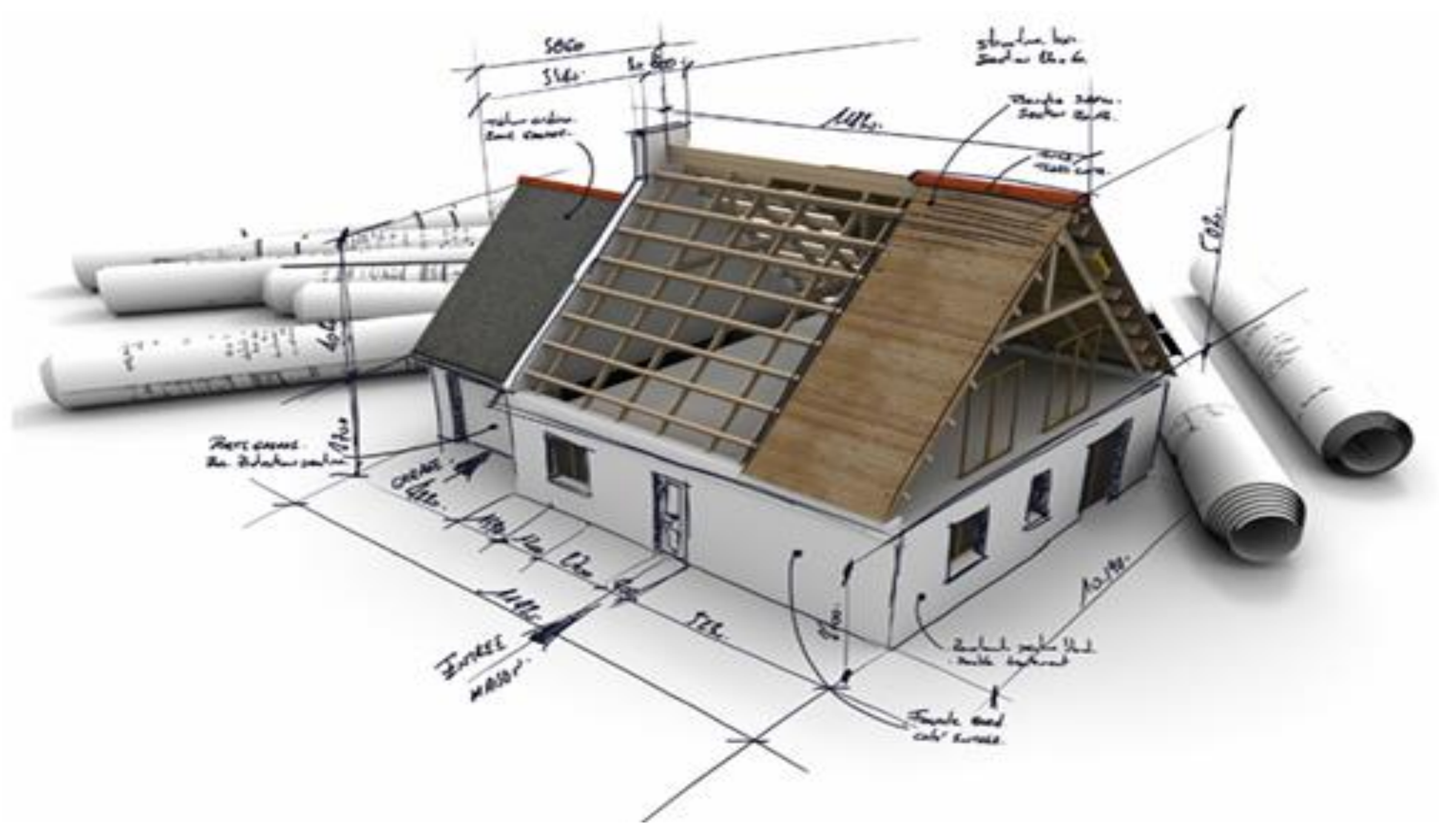
Fundamentals of HVAC Systems

Third edition



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Fundamentals of HVAC Systems

Third edition

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Chapter5- Smoke management contents

- **1 Introduction.**
- **2 Methods for smoke control.**
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Chapter5- Smoke management

1 Introduction

- The objective of fire safety is to provide some degree of protection for a building's occupants, the building and the property inside it, and neighboring buildings.

- Two basic approaches to fire protection are (1) to prevent fire Ignition and (2) to manage fire effects. Because it is impossible to prevent fire ignition completely, managing fire effect has become significant in fire protection design. Figure (1) shows a decision Tree for fire protection.

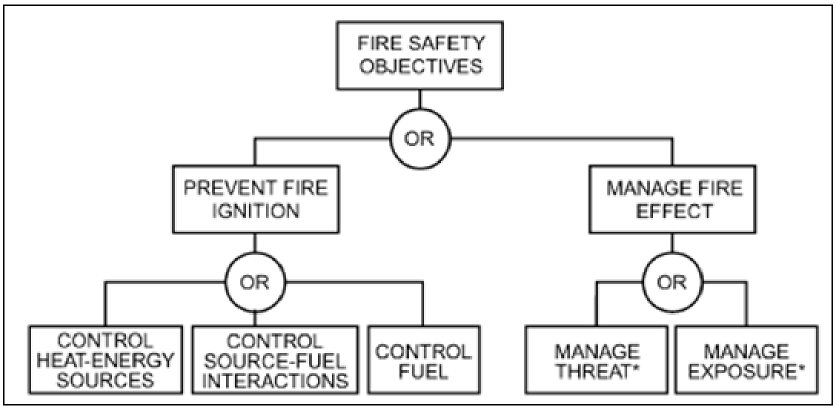


Fig (1). Simplified fire protection decision tree.

- The HVAC has traditionally been shut down when fire is discovered; this prevents fans from forcing smoke flow, but does not prevent smoke movement through ducts due to smoke buoyancy, stack effect, or wind. Subcontractors regarding appropriate work assignments.

1.1 Smoke movement

- A smoke control system must be designed so that it is not overpowered by the driving forces that cause smoke movement, which includes stack effect, buoyancy, expansion, wind, and the HVAC system. During a fire, smoke is generally moved by a combination of these forces.

1.1.1 Stack Effect

- When it is cold outside, air tends to move upward within building shafts (e.g., stairwells, elevator shafts, dumbwaiter shafts, mechanical shafts, mail chutes). This **normal stack effect** occurs because the air in the building is warmer and less dense than the outside air. Normal stack effect is large when outside temperatures are low, especially in tall buildings.

- When the outside air is warmer than the building air, there is a natural tendency for downward airflow, or **reverse stack effect**, in shafts. At standard atmospheric pressure, the pressure difference due to either normal or reverse stack effect is expressed as.

$$\Delta p = 7.64\left(\frac{1}{T_o} - \frac{1}{T_i}\right)h$$

where

Δp = pressure difference, in. of water
T_o = absolute temperature of outside air, °R
T_i = absolute temperature of air inside shaft, °R
h = distance above neutral plane, ft

- Figure (2) indicate the pressure difference between a building shaft and the outside. A positive pressure difference indicates that the shaft pressure is higher than the outside pressure, and a negative pressure difference indicates the opposite.

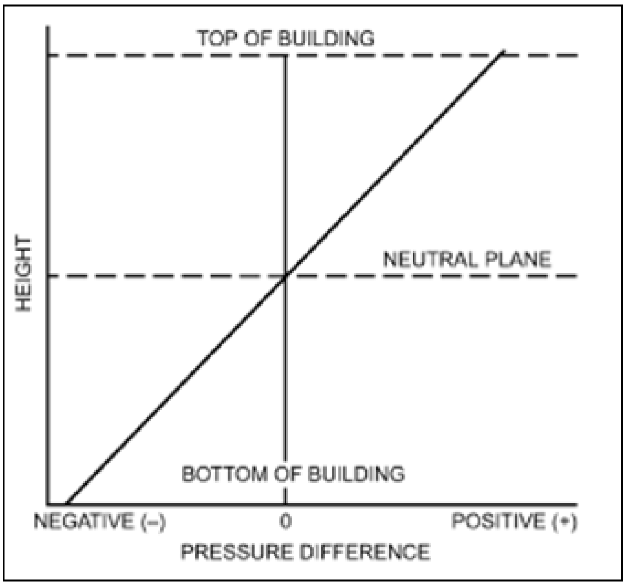


Fig (2).pressure difference between building shaft
And outside due to normal stack effect.

- Figure (3) illustrates the air movement in buildings caused by both normal and reverse stack effect. Also figure (3) can be used to determine the pressure difference due to stack effect. For normal stack effect, $\Delta p/h$ is positive, and the pressure difference is positive above the neutral plane and negative below it. For reverse stack effect, $\Delta p/h$ is negative, and the pressure difference is negative above the neutral plane and positive below it.

