and the equation of the new system curve needs to be redeveloped following the steps discussed above.

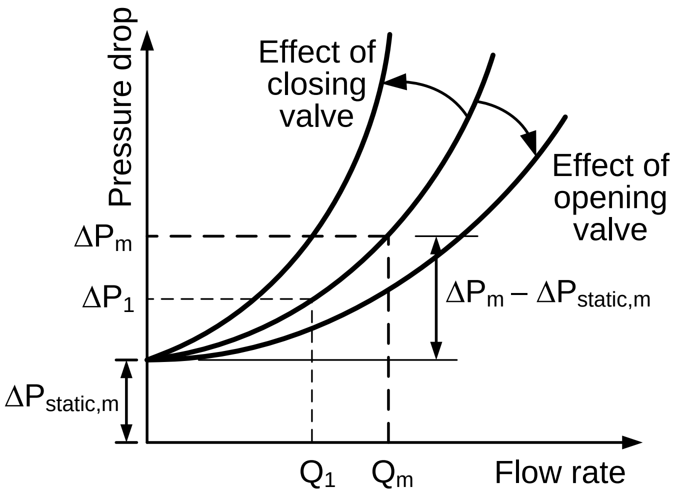


Figure 2.8 Changes of the system pressure drop with the flow rate of water and the movement of system curve with the opening of the flow modulating valve

2.3 Pump Sizing

The steady and unit mass flow rate of fluid in a perfectly insulated piping system as shown in Figure-2.9 is governed by the first law of thermodynamics, which leads to the equation:

$$\frac{P_1}{\rho_1 g}+\frac{V_1^2}{2g}+Z_1+\frac{\dot{W}_{pump}}{g}=\frac{P_2}{\rho_2 g}+\frac{V_2^2}{2g}+Z_2+e_{loss} \tag{2.20}$$

where

P = static pressure, Pa

ρ = density of the flowing fluid, 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

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

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

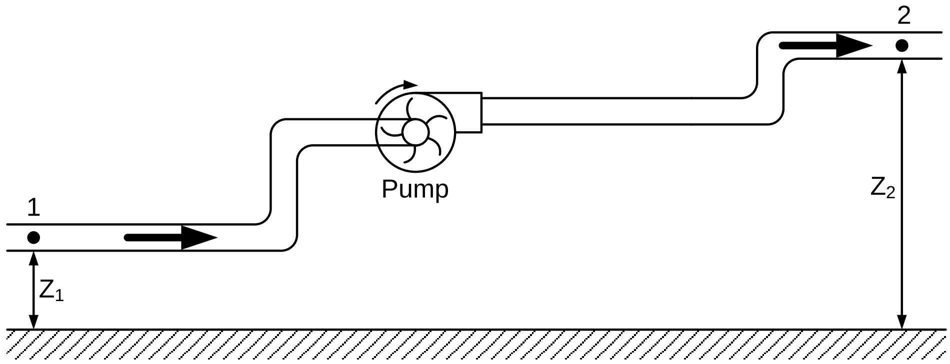


Figure 2.9 Steady flow of a fluid in a perfectly insulated piping system

Eq. (2.20) can be applied across the chilled or condenser water pumps to calculate the required power of the pumps for the unit mass flow rate of water. Applying Eq. (2.20) across the inlet and outlet of the chilled water pump as shown in Figure-2.10 gives:

$$\frac{P_{in}}{\rho g}+\frac{V_{in}^2}{2g}+Z_{in}+\frac{\dot{W}_{pump}}{g}=\frac{P_{out}}{\rho g}+\frac{V_{out}^2}{2g}+Z_{out}+e_{loss} \tag{2.21}$$

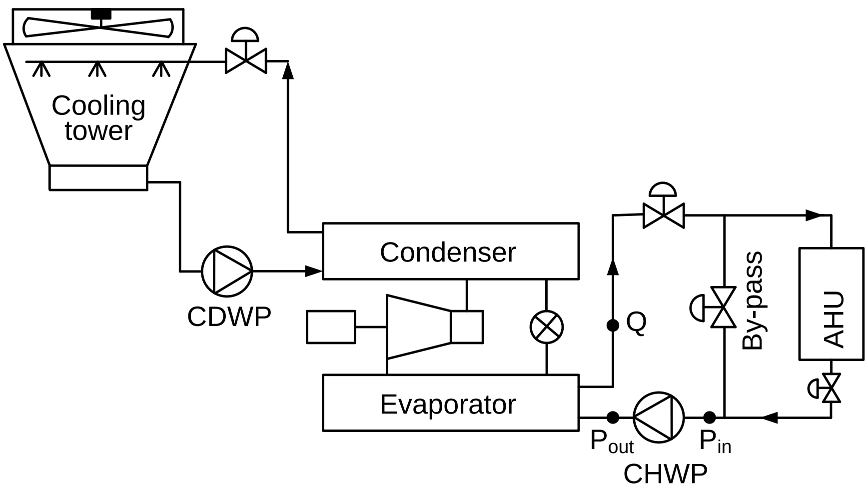


Figure 2.10 Central air-conditioning chilled water system

Eq. (2.21) can be simplified based on the following assumptions:

i. $V_{in} = V_{out}$ as the diameter of the pipe at the inlet and outlet of the pump is the same

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

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

Simplified form of Eq. (2.21) becomes:

$$\dot{W}_{pump} = \frac{P_{out} - P_{in}}{\rho} \tag{2.22}$$

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

$$W_{pump} = \frac{m(P_{out} - P_{in})}{\rho} \tag{2.23}$$

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

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

$$W_{pump} = Q(P_{out} - P_{in}) \tag{2.24}$$

where

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

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

m = mass flow rate of water, kg/s

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

W_{pump} = pump output power, Watts

If the input and output pressures are expressed in kPa, the unit of pump output power expressed in Eq. (2.23) and (2.24) will be kW.

Pumps are connected to the motor using different types of coupling systems such as flange, universal coupling, gear box etc. as shown in Figure-2.11. If the efficiency of the pump is η_{pump}, input power to the pump can be calculated as:

$$W_{pump,input} = \frac{Q(P_{out} - P_{in})}{\eta_{pump}} \tag{2.25}$$

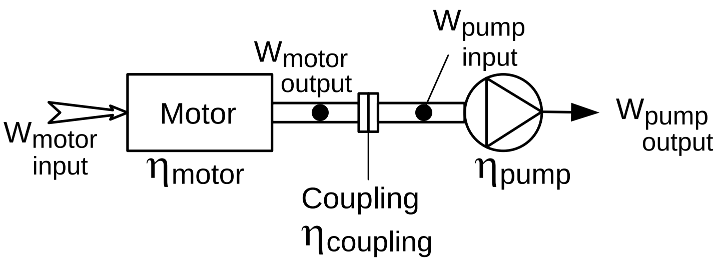


Figure 2.11 Coupling system of pump and motor

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

$$W_{motor,input} = \frac{Q(P_{out} - P_{in})}{\eta_{motor}\eta_{coupling}\eta_{pump}} \quad (2.26)$$

where

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

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

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

η_{pump} = efficiency of pump, dimensionless

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

η_{motor} = efficiency of motor, dimensionless

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

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

2.4 Pump Characteristic Curves

The relationship between the flow rate of fluid and the pressure developed by a pump is known as pump characteristic curve or pump curve. A pump characteristic curve shows the variation of the developed pressure at a particular operating speed as its discharge is throttled from zero to the full flow. Pump manufacturers usually include the variation of pump input power and efficiency in the pump characteristic curve as shown in Figure-2.12. Pump curve (pressure vs. flow rate) could be flat or steep. Pumps with flat curve are suitable for variable flow closed loop systems such as chilled water loop of central air-conditioning chilled water system. However, carefully selected pumps with steep curve are suitable for constant flow open loop systems such as condenser water loop of central air-conditioning chilled water system.

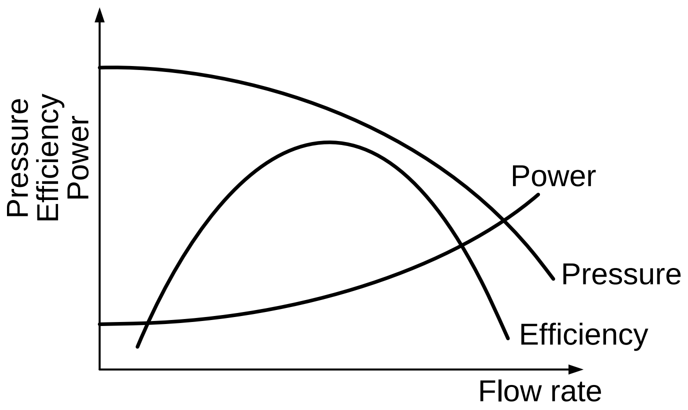


Figure 2.12 Pump input power, efficiency and characteristic curves

Pump characteristic curve moves downward if the diameter or speed of the impeller is reduced as shown in Figure-2.13.

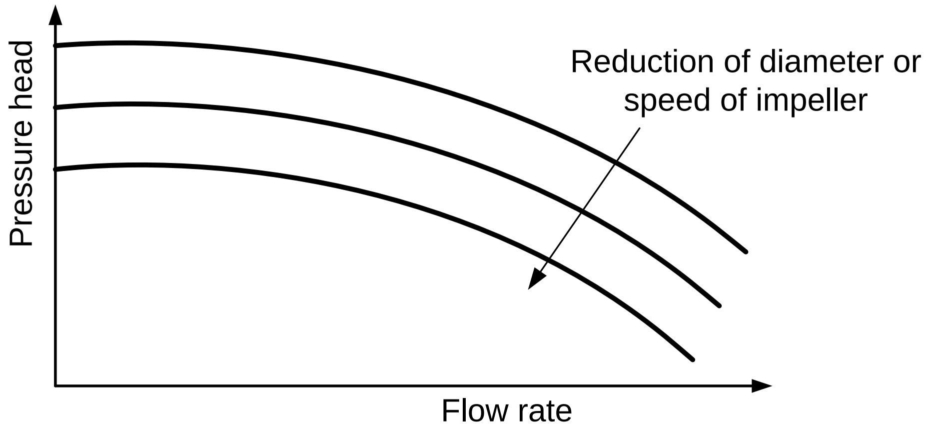


Figure 2.13 Pump characteristic curves for different diameters or speeds of impeller

Pump performance is most commonly presented by the manufacturers in the form of a set of curves of several different impellers at a particular speed as shown in Figure-2.14. If the pump is operated at different speeds, for each speed a different set of curves is given for each impeller diameter.

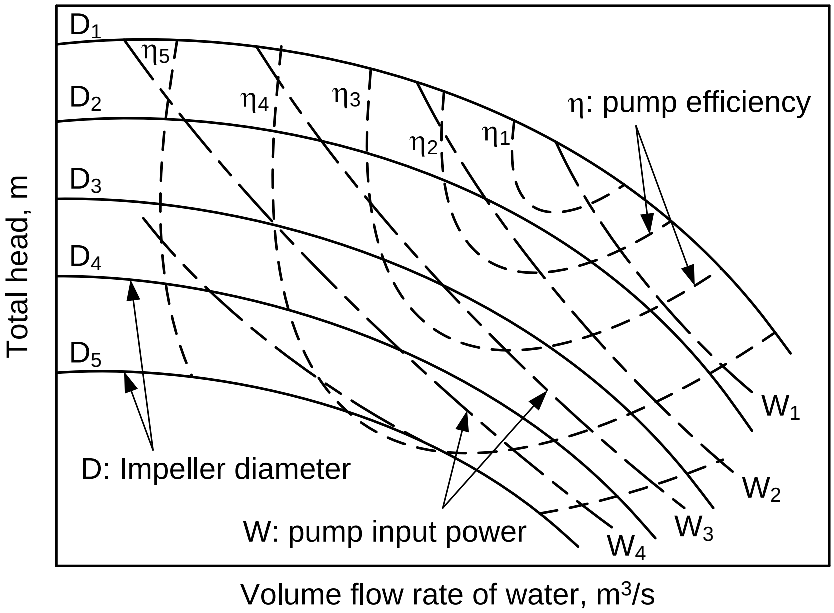


Figure 2.14 Pump performance curves

2.5 Pump Operating Point

When a pump is selected for a particular application, the intersection of the system curve and the pump curve represents the operating point. Figure-2.15 shows the system curve of a chilled water closed loop piping system, characteristic curve of the chilled water pump and the operating point. The pump will circulate water at a rate of Q_1 with the pressure head of ΔP_1, which is equal to the corresponding total pressure drop of the chilled water piping system. Intersections of the vertical line from flow rate of Q_1 with the efficiency curve and power curve represent the corresponding efficiency η_1 and input power of the pump W_1 respectively.

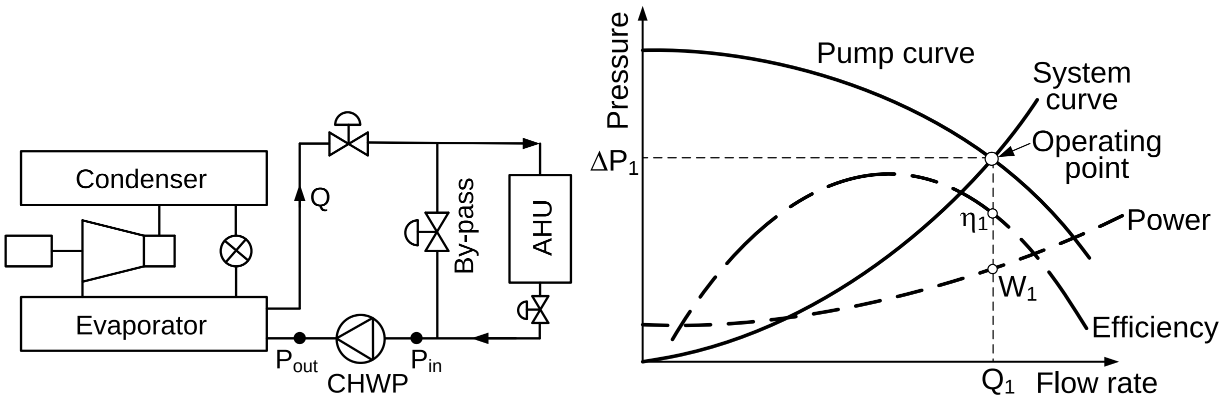


Figure 2.15 Pump characteristic curves, system curve and operating point

Similarly, if the system curve is drawn on the pump performance curves (Figure-2.14), the flow rates and the corresponding pressure heads to be generated by the pumps of different impeller diameters are shown in Figure-2.16. If impeller diameter of D_2 is selected, the pump will deliver water at a rate of Q_1 and will generate total pressure head of ΔP_1. Pump operating efficiency will be between η_1 and η_2 and pump power consumption will be between W_1 and W_2 (Figure-2.16). The values of the efficiency and pump power consumption could be estimated using interpolation. However, if impeller diameter of D_3 is selected, the pump will deliver water at a rate of Q_2 and will generate total pressure head of ΔP_2. Appropriate size of impeller is selected based on the design flow rate and corresponding total pressure drop of the piping system.

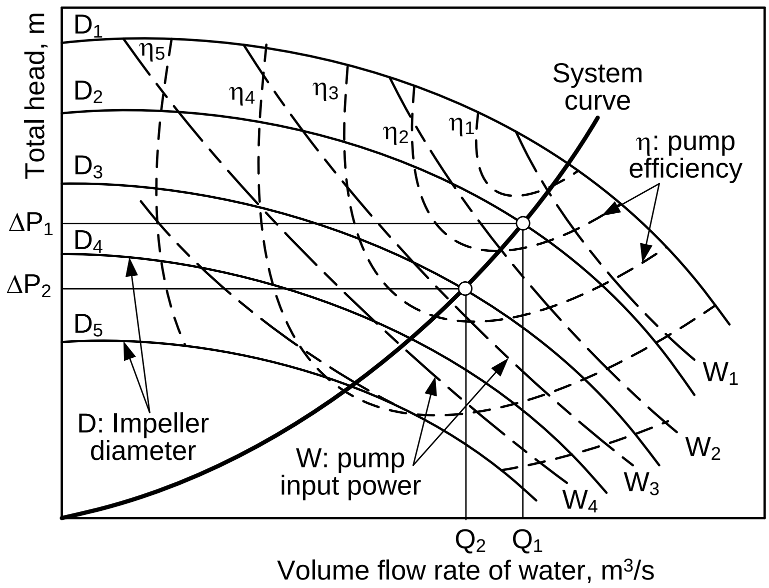


Figure 2.16 Pump performance curve, system curve and operating points

Effect of Partially Closing of Valve: Figure-2.17 shows that the system curve intersects with the pump curve at point-1 when the flow modulating valve of the piping system is fully open. The pump circulates water at a rate of Q_1 and the corresponding total pressure drop of the piping system is ΔP_1. Under this operating condition, the power consumption of the pump is:

$$W_{pump,input} = \frac{Q_1 \Delta P_1}{\eta_{pump}} \tag{2.27}$$

If the flow rate of water Q_1 is higher than the design value, the flow rate of water can be reduced by partially closing the flow modulating valve. Figure-2.5 shows that the system curve moves to the left if the flow modulating valve is partially closed. The system curve as well as the operating point-1 of Figure-2.17 will therefore move to the left at point-2 due to the partial closing of the flow modulating valve. Pump curve will not change its position as the speed of the pumps is not changed. The flow rate of water at point-2 will reduce from Q_1 to Q_2. However, the total pressure drop of the piping system will increase from ΔP_1 to ΔP_2 as shown in Figure-2.17. The power consumption of the pump at operating point-2 will be:

$$W_{pump,input} = \frac{Q_2 \Delta P_2}{\eta_{pump}} \tag{2.27}$$

As the flow rate of water decreases but the total pressure drop of the piping system increases due to the closing of the flow modulating valve, overall power consumption of the pump remains almost the same and no significant energy saving is achieved due to the reduction of the flow rate from Q_1 to Q_2.

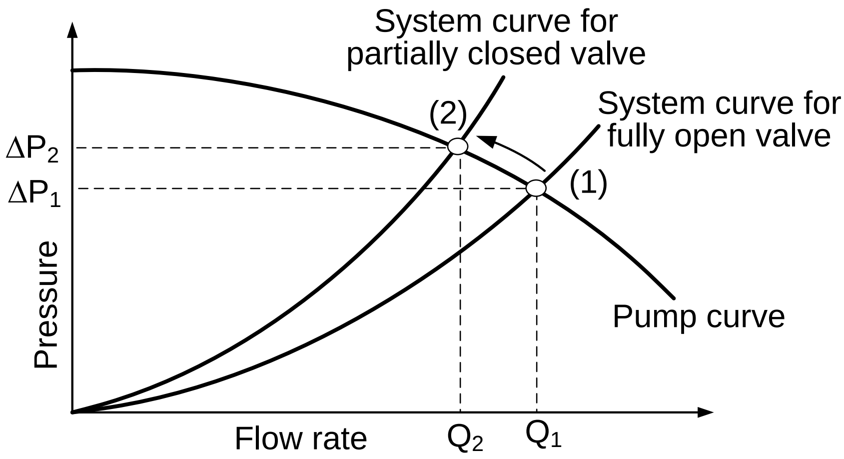


Figure 2.17 Effect of closing flow modulating valve of piping system

Effect of Pump Speed Reduction: If the flow rate of water Q_1 shown in Figure-2.17 is higher than the design value, the flow rate of water can also be reduced by reducing the speed of the impeller of the pump. As presented in Figure-2.13, the pump curve moves downward if the speed of the impeller of pump is reduced. The pump curve as well as the operating point-1 of Figure-2.18 will therefore move down to point-3 due the reduction of the speed of impeller. In this case, system curve will not change its position as the opening of the valve is not changed. The flow rate of water at point-3 will reduce from Q_1 to Q_2. At the same time, the total pressure drop of the piping system will decrease from ΔP_1 to ΔP_3 as shown in Figure-2.18. The power consumption of the pump at operating point-3 will be:

$$W_{pump,input} = \frac{Q_2 \Delta P_3}{\eta_{pump}} \tag{2.28}$$

As the flow rate of water as well as the total pressure drop of the piping system decrease due to the reduction of the speed of impeller, overall power consumption of the pump will drop significantly.

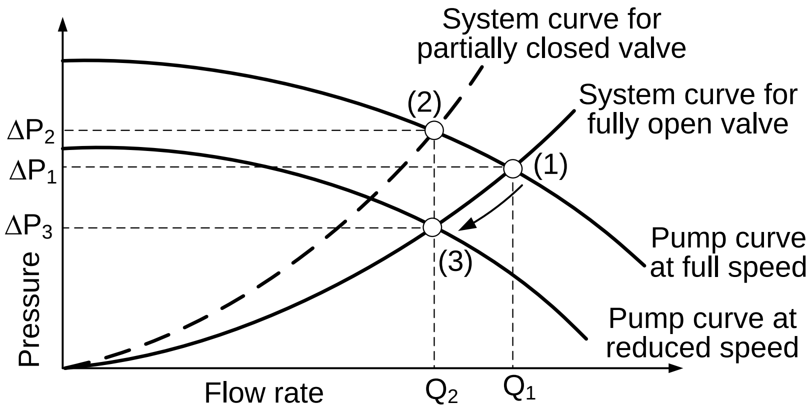


Figure 2.18 Effect of reducing pump speed of piping system

Pumps in Parallel and Series: Pumps are often connected in parallel and sometimes in series as shown in Figure-2.19 to accommodate variable flow of water and total pressure head requirements of the piping system or to provide redundancy in case of any pump failure. Variable speed drives (VSDs) can also be used in conjunction with parallel or series pump configuration to provide even more flexibility in operation. Capacity of the pumps could be the same or different.

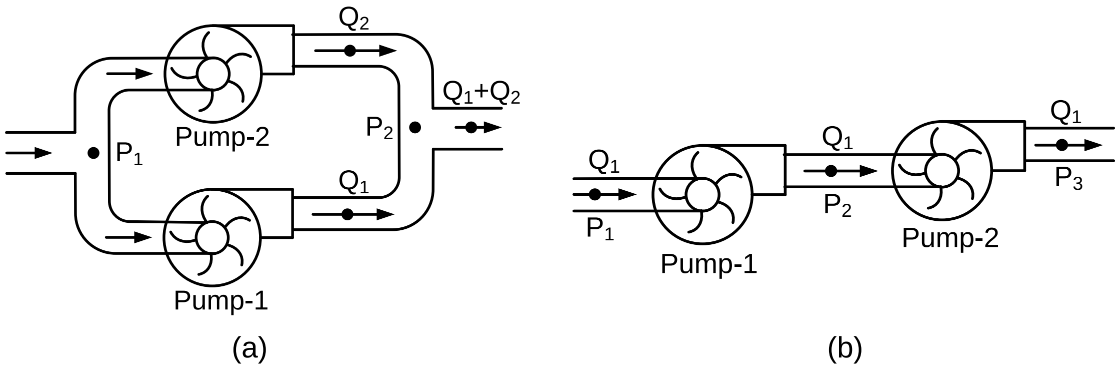
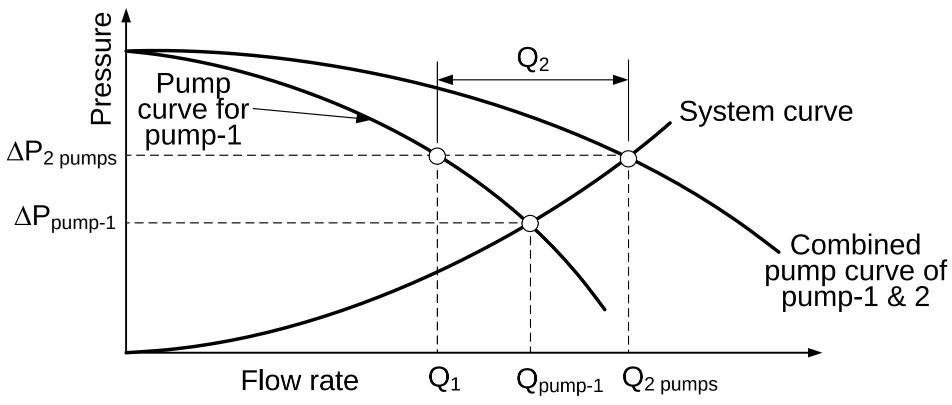


Figure 2.19 Configuration of pumps (a) Pumps in parallel and (b) Pumps in series

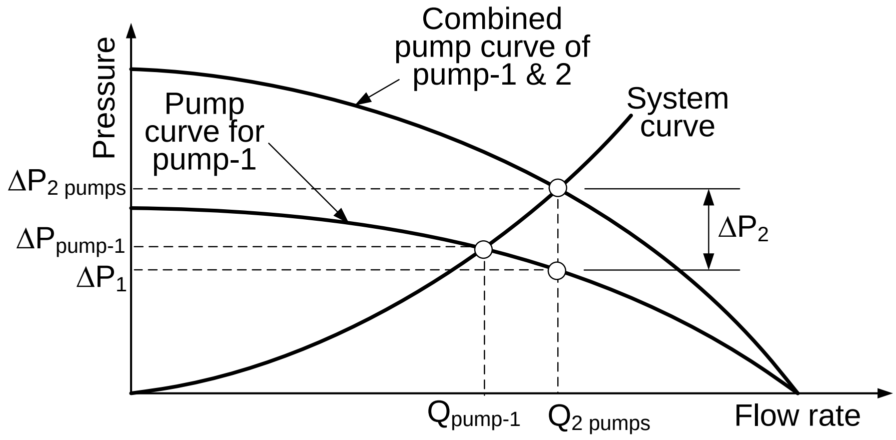
If pumps are connected in parallel configuration between header pipes of pressure P_1 and P_2 as shown in Figure-2.19 (a), each pump will generate equal pressure head of magnitude (P_2 - P_1). Total flow rate of water will be equal to the summation of flow of operating pumps. Therefore, parallel configuration is suitable for high flow rate applications. On the other hand, if pumps are connected in series configuration, total flow rate of water will be equal to the flow rate of each pump. However, pressure of the flowing fluid will increase after each pump. For the series configuration shown in Figure-2.19 (b), the pressure head generated by pump-1 and pump-2 will be (P_2 - P_1) and (P_3 - P_2) respectively. Total pressure head generation by the system will be (P_3 - P_1). Hence, series configuration is suitable for piping systems of high pressure loss.

Interactions of the pump curve and system curve for the parallel and series pumping configurations are shown in Figure-2.20. For the parallel configuration, when two pumps are operated (Figure-2.19.a), total flow rate of water through the piping system becomes $Q_{2\ pumps}$, which is equal to (Q_1 + Q_2) and the corresponding total pressure loss of the piping system reaches to $\Delta P_{2\ pumps}$ as presented in Figure-2.20 (a). To overcome the total pressure loss of the piping system, both pumps will generate equal pressure head of $\Delta P_{2\ pumps}$, which is equal to (P_2 - P_1) as shown in Figure-2.19 (a). If only pump-1 is operated, flow rate of water through the piping system will drop to $Q_{pump\text{-}1}$ and the corresponding pressure loss of the piping system will drop as well to $\Delta P_{pump\text{-}1}$. Note that the flow rate of pump-1 is increased from Q_1 to $Q_{pump\text{-}1}$ when only pump-1 is operated due to the reduction of pressure loss of the piping system (equals to required pump head) from $\Delta P_{2\ pumps}$ to $\Delta P_{pump\text{-}1}$.

Similarly, for the series configuration, when two pumps are operated (Figure-2.19.b), flow rate of water through each pump is $Q_{2\ pumps}$ (= Q_1) as shown in Figure-2.20 (b) and the total pressure head generated by the pumps is $\Delta P_{2\ pumps}$, which is equal to (P_3 - P_1). Pressure head generated by pump-1 and pump-2 are ΔP_1 and ΔP_2, which are equal to (P_2 - P_1) and (P_3 - P_2) respectively. If only pump-1 is operated, flow rate of water through the piping system will drop to $Q_{pump\text{-}1}$ but the generated head by pump-1 will increase to $\Delta P_{pump\text{-}1}$. Note that the pressure head generated by pump-1 is increased from ΔP_1 to $\Delta P_{pump\text{-}1}$ when only pump-1 is operated due to the reduction of flow rate from $Q_{2\ pumps}$ to $Q_{pump\text{-}1}$.



(a)



(b)

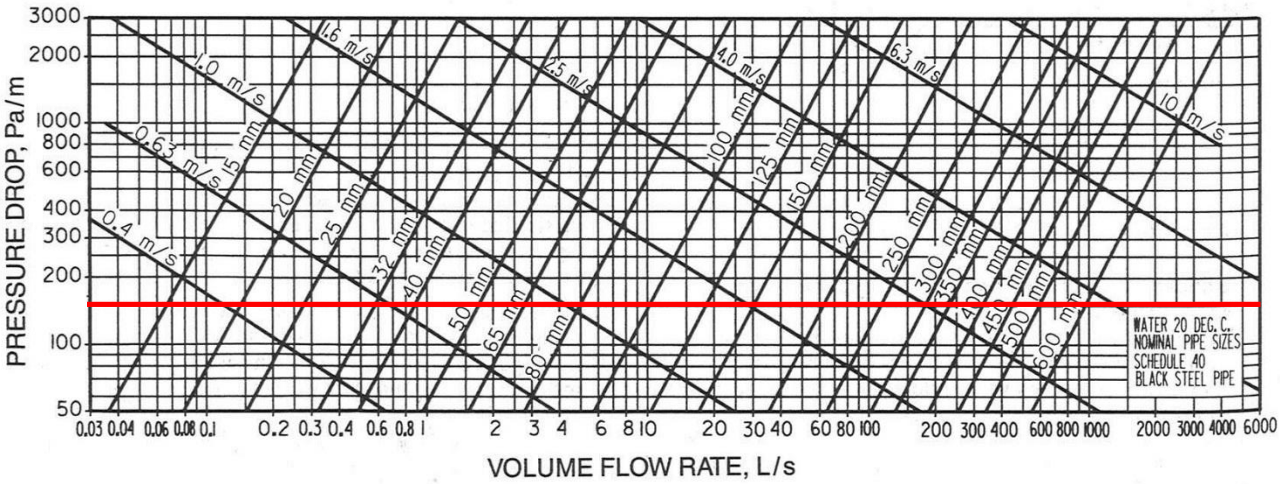
Figure 2.20 System and pump characteristic curves (a) pumps are connected in parallel and (b) pumps are connected in series

2.6 Losses in Piping Systems

Head losses in piping systems are broadly divided into two groups:

i. Friction loss and
ii. Dynamic loss

Friction Loss: Head loss in the straight pipe due to the friction between the flowing fluid and the wall of the pipe is known as friction loss. Friction loss mainly depends on (a) surface roughness of internal surface of the pipe, (b) pipe length, (c) pipe diameter and (d) volume flow rate of water. Surface roughness depends on the piping material. Friction loss can be calculated using Darcy-Weisbach Eq. (2.1). To facilitate the computation of friction loss and the actual pipe sizing, friction loss chart for pipes of different materials have been developed. Friction loss chart for black steel pipe based on 20^{o}C is shown in Figure-2.21. Frictional head loss per unit length of the pipe can be obtained directly from the chart corresponding to the flow rate of water and nominal pipe size. Similarly, when the frictional head loss and flow rate of water are specified, a pipe size and water flow velocity may conveniently be obtained using the chart. It is obvious from the chart that if a pipe of larger diameter is selected for a specific flow rate of water, pressure loss per unit length of pipe and consequently pump power consumption will drop. However, piping cost will increase due to the selection of pipe of larger diameter. Usually, friction loss of about 150 Pa/m is used to select the pipe diameter.



(Source: 1989 ASHRAE Handbook Fundamentals)

Figure 2.21 Friction loss chart for black steel pipe

Dynamic Losses: Head losses due to the resistance applied to the flowing fluid by different fittings of the pipe such as elbow, tee-joint, valve, strainer, sensors, exchanger coils etc. are known as the dynamic losses. These losses are calculated using Eq. (2.4). The loss coefficient C_o use in Eq. (2.4) depends on the type and size of the fittings. A list of the loss coefficient C_o for few selected fittings is presented in Table-2.1. The loss coefficient for different types of fittings as shown in Table 2.1 is significantly different. 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 pump power consumption.

Table 2.1 Loss coefficient C_o for different fittings at fully open condition

Nominal Pipe Dia., mm	90° Elbow Standard	90° Elbow Long	45° Elbow Long	Return Bend Standard	Return Bend Long-Radius	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve
25	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	-	4.8	2.0
32	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	-	3.7	2.0
40	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	-	3.0	2.0
50	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
65	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
80	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
100	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
150	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
200	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
250	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
300	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

(Source: 1989 ASHRAE Handbook Fundamentals)

Example 2.1

Cooling water is supplied at a rate of 500 CMH from a cooling tower to a heat exchanger as shown in Figure-2.22. Pressure loss across the heat exchanger is 7 m of water. The diameter and total length of the pipe are 250 mm and 150 m respectively. The piping system consists of 6 nos. of fully open gate valves, 2 nos. of strainers and 20 nos. of 90^o standard elbows. Loss coefficient for the strainer is 8.0. The efficiencies of the pump, coupling and motor are 0.6, 0.98 and 0.8 respectively. Estimate the total head and input power to the motor of the pump to maintain the flow. Recommend design modifications and estimate potential annual energy savings. Make the necessary assumptions.

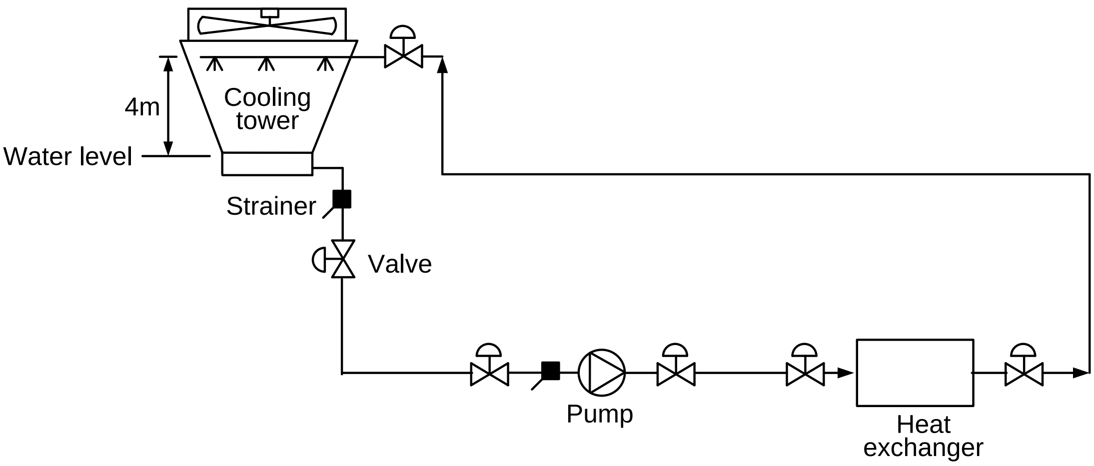


Figure 2.22 Cooling system of heat exchanger

Solution:

Water flow rate Q = 500 CMH

= 500/3600 = 0.139 m^3/s

= 0.139 x 1000 = 139 L/s

Diameter of pipe D = 250 mm = 0.25 m

Cross sectional area of pipe $A = \frac{\pi D^2}{4} = \frac{\pi (0.25)^2}{4} = 0.049 m^2$

Water flow velocity V = Q/A = 0.139/0.049 = 2.837 m/s

From friction loss chart (Figure-2.21), for Q = 139 L/s and D = 250 mm, friction loss per unit length of pipe ΔP = 220 Pa/m

Length of pipe L = 150 m

(a) Total friction loss $\Delta P_{friction}$ = ΔP x L = 220 x 150 = 33000 Pa = **33 kPa**

(b) Head loss for 6 nos. of fully open gate valves, $\Delta P_{valve} = 6xC_o\left(\frac{\rho V^2}{2}\right)$

From Table 2.1, for fully open gate valve of nominal diameter 250 mm, C_o = 0.06

Therefore, $\Delta P_{valve} = 6x0.06\left(\frac{1000x2.837^2}{2}\right) = 0.36x4024.28 = 1448.7Pa$ = **1.45 kPa**

(c) Head loss for 2 nos. of strainers, $\Delta P_{strainer} = 2xC_o\left(\frac{\rho V^2}{2}\right)$

Given loss coefficient for the strainer C_o = 8.0

Therefore, $\Delta P_{strainer} = 2x8.0\left(\frac{1000x2.837^2}{2}\right) = 64388Pa$ = **64.4 kPa**

(d) Head loss for 20 nos. of 90° standard elbows, $\Delta P_{elbow} = 20xC_o\left(\frac{\rho V^2}{2}\right)$

From Table 2.1, for 90° standard elbows of nominal diameter 250 mm, C_o = 0.25

Therefore, $\Delta P_{elbow} = 20x0.25\left(\frac{1000x2.837^2}{2}\right) = 20121Pa$ = **20.1 kPa**

(e) Head loss across the heat exchanger $\Delta P_{exchanger}$ = 7 m of water
= Hρg
= 7x1000x9.81
= 68670 Pa = **68.7 kPa**

(f) Static head across cooling tower ΔP_{CT} = 4 m of water
= Hρg
= 4x1000x9.81
= 39240 Pa = **39.2 kPa**

Pump needs to generate total head

$\Delta P_{total} = \Delta P_{friction} + \Delta P_{valve} + \Delta P_{strainer} + \Delta P_{elbow} + \Delta P_{exchanger} + \Delta P_{CT}$
= 33 + 1.45 + 64.4 + 20.1 + 68.7 + 39.2 = 226.85 kPa

Given that efficiencies of the pump, coupling and motor are 0.6, 0.98 and 0.8 respectively.

Input power to the motor of the pump $W_{motor,input} = \frac{Q\Delta P_{total}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$

$$W_{motor,input} = \frac{0.139x226.85}{0.8x0.98x0.6} = 67kW$$

Recommend Design Modifications: Breakdown of existing total head shows that major head losses are in 2 nos. of strainers (64.4 kPa) and heat exchanger (68.7 kPa). 2 nos. of strainers are not necessary. 1 no. of strainer can be removed. As a result, head loss for the strainer will be $\Delta P_{strainer}$ = 64.4/2 = 32.2 kPa.

Head loss of existing heat exchanger is 7 m of water, which is quite high.

New heat exchanger of head loss of 4 m of water can be proposed.

Therefore, head loss of proposed new heat exchanger

$$\Delta P_{exchanger} = 68.7x4/7 = 39.3 \text{ kPa}$$

New pump and motor of efficiencies of 0.8 and 0.92 respectively can be proposed.

New total head after modification

$$\Delta P_{total} = \Delta P_{friction} + \Delta P_{valve} + \Delta P_{strainer} + \Delta P_{elbow} + \Delta P_{exchanger} + \Delta P_{CT}$$

$$= 33 + 1.45 + 32.2 + 20.1 + 39.3 + 39.2 = 165.25 \text{ kPa}$$

Input power to the new motor with the new pump

$$W_{motor,input} = \frac{Q\Delta P_{total}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$$

$$W_{motor,input} = \frac{0.139x165.25}{0.92x0.98x0.8} = 31.8kW$$

Percentage reduction of pump power = 100 x (67-31.8) / 67 = 52.5%

If the system is operated for 24 hrs per day and 365 days a year, annual energy savings

= (67-31.8) x 24 x 365 = 308,350 kWh/year

2.7 Affinity Laws

The performance of centrifugal pumps can be modified by changing the rotational speed or impeller diameter. The *affinity laws for pumps* relate the flow rate, pressure developed across the pump and shaft power of the pump to the new and old speeds or impeller diameters. Within a given pump casing / housing, effects of changing the pump rotational speed *N* and impeller diameter *D* are presented in Table 2.2

Table 2.2 Affinity laws for specific pump 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 2.2, for constant impeller diameter:

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

$$Q_2 = Q_1\left(\frac{N_2}{N_1}\right) \tag{2.29}$$

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

$$\Delta P_2 = \Delta P_1\left(\frac{N_2}{N_1}\right)^2 \tag{2.30}$$

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

$$W_2 = W_1\left(\frac{N_2}{N_1}\right)^3 \tag{2.31}$$

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

Efficiency of the pump at the old speed N_1 is

$$\eta_1 = \frac{Q_1 \Delta P_1}{W_1} \tag{2.32}$$

Efficiency of the pump at the new speed N_2 is

$$\eta_2 = \frac{Q_2 \Delta P_2}{W_2} \tag{2.33}$$

Combining Eqs. (2.32) and (2.33) gives:

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

$$\frac{\eta_2}{\eta_1} = Q_1\left(\frac{N_2}{N_1}\right)\Delta P_1\left(\frac{N_2}{N_1}\right)^2 \frac{1}{W_1}\left(\frac{N_1}{N_2}\right)^3 \frac{W_1}{Q_1 \Delta P_1} = 1 \tag{2.34}$$

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

2.8 Constant and Variable Flow Systems

Pump Oversizing: Pumps are sized to achieve the design flow rate of water while overcoming the total pressure drop of the piping systems which includes friction losses in the piping and dynamic losses by different fittings of the pipe such as elbow, tee-joint, valve, strainer etc. Losses are estimated using specification of components and published research data. Safety factors are added to the design calculation due to the uncertainty of the data and provision for possible changes during installation to suit site constraints. If high safety factor is used, the selected pumps will be oversized for the particular application. During operation, the oversized pump will have an actual system curve that is quite different from the design system curve. As a result, the actual flow rate of water will be higher than the required flow rate. Water flow rate can be reduced to the design value by artificially adding resistance in the piping system using the flow balancing valve or modulating the speed of pump impeller using a variable speed drive (VSD). A case of the oversizing of chilled water pump (CHWP) is shown in Figure-2.23 and the effect of reducing the flow rate of chilled water to the design flow using flow balancing valve and modulating the speed of pump using VSD is illustrated in the following example:

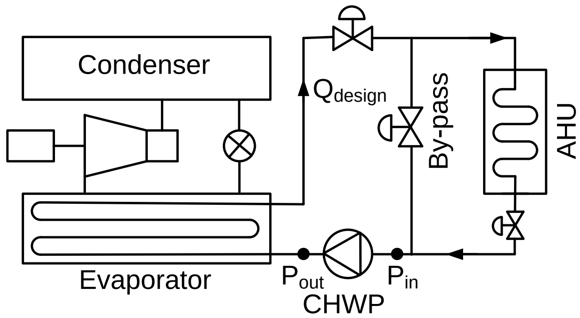


Figure 2.23 Chilled water pumping system

Assuming that at the design flow rate Q_{design}:

Head loss in evaporator of chiller	= 0.5 bar
Head loss in piping system (friction losses)	= 1.5 bar
Head loss in valves and fittings (dynamic losses)	= 0.4 bar
Head loss in AHU coils	= 0.5 bar

Total head loss = 0.5 + 1.5 + 0.4 + 0.5 = 2.9 bar

Total head loss with 25% safety factor = 2.9 x 1.25 = 3.7 bar

Calculated total head loss with 25% safety factor ΔP_{design} = 3.7 bar

The intersection of the pump curve and the calculated system curve represents the design operating point as shown in Figure-2.24.

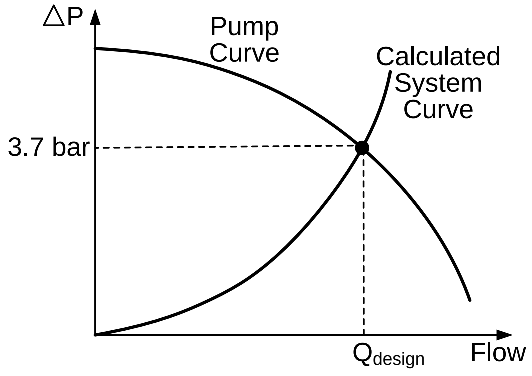


Figure 2.24 Design operating point

If a high safety factor is incorporated, the calculated total head loss (ΔP_{design} = 3.7 bar) will be greater than the actual total head loss and the selected pump will be oversized. Consequently, the operating point will move along the pump curve to the actual operating point where the actual system curve intersects with the pump curve, leading to a flow rate Q_{actual} higher than Q_{design} as shown in Figure-2.25. The required operating point is shown on the actual system curve corresponding to the design flow rate Q_{design}.

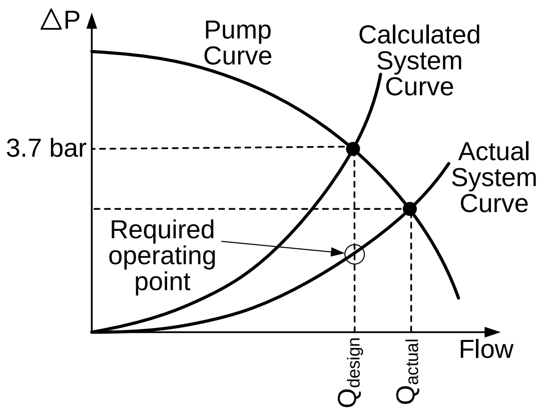


Figure 2.25 Actual operating point

Actual system curve can be moved back to the calculated system curve as shown in Figure-2.26 by adding sufficient resistance using the balancing valve to get the design flow. As a result, flow rate will be reduced from the actual Q_{actual} to the design Q_{design}. However, power consumption of the pump will be high due to the resistance introduced by the flow balancing valve which should not be accepted from an energy-efficiency point of view. Under this strategy, pump power consumption (represented by the hatched area ABCD) can be calculated as:

$$W_{pump,input} = \frac{Q_{design} \Delta P_{design}}{\eta_{pump}} \tag{2.35}$$

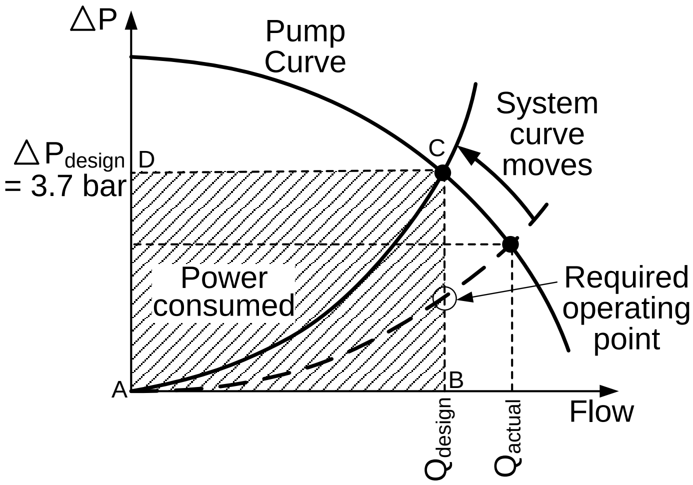


Figure 2.26 Pump power consumption if flow is reduced to the design flow using modulating valve

Alternatively, if the pump curve is moved downward to the required operating point as shown in Figure-2.27 by reducing the speed of the impeller using VSD, both the flow rate and the total head loss will drop from Q_{actual} to Q_{design} and ΔP_{design} to ΔP_{final} respectively. Under this strategy, pump power consumption (represented by the lower part of the hatched area ABEF) can be calculated as:

$$W_{pump,input} = \frac{Q_{design} \Delta P_{final}}{\eta_{pump}} \tag{2.36}$$

The reduction of pump power consumption due to the reduction of impeller speed in comparison to the partial closing of the flow balancing valve (represented by the upper part of the hatched area FECD of Figure-2.27) can be estimated as:

$$\Delta W_{pump,input} = \frac{Q_{design} \left(\Delta P_{design} - \Delta P_{final} \right)}{\eta_{pump}} \tag{2.37}$$

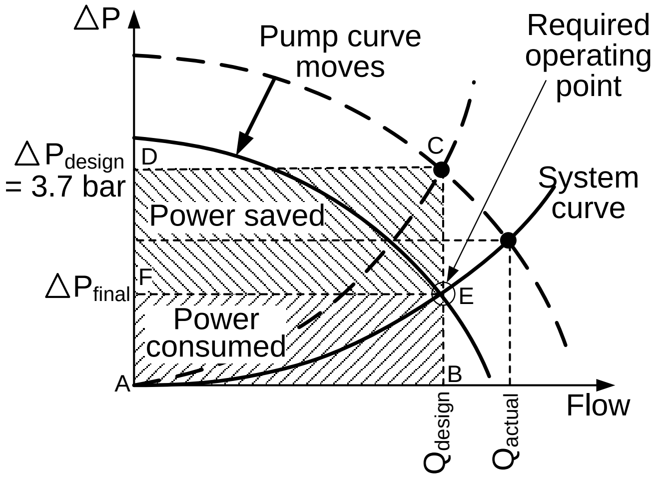


Figure 2.27 Pump power consumption if flow is reduced to the design flow by reducing the speed of impeller

Constant Speed Pumping System: Pumps of central air-conditioning systems are sized to satisfy the peak cooling load. However, the cooling load of the buildings and industrial processes varies with time. Usually, 2-way modulating valves are installed at the outlet of the cooling coil of each AHU to modulate the flow rate of chilled water through the AHU. Under part load condition, the 2-way modulating valves are partially closed to reduce the flow of chilled water through the cooling coils and thereby control the amount of cooling delivered by the coils as shown in Figure-2.28. If chilled water pumps are operated at constant speed, the operating point will move to the left along the pump curve as shown in Figure-2.29 due to the increase of the system pressure caused by partial closing of the flow modulating valves. Under part load conditions, the flow rate of chilled water will reduce but the system pressure will increase.

Power consumption of pumps at full load condition can be calculated as:

$$W_{pump, full\ load} = \frac{Q_{full\ load} \Delta P_{full\ load}}{\eta_{pump}} \tag{2.38}$$

Similarly, power consumption of pumps at part load condition can be determined as:

$$W_{pump, part\ load} = \frac{Q_{part\ load} \Delta P_{part\ load}}{\eta_{pump}} \tag{2.39}$$

If pumps are operated at constant speed, power consumption of pumps does not drop significantly under part load with respect to full load operation as chilled water flow rate decreases but the system pressure increases.

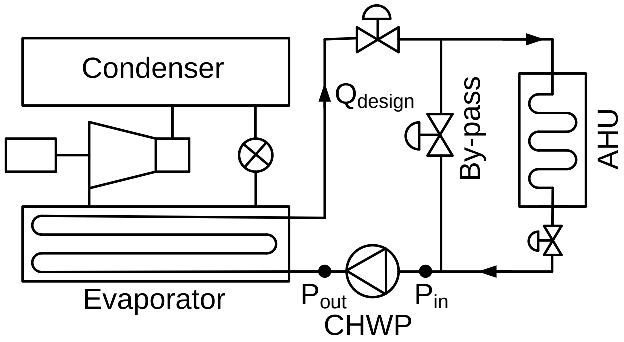


Figure 2.28 Constant speed chilled water pumping system

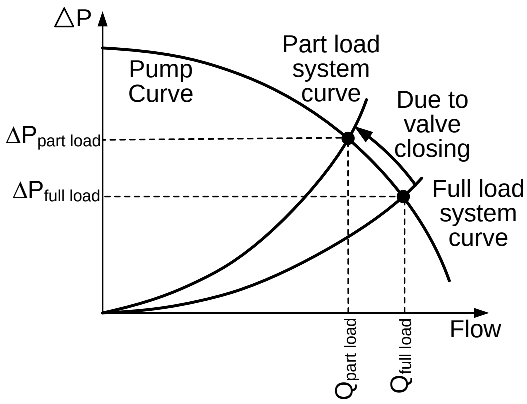


Figure 2.29 Operating points at full and part load conditions for constant speed pumps

Variable Speed Pumping System: If the speed of the pump is varied using variable speed drive (VSD) based on constant differential pressure (ΔP) as shown in Figure-2.30, the operating points at full and part load operations are presented in Figure-2.31. Under part load operation, the modulating valve located at the outlet of each AHU will be closed partially to restrict the flow of chilled water and reduce the cooling effect. As the modulating valve starts to close, the system pressure increases. The controller then reduces the speed of the pump to maintain the constant differential pressure (ΔP). This results in significant energy savings. If the differential pressure (ΔP) sensor is located in the plant room near the pump, the pressure difference across the pump will remain constant at full and part load operations. However, if the differential pressure (ΔP) sensor is located across the furthest critical AHU, the pressure difference across the pump will drop under part load operation due to the reduction of piping losses at low flow rate, which will result in further energy savings.

Power consumption of pumps at full load condition can be calculated as:

$$W_{pump,\, full\, load} = \frac{Q_{full\, load}\Delta P_{full\, load}}{\eta_{pump}} \tag{2.40}$$

Similarly, power consumption of pumps at part load condition can be determined as:

$$W_{pump,\, part\, load} = \frac{Q_{part\, load}\Delta P_{full\, load}}{\eta_{pump}} \tag{2.41}$$

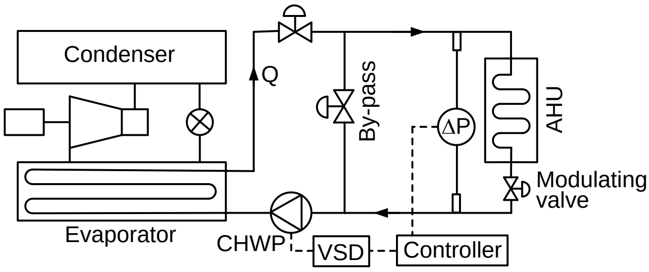


Figure 2.30 Differential pressure controlled variable speed chilled water pumping system

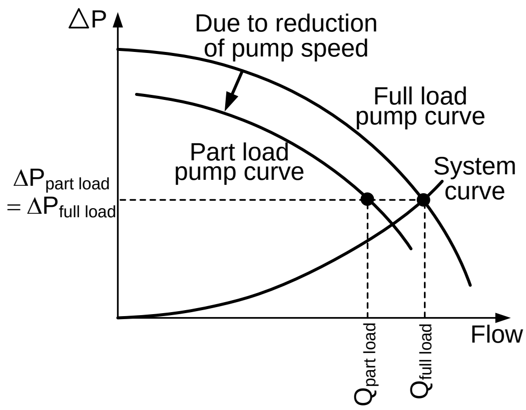


Figure 2.31 Operating points at full and part load conditions for variable speed pumps controlled based on constant differential pressure

If a chilled water system is designed to support the cooling load of relatively large open spaces (such as open space of retail mall), the speed of the chilled water pumps can be modulated based on chilled water return temperature as shown in Figure-2.32. For large open spaces, cooling load of each AHU may change in similar manner. Chilled water return temperature will drop below pre-set value if the cooling load of the spaces drops. The controller then reduces the speed of the pump to reduce the flow rate of chilled water and the corresponding cooling effect. There is no flow modulating valve for this system. As a result, the system curve remains unchanged. Pump operating point moves along the system curve with the change of the speed of pump as shown in Figure-2.33, resulting in a drop in both water flow rate and system pressure. Consequently, maximum energy savings under part load operation can be achieved. However, if the cooling load of the spaces does not follow the same pattern, comfort condition of each space may not be achieved.

Power consumption of pumps at full load condition can be determined as:

$$W_{pump,\, full\, load} = \frac{Q_{full\, load} \Delta P_{full\, load}}{\eta_{pump}} \tag{2.42}$$

Similarly, power consumption of pumps at part load condition can be calculated as:

$$W_{pump,\, part\, load} = \frac{Q_{part\, load} \Delta P_{part\, load}}{\eta_{pump}} \tag{2.43}$$

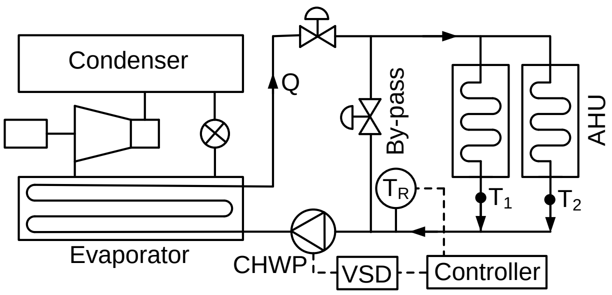


Figure 2.32 Differential pressure controlled variable speed chilled water pumping system

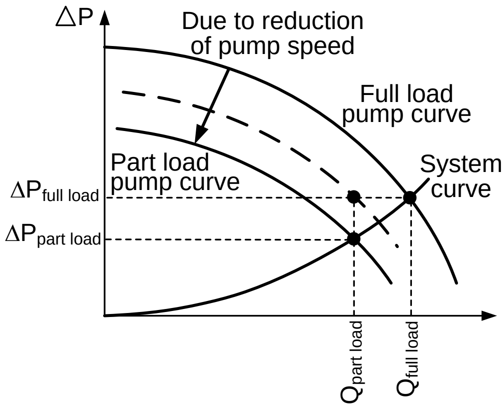


Figure 2.33 Operating points at full and part load conditions for variable speed pumps controlled based on chilled water return temperature

2.9 Optimisation Strategies for Pumping Systems

Pump Serving Multiple Processes: Often pumping systems are designed to supply water to different processes as shown in Figure-2.34. However, different processes normally have different operating schedules. Automatic isolation valve should be installed for each process and interlocked with the operation of individual processes. If a process is not in operation, the isolation valve of the process will close the line to avoid the wastage of water circulation through the process. However, power consumption of the constant speed pump will not drop remarkably. Due to the closing of the isolation valve, total water flow rate will decrease but the pressure at the discharge of the pump will increase. High pressure water will flow through the remaining operating processes.

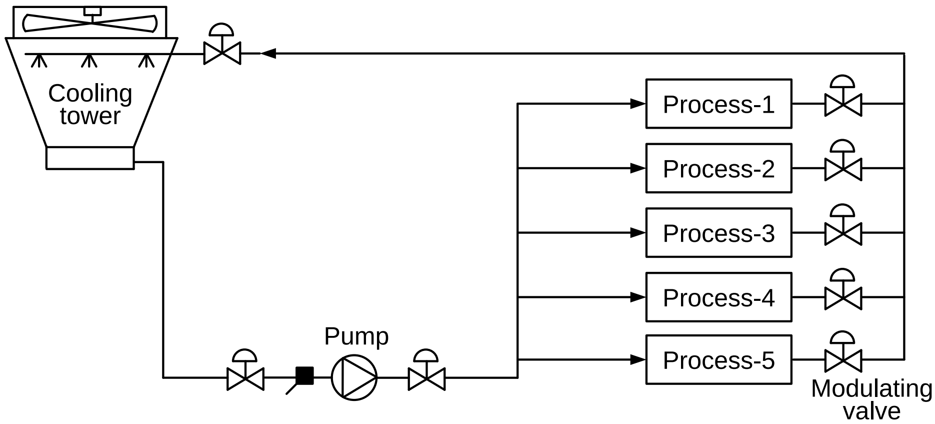


Figure 2.34 Pumping system serving multiple processes

To reduce the energy consumption of the pump, pressure sensor should be installed as shown in Figure-2.35 and the speed of the pump should be modulated to maintain the pre-set pressure. If a process is not in operation, the corresponding isolation valve will close the water flow line. As a result, the flow rate of water will decrease and the discharge pressure of the pump will increase. The developed high pressure will be sensed by the pressure sensor and feedback provided to the controller. The controller will modulate the speed of the pump to maintain the pre-set pressure resulting in significant savings in pumping energy.

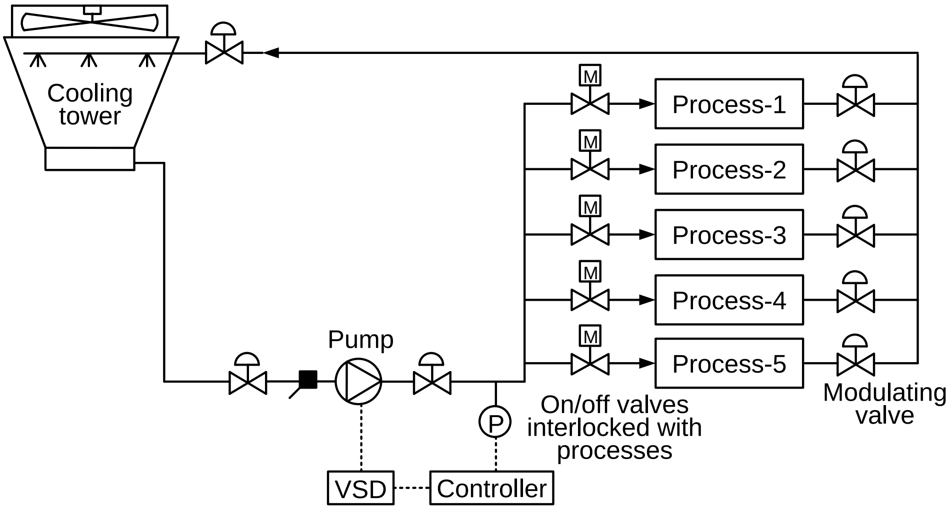


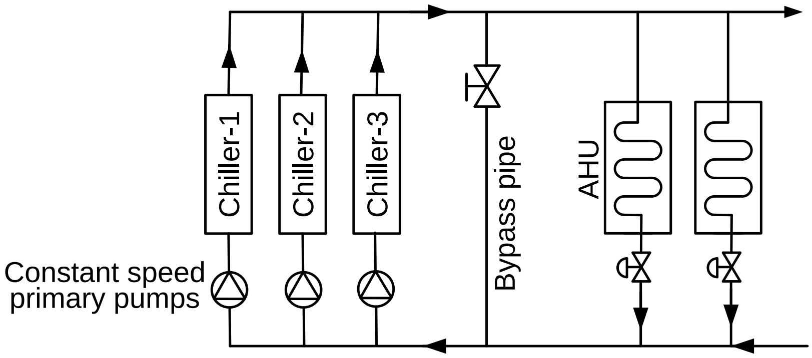
Figure 2.35 Control of pumping system serving multiple processes

Example:

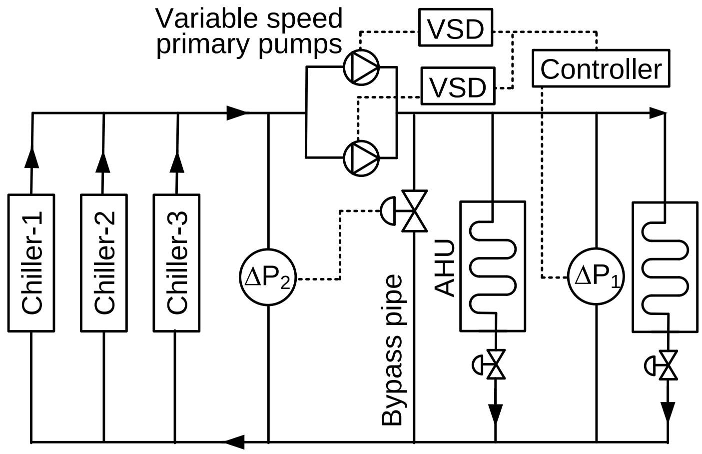
Cooling load of a building is served by two chillers of capacity 500 RT each. Cooling loads and corresponding periods are shown in Table 2.3. The chilled water flow rate for each 500 RT chiller at full load condition is 0.0756 m^3/s. When 2 nos. of 500 RT chillers are operating at full load, the pressure head for primary loop is 80 kPa and secondary loop is 200 kPa. Efficiency of pumps and motors are 65% and 90% respectively. Compare the pumping power consumed for three options (Figure-2.36): (a) Constant speed primary pumping system, (b) Variable speed primary pumping system and (c) Constant speed primary and variable speed secondary pumping system. Make the necessary assumptions.

Table 2.3 Cooling loads and corresponding periods

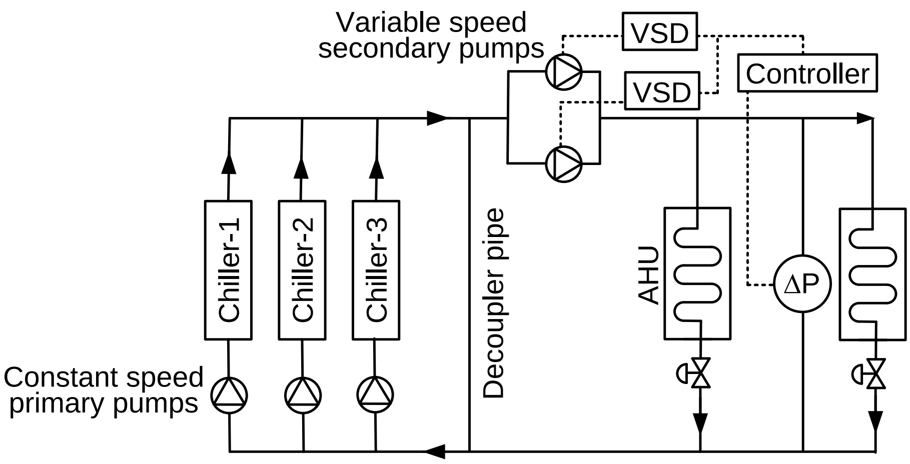
Loading	Building cooling load (RT)	Daily operating hours
100%	1000	3
90%	900	8
80%	800	4
70%	700	3
60%	600	3
50%	500	2
40%	400	1



Option-a: Constant speed primary pumping system



Option-b: Variable speed primary pumping system



Option-c: Constant speed primary and variable speed secondary pumping system

Figure 2.36 Three different options for pumping of chilled water

Solution:

Chilled water flow rate, Q = 0.0756 m^3/s per chiller

Pressure head for primary loop, $\Delta P_{primary}$ = 80 kPa

Pressure head for secondary loop, $\Delta P_{secondary}$ = 200 kPa

Total pressure head, $\Delta P_{total} = \Delta P_{primary} + \Delta P_{secondary}$ = 80 + 200 = 280 kPa

Efficiency of pump, η_{pump} = 0.65 and

Efficiency of motor, η_{motor} = 0.9

Option-a: Constant speed primary pumping system

Input power to motor of pump: $W_{motor,input} = \dfrac{Q\Delta P}{\eta_{motor}\eta_{coupling}\eta_{pump}}$

At full load, input power to motors of two pumps:

$$W_{motor,input} = \frac{Q\Delta P_{total}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$$

$$W_{motor,input} = \frac{2x0.0756x280}{0.65x1x0.9} = 72.4\,kW$$

Assuming that the modulating valves of secondary loop will be closed partially to regulate the flow of chilled water at part load conditions and pressure head of secondary loop will remain unchanged.