- If the leakage paths are uniform with height, the neutral plane is near the mid height of the building. However, when the leakage paths are not uniform, the location of the neutral plane can vary considerably.

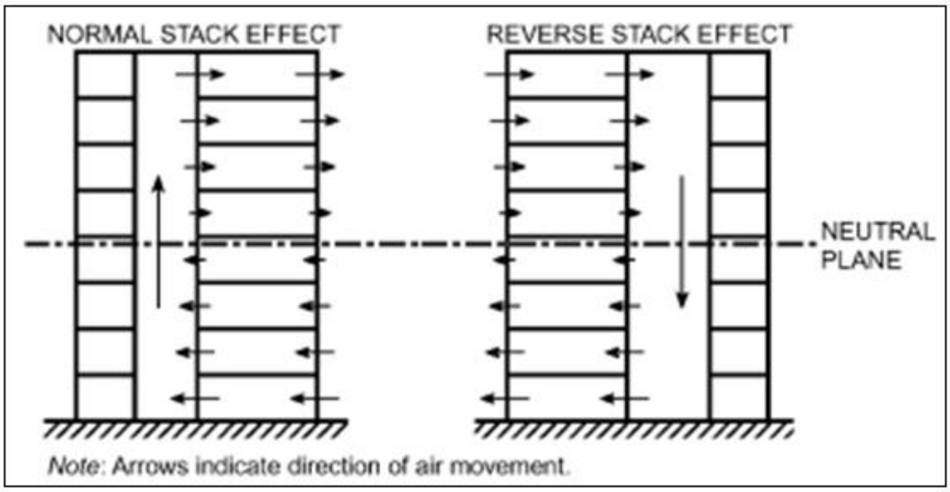


Fig (3). Air movement due to normal and reverse stack effect.

- Smoke movement from a building fire can be dominated by stack effect. If the fire is below the neutral plane, smoke moves with the building air into and up the shafts. This upward smoke flow is enhanced by buoyancy forces due to the temperature of the smoke. Once above the neutral plane, the smoke flows from the shafts into the upper floors of the building .if the floors below the neutral plane (except the fire floor) remain relatively smoke-free until the quantity of smoke produced is greater than can be handled by stack effect flows.

- Air currents caused by reverse stack effect tend to move relatively cool smoke down. In the case of hot smoke, buoyancy forces can cause smoke to flow upward, even during reverse stack effect conditions.

1.1.2 Buoyancy

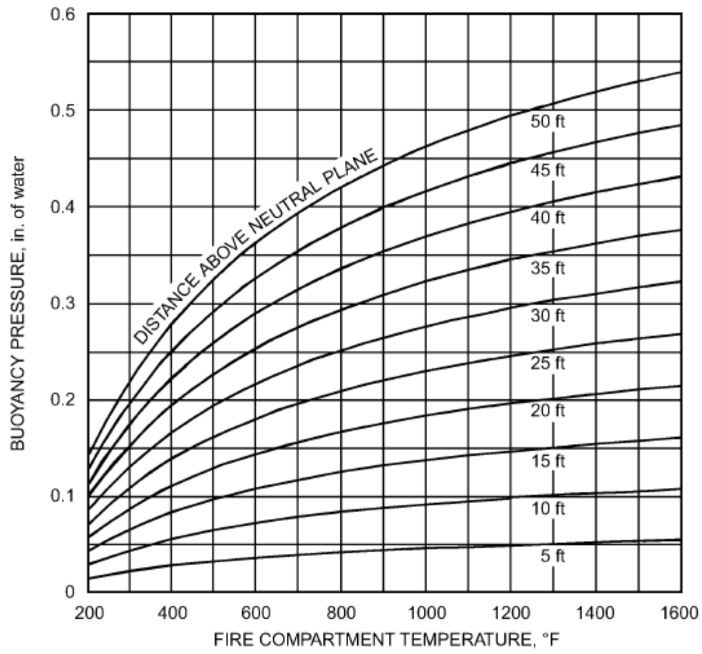
- High-temperature smoke has buoyancy because of its reduced density. At sea level, the pressure difference between a fire compartment and its surroundings can be expressed as follows:

$$\Delta p = 7.64\left(\frac{1}{T_s} - \frac{1}{T_f}\right)h$$

where

Δp = pressure difference, in. of water
T_s = absolute temperature of surroundings, °R
T_f = average absolute temperature of fire compartment, °R
h = distance above neutral plane, ft

- The pressure difference due to buoyancy can be obtained from the following graph.



Pressure Difference due to Buoyancy

- Much larger pressure differences are possible for tall fire compartments where the distance *h* from the neutral plane can be larger. In sprinkler-controlled fires, the temperature in the fire room remains at that of the surroundings except for a short time before sprinkler activation. Sprinklers are activated by the **ceiling jet**; a thin (2 to 4 in.) layer of hot gas under the ceiling. The maximum temperature of the ceiling jet depends on the location of the fire, the activation temperature of the sprinkler, and the thermal lag of the sprinkler heat-responsive element. For most residential and commercial applications, the ceiling jet is between 180 and 300°F. In Equation (2), T_f is the average temperature of the fire compartment .For a sprinkler-controlled fire.

$$T_f = \frac{T_s(H - H_j) + T_j H_j}{H}$$

where

H = floor to ceiling height, ft
H_j = thickness of ceiling jet, ft
T_j = absolute temperature of ceiling jet, °R

1.1.3 Expansion

- The energy released by a fire can also move smoke by expansion. In a fire compartment with only one opening to the building, building air will flow in, and hot smoke will flow out. The ratio of volumetric flows can be expressed as a ratio of absolute temperatures:

$$\frac{Q_{out}}{Q_{in}} = \frac{T_{out}}{T_{in}}$$

Q_{OUT} = volumetric flow rate of smoke out of fire compartment. (Cfm)

Q_{in} = volumetric flow rate of smoke out of air into fire compartment. (Cfm)

T_{OUT} = absolute temperature of smoke leaving fire compartment. (R)

T_{in} = absolute temperature of air into fire compartment. (R)

1.1.4 Wind

- Wind can have a pronounced effect on smoke movement within a building. The pressure wind exerts on a surface can be expressed as:

$$p_w = 0.00643 C_w \rho_o V^2$$

where

p_w = pressure exerted by wind, in. of water
C_w = dimensionless pressure coefficient
ρ_o = outside air density, lb_m/ft^3
V = wind velocity, mph

- The pressure coefficients C_W are in the range of –0.8 to 0.8, with positive values for windward walls and negative values for leeward walls. The pressure coefficient depends on building geometry and varies locally over the wall surface. In general, wind velocity increases with height from the surface of the earth. Figure (4) indicate the compartment fire scenario of opposing wind with downward flow.

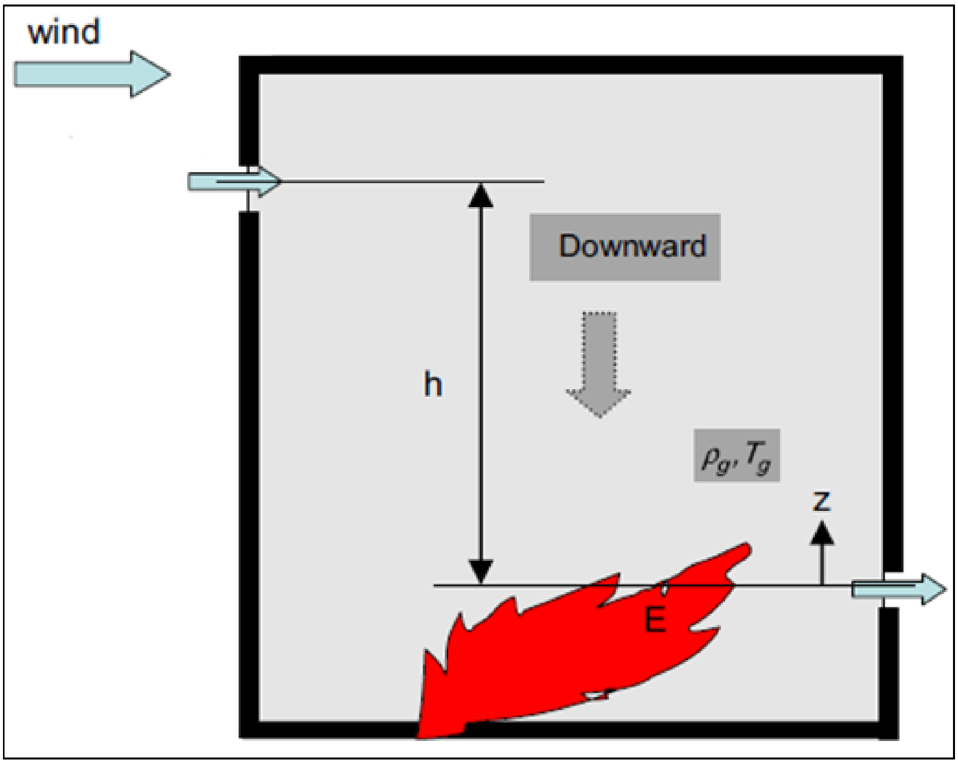
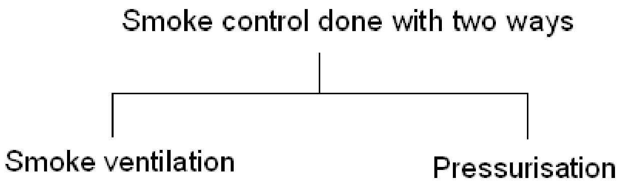


Fig (4). The compartment fire scenario of opposing wind with downward flow.