Input power to motors of one pump: $W_{motor,input} = \dfrac{0.0756x280}{0.65x1x0.9} = 36.2\,kW$

Option-b: Variable speed primary pumping system

At full load, input power to motors of two pumps = 36.2x2 = 72.4 kW

Assuming that chilled water flow of the secondary loop will mainly be modulated by VSDs under part load operations and closing of modulating valve is negligible.

At 90% loading, input power to motors of two pumps:

$$W_{90\%} = W_{full}\left(\frac{N_2}{N_1}\right)^3 = W_{full}\left(\frac{0.9N_1}{N_1}\right)^3 = W_{full}(0.9)^3 = 72.4(0.9)^3 = 52.8kW$$

At 80% loading, input power to motors of two pumps:

$$W_{80\%} = W_{full}(0.8)^3 = 72.4(0.8)^3 = 37.1kW$$

At 70% loading, input power to motors of two pumps:

$$W_{70\%} = W_{full}(0.7)^3 = 72.4(0.7)^3 = 24.8kW$$

At 60% loading, input power to motors of two pumps:

$$W_{60\%} = W_{full}(0.6)^3 = 72.4(0.6)^3 = 15.6kW$$

At cooling load of 500 RT, 1 no. of chiller and 1 no. of pump will be operated (1 no. of chiller and 1 no. of pump will be turned off)

Assuming negligible pressure loss in piping of primary loop, pressure loss across chiller will be 80 kPa as 100% water will flow through one chiller

Pressure loss in secondary loop:

$$\Delta P_{secondary,\ 50\%} = \Delta P_{secondary,\ 100\%}\left(\frac{N_2}{N_1}\right)^2 = \Delta P_{secondary,\ 100\%}\left(\frac{0.5N_1}{N_1}\right)^2$$

$$\Delta P_{secondary,\ 50\%} = 200(0.5)^2 = 50\ \ kPa$$

Total pressure loss: $\Delta P_{total} = \Delta P_{primary} + \Delta P_{secondary,\ 50\%}$ = 80 + 50 = 130 kPa

Input power to motor of one pump: $W_{motor,input} = \dfrac{Q\Delta P_{total}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$

$$W_{motor,input} = \frac{0.0756x130}{0.65x1x0.9} = 16.8\,kW$$

When building cooling load is 400 RT, chiller loading = 400/500 = 80%

Input power to motor of one pump at 80% loading:

$$W_{80\%} = 16.8(0.8)^3 = 8.6kW$$

Option-c: Constant speed primary and variable speed secondary pumping system

Input power to motor of one primary pump: $W_{motor,\,primary} = \dfrac{Q\Delta P_{primary}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$

$$W_{motor,\,primary} = \frac{0.0756x80}{0.65x1x0.9} = 13.3\,kW$$

Input power to motor of one secondary pump:

$$W_{motor,secondary} = \frac{Q\Delta P_{secondary}}{\eta_{motor}\eta_{coupling}\eta_{pump}}$$

$$W_{motor,secondary} = \frac{0.0756x200}{0.65x1x0.9} = 25.8\,kW$$

Input power to motor of 2 nos. of secondary pumps = 2x25.8 = 51.6 kW

At 90% loading, input power to motors of two pumps: $W_{90\%} = 51.6(0.9)^3 = 37.6kW$

At 80% loading, input power to motors of two pumps: $W_{80\%} = 51.6(0.8)^3 = 26.4kW$

At 70% loading, input power to motors of two pumps: $W_{70\%} = 51.6(0.7)^3 = 17.7kW$

At 60% loading, input power to motors of two pumps: $W_{60\%} = 51.6(0.6)^3 = 11.1kW$

At 50% loading, input power to motors of two pumps: $W_{50\%} = 51.6(0.5)^3 = 6.5kW$

At 40% loading, input power to motors of two pumps: $W_{40\%} = 51.6(0.4)^3 = 3.3kW$

Note:

i. Usually, input frequency of motor can safely be reduced from 50 Hz to 30Hz (meaning (50-30)/50 x 100% = 40% reduction in speed).

ii. Input frequency of motor can be reduced to 25 hz, if ambient temperature is low or provide additional cooling to motor

iii. Motor efficiency usually remains constant from 100% to about 40% loading. Motor efficiency usually drops drastically if motor loading is below 40%.

Table 2.4 Summary of pumps power consumption for option-a, b and c

Loading	Building cooling load (RT)	Daily operating hours	Pumping power (kW)				kWh consumption per day		
			Option-a	Option-b	Option-c		Option-a	Option-b	Option-c
					Primary	Secondary			
100%	1000	3	72.4	72.4	26.6	51.6	217.2	217.2	234.6
90%	900	8	72.4	52.8	26.6	37.6	579.2	422.4	513.6
80%	800	4	72.4	37.1	26.6	26.4	289.6	148.4	212
70%	700	3	72.4	24.8	26.6	17.7	217.2	74.4	132.9
60%	600	3	72.4	15.6	26.6	11.1	217.2	46.8	113.1
50%	500	2	36.2	16.8	13.3	6.5	72.4	33.6	39.6
40%	400	1	36.2	8.6	13.3	3.3	36.2	8.6	16.6
						Total	1629	951.4	1262.4

Remarks:

i. Power consumption for option-b (variable primary flow pumping; VSDs on primary pumps) is the lowest.

ii. Power consumption depends on a number of factors such as cooling load profile, operating hours, pressure drop for primary and secondary loop etc.

iii. Above analysis should be conducted for individual chilled water system and the appropriate pumping system should be selected for the chilled water system.

2.10 System Serving Multiple Buildings

If a chilled water system is designed to provide air-conditioning to multiple buildings that have different pressure heads and cooling loads, power consumption of the chilled water pumps can be reduced by using primary-secondary chilled water pumping system as shown in Figure-2.37. For example, building-A is a high-rise building with higher pressure head requirements compared to the low-rise building-B. Dedicated secondary chilled water pumping systems should be installed for building-A and B (Figure-2.37). The secondary pumps should be sized to satisfy the pressure head and chilled water flow requirements for each building.

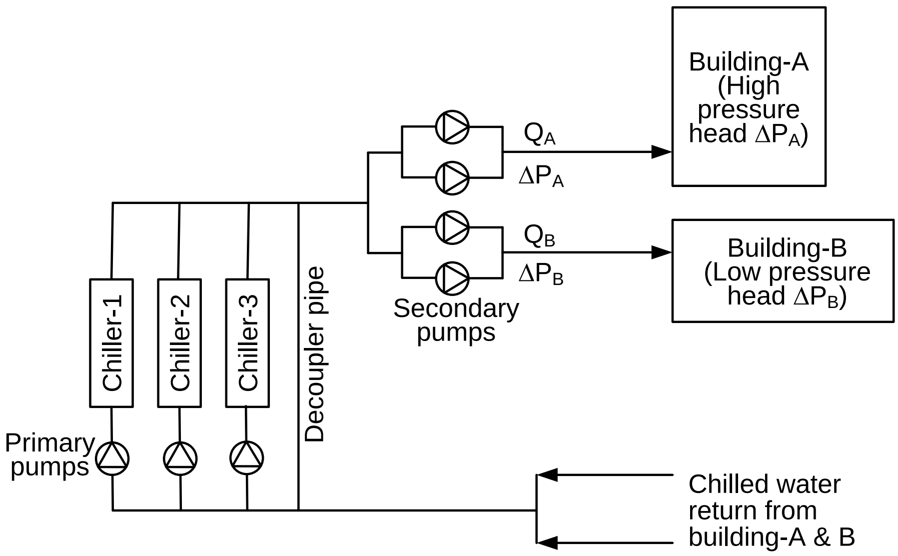


Figure 2.37 Dedicated secondary chilled water pumping systems for buildings having different pressure heads

Total power consumption of the secondary pumps can be calculated as:

$$W_{secondary\ \ pumps} = \frac{Q_A \Delta P_A}{\eta_{pump}} + \frac{Q_B \Delta P_B}{\eta_{pump}} \qquad (2.44)$$

where

$W_{secondary\ pumps}$ = input power to the motor of secondary pumps, kW

Q_A = volume flow rate of water in building-A, m^3/s

Q_B = volume flow rate of water in building-B, m^3/s

ΔP_A = pressure head of piping system for building-A, kPa

ΔP_B = pressure head of piping system for building-B, kPa

η_{pump} = efficiency of secondary pumps, dimensionless

However, if only primary chilled water pumps or primary-secondary pumps with combined secondary header as shown in Figure-2.38 is used, the pumps have to be sized to satisfy the maximum pressure head of building-A. As a result, the generated pressure head by the pumps will be higher than the required pressure for low-rise building-B. Balancing valve needs to be installed in the header pipe of building-B (Figure-2.38) and partially closed to artificially add resistance and restrict the flow of chilled water in building-B resulting in wastage of pumping energy.

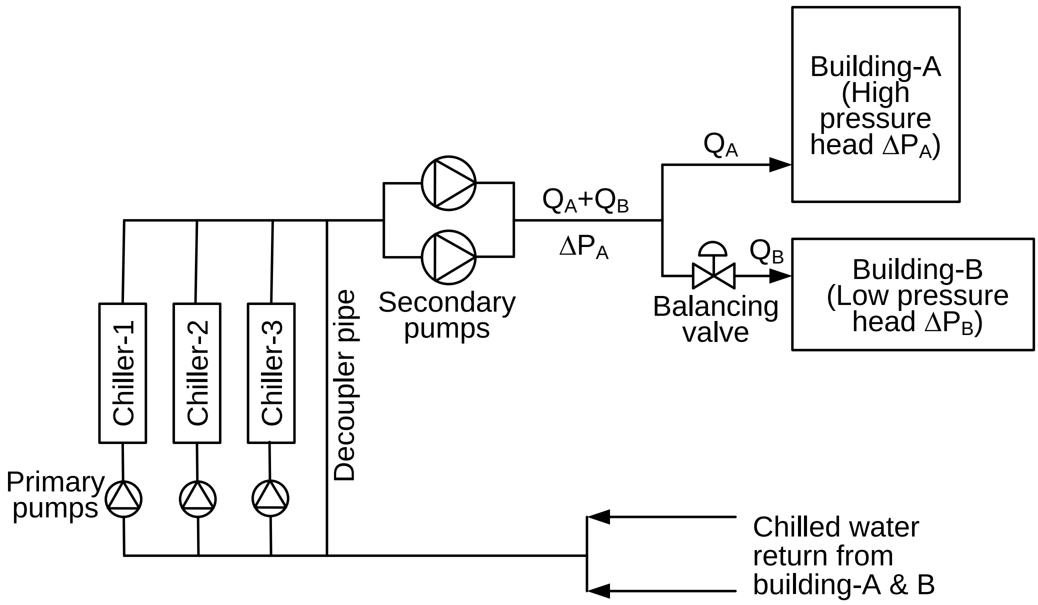


Figure 2.38 Secondary chilled water pumping systems with combined secondary header pipe

Power consumption of the secondary pumps can be calculated as:

$$W_{secondary\ \ pumps} = \frac{(Q_A + Q_B)\Delta P_A}{\eta_{pump}} \qquad (2.45)$$

Comparing Eqs. (2.44) and (2.45), the achievable power savings with the dedicated secondary chilled water pumping system (Figure-2.37) is:

$$\Delta W_{secondary\ \ pumps} = \frac{Q_B(\Delta P_A - \Delta P_B)}{\eta_{pump}} \qquad (2.46)$$

Usually, primary pumps are small and operated at constant speed to circulate water through the chillers. VSDs are installed on the secondary pumps since the size of the secondary pumps are bigger and it enables higher variations of chilled water flow with the change in the building cooling load without affecting the chiller performance. The speed of the pumps should not be reduced more than 40 percent to ensure sufficient cooling of the motors and lubrication of the pump seals.

2.11 Reset of Pump Set-Point

The efficiency of the chilled water pumping system can be improved further by varying the differential pressure set point used for controlling the VSD of chilled water pumps according to the cooling load. As the cooling load for each zone may not change by the same percentage with time, the opening of the chilled water flow modulating valves of different AHUs could be different. Building energy management system can be used to monitor the percentage opening of chilled water flow modulating valves of all AHUs and determine the value of maximum valve opening. The value of the maximum percentage of valve opening is then compared with the pre-set limit and adjusts the differential pressure set point using appropriate control strategy as shown in Figure-2.39. For example, the pre-set limit of valve opening in Figure-2.39 is 80% to 90%. If the maximum percentage of valve opening is less than 80%, differential pressure set point will be decreased by the controller while ensuring that enough chilled water is flowing through all other AHUs. On the other hand, if the maximum percentage of valve 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-2.40. The flow rate of water is varied mainly by adjusting the speed of the pumps resulting in further energy savings of pumping systems.

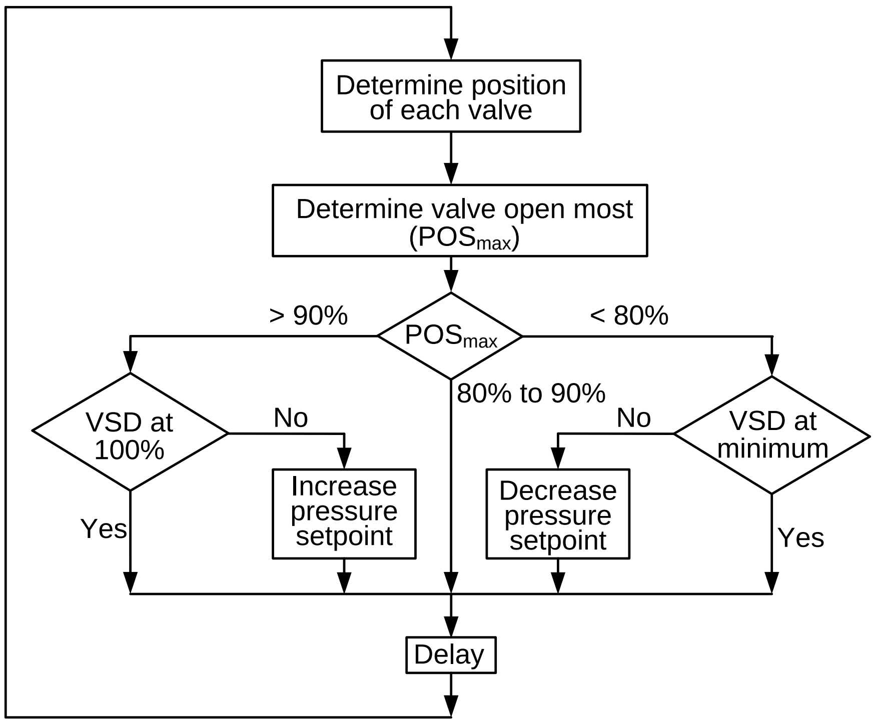


Figure 2.39 Control strategy of variable differential pressure set point for VSD

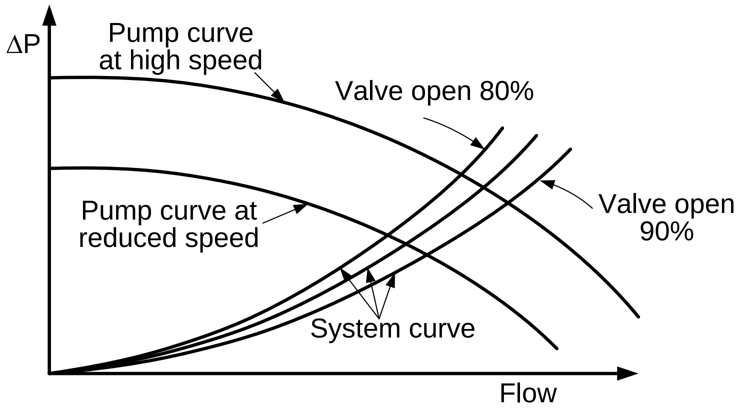


Figure 2.40 Movement of system and pump curves for pumping systems using variable differential pressure set point

2.12 Pump Performance and Operational Requirements Based on Singapore Standards

Chilled Water Pumps: Based on Singapore Standard SS553, maximum power consumption for the chilled water pumping system is 349 kW/(m^3/s). Following the standard of Air-Conditioning, Heating and Refrigeration Institute (AHRI), chilled water flow rate of 2.4 usgpm/RT is usually used for the design of chilled water system.

Therefore, energy efficiency (or specific power consumption) for the chilled water pumps:

$$349\,\frac{kW}{m^3/s}\,x\,\frac{2.4\,usgpm}{1\,RT}\,x\,\frac{0.0631\,L/s}{1\,usgpm}\,x\,\frac{1\,m^3/s}{1000\,L/s} = 0.053\,kW/RT \qquad (2.47)$$

Specific power consumption of 0.053 kW/RT includes the total power consumption of the chilled water pumping system which may consist of primary, secondary and tertiary chilled water pumps.

It is also specified in Singapore Standard SS553 that motors exceeding 15 kW should have controls and/or devices (such as VSD) that will result in motor power of no more than 30% of design wattage at 50% of design flow rate.

Condenser Water Pumps: Based on Singapore Standard SS553, maximum power consumption for the condenser water pumping system is 301 kW/(m^3/s). Following the standard of Air-Conditioning, Heating and Refrigeration Institute (AHRI), condenser water flow rate of 3.0 usgpm/RT is usually used for the design of chilled water system.

Therefore, energy efficiency (or specific power consumption) for the condenser water pumps:

$$301\,\frac{kW}{m^3/s}\,x\,\frac{3.0\,usgpm}{1\,RT}\,x\,\frac{0.0631\,L/s}{1\,usgpm}\,x\,\frac{1\,m^3/s}{1000\,L/s} = 0.057\,kW/RT \qquad (2.48)$$

It is also specified in Singapore Standard SS553 that motors exceeding 15 kW should have controls and/or devices (such as VSD) that will result in motor power of no more than 30% of design wattage at 50% of design flow rate.

Minimum Nominal Efficiency: Based on Singapore Standard SS530 : 2014, minimum nominal full-load efficiency for general purpose motors are presented in Table-2.5.

Table 2.5 Minimum nominal full-load efficiency for general purpose motors

Power (kW)	Minimum nominal full-load efficiency, %			
	Continuous use[a]		Occasional use[b]	
	2-pole	4-pole	2-pole	4-pole
	IE3[c]		IE2[c]	
15	91.9	92.1	90.3	90.6
18.5	92.4	92.6	90.9	91.2
22	92.7	93.0	91.3	91.6
30	93.3	93.6	92.0	92.3

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

[a] Continuous use: Motors are used for at least 2900 hours per year

[b] Occasional use: Motors are used for less than 2900 hours per year

[c] IE2 and IE3: Efficiency classes of single-speed, induction motors according to IEC 60034-30-1.

Chapter-3: Air Handling Systems

3. Air Handling Systems

3.1 Main Components

In buildings with central air-conditioning systems, air is treated (cooled and dehumidified) in air handling units (AHUs) and then distributed to the various parts of the building to control the relative humidity and temperature of the spaces. Main components of the AHU systems are fans, cooling coils, filters, dampers and ducting system. Chilled water pumps supply chilled water from the chilled water system to the cooling coil of the AHUs. Fans blow a mixture of the outdoor fresh air and part of the return air through the cooling coil. As a result, the air is cooled and dehumidified and then distributed to the air-conditioning spaces by overcoming the pressure losses of the ducting systems to maintain the comfort conditions. Fans provide the necessary energy for the distribution of the treated air. If the AHU systems are carefully designed, the electrical energy required to operate the system can be minimised and indoor air quality can be improved.

Learning Outcomes

Participants will be able to:

i. Determine the relative advantages and disadvantages of, as well as develop control strategies for different types of AHU systems for specific applications.
ii. Determine the ducting system characteristic curves and analyse the interaction of the fan and system characteristic curves under different operating conditions.
iii. Select fan and apply affinity laws to evaluate the performance of fan systems
iv. Identify energy saving opportunities and calculate potential energy savings.

3.2 Types of Air Handling Systems

3.2.1 All-Air Systems

For large commercial air-conditioning systems, both cooling and dehumidification of the conditioned spaces are usually furnished by air. This system is known as all-air system. Chilled water is used to transfer energy between the chillers and the air handling units (AHUs). A schematic diagram of a typical AHU is shown in Figure-3.1.

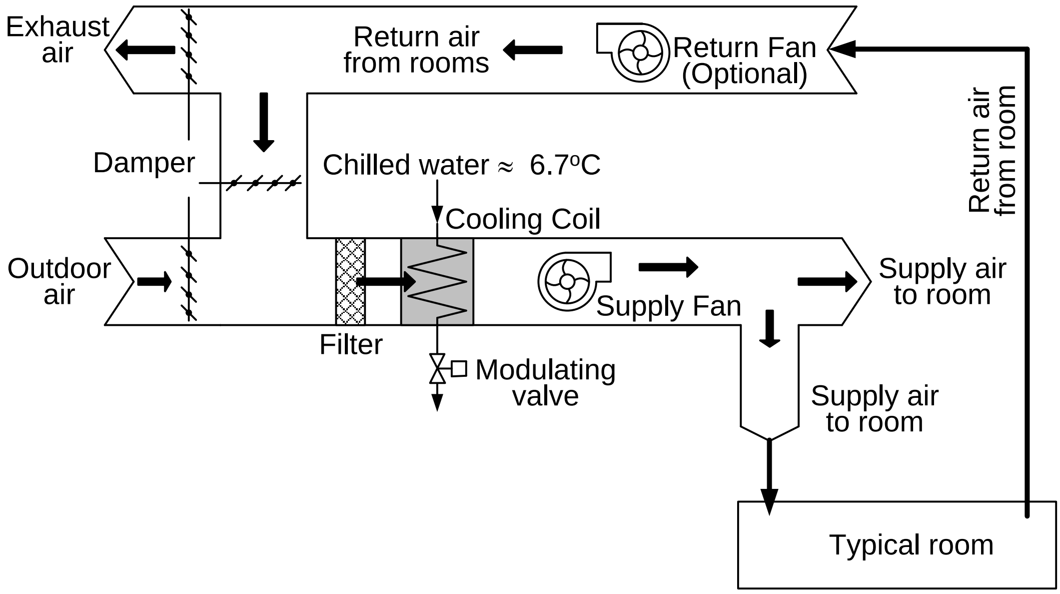


Figure 3.1 Typical all-air air handling unit

Air transfers the energy between the air handling units and the conditioned spaces. Based on the arrangement of ductwork between the air handling units and the conditioned spaces, all-air systems are classified as (a) Single-zone constant volume system, (b) Constant volume reheat system, (c) Variable-air-volume system and (d) Dual-duct system.

Single-Zone Constant Volume System: Single-zone constant volume all-air system is the simplest system that serves only one zone. A schematic diagram of a single-zone all-air system with associated dampers and controls is shown in Figure-3.2. The air handling unit can be installed either within the zone or remote from the zone. This system is generally used where reasonably uniform temperature and relative humidity can be maintained throughout the zone. Pre-set temperature of the zone is maintained by getting feedback from the room and discharge temperature sensors and adjusting the opening of the chilled water and hot water modulating valves. Hot water coil is used if the required relative humidity of the space is very low. Air is cooled to a very low temperature and most of the moisture is condensed using the cooling coil. Finally, the cold and dehumidified air is heated using the heating coil and then supplied to the space. Discharge temperature sensor could be eliminated and feedback from the room temperature sensor alone can be used to adjust the opening of the chilled water modulating valve directly, but the response for changing the zone temperature would be slower.

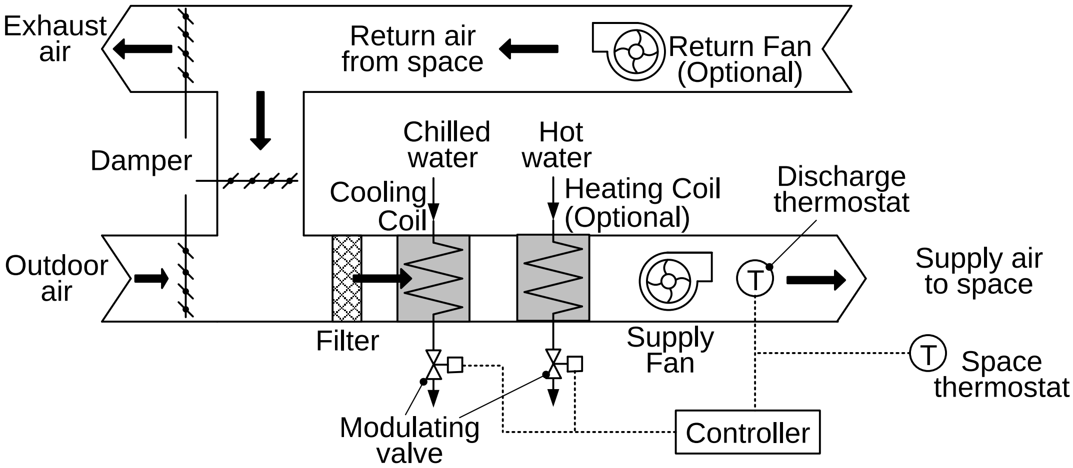


Figure 3.2 Single-zone constant volume all-air system

Constant Volume Reheat System: Constant volume reheat system is generally used in spaces or zones of unequal loading. Supply air is cooled in the air handling unit to a fixed cold temperature that is sufficiently low to support the zone having the maximum cooling load. Heating coil is installed in the branch ducts of specific zones. Temperature sensors of different zones activate their heating coils when zone temperatures fall below the pre-set value. Hot water, steam or electricity is used as the heating medium for the heating coil. Reheat system is not an energy efficient system as the supplied heat to the heating coil increases the cooling load of the spaces. A schematic diagram of a typical constant volume reheat system with components is shown in Figure-3.3.

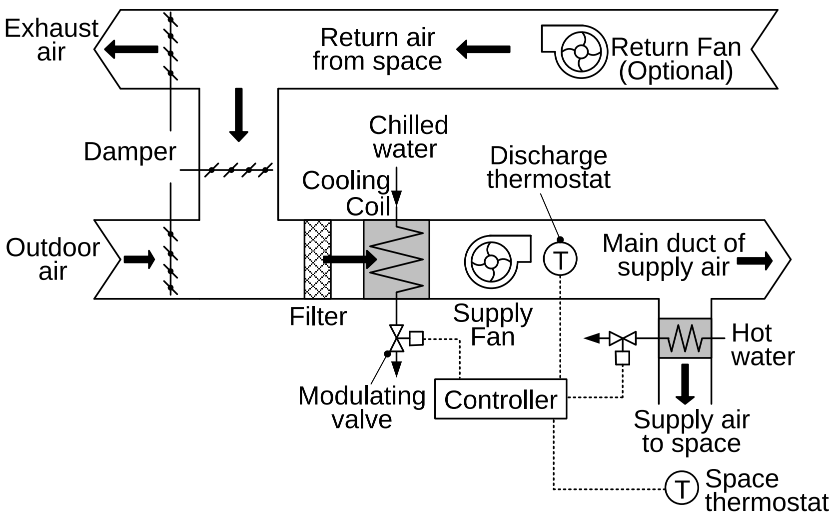


Figure 3.3 Schematic diagram of typical constant volume reheat system

Variable Air Volume System: Figure-3.4 shows a typical variable-air-volume (VAV) single duct system. Each zone has its own air flow modulating motorised damper known as variable air volume (VAV) box. Based on pre-set temperature of each zone, temperature sensor of individual zone controls the opening of the motorised damper and the volume flow rate of air to each zone. If the motorised damper is partially closed due to the low cooling load of the spaces, air pressure in the main duct increases which is sensed by static pressure sensor of main duct. Temperature of the discharge air is also monitored by the temperature sensor located in the main duct. Based on the feedback of the static pressure sensor and the temperature sensor of the main duct, the controller regulates the flow of air by modulating the speed of the blower using variable speed drive and regulates the flow of chilled water in the cooling coil by adjusting the opening of the modulating valve. Fan power consumption drops significantly at part cooling load condition due to the reduction of fan speed in relation to the volume of air being circulated.

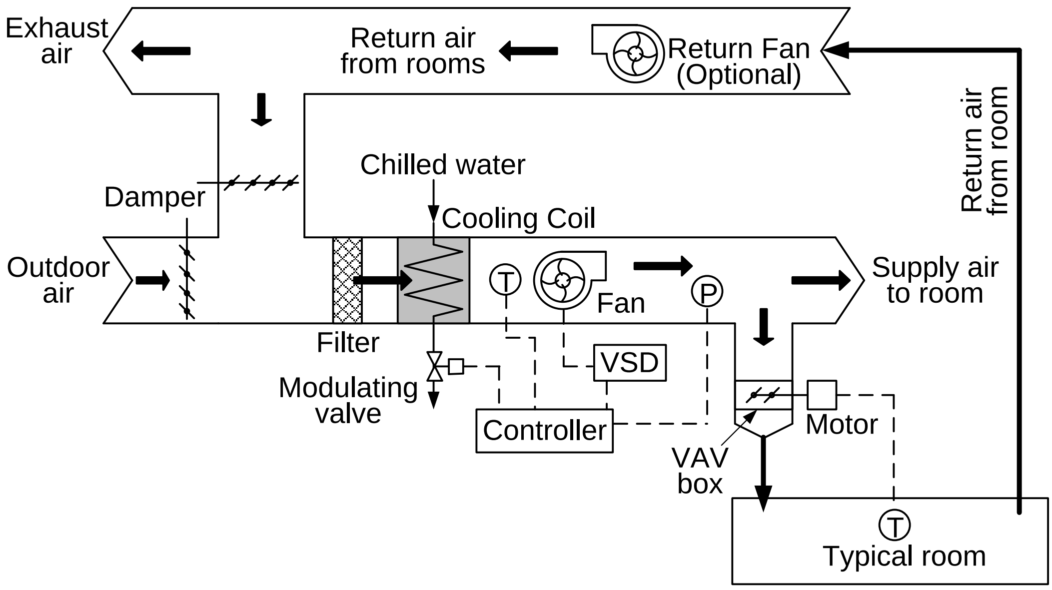


Figure 3.4 Schematic diagram of typical variable air volume system

Dual-Duct System: In the dual-duct systems, the central AHU supplies cold and dehumidified air through one duct while warm air is supplied through the other duct in each space as shown in Figure-3.5. The temperature of individual spaces is maintained by modulating the flow of cold and warm air in proper proportions. Dual-duct system is used in hospital, large laboratories or production areas where sensible heat load varies significantly and needs to get prompt and opposite temperature response. Due to the mixing of the warm and cold air, the dual-duct systems are generally energy inefficient. To save energy in dual-duct system, the controller should reset the cold air supply temperature to the highest acceptable value and the warm air supply temperature to the lowest acceptable value. Energy performance of the dual-duct system can be improved further by incorporating the variable air volume system using the VAV box as shown in Figure-3.5.

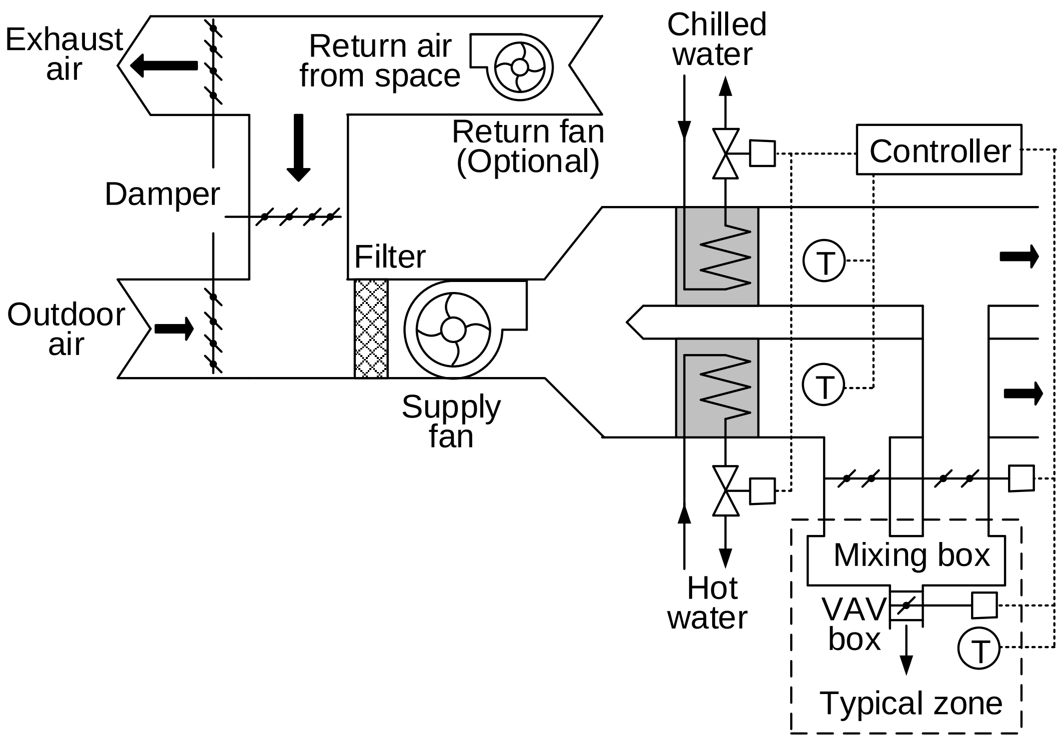


Figure 3.5 Schematic diagram of typical dual-duct system

3.2.2 Air-Water Systems

In air-water system, both conditioned fresh air and chilled water are distributed to the air-conditioning spaces. Each air-conditioning space or zone contains required number of fan-coil units as shown in Figure-3.6. Chilled water is supplied to the heat exchanger coil of the fan-coil units which are mainly used to carry away the sensible heat load of spaces. A central Primary Air Handling Unit (PAU) is used to cool and dehumidify outdoor fresh air and supply to the conditioned spaces using ducts. This conditioned fresh air provides the ventilation required and carries away the moisture resulting from the latent load of the spaces. As the specific heat capacity and density of chilled water are much greater than air, the space required to distribute the chilled water using pipes is much less than the required air distribution ducts of all-air systems to accomplish the same air-conditioning task. The size of the fresh air duct of air-water system is quite small. If the fresh air requirement for the ventilation of the spaces is balanced with the required exhaust rate, the return air duct can be eliminated. Moreover, the power consumption of chilled water distribution pumps of air-water systems is significantly less than the fan power consumption of all-air systems to supply and return the air required for the cooling and ventilation of the spaces. Temperature of individual spaces or zones can be controlled by modulating the chilled water flow rate or regulating the space air flow rate through the heat exchanger coil of the fan-coil units. Some condensation of moisture will take place in the heat exchanger coil of the fan-coil units. Condensate drainage system should be designed for the fan-coil units.

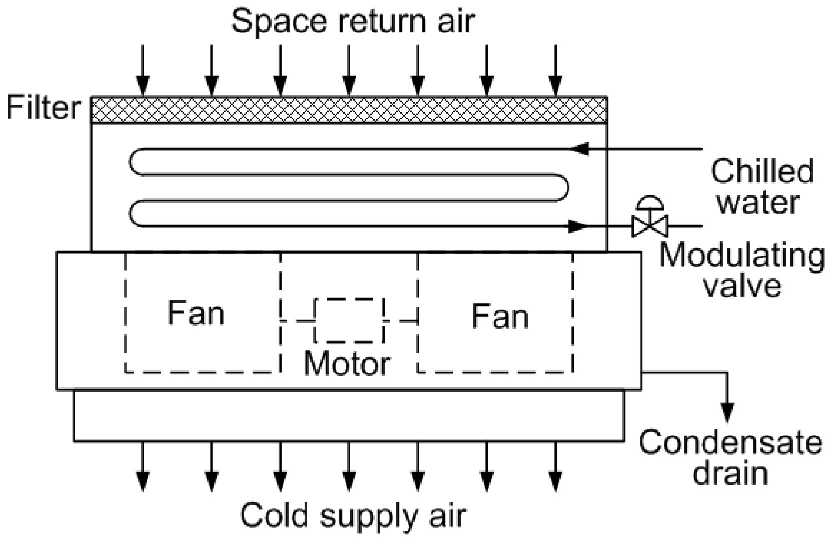


Figure 3.6 Schematic diagram of fan-coil unit

3.2.3 All-Water Systems

In all-water system, each air-conditioning space or zone contains required number of fan-coil units. Only chilled water is supplied to the heat exchanger coil of the fan-coil units to provide required cooling and dehumidification. Unconditioned fresh air is supplied by an opening through the wall or by infiltration for the purpose of ventilation. It does not require any ventilation fresh air duct resulting in considerable space savings throughout the building. As the fan-coil system usually does not have the provision for positive ventilation air, all-water systems may not meet stringent indoor air quality (IAQ) requirements of air-conditioning spaces. Similar to the air-water systems, temperature of individual spaces or zones can be controlled by modulating the chilled water flow rate or regulating the space air flow rate through the heat exchanger coil of the fan-coil units. Proper condensate drainage system should also be designed for the fan-coil units.

3.3 Types of Fans and Their Performances

Common types of fans used in the AHU systems are (a) forward curved, (b) radial, (c) backward curved and (d) axial flow as shown in Figure-3.7. The function of the fans is to supply the required amount of conditioned air to the air-conditioning spaces by overcoming the pressure losses of the filters, cooling and heating coils, dampers, bends and fittings of the ducting systems. Velocity components of air for different types of fans are presented in Figure-3.8. As shown in Figure-3.8, the required velocity of backward curved fan is the highest to attain the same absolute velocity of air. Typical performances of different types of fans are summarised in Table 3.1. The efficiency of the backward curved fans is relatively high in comparison to the other types of fans. Backward curved fans also develop relatively high pressure head. However, the flow rate is comparatively low. Hence, backward curved fans are generally selected for high pressure head (such as long duct with a lot of bends and fittings) and low flow rate applications.

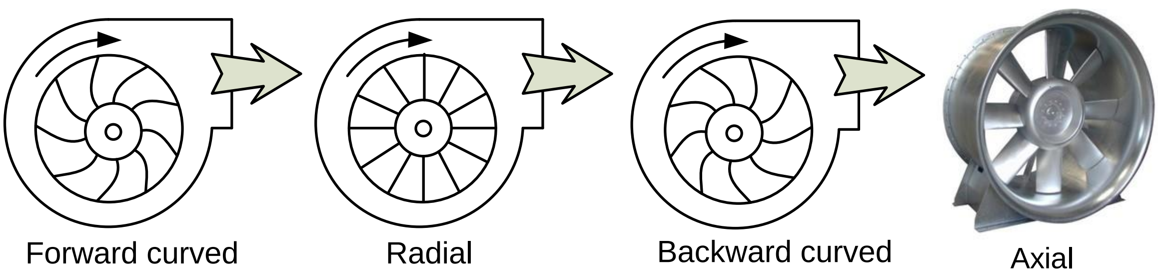


Figure 3.7 Common types of fans

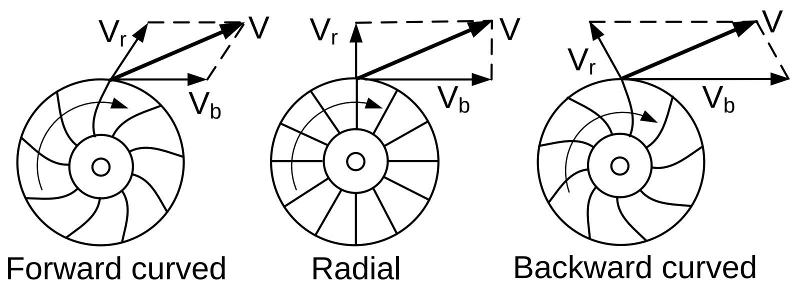


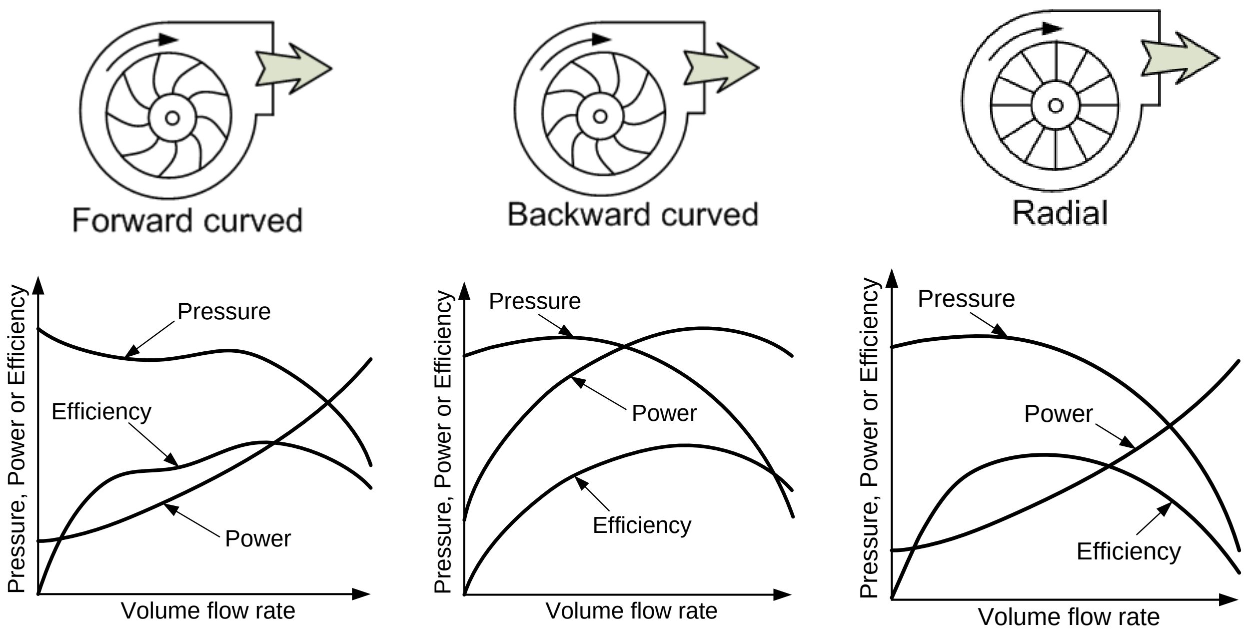
Figure 3.8 Velocity components of air for different types of fans [V = Absolute velocity of air leaving blade (Shown equal for all three blade types); V_r = Velocity of air leaving blade relative to blade; V_b = Velocity of blade tip]

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.

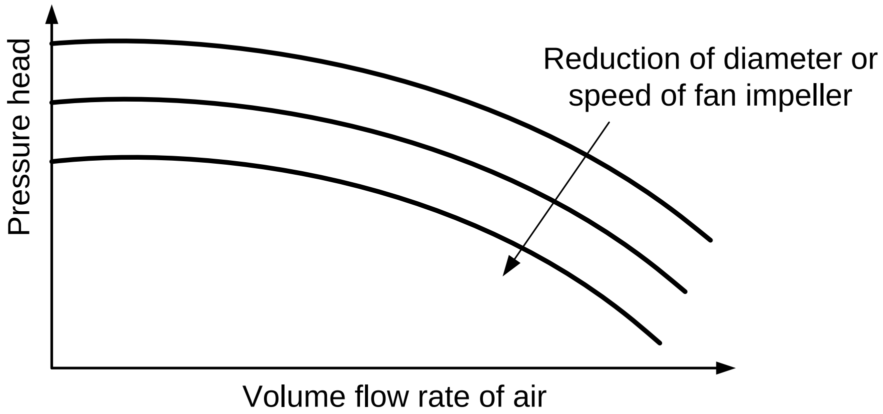


Figure 3.10 Fan characteristic curves for different diameters or speeds of impeller

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.

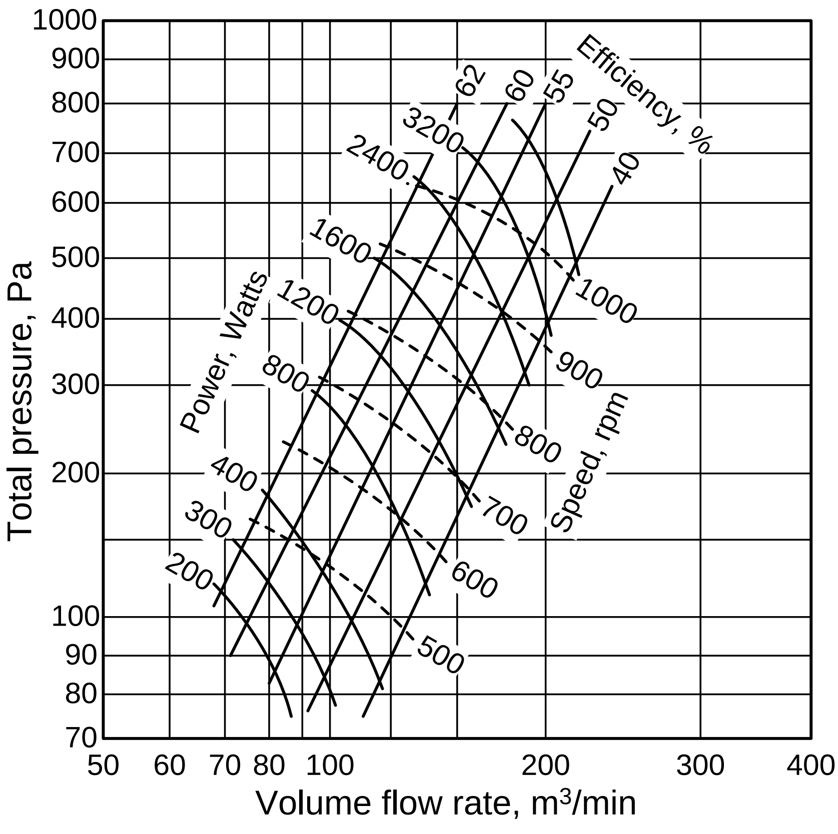


Figure 3.11 Fan performance curves at different speeds

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:

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

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.

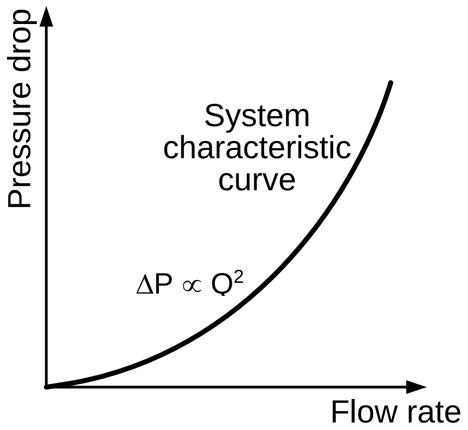


Figure-3.12 System curve for ducting system

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:

$$\Delta P_m = C Q_m^2$$

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

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:

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

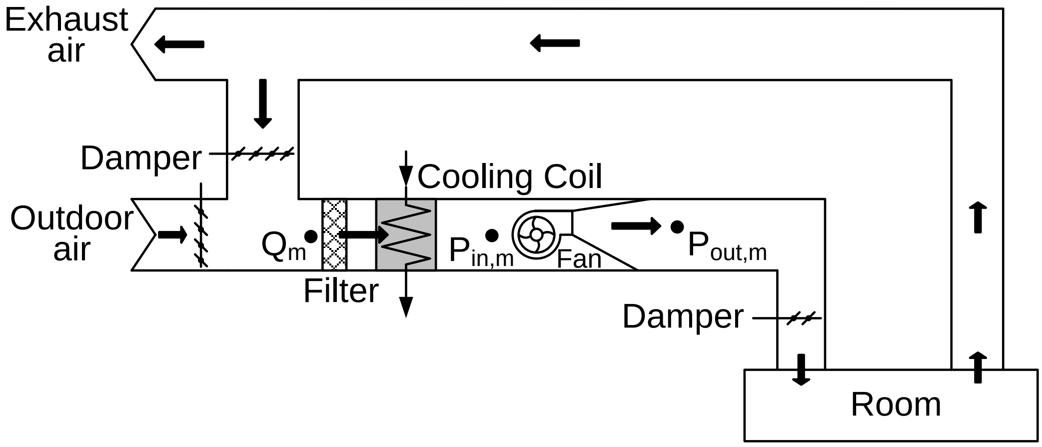


Figure 3.13 Measurement of pressure and air flow rate in ducting system

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.

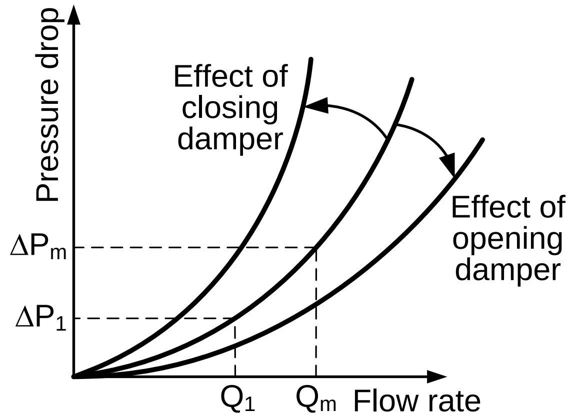


Figure 3.14 Changes of the system pressure drop with the flow rate of air and the movement of system curve with the opening of the damper

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

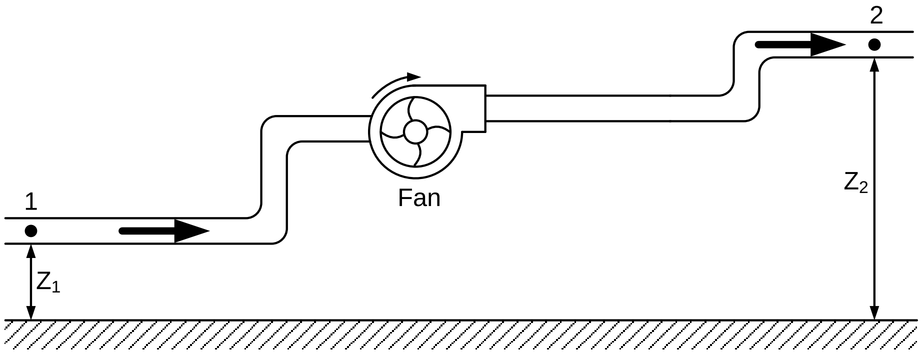


Figure 3.15 Steady flow of air in a perfectly insulated ducting system

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:

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

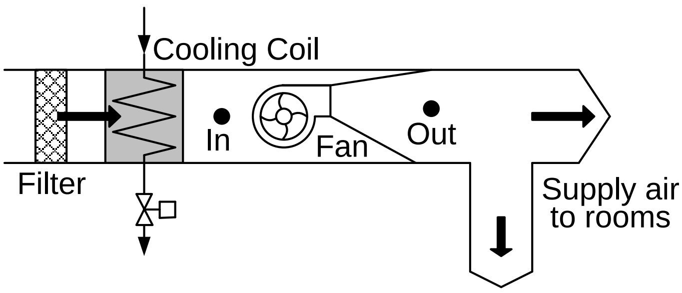


Figure 3.16 Inlet and outlet of fan of an AHU system

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:

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

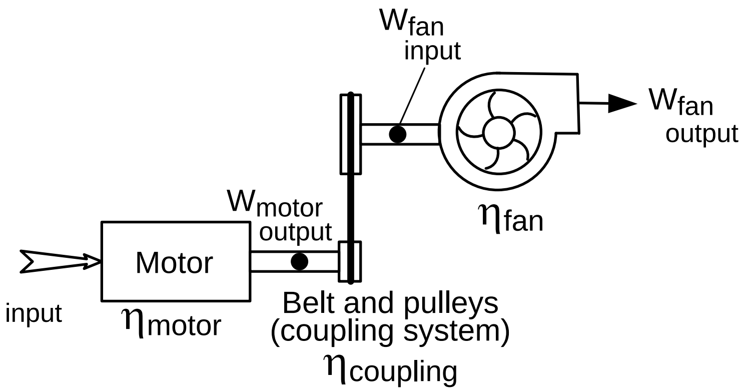


Figure 3.17 Belt and pulley system for the coupling of fan and motor

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.

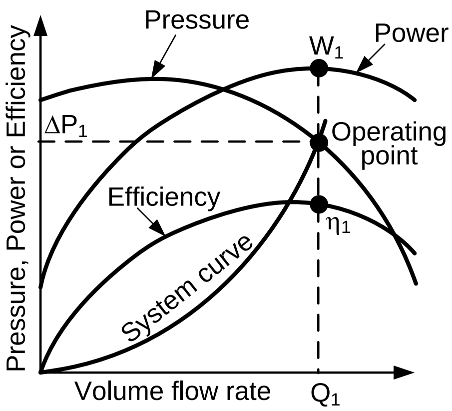


Figure 3.18 Fan characteristic curves, system curve and operating point

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.

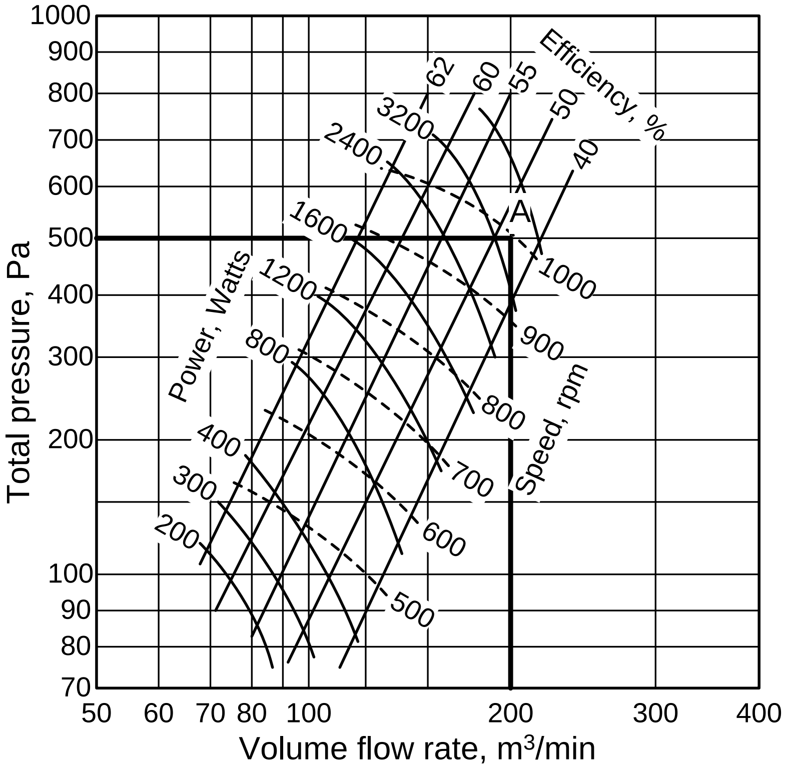


Figure 3.19 Fan performance curves and operating point

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:

$$W_{fan,input} = \frac{Q_1 \Delta P_1}{\eta_{fan}} \tag{3.20}$$

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.

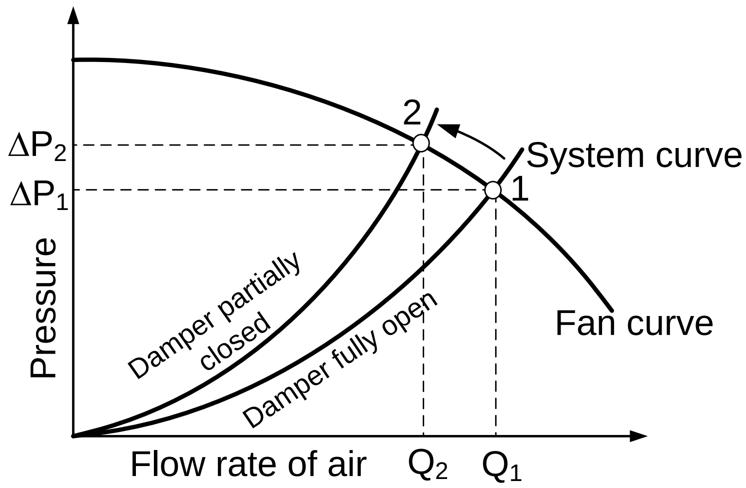


Figure 3.20 Effect of closing damper of the ducting systems

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.

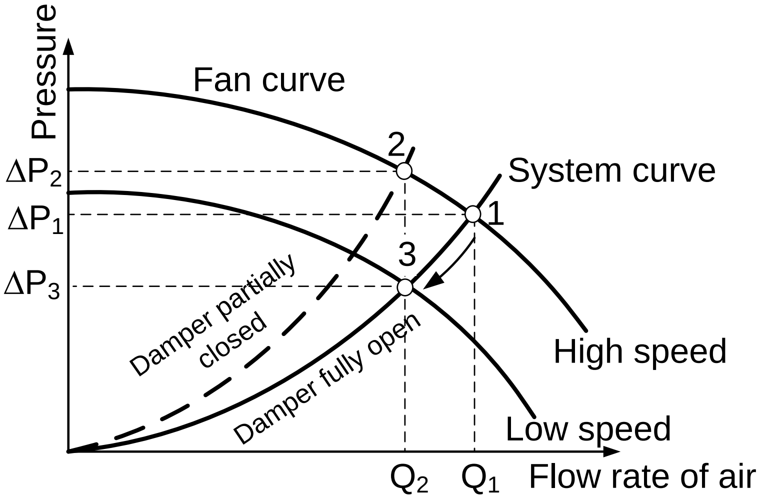
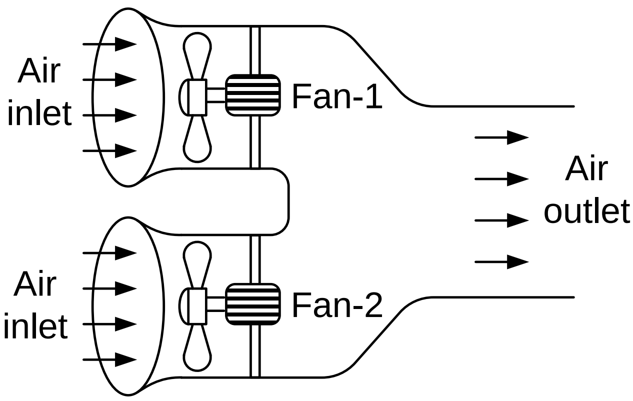
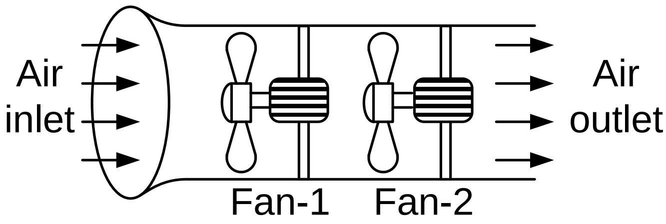


Figure 3.21 Effect of closing damper of ducting systems

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 $Q_{fan\text{-}1}$.

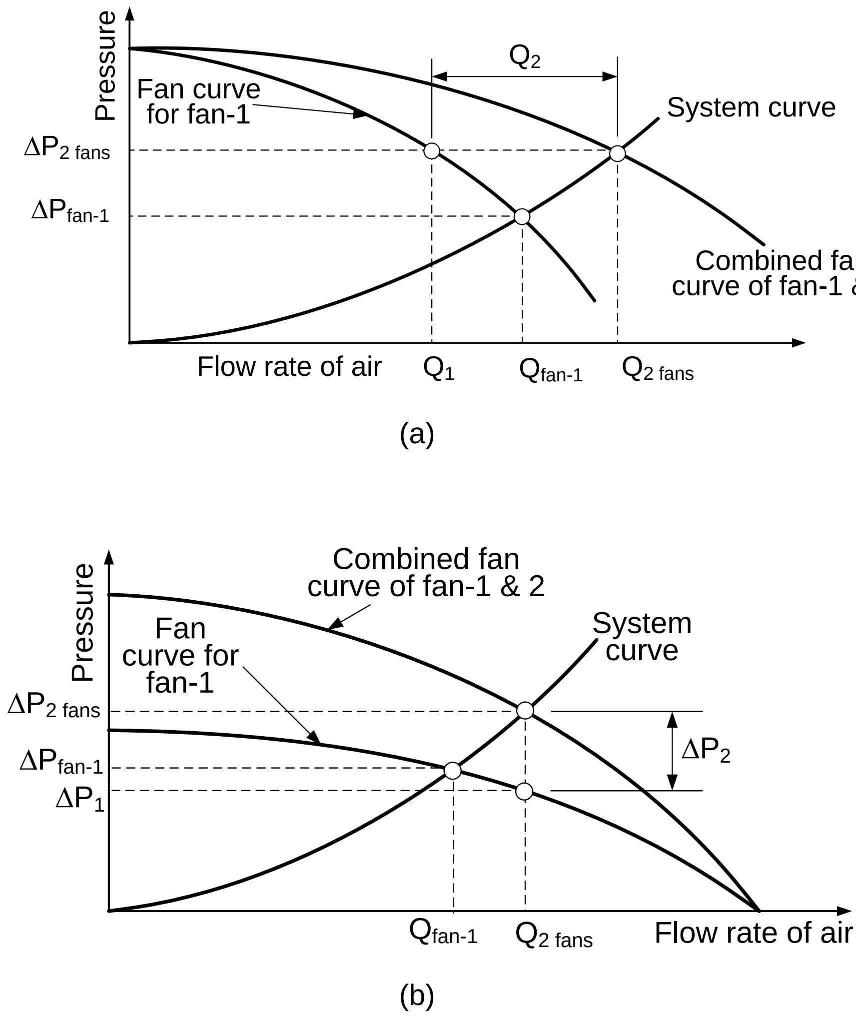


Figure 3.23 Fan and system characteristic curves (a) fans are connected in parallel and (b) fans are connected in series

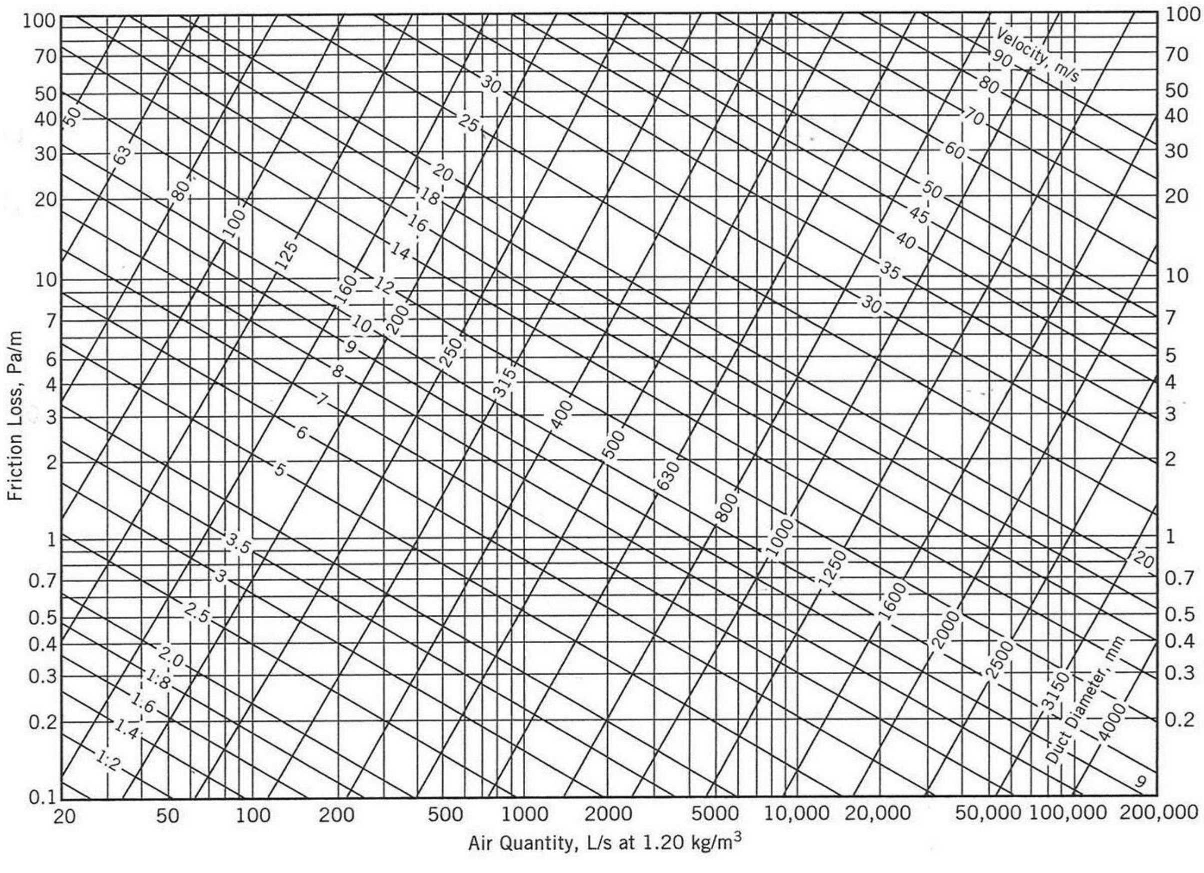
3.7 Losses in Ducting Systems

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).