1.1.5 HVAC Systems

- Before methods of smoke control were developed, HVAC systems were shut down when fires were discovered because the systems frequently transported smoke during fires.

2 Methods for smoke control.



2.1 Smoke ventilation

- In high rise, multi room buildings, where the Staircases, Lift Lobbies and Corridors provide the escape route, SMOKEEXTRACTION may only serve to worsen the situation. A SMOKE EXTRACTION system, illustrated in Fig 1, will provide negative pressure in the escape routes which will tend to draw the smoke into the very spaces requiring protection as shown in figure(5).

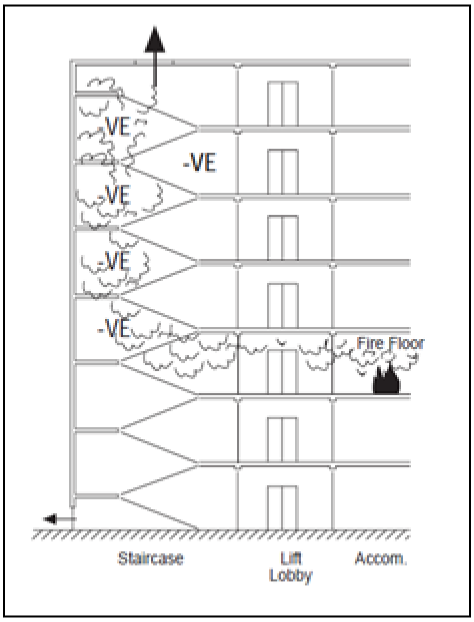


Figure (5). Smoke control by ventilation

2.2 Pressurization

2.2.1 Why Pressurization?

- The objective of any SMOKE CONTROL system is to keep the smoke and toxic Gases out of the escape route long enough to allow the occupants to escape or seek a safe refuge. In addition an adequate smoke control system will help the fire fighters deal both with the fire and any residue smoke.
It is possible to hold back smoke from a fire by simply supplying clean air into the escape routes, thereby developing excess or POSITIVE pressure in the spaces requiring protection. Fig (6).illustrates this method which is known as pressurization.

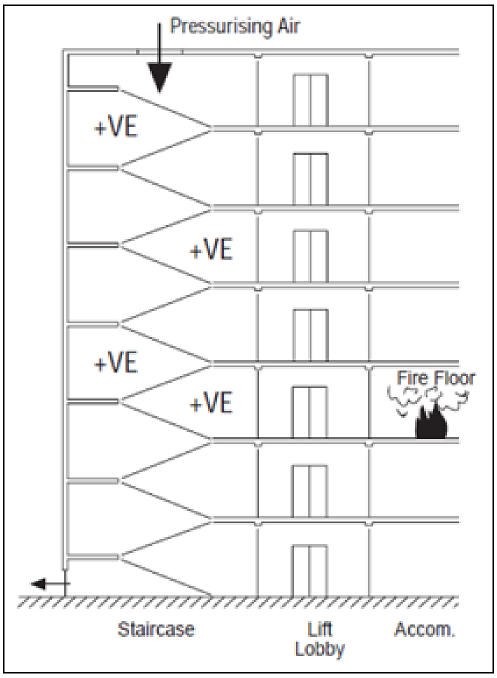


Fig (6) . Smoke control by pressurization

2.2.2 Spaces to be pressurized

2.2.2.1 Stair case only.

- The SMOKE CONTROL system will provide protection to the vertical part of the escape route only. It should only be used when the STAIRCASE is entered direct from the accommodation or via a simple lobby (i.e. a lobby without LIFTS, TOILETS or other possible air escape routes) as shown in figure (7). During building fires, some stairwell doors are opened intermittently during evacuation and firefighting, and some doors may even be blocked open. Stairwell pressurization systems may be single- or multiple injection systems.

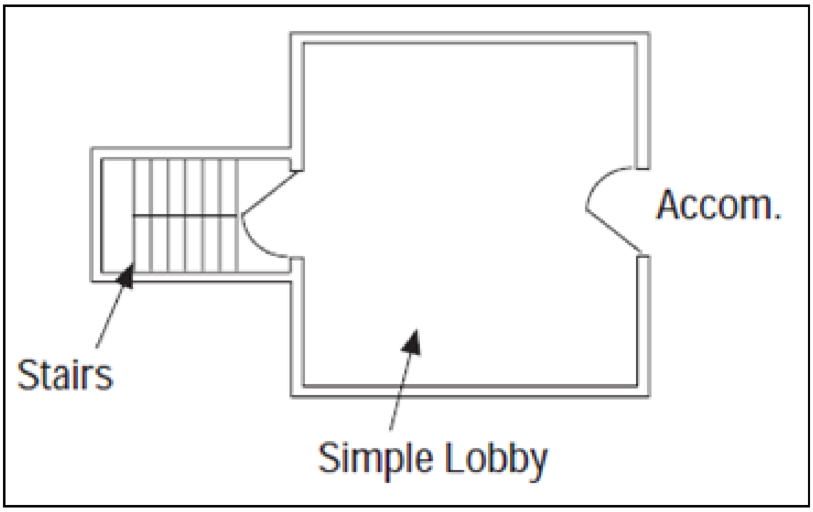


Fig (7). Stair case with simple lobby

2.2.2.1.1 Single-injection system

- Supplies pressurized air to the stairwell at one location, usually at the top. For tall stairwells, single-injection systems can fail when a few doors are open near the air supply injection point. Such a failure is especially likely in bottom-injection systems when a ground-level stairwell door is open. Propeller fans used for air injection which using it has ***some advantages as following***.

  - ✓ They have a relatively flat pressure response curve with respect to varying flow. Therefore, as doors are opened and closed, propeller fans quickly respond to airflow changes in the stairwell

✓ They Are less costly than other types of fans and can provide adequate smoke control with lower installed costs.

- They often require windshields at the intake because the Operate at low pressures and are readily affected by wind pressure on the building. This is less critical on roof where the Fans are often protected by parapets and where the direction of the wind is at right angles to the axis of fan .the most common injection point is at the top of the Stairwell as shown in figure(8).

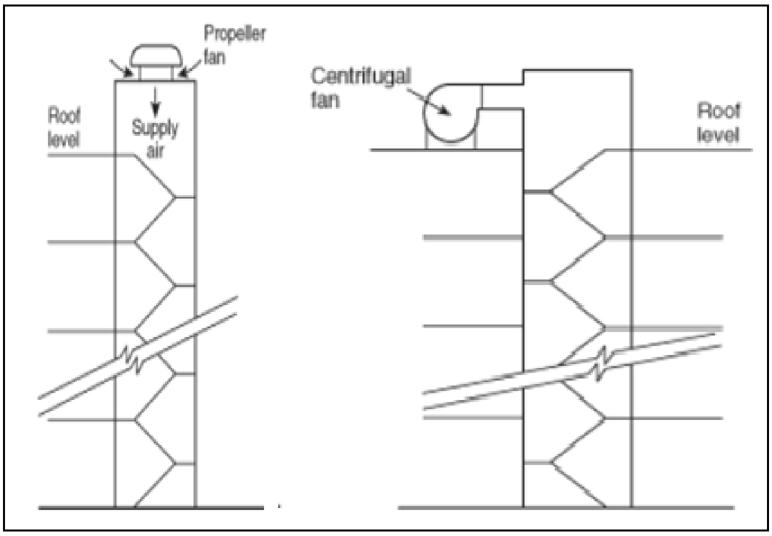


Fig (8). Stair well pressurization by top injection.

2.2.2.1.2 Multiple injection systems

- Supply air can be supplied at a number of locations over the height of the stairwell. This can do with separate duct or with duct shaft to eliminate the expense of a separate duct shaft by locating the supply duct in the stairwell itself. Figure (9) indicate multiple injection systems.

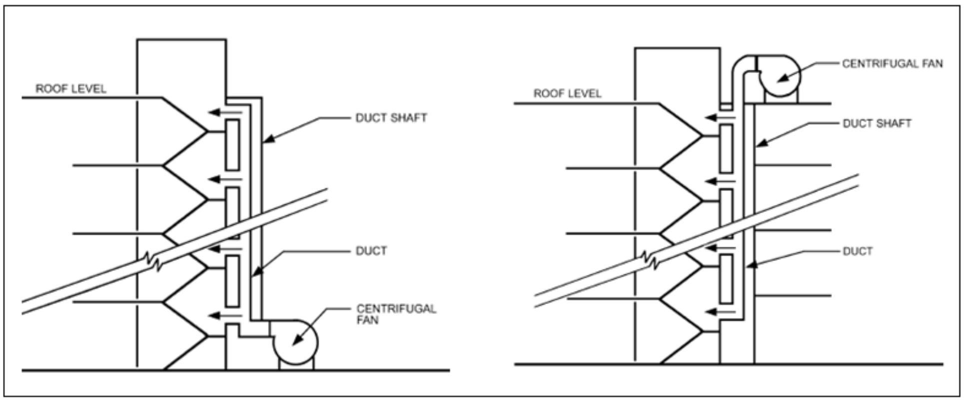


Fig (9). Stair well pressurization by multi injection.

- Compartmentation of the stairwell into a number of sections is one alternative to multiple injection as shown in figure (10). When the doors between compartments are open, the effect of compartmentation is lost. For this reason, compartmentation is inappropriate for densely populated buildings where total building evacuation by the stairwell is planned in the event of fire. When a staged evacuation plan is used and when the system is designed to operate successfully with the maximum number of doors between compartments open, compartmentation can effectively pressurize tall stairwells.

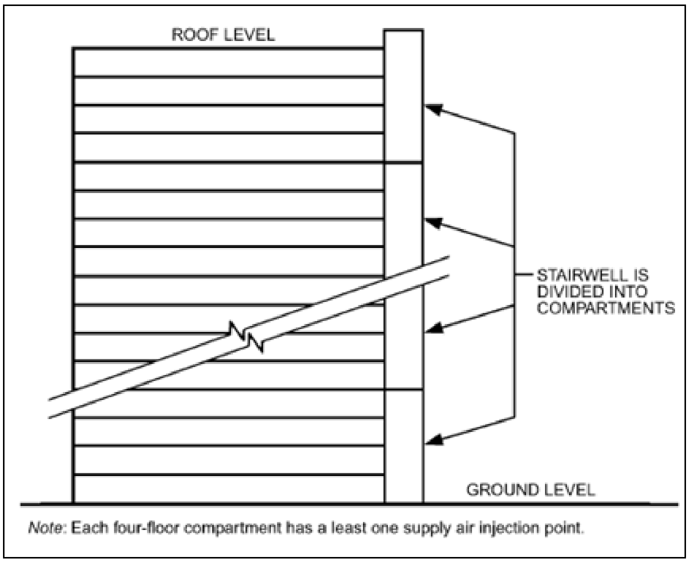


Fig (10). Compartmentation of pressurized stair well

2.2.2.2 Stair case & lobby.

- Two duct systems, from a common fan, required both for STAIRCASE and LIFT LOBBY, used where the LOBBY provides outlets from LIFTS, contains TOILETS or other ancillary rooms Fig (11).

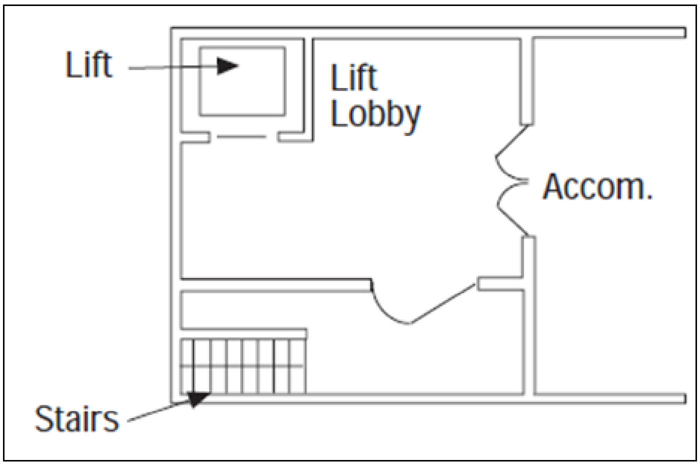


Fig (11).stair case with lift lobby.

2.2.2.3 Stair case, lift lobby, and corridor.

- Extending the LOBBY pressurization system into the CORRIDOR using additional outlets in the corridor Fig (12). Used only where the construction of the corridor has afire resistance of 30 minutes or more.

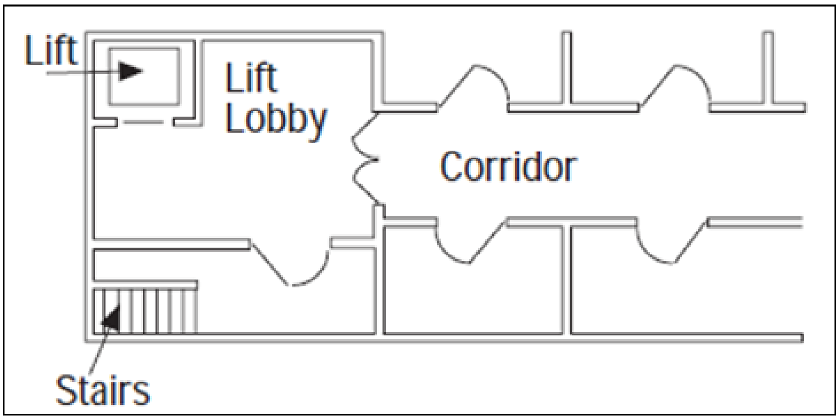


Fig (12). Stair case, lift lobby, and corridor.

2.2.2.4 Lift shaft

- The STAIRCASE and LIFT SHAFT can be pressurized using a common fan with separate ducting.

2.2.3 Element of pressurization system.

- A pressurization System has two main components as illustrated in Fig (13).

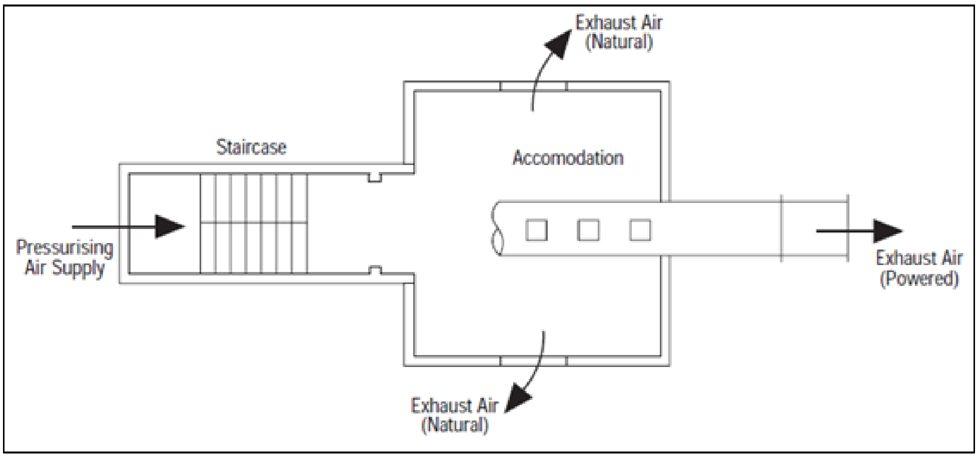


Fig (13). Element of pressurization system.

2.2.3.1 A Supply Air System: designed to blow into the protected spaces a sufficient Quantity of air to maintain the required pressure level or air velocity. This will always be fan powered.

- Amount of supply air to maintain the pressurization level can be calculated with two approaches.
- **First approach:** For supply air volume calculation.

$$Q = 0.83 A_E P^{\frac{1}{n}}$$

Q = The volume of air required (m^3/sec)
A_E = Leakage area from the space (m^2)
P = Pressure differential (Pa)
n = Leakage factor

For large leakage areas -Doors etc - n = 2
For small leakage areas - Window cracks - n = 1.6

For the purpose of a Pressurisation System designed to hold the smoke behind doors the formula becomes,

$$Q = 0.83 A_E P^{\frac{1}{2}}$$

- Where the pressure levels obtained from the following design pressure table.