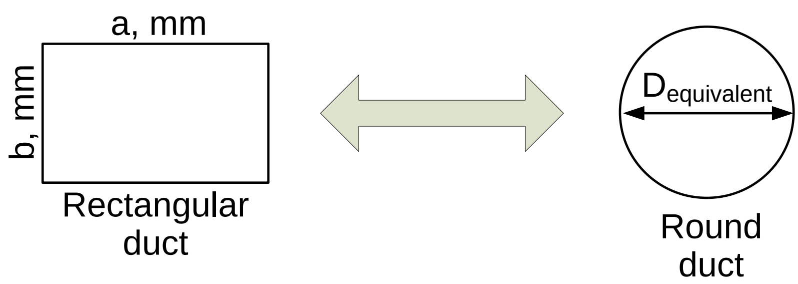
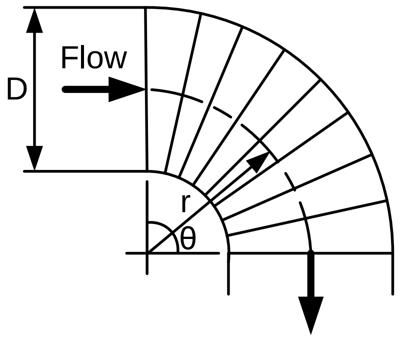


Figure 3.25 Rectangular and circular ducts of equivalent friction loss

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.

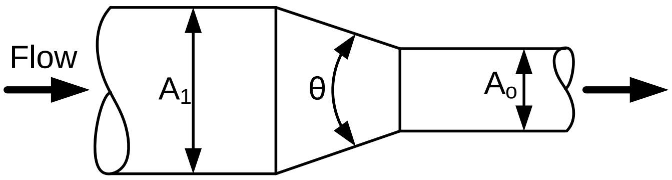
Table 3.2 Loss coefficients for pleated elbow (r/D = 1.5)



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)*

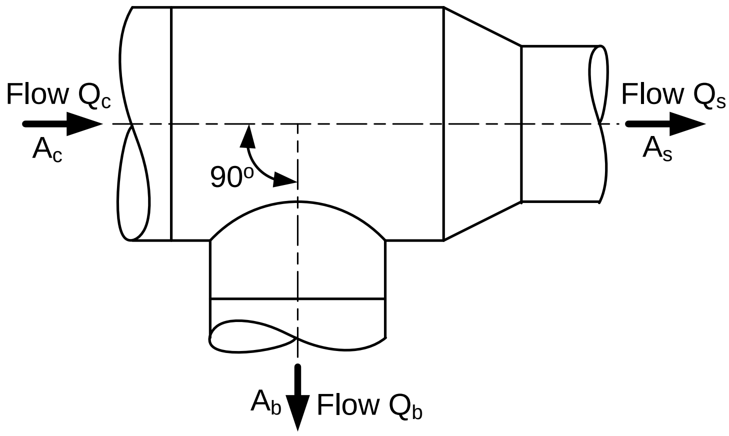
Table 3.3 Loss coefficients for round to round transition



A_o/A_1	C_o						
	$\theta = 10^o$	20^o	45^o	90^o	120^o	150^o	180^o
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 90° 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 90° 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.

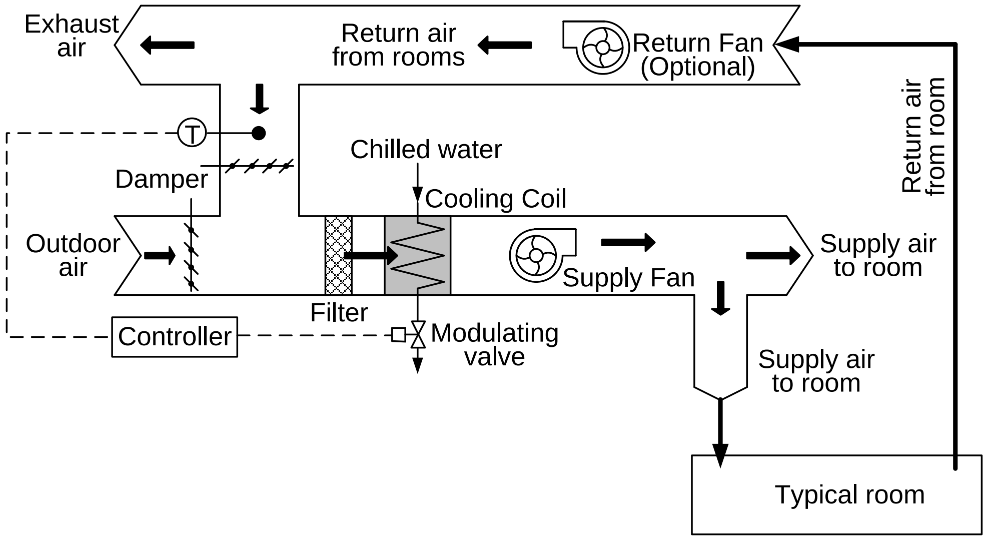


Figure 3.26 Schematic diagram of a typical constant air volume (without reheat coil) system

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.

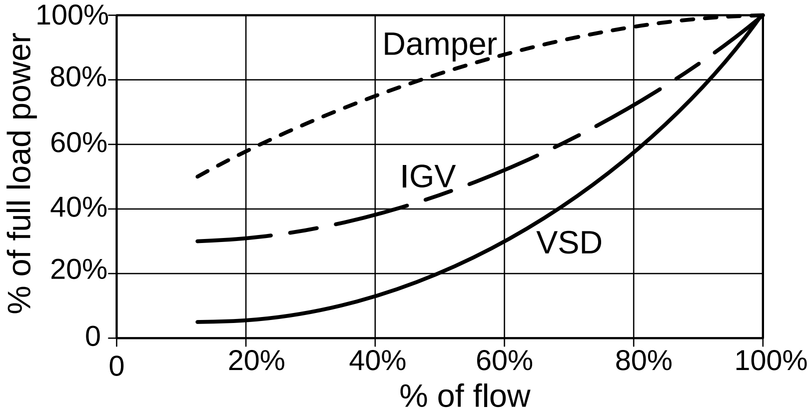
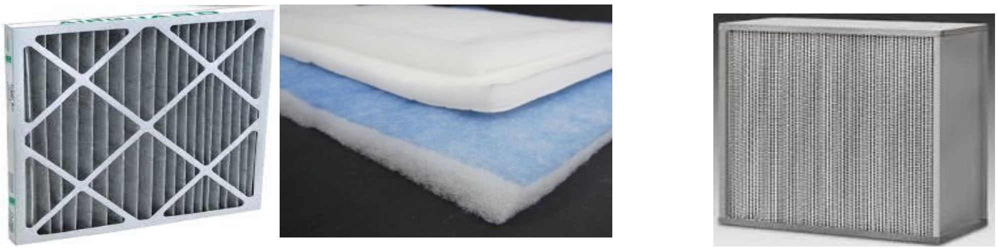


Figure 3.27 Fan power consumption for different VAV systems

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

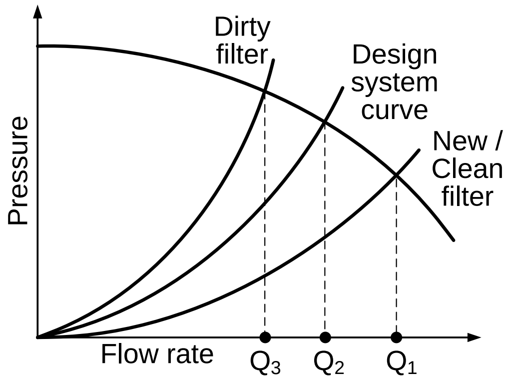


Figure 3.29 Effect of filter conditions on flow rate of air

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.

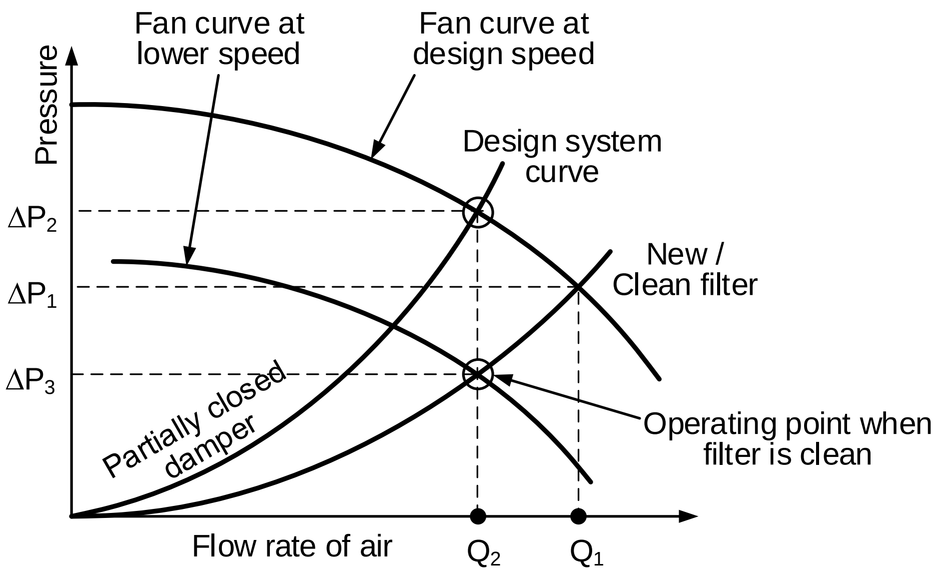


Figure 3.30 Effect of closing damper and fan speed with clean filter

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.

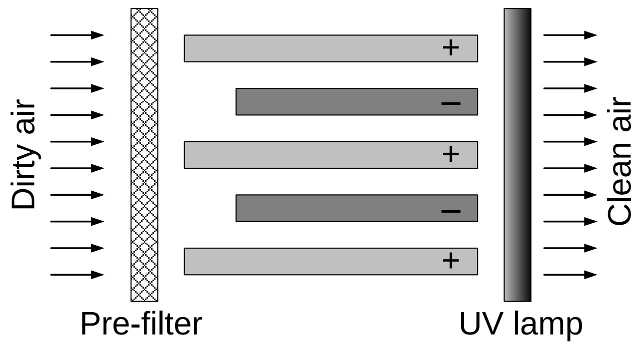


Figure 3.31 Schematic diagram of electronic air filter

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: $W_1 = \dfrac{Q\Delta P_1}{\eta}$

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

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

$W_2 = W_1 \dfrac{\Delta P_2}{\Delta P_1}$ = 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.

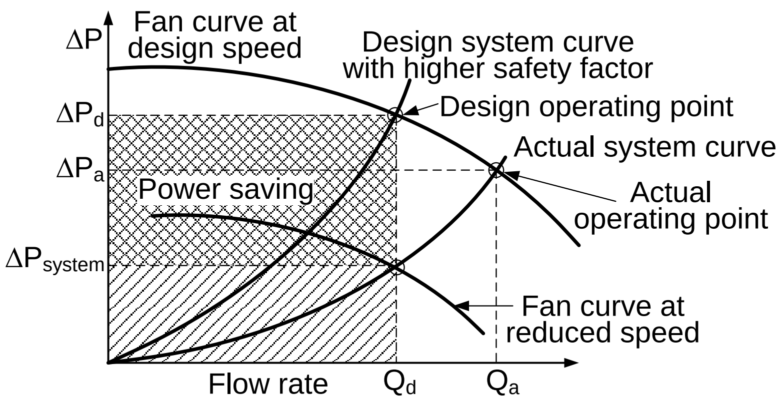


Figure 3.32 Fan performances due to use of high safety factors

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:
$N_{f,required} = N_{f,present} \times (Q_{required} / Q_{present})$ = 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: $D_{m,present} N_{m,present} = D_{f,present} N_{f,present}$

$$200 x 960 = D_{f,present}\, x 800$$

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:

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

$$200 x 960 = D_{f,required}\, x 600$$

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

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.

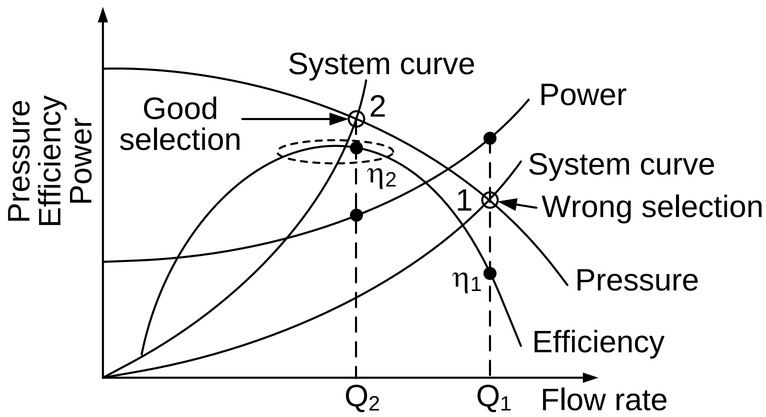


Figure 3.33 Fan efficiencies at different operating points

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.

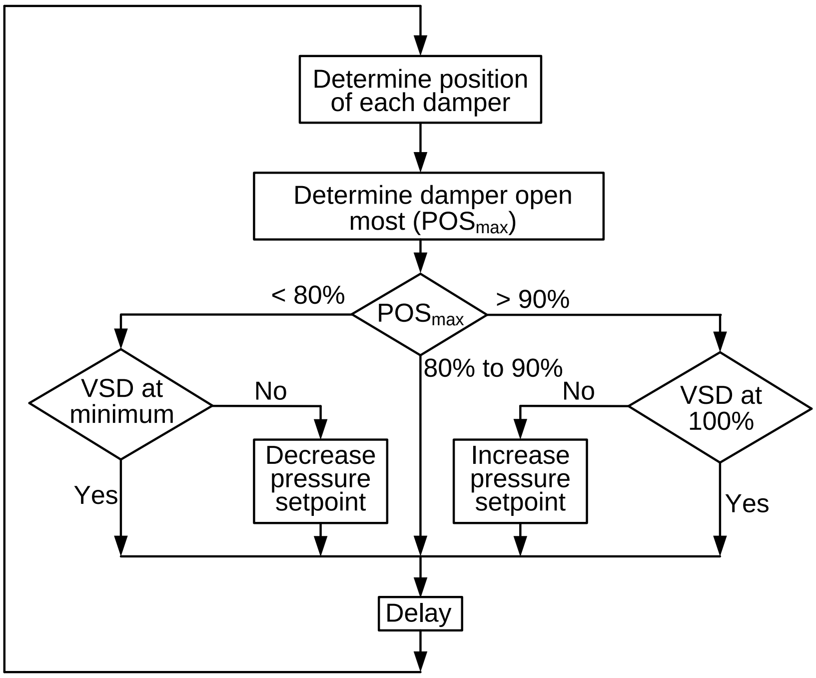


Figure 3.34 Control strategy of variable differential pressure set point for VSD of fans

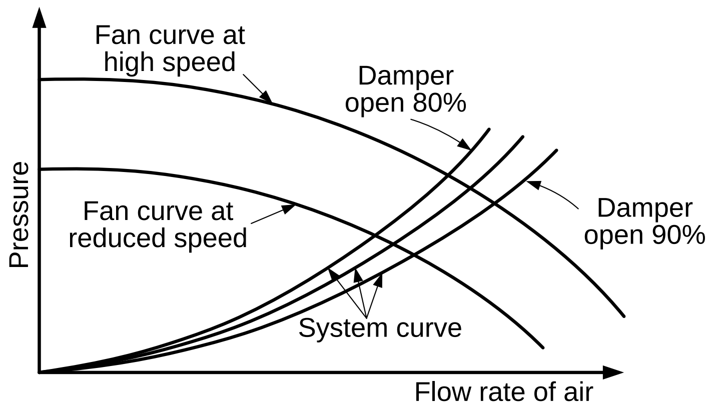


Figure 3.35 Movement of system and fan curves for fan systems using variable differential pressure set point

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.

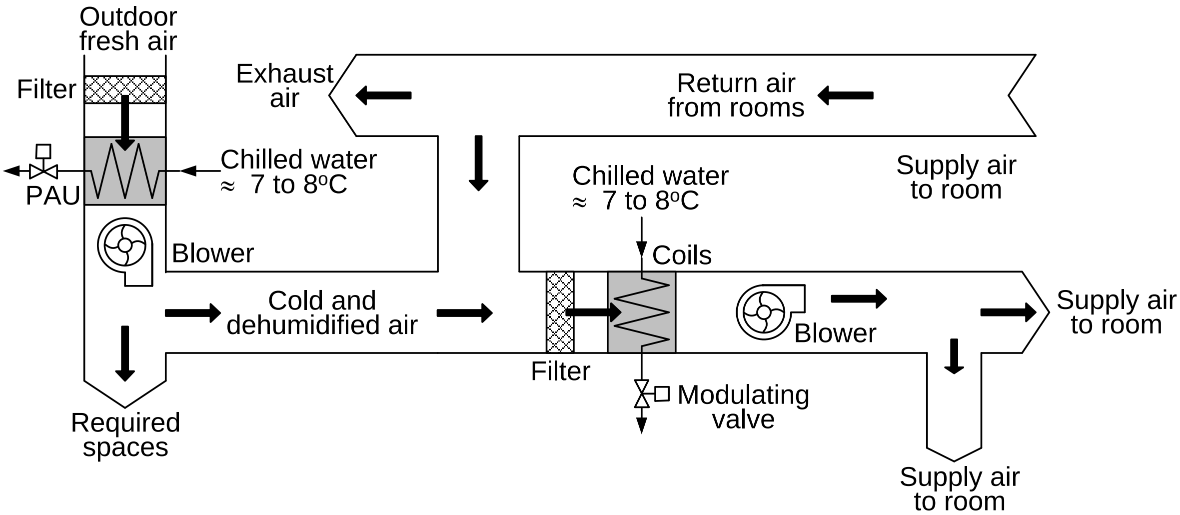


Figure 3.36 Schematic diagram of a primary air handling system

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.

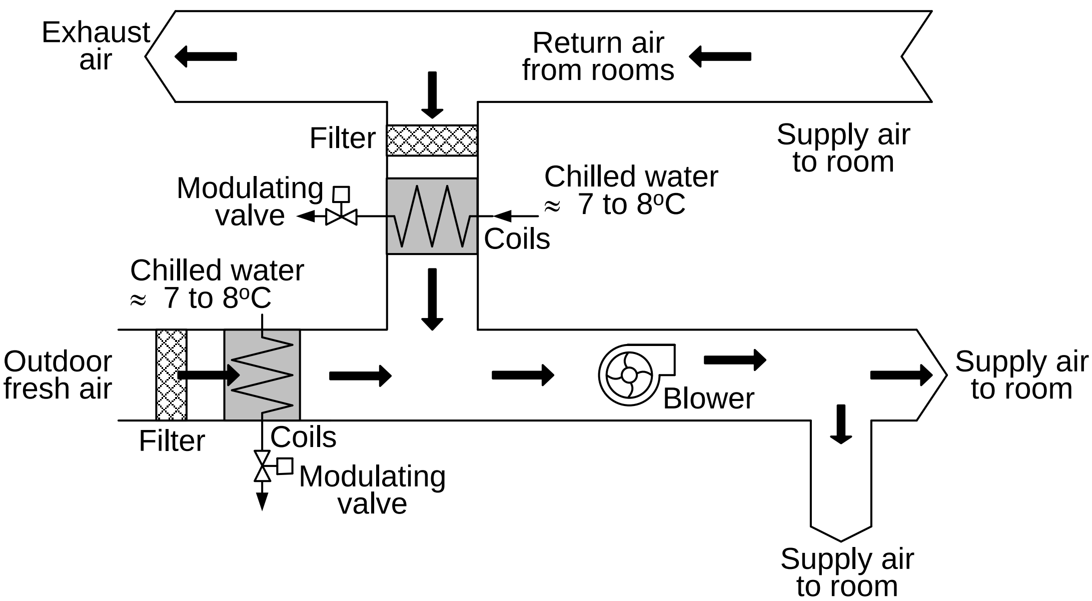


Figure 3.37 Schematic diagram of dual-path air handling system

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.

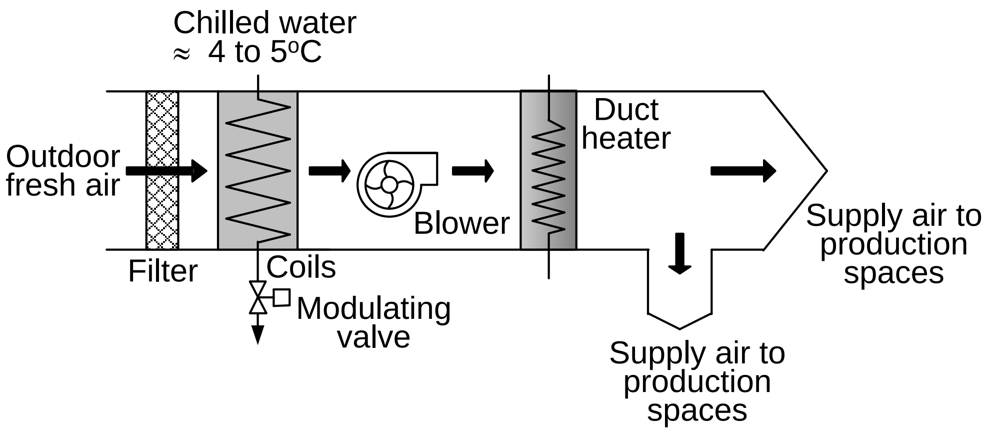


Figure 3.38 Schematic diagram of make-up air unit

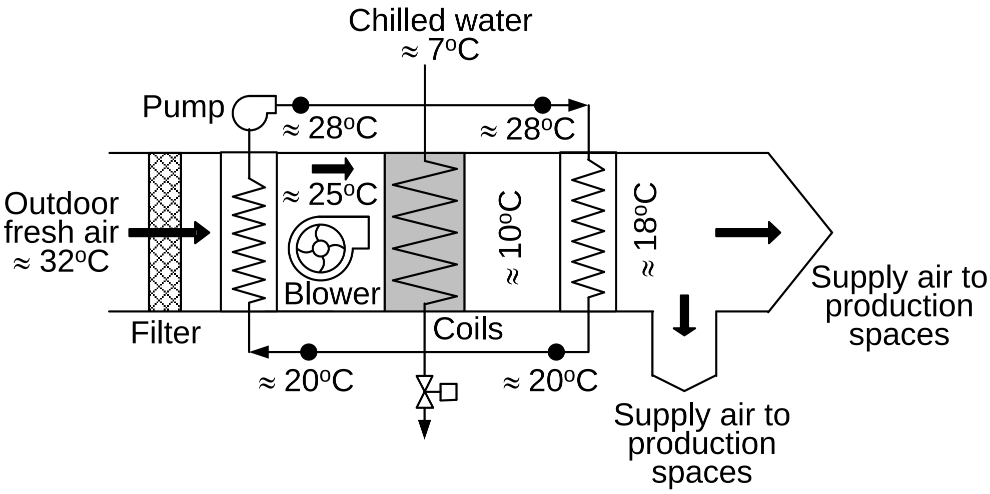


Figure 3.39 Example of temperature distribution in run-around coil systems

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.

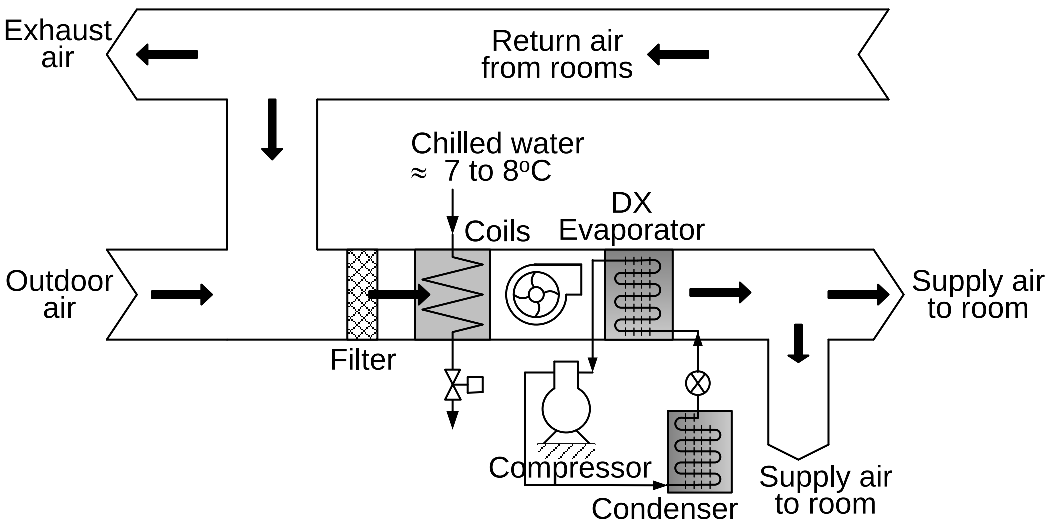


Figure 3.40 Schematic diagram of AHU with direct expansion refrigeration system

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°C (depending on the design of the cooling coil) can be used in the AHU system. This results in significant improvement of chiller efficiency.

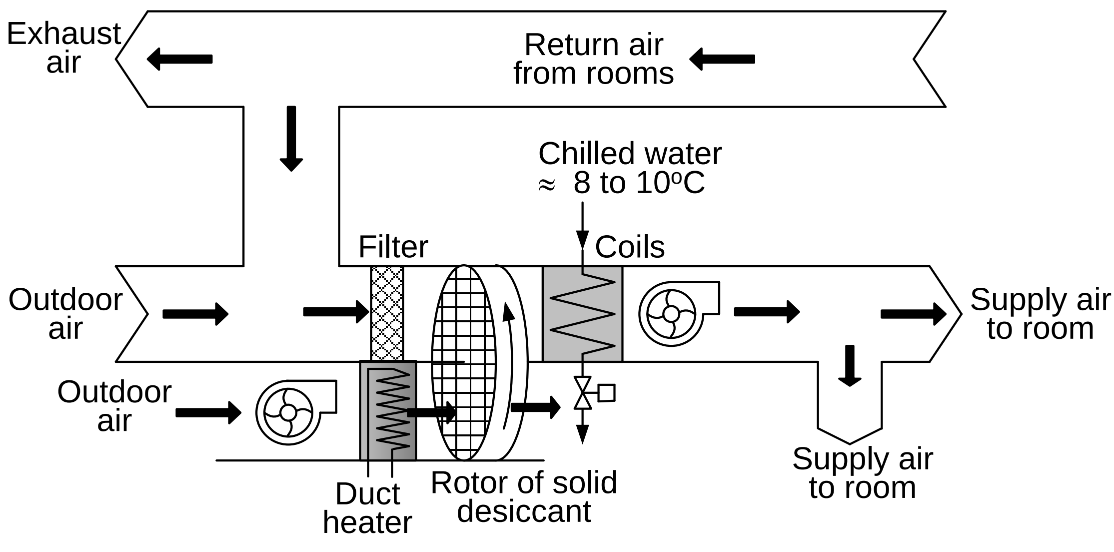


Figure 3.41 Schematic diagram of AHU with solid desiccant systems

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.

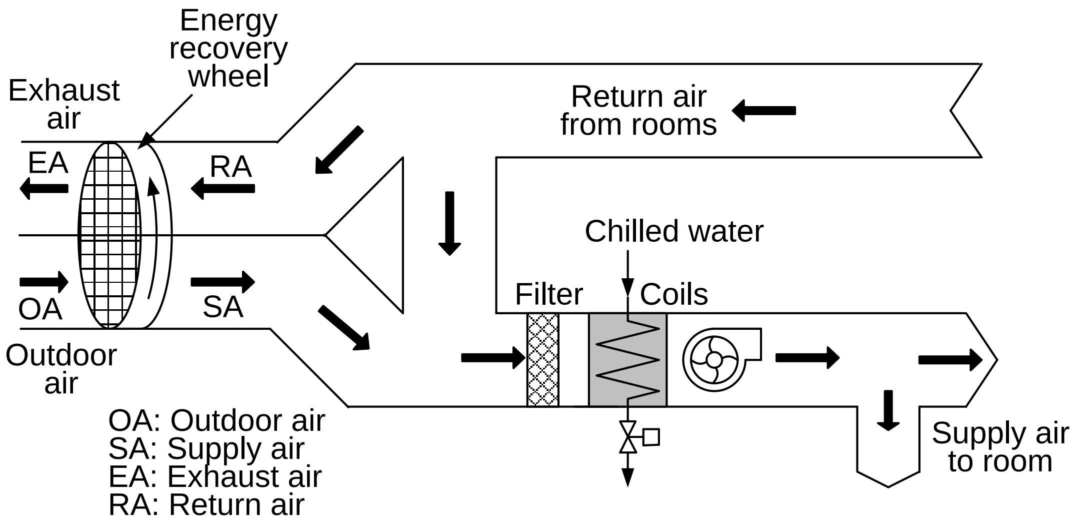


Figure 3.42 Energy recovery wheel installed in AHU duct

Efficiency of sensible energy recovery process is expressed as:

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

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:

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

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:

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

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:

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

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.

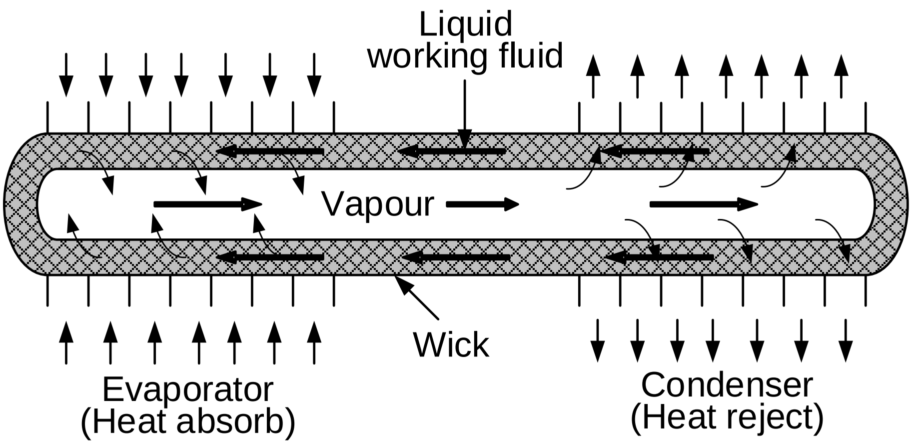


Figure 3.43 Typical heat pipe

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.

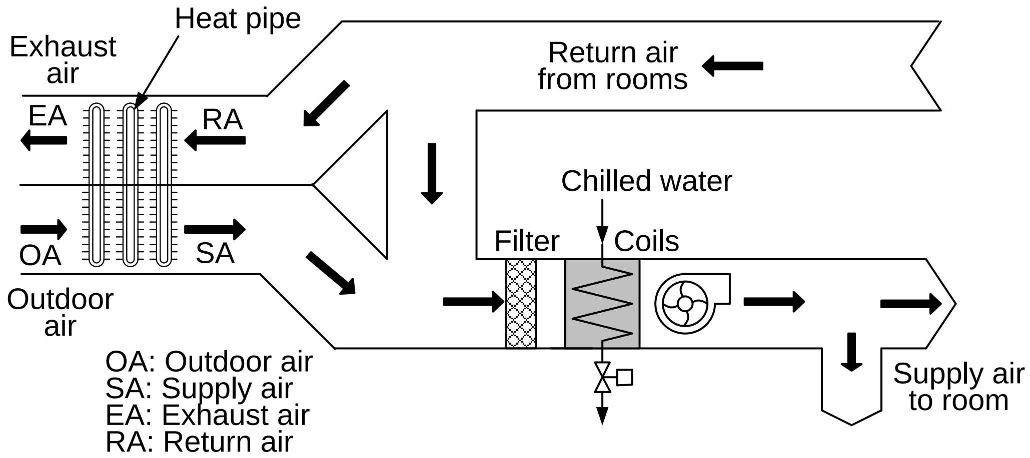


Figure 3.44 Heat pipe system installed in AHU duct.