Building Height (m)	Fire Pressure (Pa)	Wind/Stack Effect (Pa)	Design Pressure (Pa)
5	8.5	8.0	25
25	8.5	10.5	25
50	8.5	13.0	50
100	8.5	19.5	50
150	8.5	29.5	50

- The effective door leakage areas can be estimated by using the values given in typical leakage areas around doors table. These values only apply to the door types and sizes shown in following table.

Type of Door	Size	Crack Length (m)	Leakage Area (m^2)
Single Leaf in Frame Opening into Pressurised Space	2 m x 800 mm	5.6	0.01
Single Leaf in Frame Opening Outwards	2 m x 800 mm	5.6	0.02
Double Leaf with or without Central Rebate	2 m x 1.6 m	9.2	0.03
Lift Door	2 m High x 2 m Wide	8.0	0.06

- **For single openings**, one door, A_e = net free area of the opening. **For several openings**, or doors, situated in **PARALLEL** around a pressurized space as shown in figure (14) A_e obtained from the following equation.

$$A_e = A1 + A2 + A3 + A4...$$

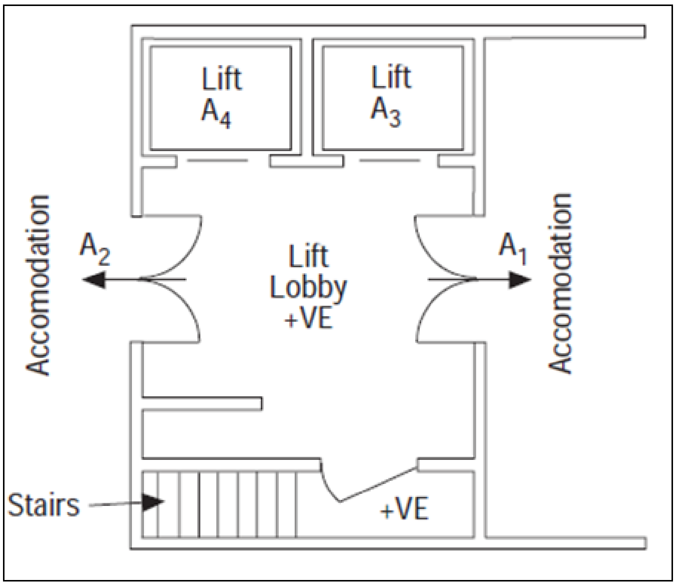


Fig (14).doors in parallel.

- **For several opening - or Doors - situated in SERIES** along an escape route – Figure (15).

$$A_E = \left[\frac{1}{(A_1)^2} + \frac{1}{(A_2)^2} + \frac{1}{(A_3)^2} + \frac{1}{(A_4)^2} \dots\dots \right]^{-1/2}$$

- For two doors in series the above equation can be simplified as shown in following equation.

$$A_E = \frac{(A_1 \times A_2)}{(A_1^2 + A_2^2)^{\frac{1}{2}}}$$

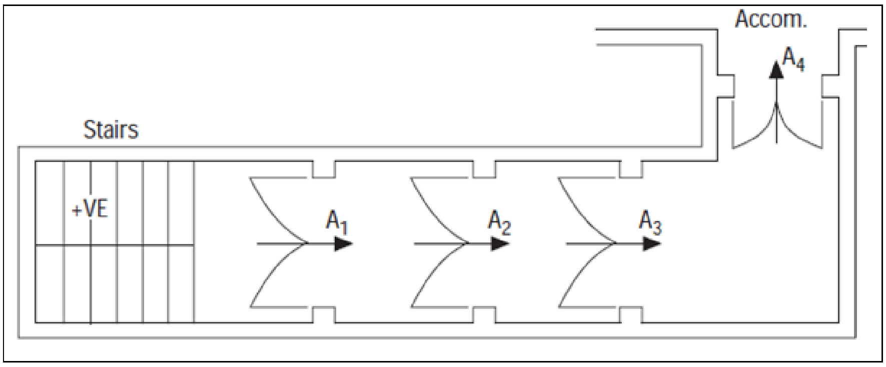


Fig (15).doors in series.

- ***For example*** the following figures indicate some of doors connected in parallel and series.

as an example. The figure shows that A_2 and A_3 are in parallel; therefore, their effective area is

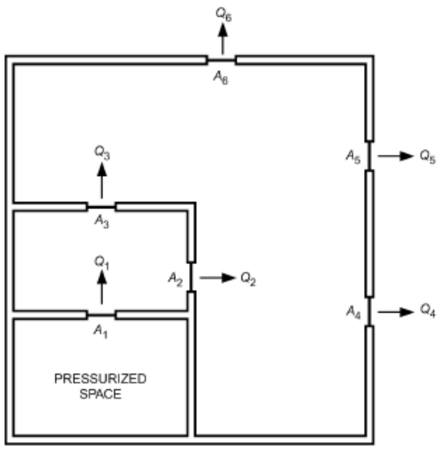
$$(A_{23})_e = A_2 + A_3$$

Areas A_4, A_5, and A_6 are also in parallel, so their effective area is

$$(A_{456})_e = A_4 + A_5 + A_6$$

These two effective areas are in series with A_1. Therefore, the effective flow area of the system is given by

$$A_e = \left[\frac{1}{A_1^2} + \frac{1}{(A_{23})_e^2} + \frac{1}{(A_{456})_e^2}\right]^{-0.5}$$



Combination of Leakage Paths in Parallel and Series

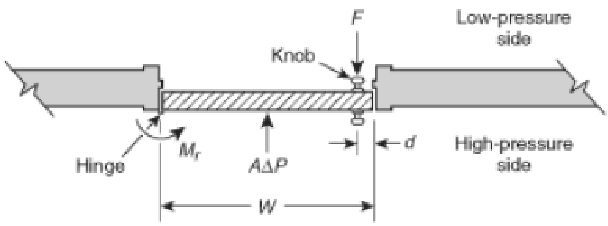
✓ **Foot note**

The maximum allowable pressure difference should not cause excessive door-opening forces. The force required to open a door is the sum of the forces to overcome the pressure difference across the door and to overcome the door closer. This can be expressed as

$$F = F_{dc} + \frac{WA\Delta p}{2(W-d)}$$

where

F = total door-opening force, N
F_{dc} = force to overcome door closer, N
W = door width, m
A = door area, m^2
Δp = pressure difference across door, Pa
d = distance from doorknob to edge of knob side of door, m



The minimum allowable pressure difference across a boundary of a smoke control system might be the difference such that no smoke leakage occurs during building evacuation. In this case, the smoke control system must produce sufficient pressure differences to overcome forces of wind,

Stack effect, or buoyancy of hot smoke. Evaluation of these pressure differences depends on evacuation time, rate of fire growth, building configuration, and the presence of a fire suppression system.

- **Supply air volume for first approach does according to three modes.**

✓ **Mode 1 - DECTECTION PHASE**: To raise pressure differential in the protection Space staircase, corridor etc., by the required amount (50Pa in the UK) when **all doors are closed** Figure (16).

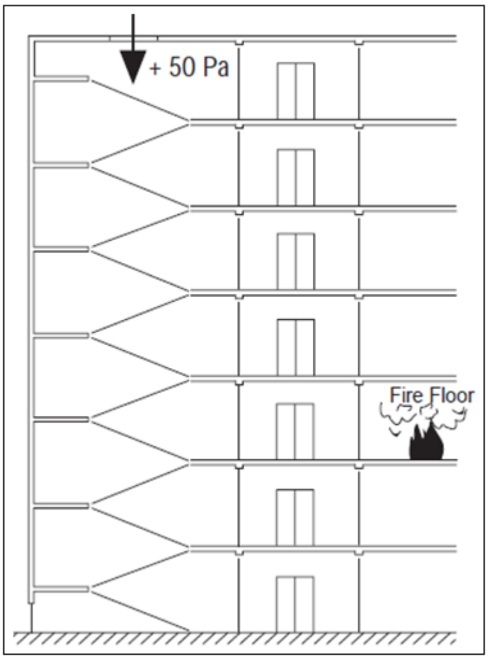


Fig (16).detection phase all doors are closed (mode1).

- ✓ **Mode 2 - ESCAPE PHASE** - To maintain a specified AIR VELOCITY (0.75m/sec) through the OPEN DOOR(S) onto the fire floor with various other doors open, OR a PRESSURE DIFFERENCE OF 10+ Pa with the fire floor door(s) closed and various other doors open (fig 17).

- ✓ **Mode 3 - FIRE FIGHTING PHASE** - To maintain a specified AIR VELOCITY (2.0 m/sec) through the OPEN DOOR(S) onto the fire floor with various other doors open.

- ✓ **ALL pressurization systems for SMOKECONTROL** have a Detection Phase (Mode 1) + escape or firefighting phase. Classifies the ESCAPE PHASE relative to building type and use as indicates in the following table.

System Class	Area of use
A	Residential, sheltered housing and buildings designed for three door protection (Fig 5)
B	Protection of firefighting shafts (Fig 6)
C	Commercial premises (using simultaneous evacuation)(Fig 7)
D	Hotels, hostels and institutional-type buildings, excluding buildings designed to meet call A (Fig 8)
E	Phased evacuation (Fig 9)

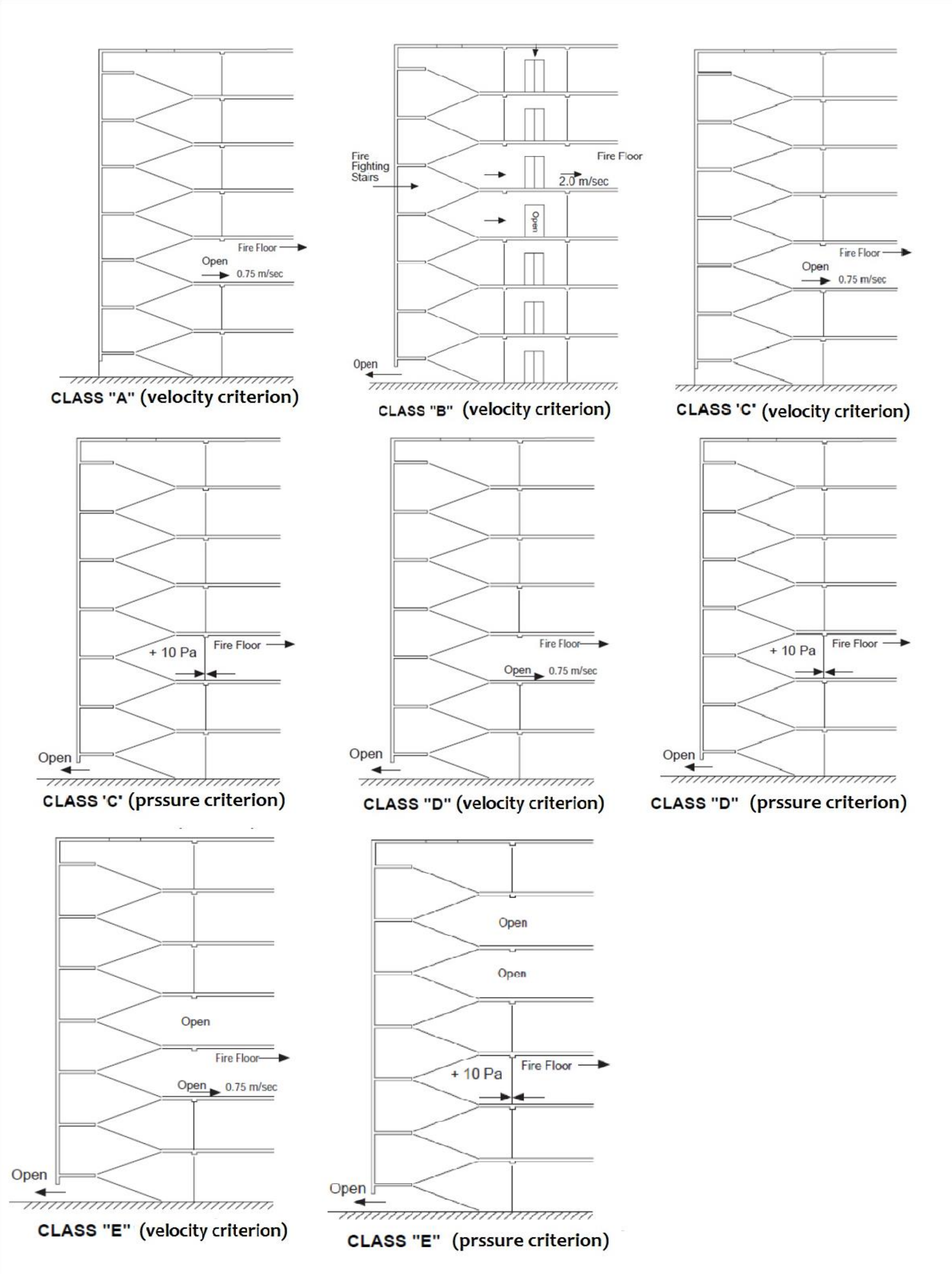


Fig (17). Escape phase, and firefighting phase.

- **Second approach.**

✓ This method closely approximates the performance of pressurized stairwells in buildings without elevators. It is also useful for buildings with vertical .The pressure difference ΔP_{sb} between the stairwell and the building can be expressed as:

$$\Delta p_{sb} = \Delta p_{sbb} + \frac{By}{1 + (A_{sb}/A_{bo})^2}$$

where

Δp_{sbb} = pressure difference between stairwell and building at stairwell bottom, Pa
B = $3460(1/T_o - 1/T_s)$ at sea level standard pressure
y = distance above stairwell bottom, m
A_{sb} = flow area between stairwell and building (per floor), m^2
A_{bo} = flow area between building and outside (per floor), m^2
T_o = temperature of outside air, K
T_s = temperature of stairwell air, K

✓ For a stairwell with no leakage directly to the outside, the flow rate of pressurization air is

$$Q = 0.559 N A_{sb}\left(\frac{\Delta p_{sbt}^{3/2} - \Delta p_{sbb}^{3/2}}{\Delta p_{sbt} - \Delta p_{sbb}}\right)$$

where

Q = volumetric flow rate, m^3/
N = number of floors
Δp_{sbt} = pressure difference from

✓ Leakage areas for wall, and floors of commercial buildings calculated from following table.

Construction Element	Wall Tightness	Area Ratio
		A/A_w
Exterior building walls[a] (includes construction cracks and cracks around windows and doors)	Tight	0.50×10^{-4}
	Average	0.17×10^{-3}
	Loose	0.35×10^{-3}
	Very Loose	0.12×10^{-2}
Stairwell walls[a] (includes construction cracks but not cracks around windows or doors)	Tight	0.14×10^{-4}
	Average	0.11×10^{-3}
	Loose	0.35×10^{-3}
Elevator shaft walls[a] (includes construction cracks but not cracks around doors)	Tight	0.18×10^{-3}
	Average	0.84×10^{-3}
	Loose	0.18×10^{-2}
		A/A_f
Floors[b] (includes construction cracks and gaps around penetrations)	Tight	0.66×10^{-5}
	Average	0.52×10^{-4}
	Loose	0.17×10^{-3}

- **An Exhaust Air System:** to enable the pressurizing air to escape from the unpressurised areas of the building via the fire floor which provide a LOW RESISTANCE route for the supply air to leave the building via the fire floor. This can be achieved by one of four methods:

  - ✓ Via the leakage provided by the window cracks on the outside of the building. In practice this is unrealistic. The area provided is unlikely to be sufficient.
  - ✓ Through automatically opened windows or vents around the perimeter of the Building. This is a possibility where the area concerned has sufficient outside wall space to accommodate the vent area necessary. Operating Mode 2 would require almost 0.5m2 of vent area on every floor for each pressurized staircase.
  - ✓ The provision of a vertical duct through the building with a damper arranged to open automatically on the FIRE FLOOR. See Fig (18).This method is often the best solution although to ensure a low resistance path.

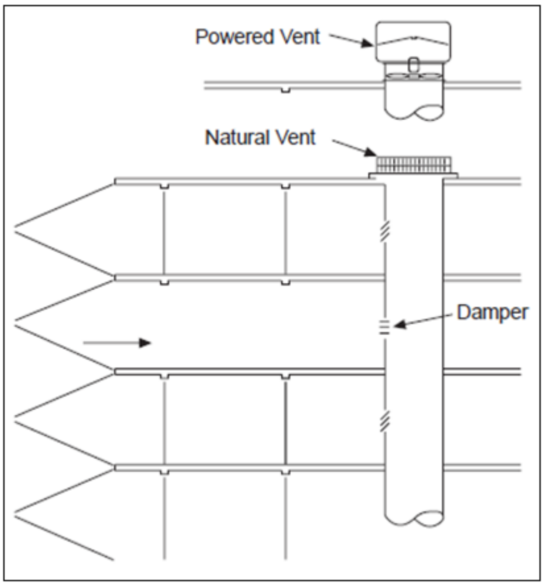


Fig (18).exhaust system by vertical duct.