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.

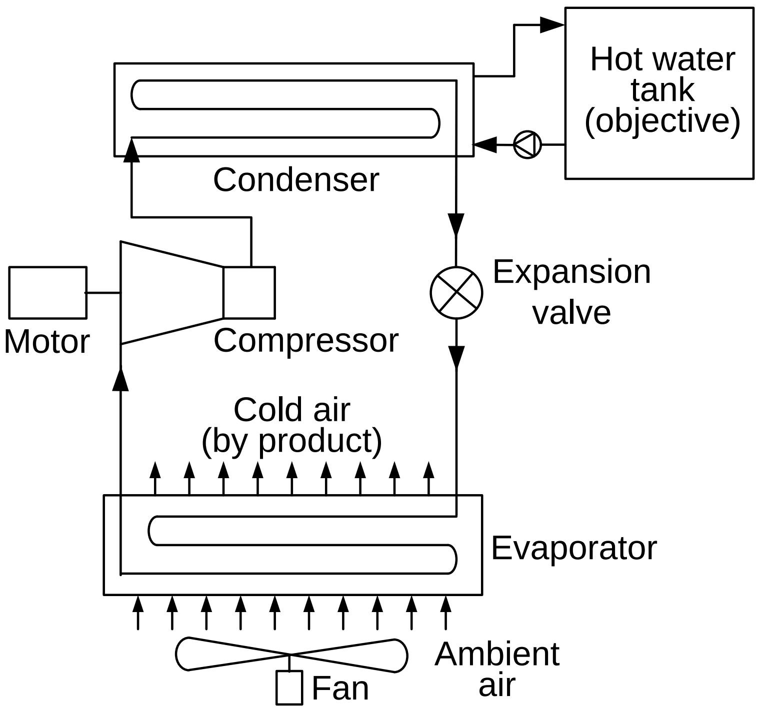


Figure 3.45 Typical heat pump system

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:

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

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

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.

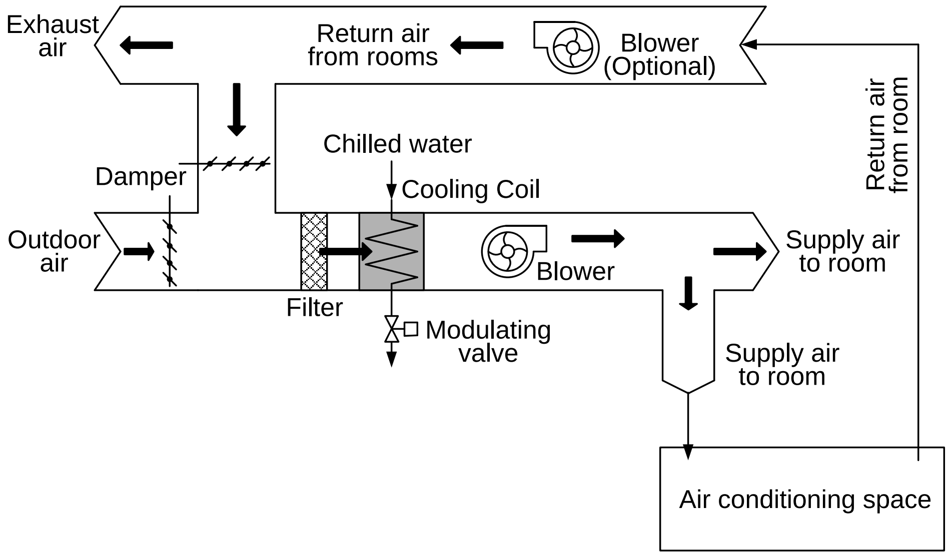


Figure 4.1 Typical air handling unit (AHU) and ducting system

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: °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: ^oC): 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.

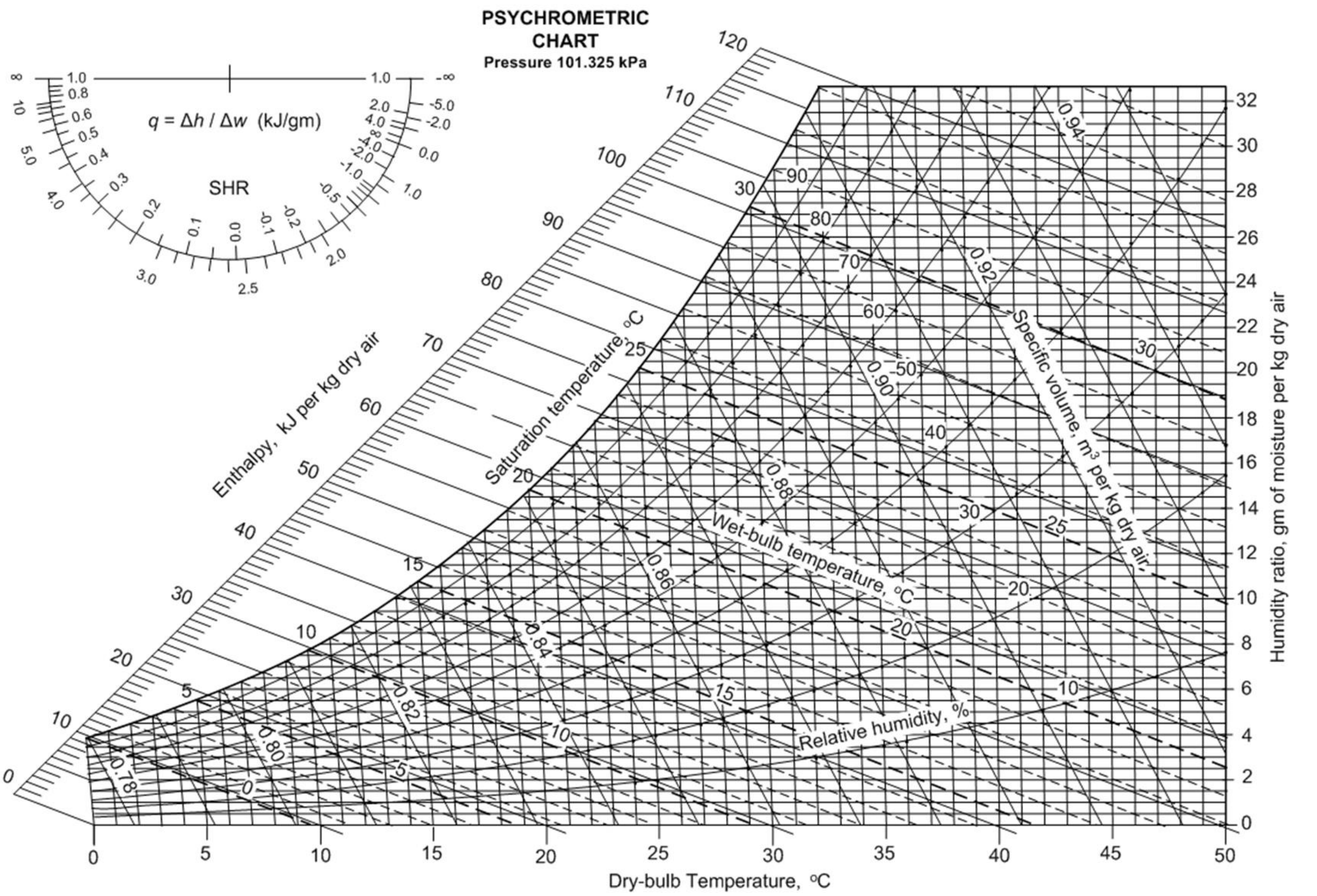


Figure 4.2 Psychrometric chart

Dew-point Temperature, T_{dp} (Unit: ^oC): 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.

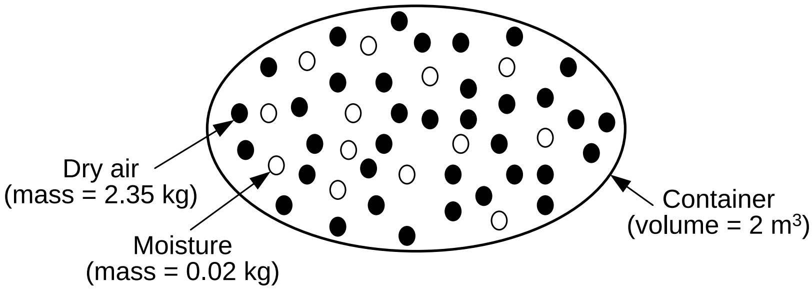


Figure 4.3 Air vapour mixture in a control volume

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°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°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°C and moisture-holding capacity of the air is increased to 0.064 kg at the temperature of 30°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°C to 30°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.

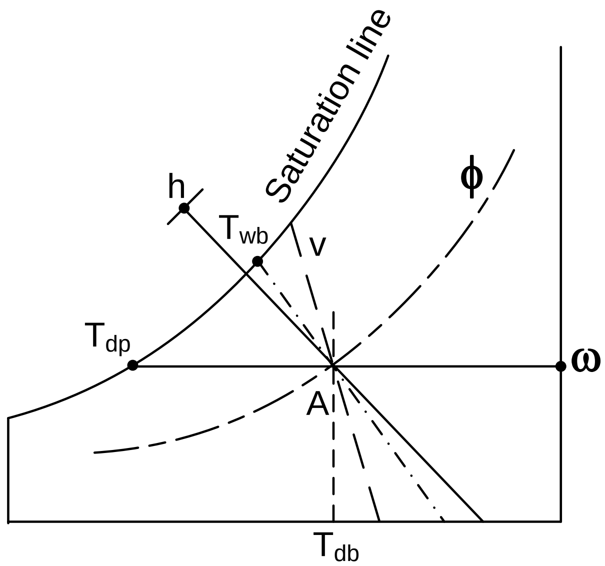


Figure 4.4 Psychrometric chart showing different property lines

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

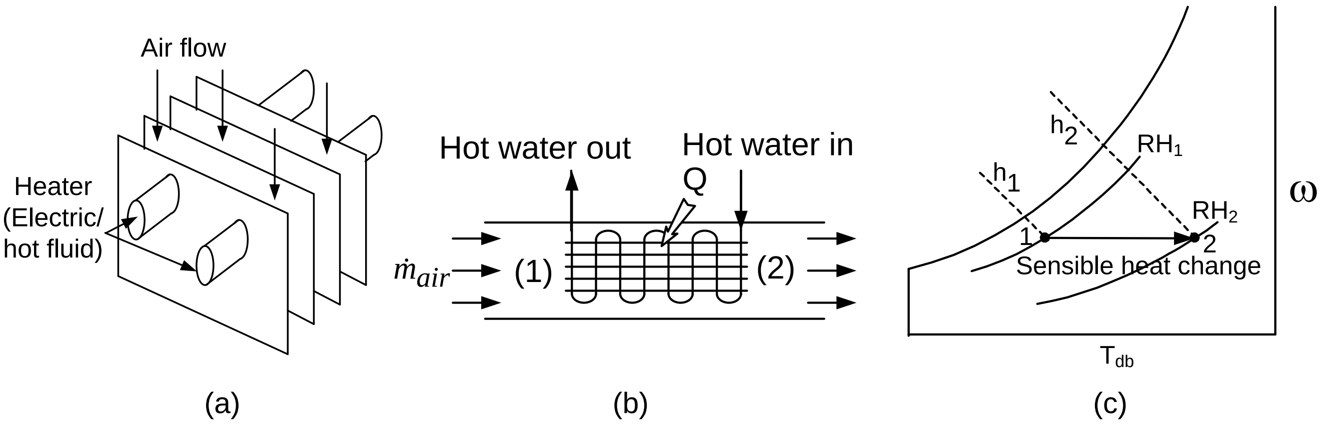


Figure 4.5 Sensible heating process: (a) Typical finned heating coil, (b) States of air in heating process and (c) States of air on Psychrometric chart

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 $Q = \dot{m}_{air}\left(h_1 - h_2\right)$

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

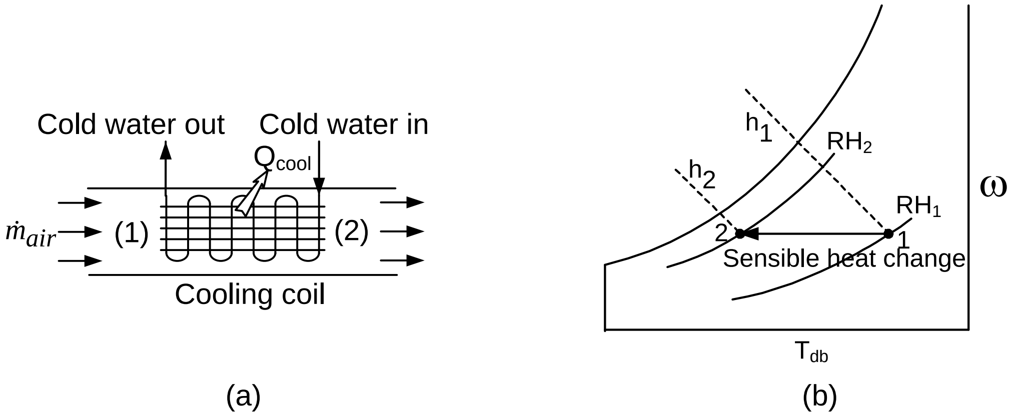


Figure 4.6 Sensible cooling process: (a) States of air in cooling process and (b) States of air on Psychrometric chart

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.

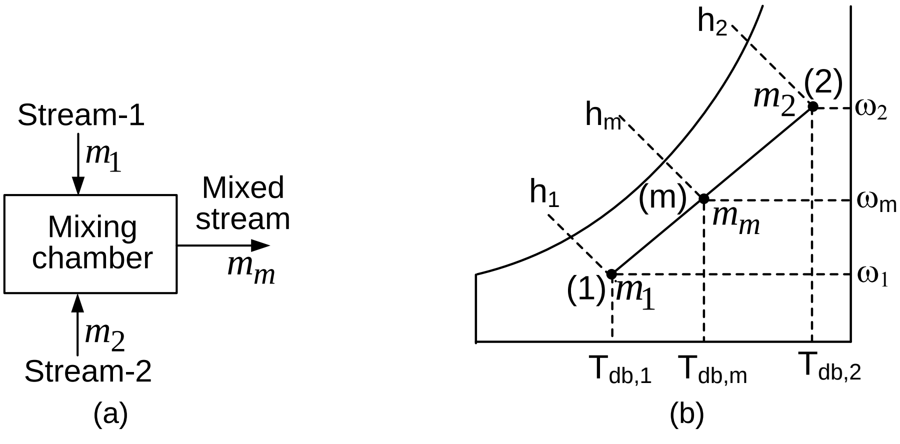


Figure 4.7 Mixing process of air streams: (a) Mixing of two air steams and (b) States of air streams on Psychrometric chart

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, °C

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).

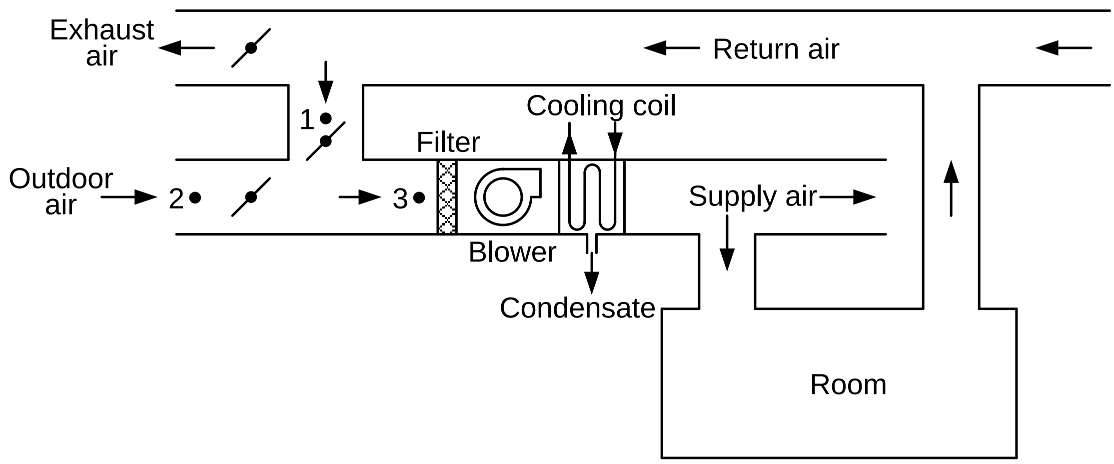


Figure 4.8 Mixing process of return and outdoor fresh air streams

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, °C	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^oC 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

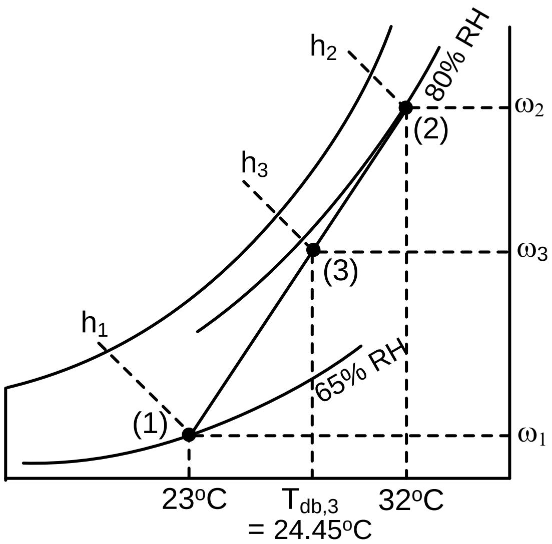


Figure 4.9 Mixing process of return and outdoor fresh air on Psychrometric chart

For dry-bulb temperature of 32^oC 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°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°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.

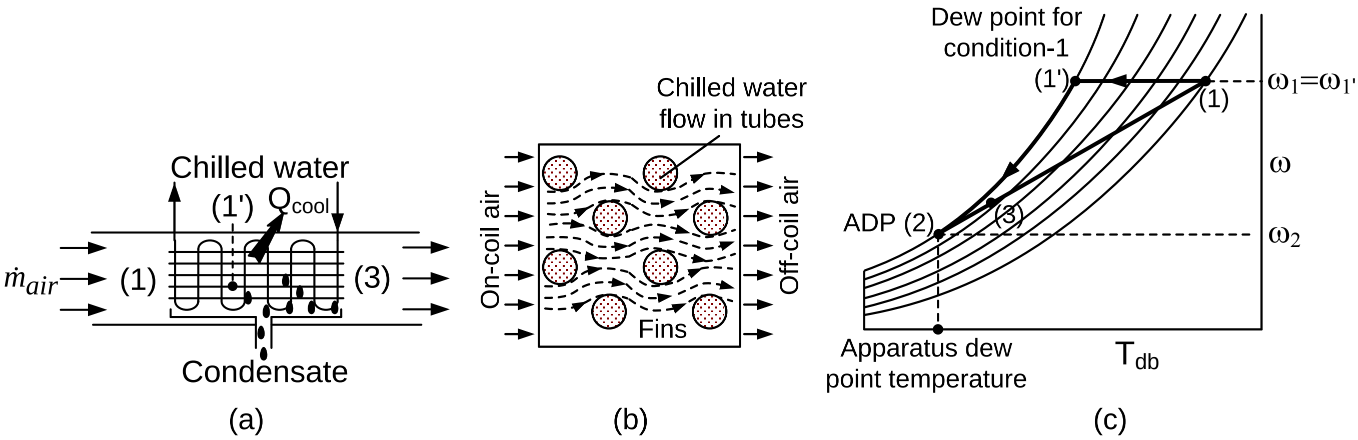


Figure 4.10. Cooling of moist air below dew point: (a) Process of cooling coil, (b) Flow of air inside cooling coil and (c) States of air on Psychrometric chart

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.

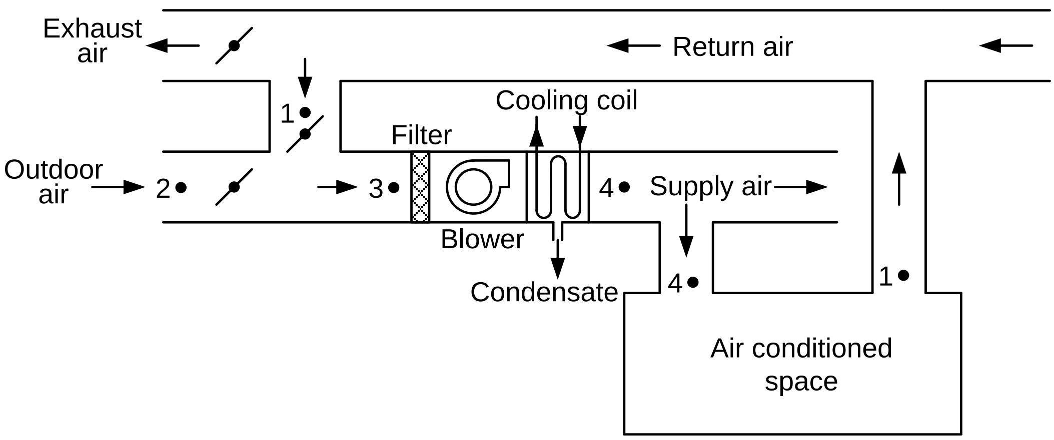


Figure 4.11 Typical AHU and ducting system

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 = $m_4(h_1\text{-}h_5)$

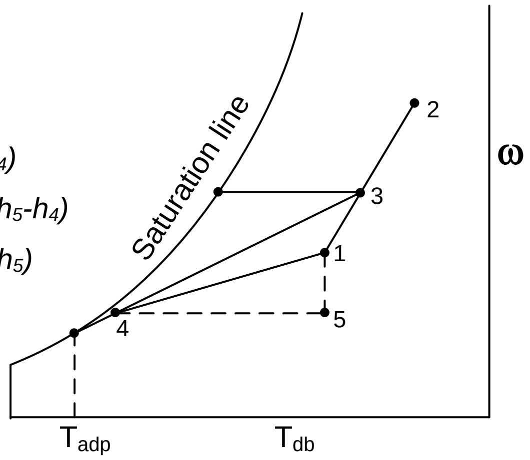


Figure 4.12 Typical AHU and space conditioning processes on the Psychrometric chart

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^oC and 95% respectively.

Calculate:

a) Heat removal rate by the chilled water. Assume condensate temperature is 14^oC

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.

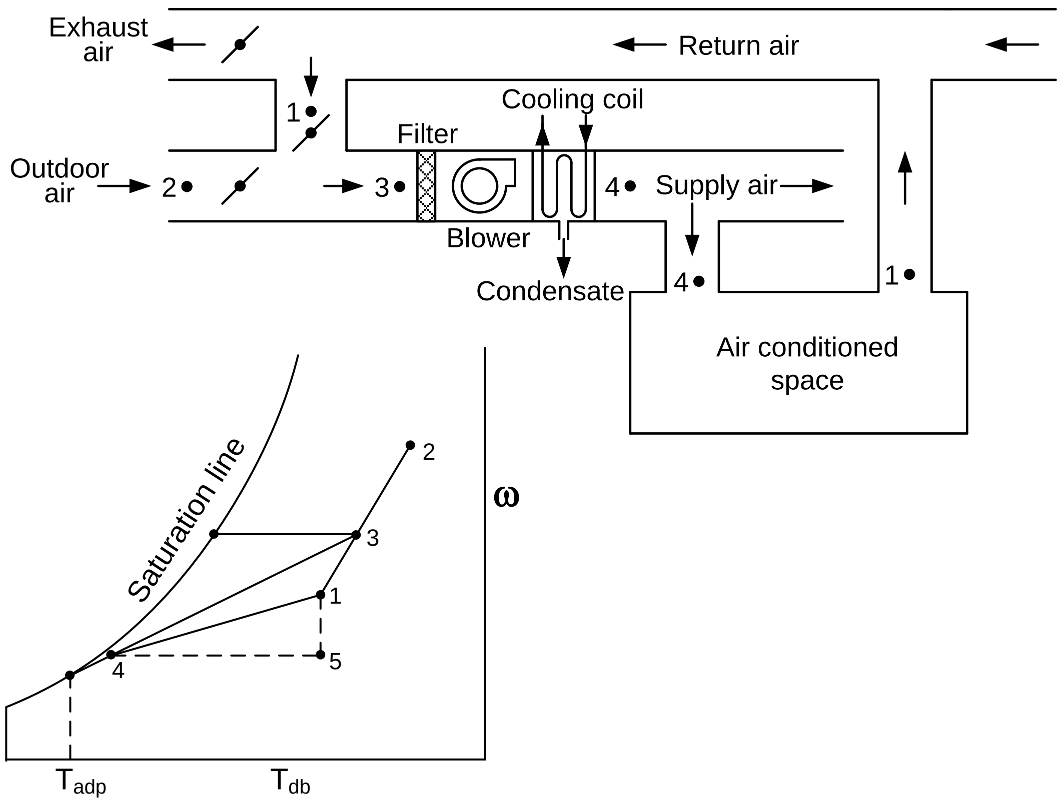


Figure 4.15 AHU system and the space conditioning processes

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 $= m_3(\omega_3 - \omega_4)/1000$

$= 3.879(13.59\text{-}9.5)/1000 = 0.0159$ kg/s

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

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

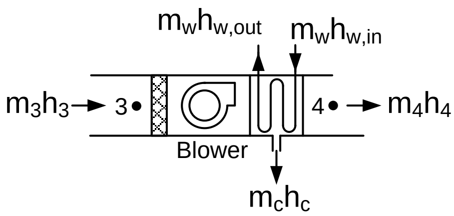


Figure 4.16 Heat transfer processes involve in AHU

a) Energy balance for the AHU gives:

$$m_w h_{w,out} + m_4 h_4 + m_c h_c = m_w h_{w,in} + m_3 h_3$$

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

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

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

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 $= m_3(\omega_1 - \omega_4)/1000$

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

Energy balance gives:

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

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

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

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

$= 3.879(24.5 - 9.5)/1000 = 0.058$ kg/s

Energy balance gives:

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

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

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.

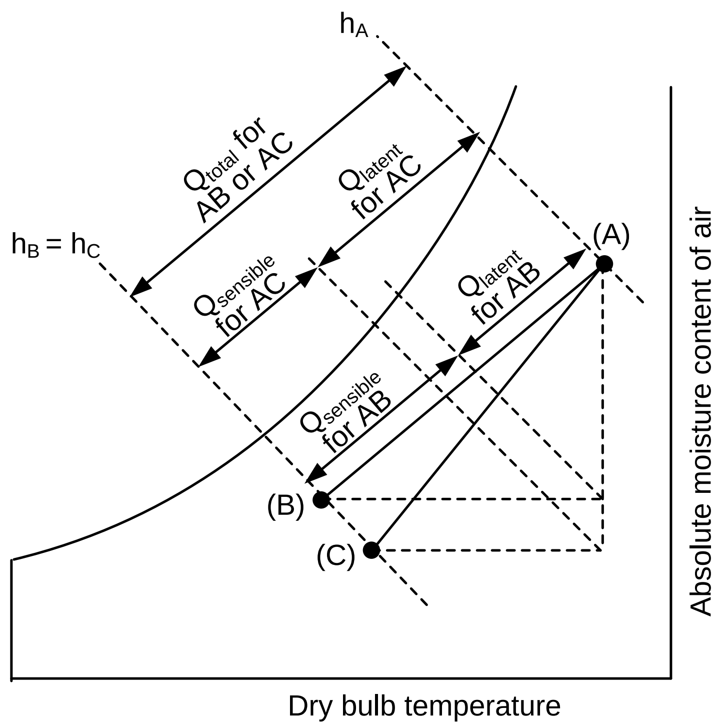


Figure 4.17 Sensible, latent and total heat removal capacity of the cooling coil of AHU

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.

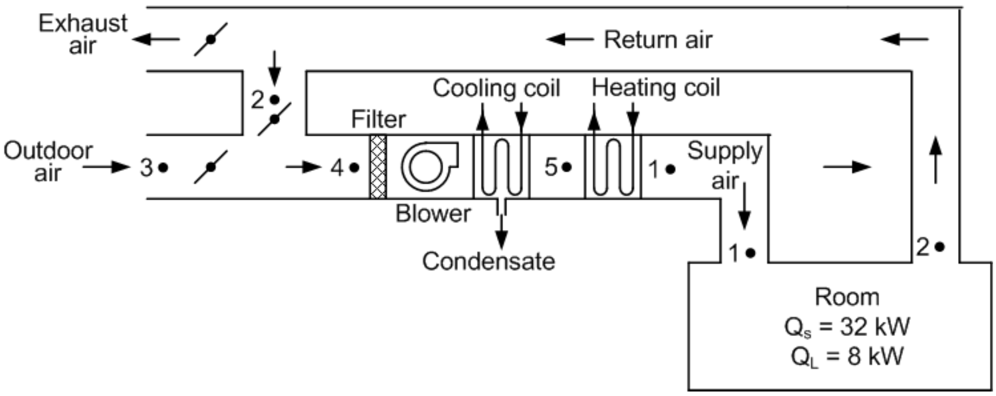


Figure 4.18 Outline diagram of the AHU system

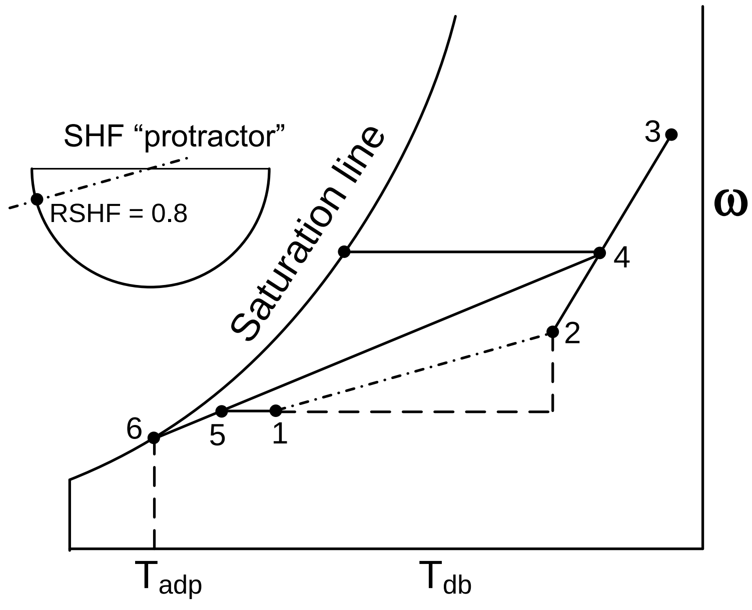


Figure 4.19 Process lines on Psychrometric chart

(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^oC_{db}, 50% RH)

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

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

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

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

Using Psychrometric chart:

h_4 = 54 kJ/kg dry air (using 28.25^oC_{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:

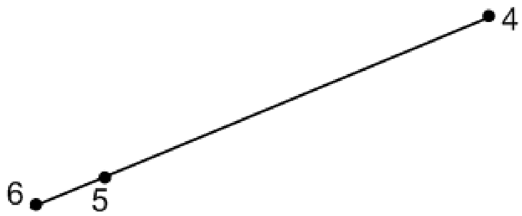
Total room load = 40 kW = M_{sa} ($h_2 - h_1$)

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)”



$T_6 = 7^oC_{db}$, $T_4 = 28.25^oC_{db}$,

Using Psychrometirc chart $T_5 = 13.2^oC_{db}$

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.

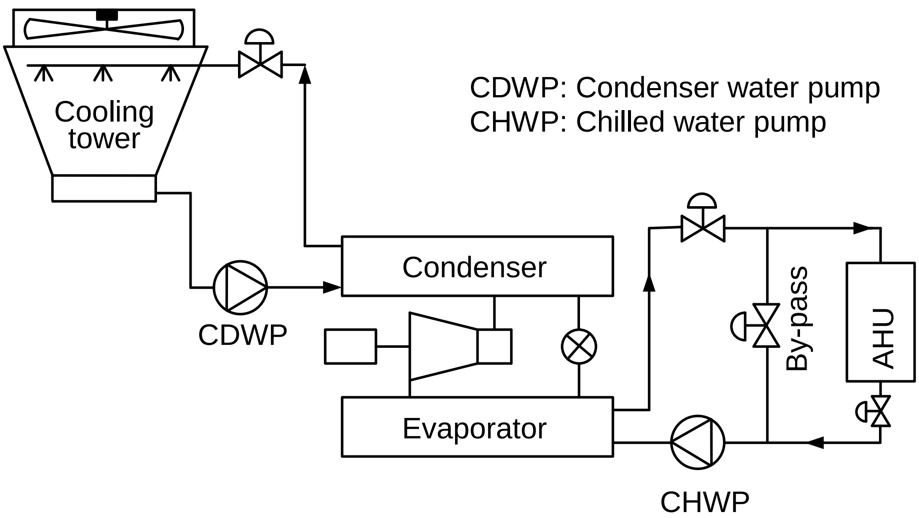


Figure 5.1 Central air-conditioning chilled water system with cooling tower.