  - ✓ Mechanical, powered extract from the unpressurised space. This can be Achieved by either:

  - ▪ Providing the vertical duct Figure (18) with an exhaust fan selected to both overcome the resistance of the duct work and handle the hot smoke. The size of the duct can then be reduced.
  - ▪ By utilizing any existing mechanical exhaust system from the unpressurised spaces.

- ✓ **A powered exhaust system from the fire room could provide two incremental advantages.**

- The exhaust fan would be selected to overcome any resistance from the exhaust vent or grill. The pressure in the staircase, when the doors are open, would-be reduced, and with it the quantity of air leaving the building via the final exit door -on systems where this is specified as OPEN. The result would be a reduced quantity of supply air to the system.

- The system can be designed to remove more air than the pressurization system is supplying. This would tend to create a negative pressure, in the fire area, relative to the rest of the building ensuring that all airflow through the building is towards the fire area. Smoke will thereby be prevented from entering unaffected parts of the building via unidentified leakage paths. Mechanical exhaust may be the only way of dealing with the high volumes of air exhaust demanded by Class B, D and E System.

3 Overpressure relief.

- Compensated system operation can also be accomplished by overpressure relief. In this Instance, pressure buildup in the stairwell as doors close is relieved directly from the Stairwell to the outside. This done by one of the following techniques.

- **Barometric dampers with adjustable counterweights** can be used to allow the damper to open when the maximum interior pressure is reached. This represents the simplest, least expensive method of overpressure relief because there is no physical interconnection between the dampers and the fan figure (19).

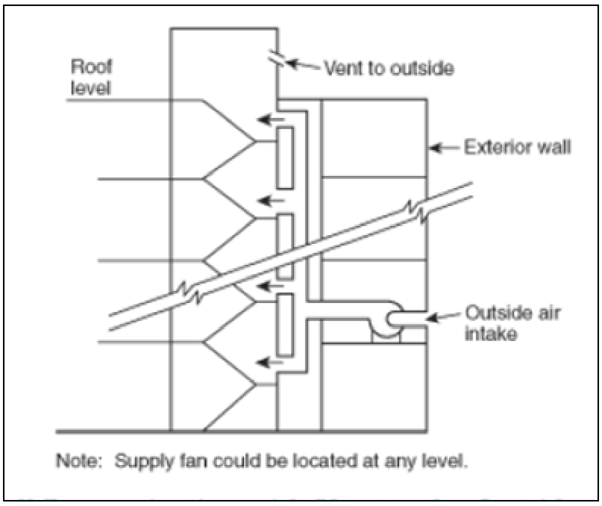


Fig (19).stair well pressurization with vent to the outside.

- **Motor-operated dampers with pneumatic or electric motor** operators are another Option for overpressure relief. These dampers are to be controlled by differential pressure controls located in the stairwell. It requires more control than the barometric dampers and therefore is more complicated and costly.

- **An automatic-opening stairwell door or vent to the outside at ground level** under normal conditions this door would be closed and, in most cases, locked for security reasons. Provisions should be made to ensure that this lock does not conflict with the automatic operation of the system.

- **An exhaust fan can be used to prevent excessive pressure when all stairwell doors are closed.** The fan should be controlled by a differential pressure sensor configured so that the fan will not operate when the pressure difference between the stairwell and the building falls below a specified level.


SESSION END

Chapter6- DX water cooled system design concepts

- **1 introduction**
- **2 Dx water cooled system components.**

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Chapter6- DX water cooled design concepts

1 introduction

- As over one third of the global CO2 emissions are attributed to the combustion of fossil fuels to meet the energy demands of buildings [1], many energy conservation projects are targeted at reducing energy consumption in this area [1-3]. Water-cooled air-conditioning systems (WACS) are in general more energy efficient than air-cooled air-conditioning systems (AACS), especially in subtropical climates where the outdoor air is hot and humid. This has led to a lot of recent investigations on widening the application of the more energy-efficient water-cooled air-conditioning systems (WACS) and district cooling systems (DCS) to buildings.

- The temperature of water from the city mains is around 27oC in summer and that from a cooling tower just slightly higher but still significantly lower than the summer outdoor air temperature. Water-cooled condensers have a higher heat transfer coefficient than air-cooled condensers, which leads to a lower condensing temperature. The lower condensing temperature educes the pressure ratio across the compressor, and thus reduces the compressor power consumption and thereby increases the COP and capacity. Hence, water-cooled air-conditioners can be much more energy efficient than air-cooled air-conditioners.

- Also water cooled system can be more applicable for serving buildings have architectural and structural problems.

- Due to presence of cooling tower, pump, and piping systems make water cooled system high initial cost.

- Very high maintenance costs due to frequent cleaning and water treatment requirements.

2 Dx water cooled system components.

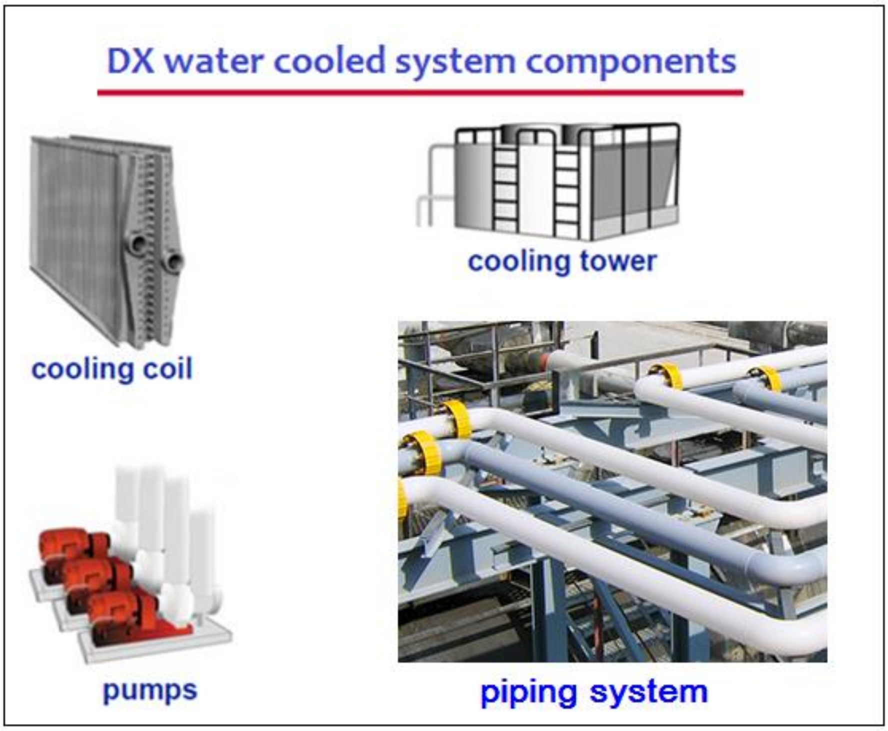


Fig (1).dx water cooled system components.

2.1 DX water cooled unit

- In Water cooled systems Heat is rejected to the outside atmosphere via a *cooling tower* instead of a fluid cooler.
- Also water cooled unit can be available in vertical and horizontal position. For water cooled unit specification see appendix.

2.2 Cooling towers

2.2.1 Principle of operation.

- A cooling tower cools water by a combination of heat and mass transfer. The water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill, which exposes a very large water surface area to atmospheric air. Atmospheric air is circulated by (1) fans, (2) convective currents, (3) natural wind currents, or (4) induction effect from sprays. A portion of the water absorbs heat to change from a liquid to a vapor at constant pressure. This heat of vaporization at atmospheric pressure is transferred from the water remaining in the liquid state into the airstream. The following figure indicates relation between temperature of water and air in counter flow cooling tower.

- The difference between the leaving water temperature and the entering air wet-bulb temperature is the approach to the wet bulb or simply the approach of the cooling tower fig (2). The approach is a function of cooling tower capability, and a larger cooling tower produces a closer approach (colder leaving water) for a given heat load, flow rate, and entering air condition. Thus, the amount of heat transferred to the atmosphere by the cooling tower is always equal to the heat load imposed on the tower.