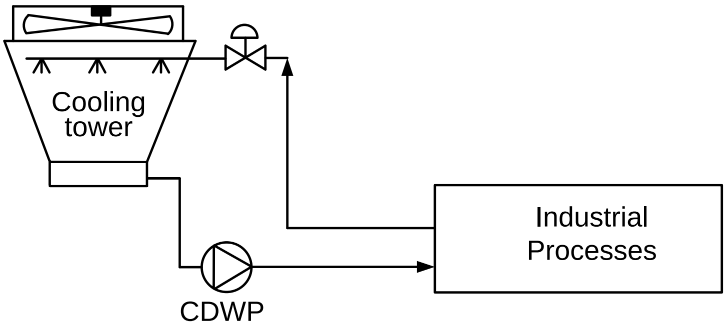
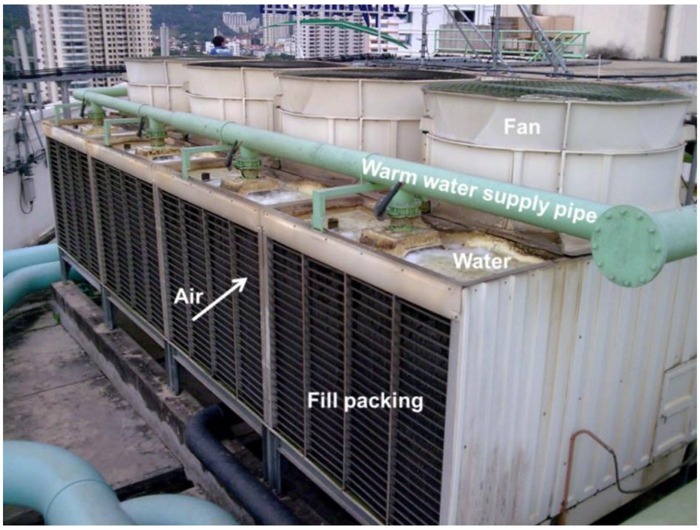
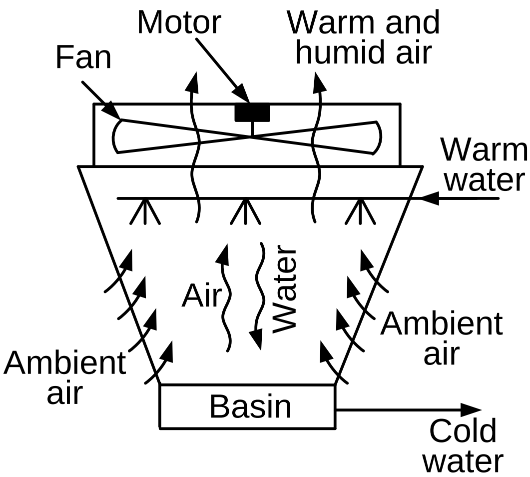


Figure 5.2 Cooling tower for cooling of industrial processes.





(a) (b)

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.

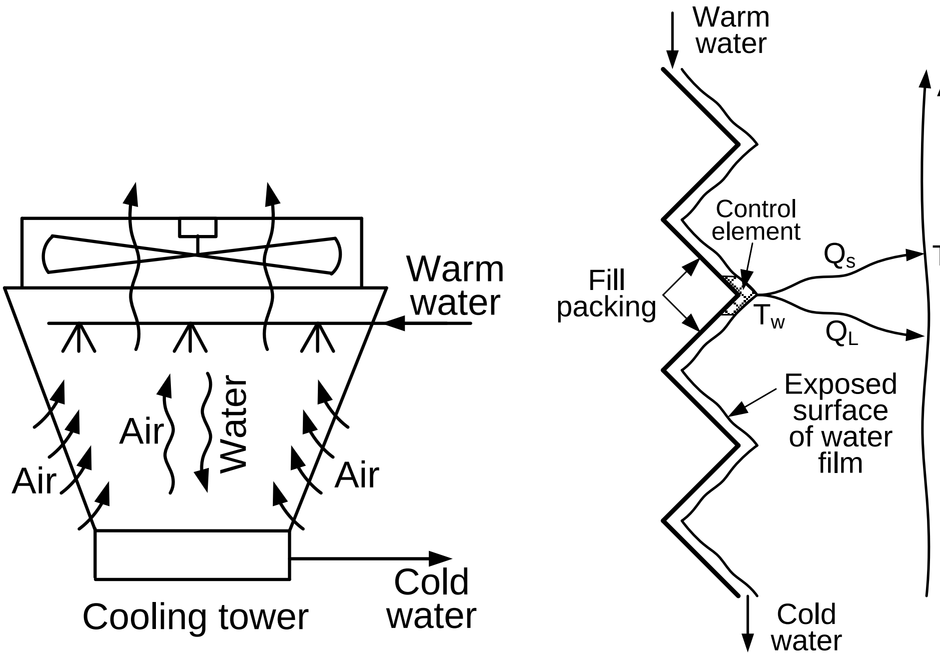


Figure 5.4 Heat transfer processes of cooling towers

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

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

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

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°C and leaves at 30°C. Ambient air flows in cross flow configuration through the packing materials of the cooling tower at an average temperature of 32°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, °C	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°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°C and RH = 70%.

Saturated vapour pressure at 32°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:

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

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

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

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

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

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.

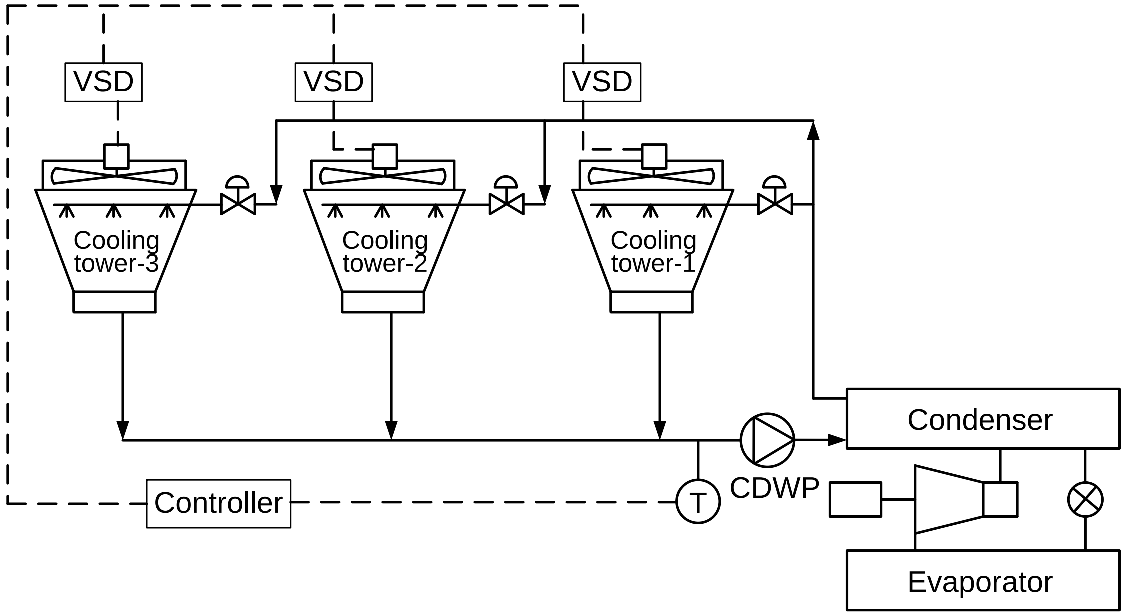


Figure 5.5 Control of cooling tower capacity based on water leaving temperature from the cooling tower

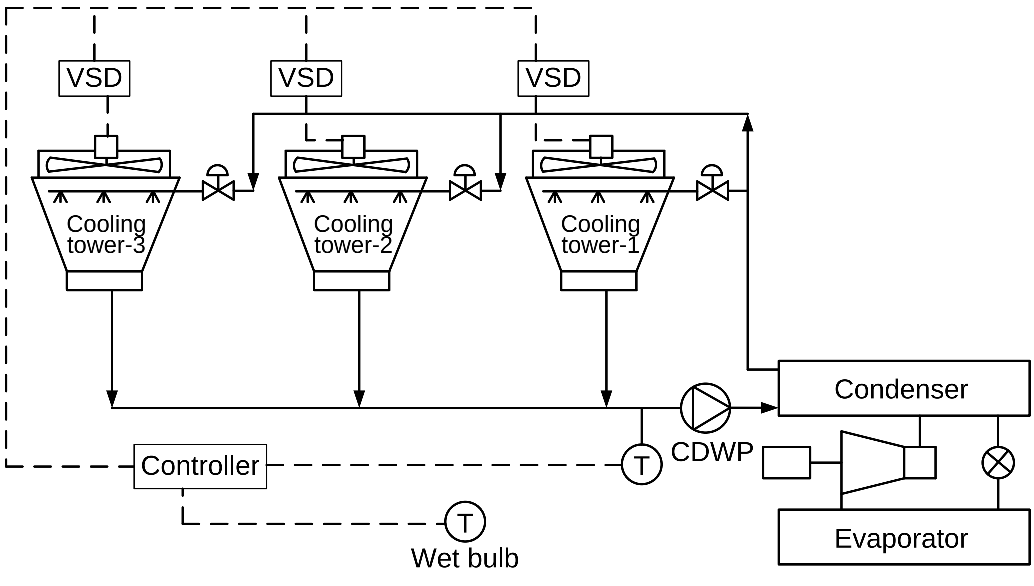


Figure 5.6 Control of cooling tower capacity based on approach temperature

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.

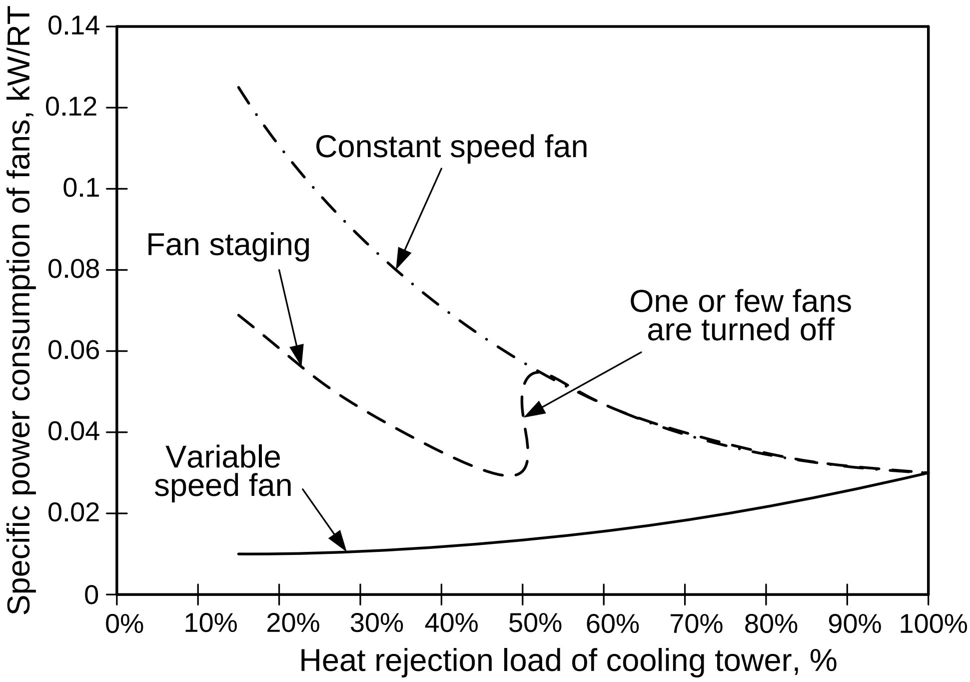


Figure 5.7 Specific power consumption of cooling tower fans for different operating strategies

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.

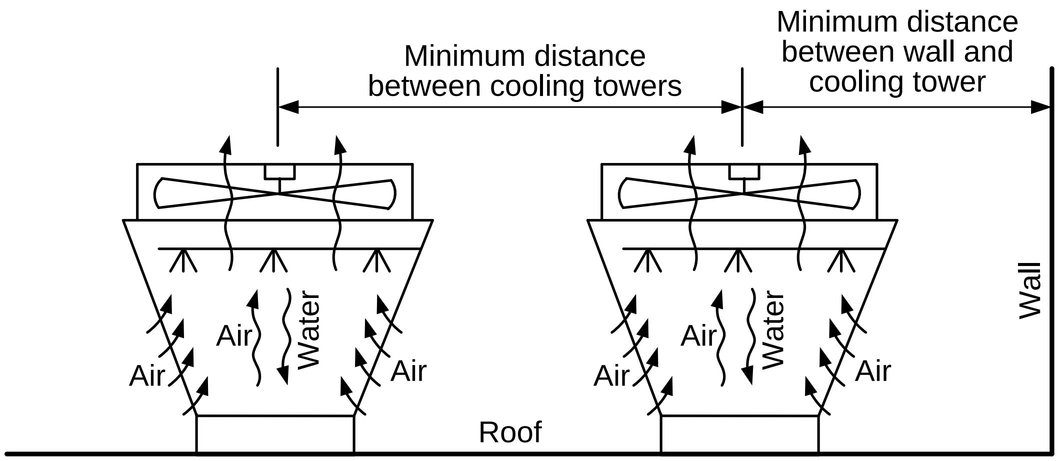


Figure 5.7 Minimum distance between the cooling towers and the surrounding obstructions

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.

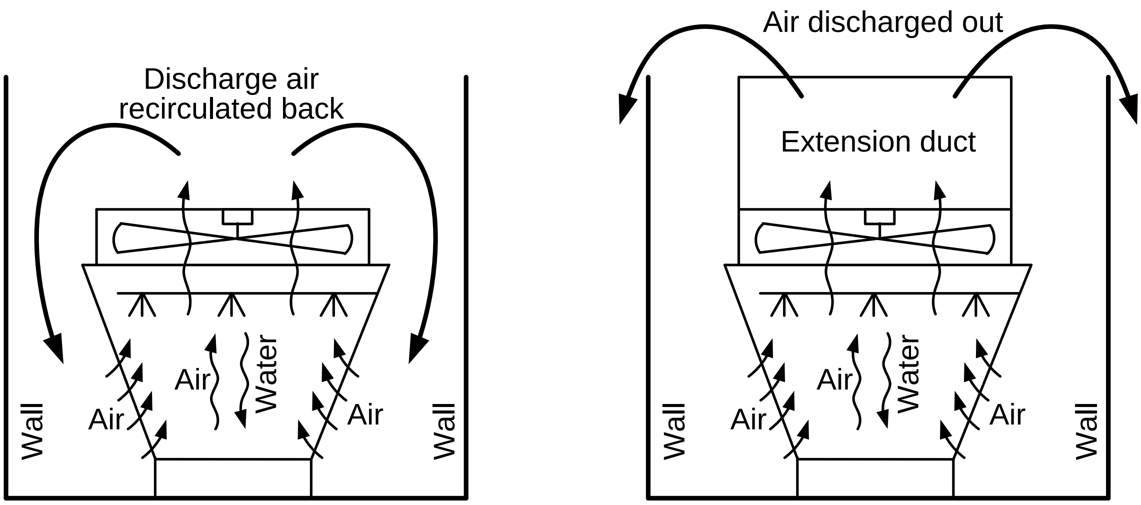


Figure 5.8 Extension duct to direct the warm and humid discharged air flow away from the air intakes

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^{o}C at a rate of 0.1 m^3/s. The water is cooled to 29^{o}C in the cooling tower by air which enters the tower at 1 atm, 28^{o}C, 70% RH and leaves at 32^{o}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^{o}C and 29^{o}C are 146.68 kJ/kg and 121.61 kJ/kg respectively.

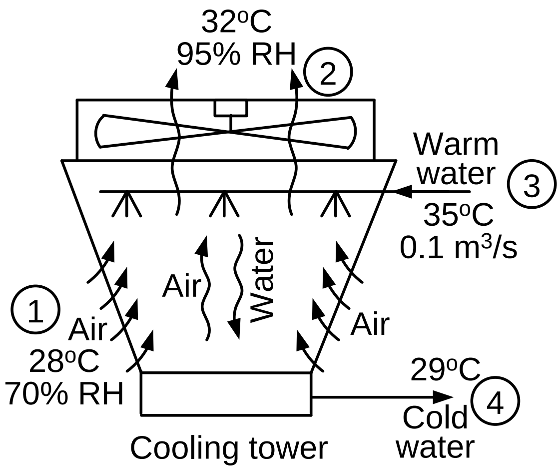


Figure 5.9 Operating parameters of cooling tower

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:

$$Q_{reject} = V\rho C_p \Delta T \tag{5.10}$$

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

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.

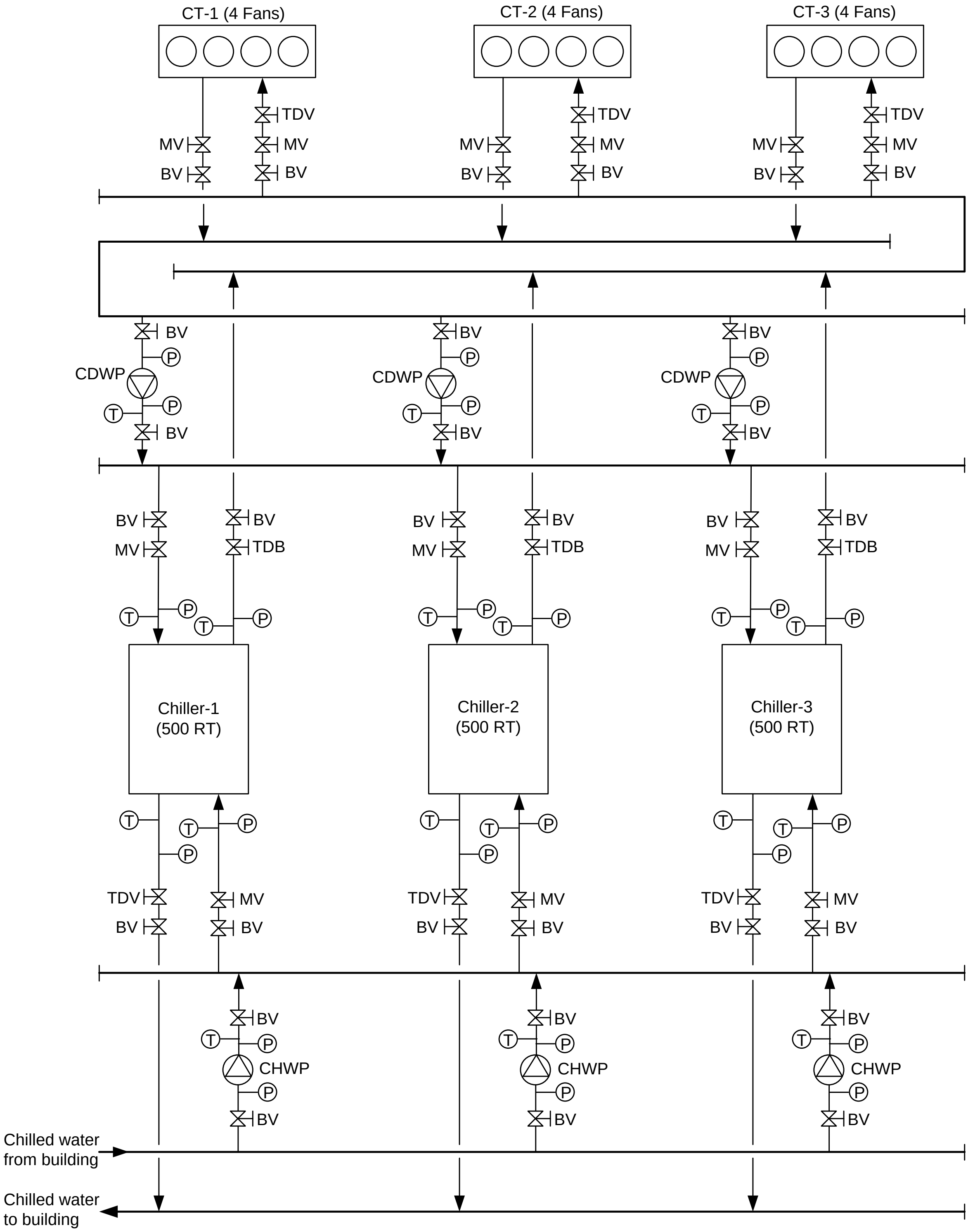


Figure 6.1 Water-cooled central chilled water system before retrofitting work (CHWP: Chilled water pump, CDWP: Condenser water pump, CT: Cooling tower, BV: Butterfly valve, MV: Motorized valve, TDV: Triple duty valve)

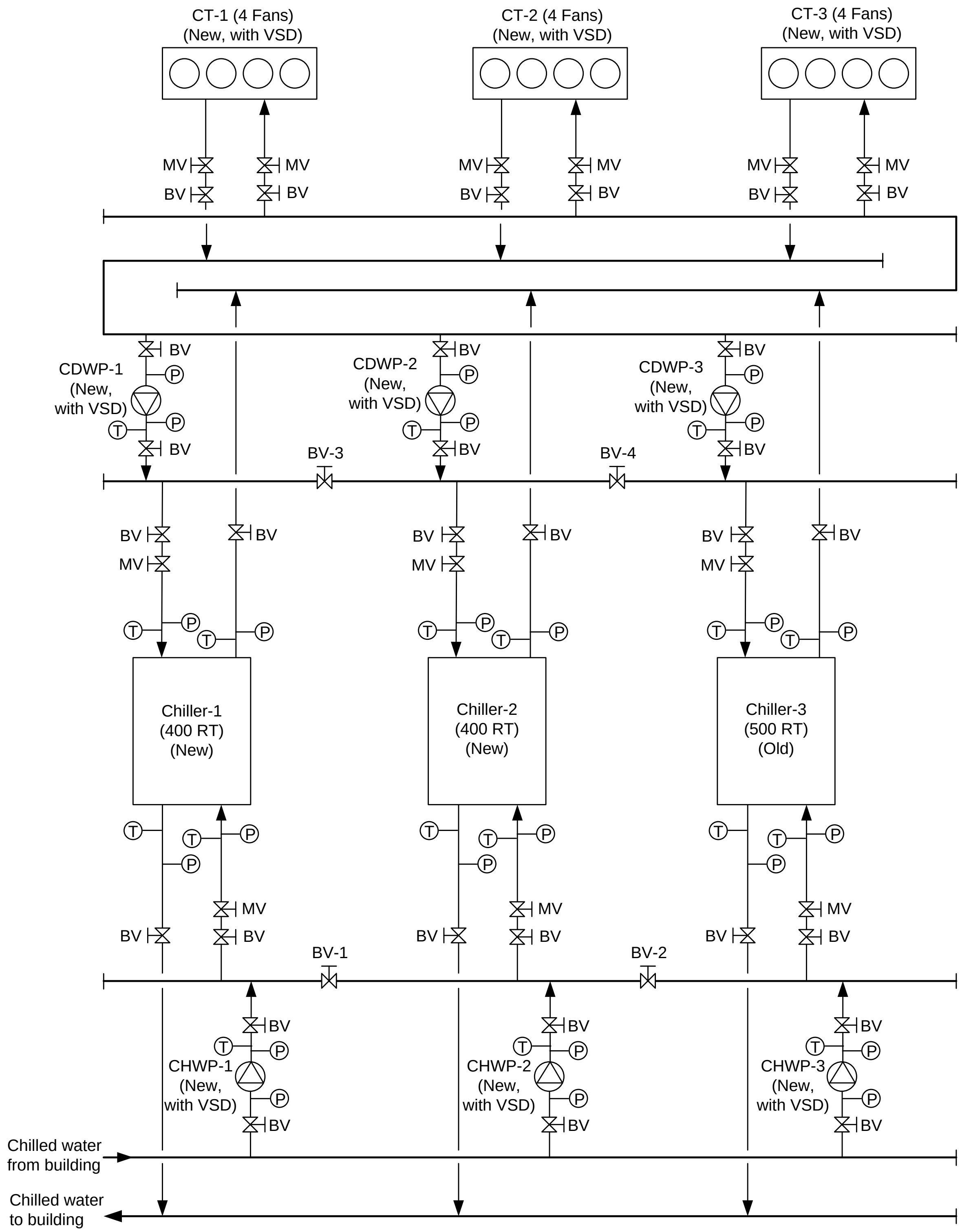


Figure 6.2 Water-cooled central chilled water system after retrofitting work (CHWP: Chilled water pump, CDWP: Condenser water pump, CT: Cooling tower, BV: Butterfly valve, MV: Motorized valve)

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

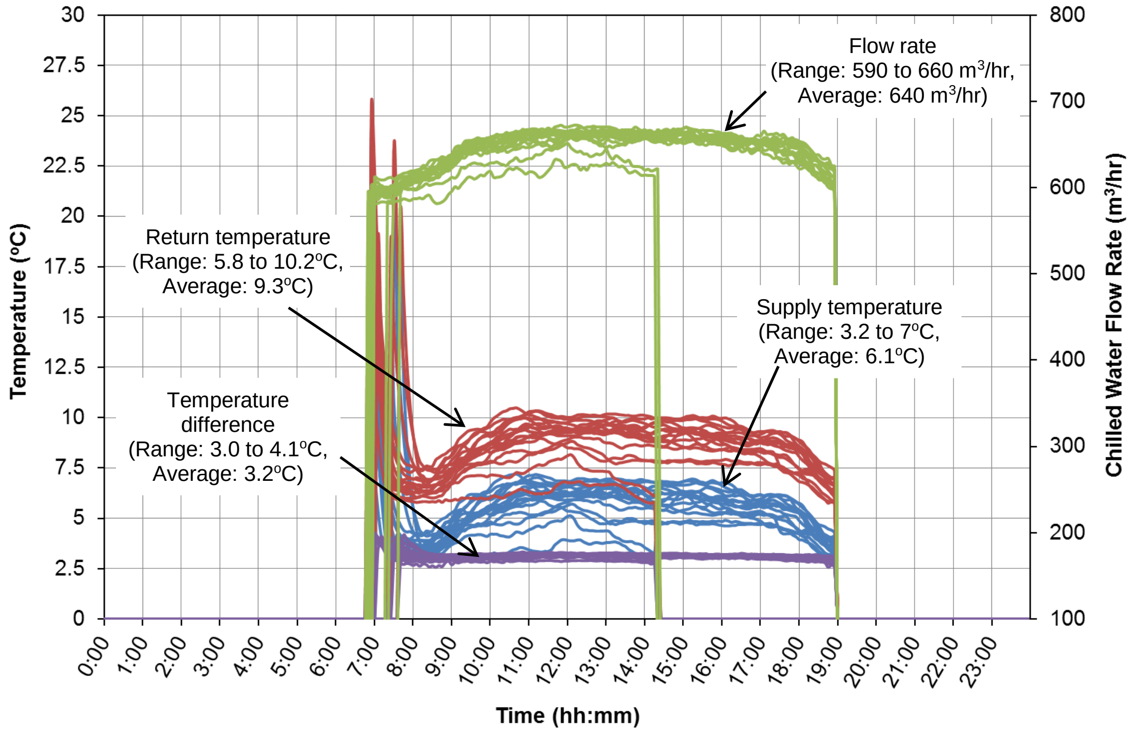


Figure 6.3 Total flow rate and temperature profiles of chilled water for 2 nos. of old chillers

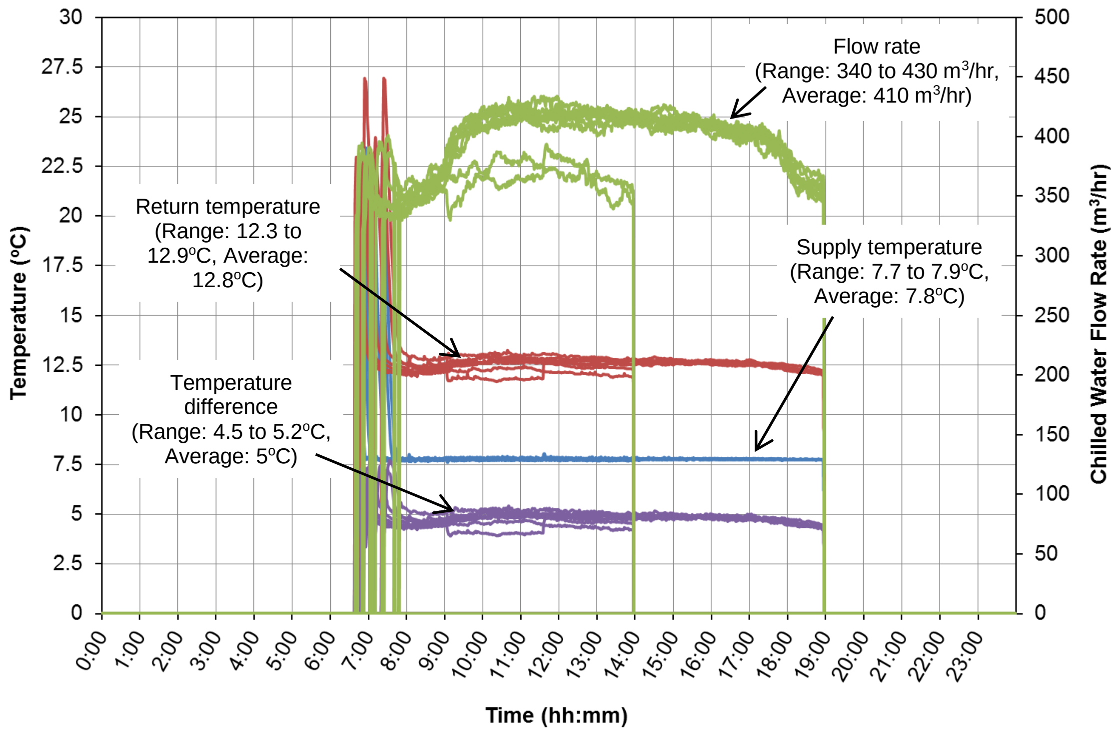


Figure 6.4 Total flow rate and temperature profiles of chilled water for 2 nos. of retrofitted chillers

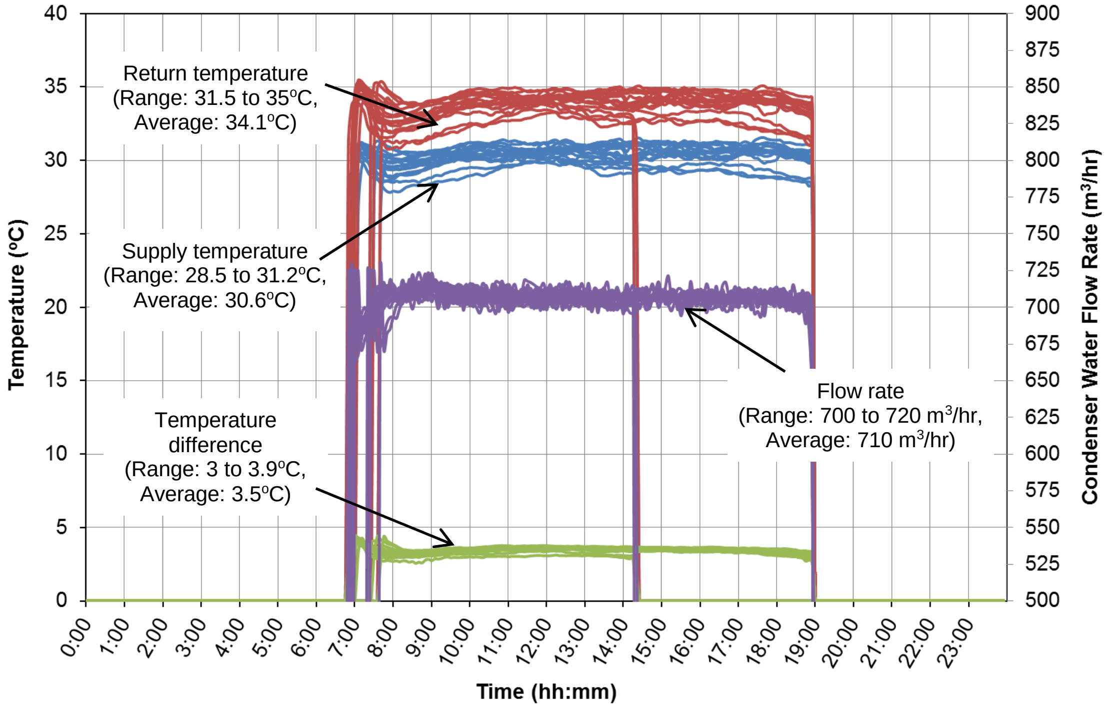


Figure 6.5 Total flow rate and temperature profiles of condenser water for 2 nos. of old chillers

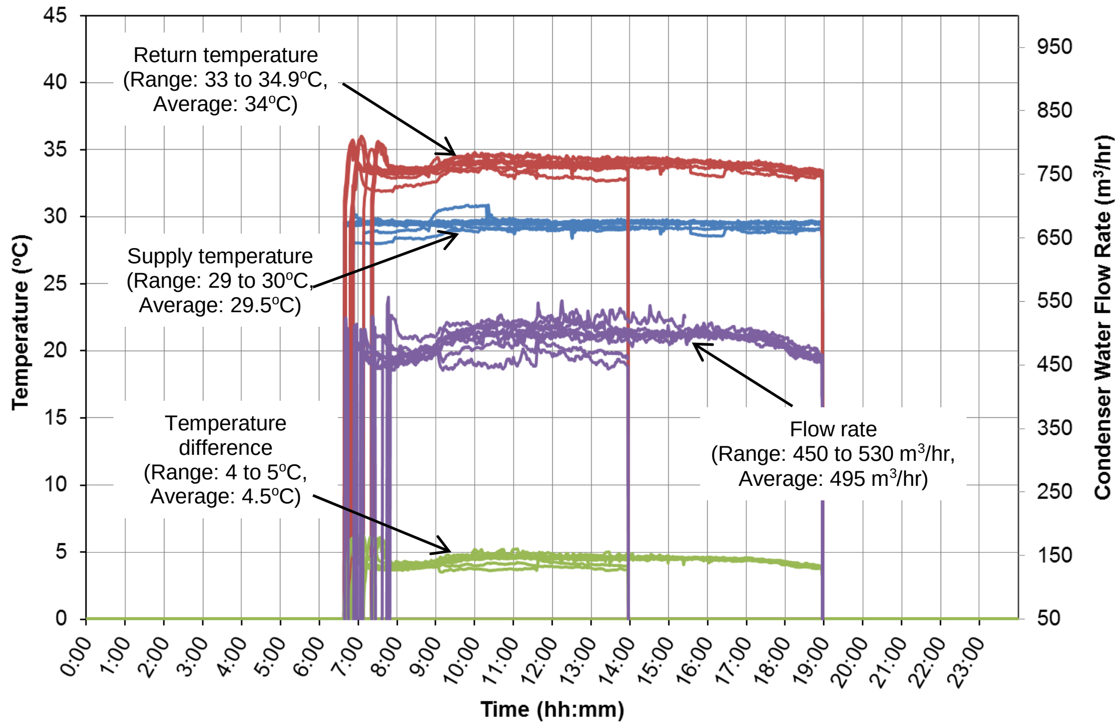


Figure 6.6 Total flow rate and temperature profiles of condenser water for 2 nos. of retrofitted chillers

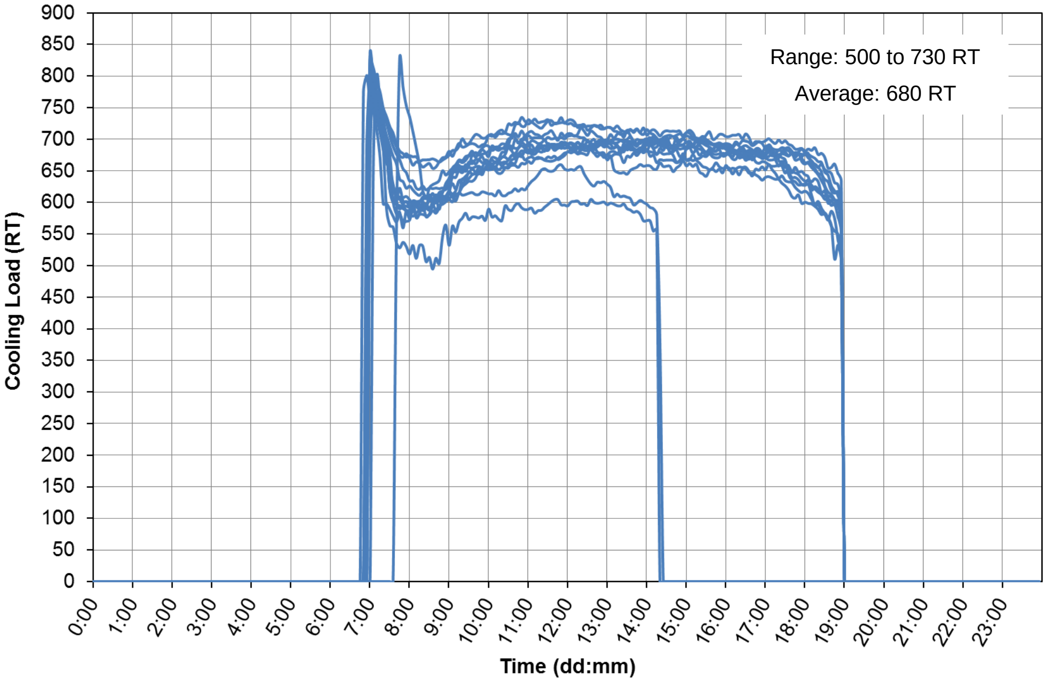


Figure 6.7 Total cooling load profile for 2 nos. of old chillers

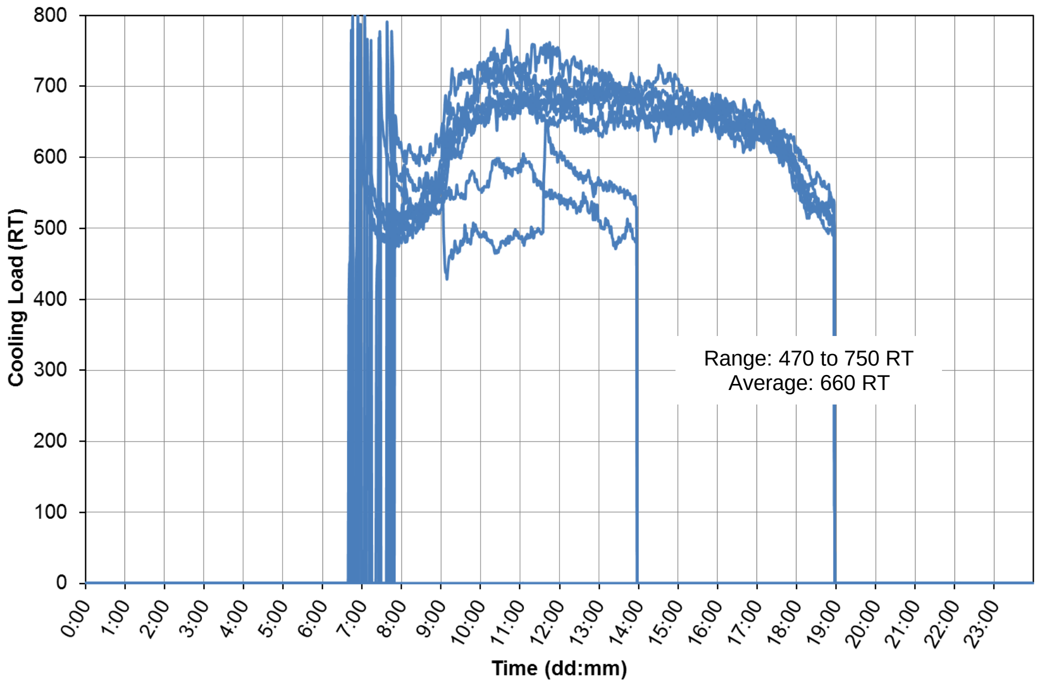


Figure 6.8 Total cooling load profile for 2 nos. of retrofitted chillers

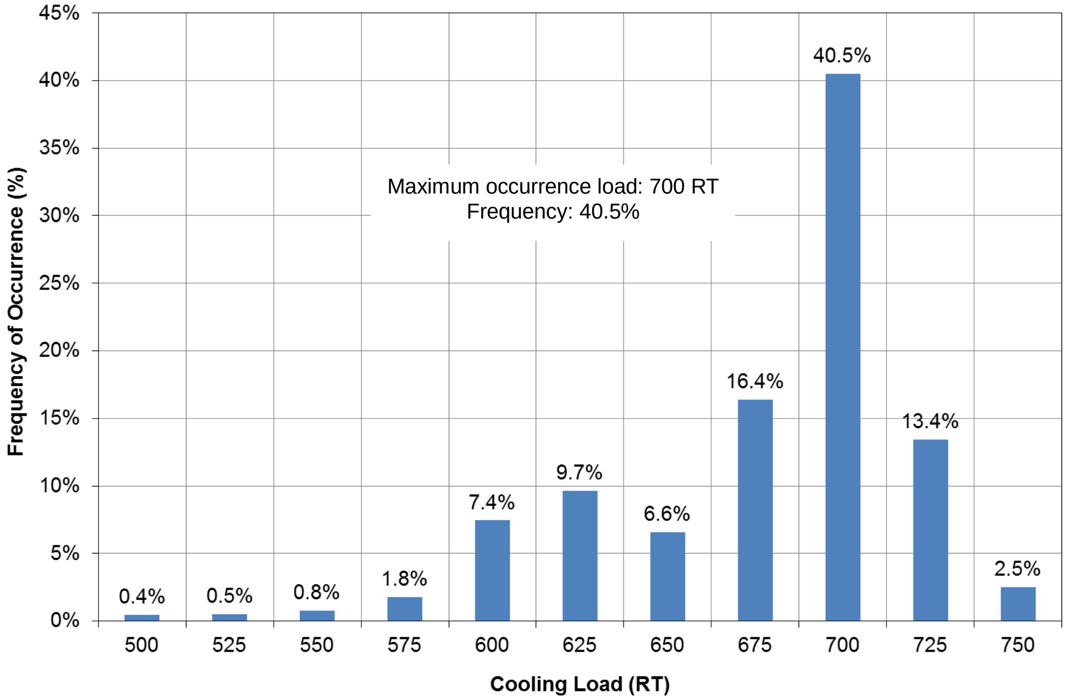


Figure 6.9 Frequency of cooling load occurrence for old chilled water system

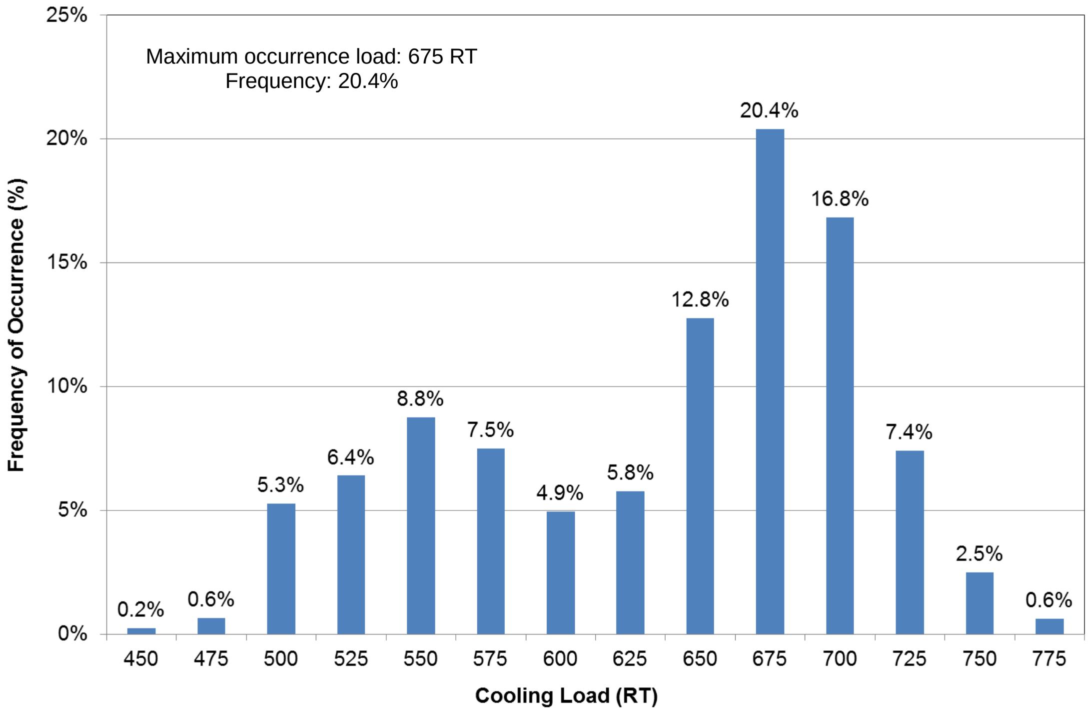


Figure 6.10 Frequency of cooling load occurrence for retrofitted chilled water system

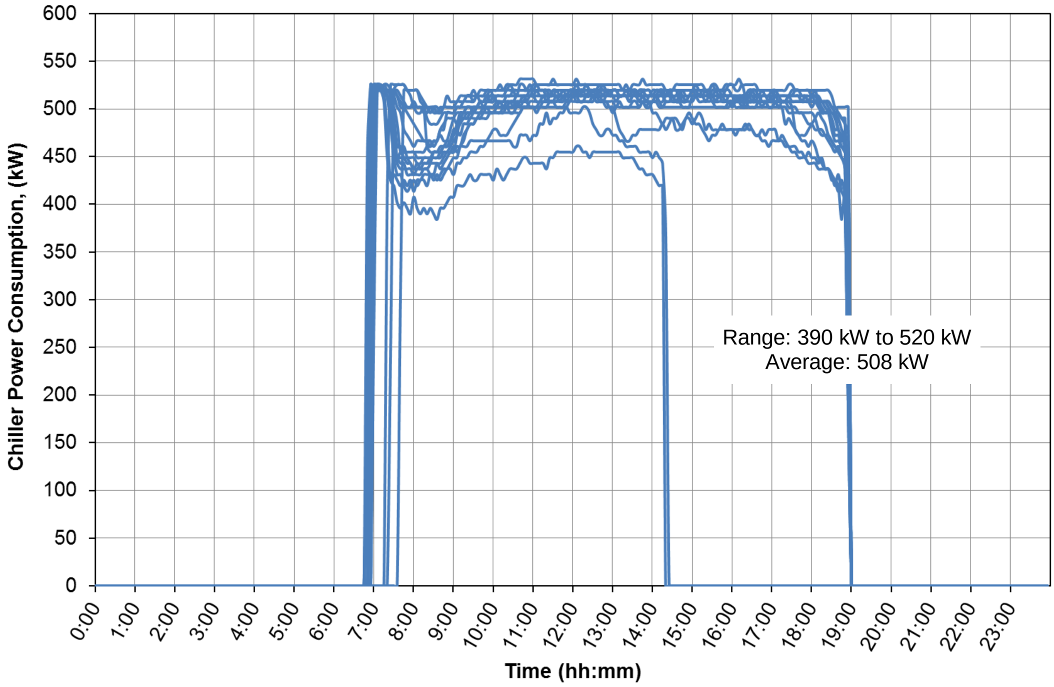


Figure 6.11 Total power consumption profile for 2 nos. of old chillers

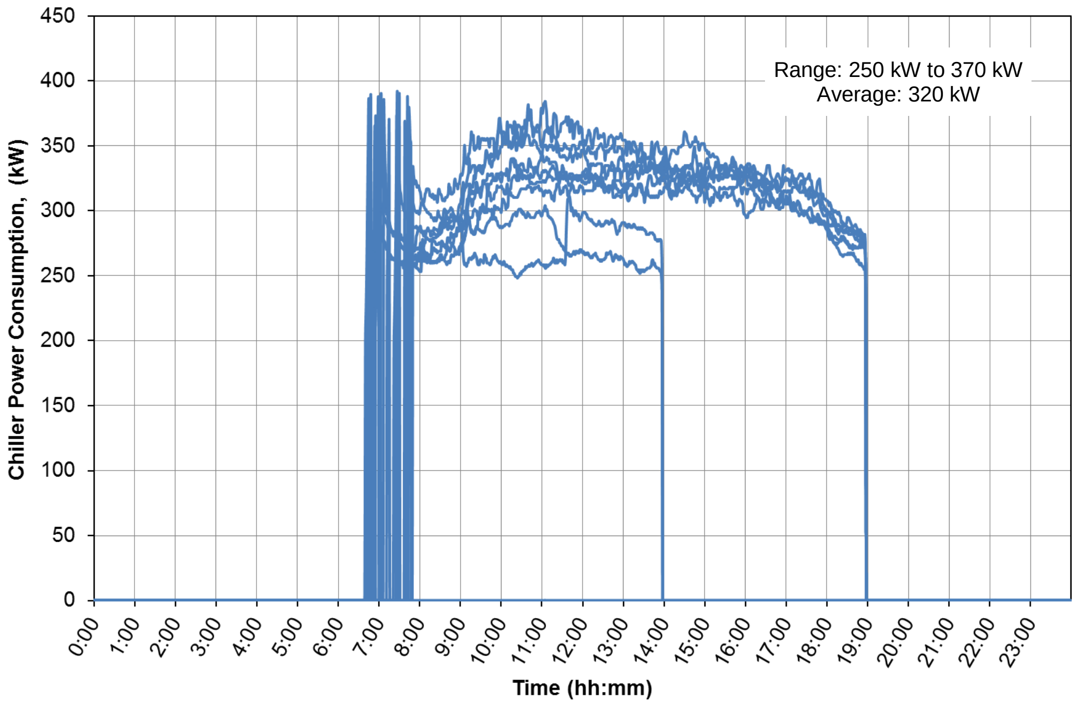


Figure 6.12 Total power consumption profile for 2 nos. of retrofitted chillers

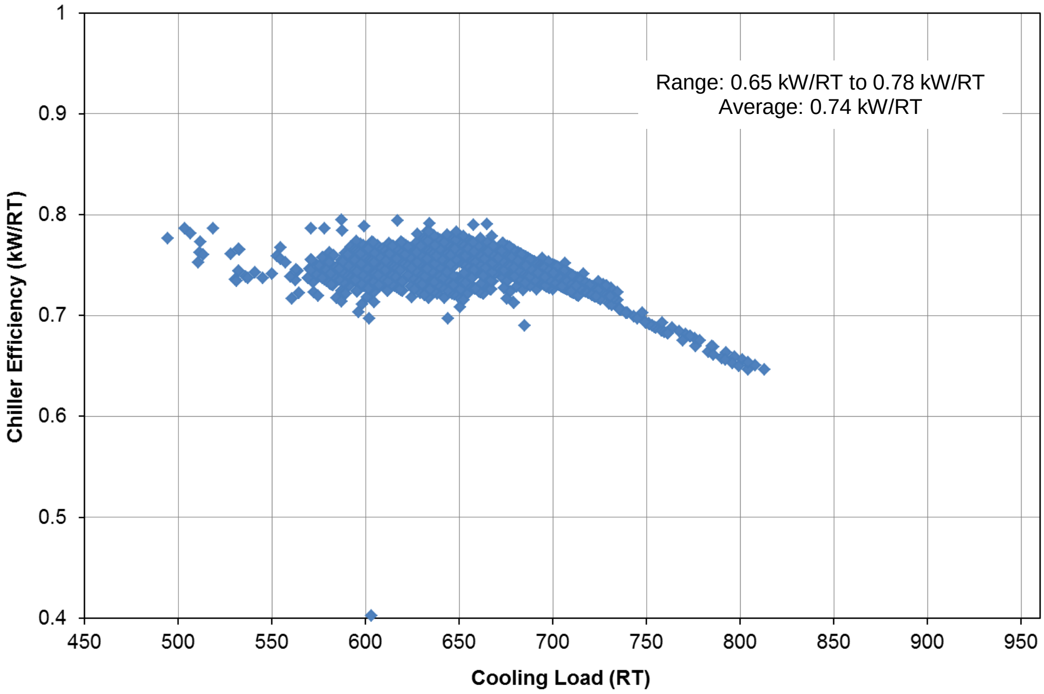


Figure 6.13 Efficiency profile for 2 nos. of old chillers

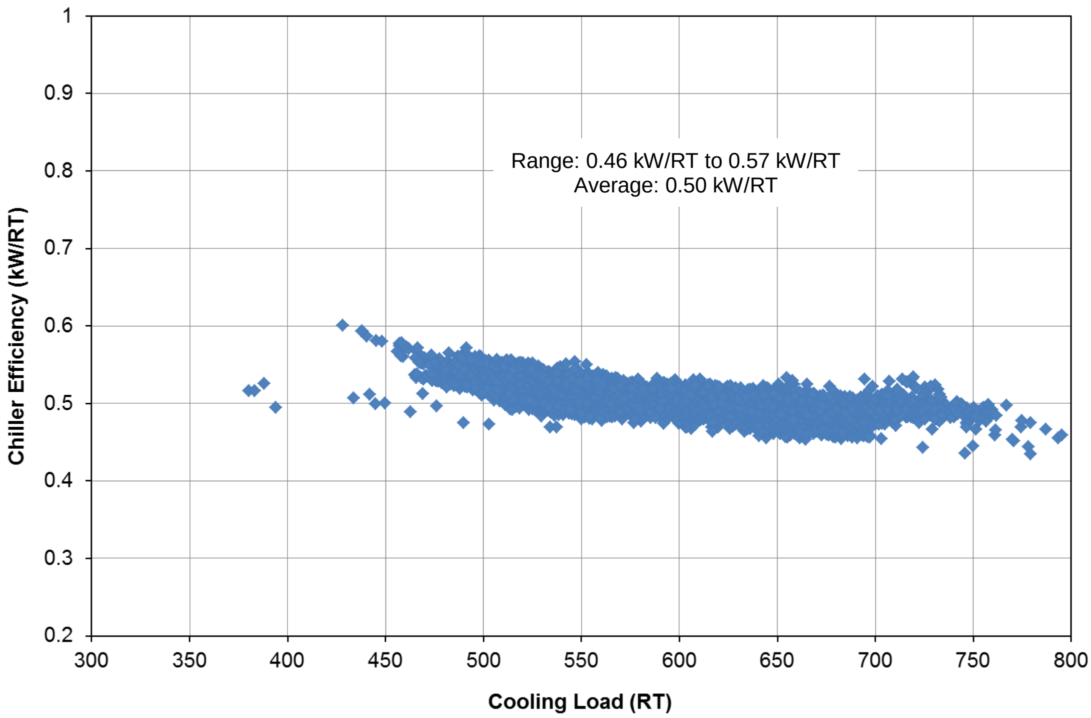


Figure 6.14 Efficiency profile for 2 nos. of retrofitted chillers

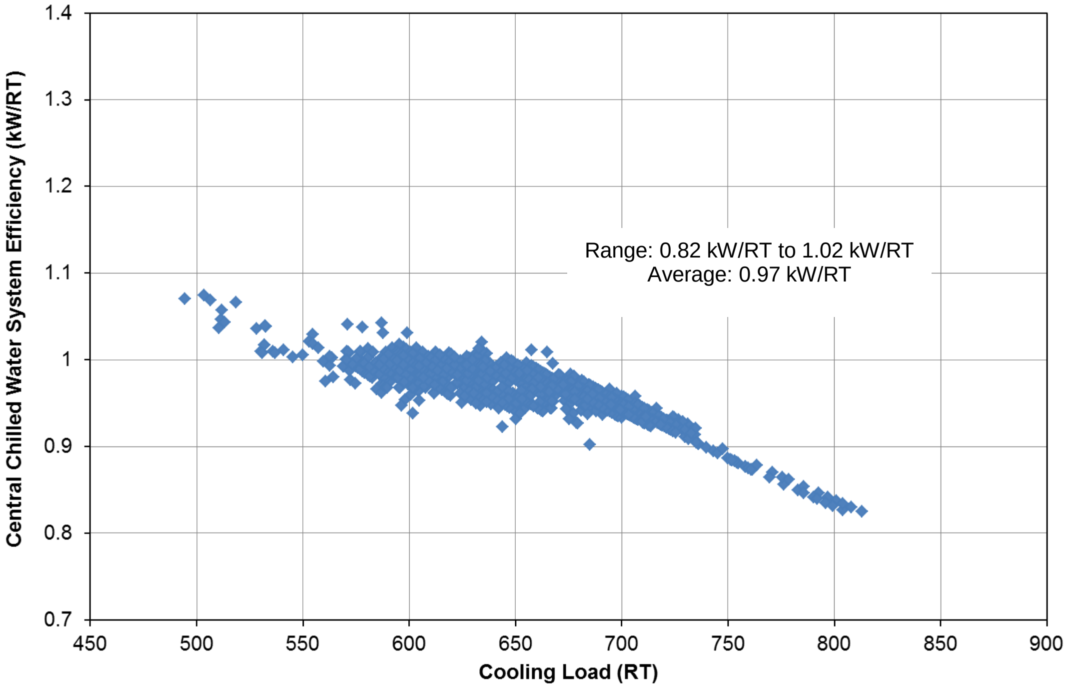


Figure 6.15 Old central chilled water system efficiency profile

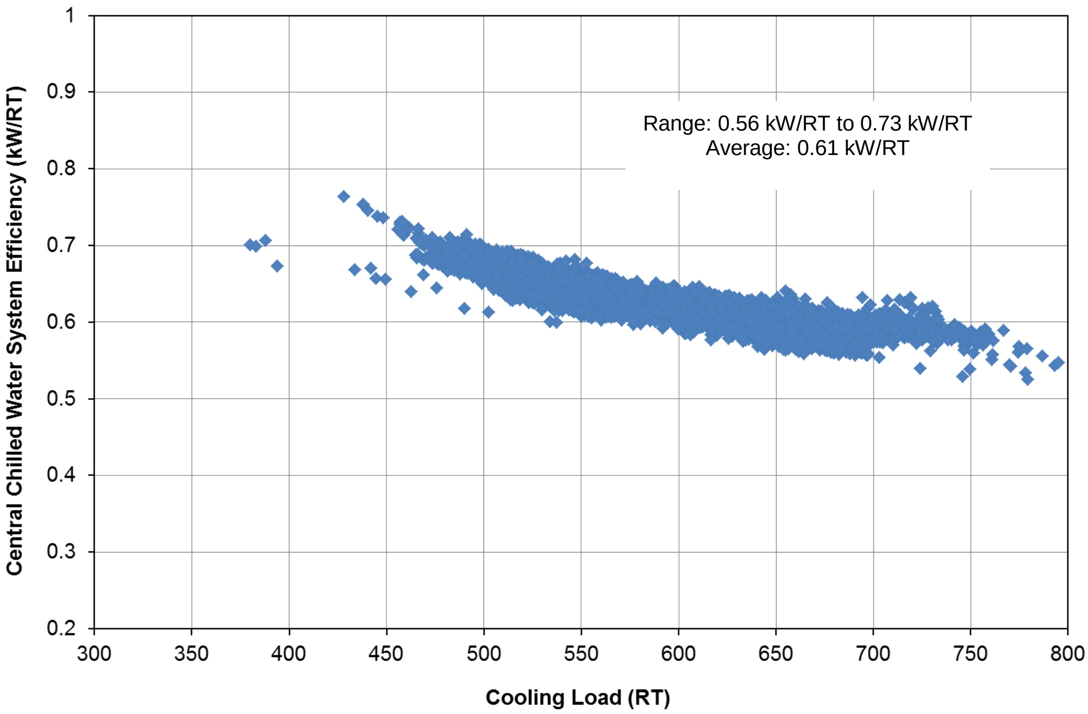


Figure 6.16 New central chilled water system efficiency profile

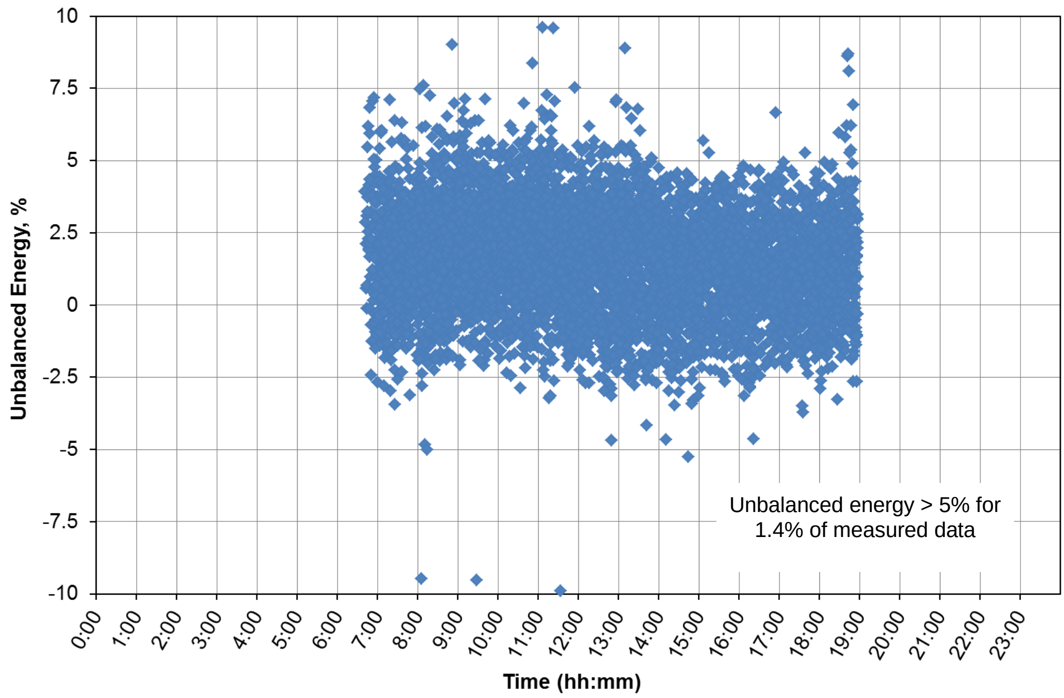


Figure 6.17 Percentage of unbalanced energy for old chilled water system

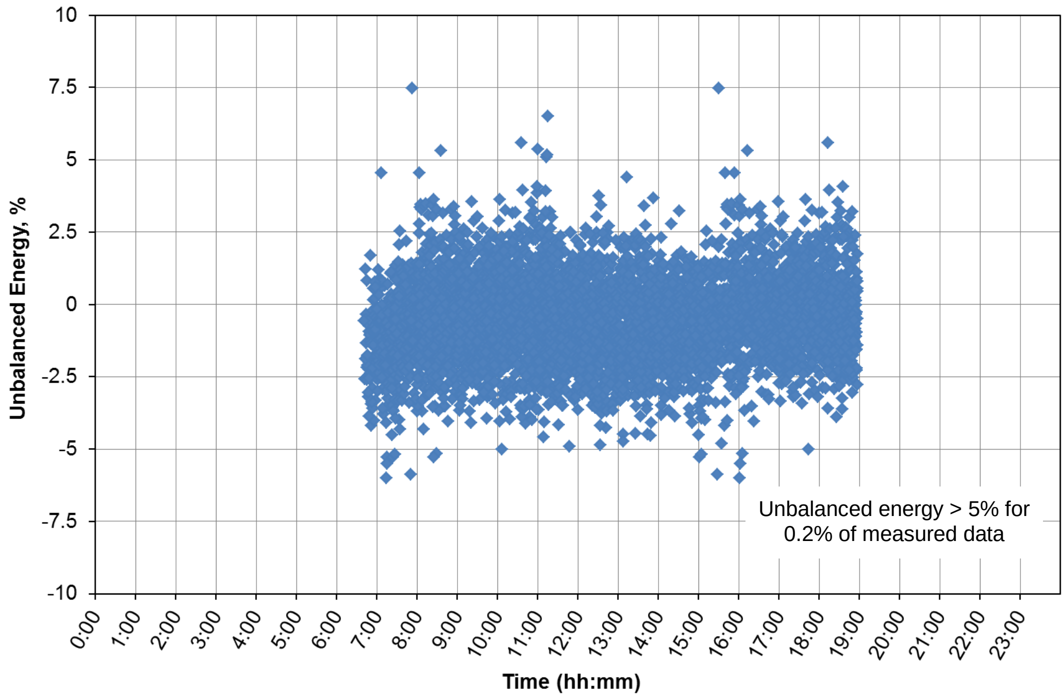


Figure 6.18 Percentage of unbalanced energy for retrofitted chilled water system

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 3.2ºC to 7.0ºC with an average value of 6.1ºC.
iv. To enhance the energy performance of the retrofitted chilled water system, chilled water supply temperature was increased to 7.8ºC. AHUs were able to maintain preset space temperature of 24ºC and relative humidity of 65% using the raised chilled water supply temperature of 7.8ºC.
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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