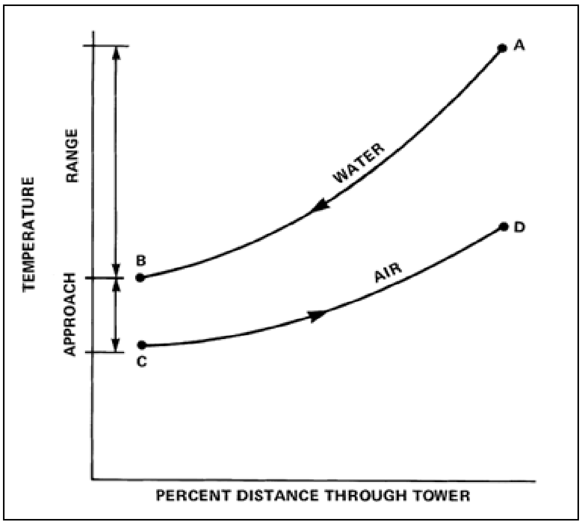


Fig (2).temperature relationship between water and air
In counter flow cooling tower.

- The thermal performance of a cooling tower depends principally on the entering air wet-bulb temperature. The entering air dry-bulb temperature and relative humidity, taken independently, have an insignificant effect on thermal performance of mechanical-draft cooling towers, but they do affect the rate of water evaporation within the cooling tower.

- Figure (3) of a psychometric analysis of the air passing through a cooling tower illustrates this effect. From this figure between two points A, B find the amount of transferred is proportional to the difference in enthalpy of the air between the entering and leaving conditions (hB – hA). Because lines of constant enthalpy correspond almost exactly to lines of constant wet-bulb temperature, the change in enthalpy of the air may be determined by the change in wet-bulb temperature of the air.

- The heating of the air, represented by Vector AB, may be separated into component AC, which represents the sensible portion of the heat absorbed by the air as the water is cooled, and component CB, which represents the latent portion. If the entering air condition is changed to Point D at the same wet-bulb temperature but at a higher dry-bulb temperature, the total heat transfer, represented by Vector DB, remains the same, but the sensible and latent components change dramatically. DE represents sensible cooling of air, while EB represents latent heating as water gives up heat and mass to the air. Thus, for the same water-cooling load, the ratio of latent heat transfer to sensible heat transfer can vary significantly.

- The ratio of latent to sensible heat is important in analyzing the water usage of a cooling tower. Mass transfer (evaporation) occurs only in the latent portion of the heat transfer process and is proportional to the change in specific humidity. Because the entering air dry-bulb temperature or relative humidity affects the latent to sensible heat transfer ratio, it also affects the rate of evaporation.

- The evaporation rate at typical design conditions is approximately 1% of the water flow rate for each 7 K of water temperature range; however, the average evaporation rate over the operating season is less than the design rate because the sensible component of total heat transfer increases as the entering air temperature decreases.

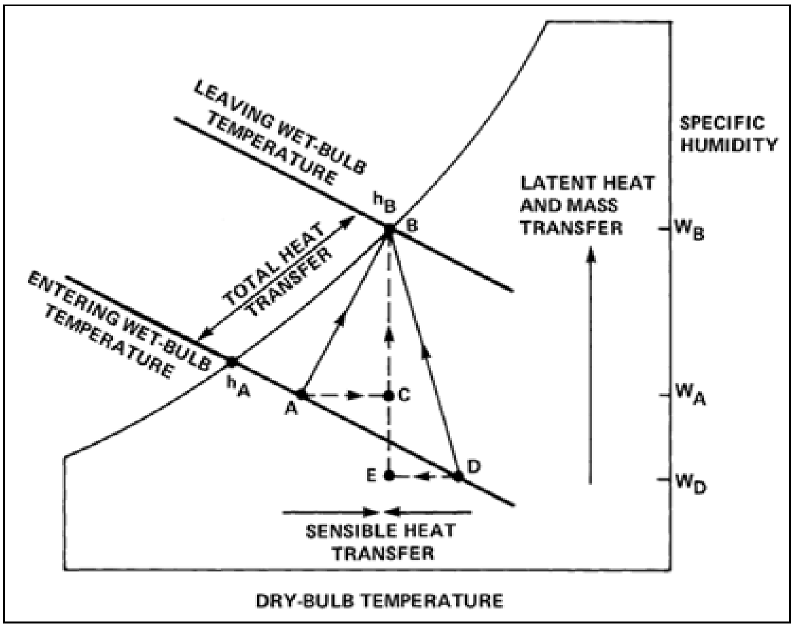


Fig (3).a psychometric analysis of the air
Passing through a cooling tower.

2.2.2 Design condition

- The thermal capability of any cooling tower may be defined by the following parameters:
  - ✓ Entering and leaving water temperatures.
  - ✓ Entering air wet-bulb or entering air wet-bulb and dry-bulb temperatures.
  - ✓ Water flow rate.

2.2.3 Cooling tower types

- **Two basic types of evaporative cooling devices are used:**

  ✓ **First of these**, the direct-contact or open cooling tower, exposes water directly to the cooling atmosphere, thereby transferring the source heat load directly to the air. As shown in the following figure. In this type of cooling towers to increase its capacity by Increasing surface and time of contact between water and air fill installed below the water distribution system. There are two types of fill used in this cooling tower as shown in figure (4).

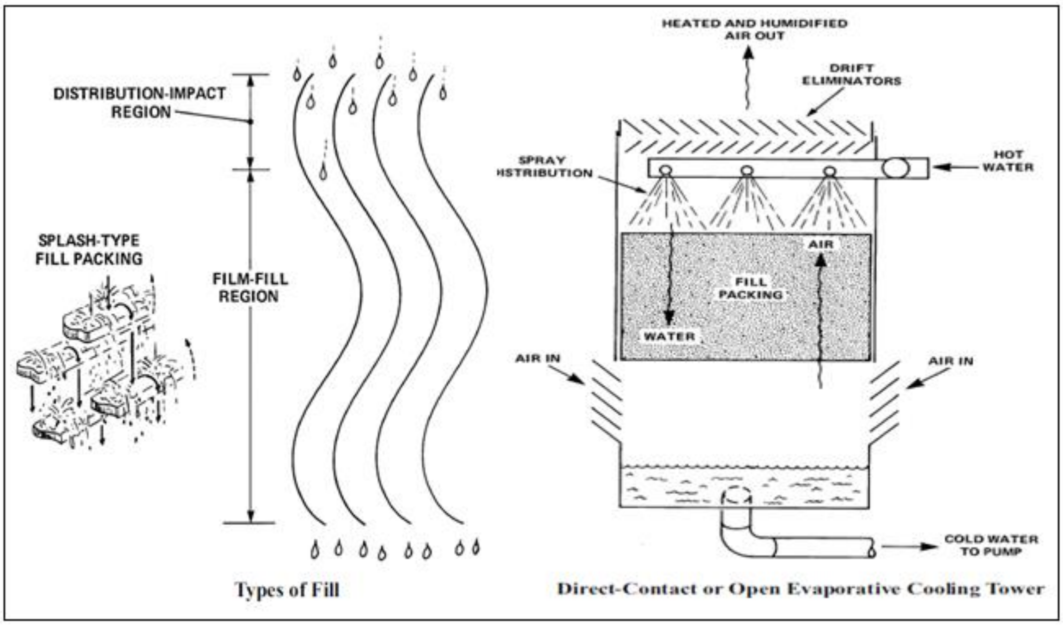


Fig (4).direct contact or open cooling tower.

  ✓ **The second type,** often called a closed-circuit cooling tower, involves indirect contact between heated fluid and atmosphere as shown in figure (5).

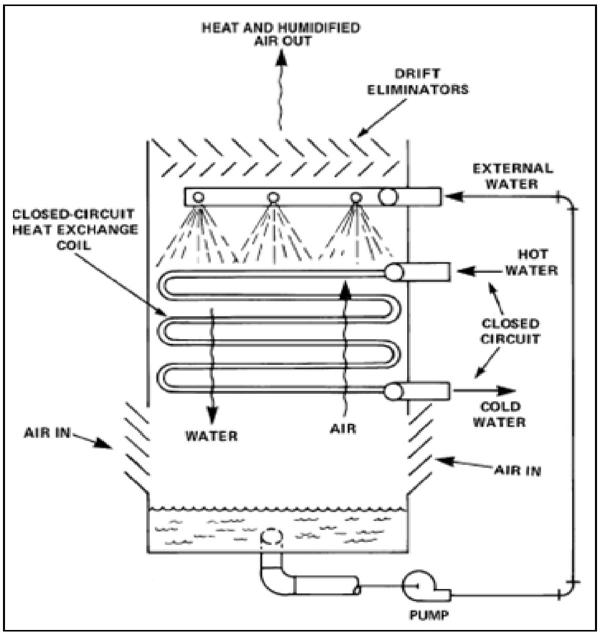


Fig (5). Non-contact or closed cooling tower.

- **Direct-Contact Cooling Towers divided into :**

  ✓ **Non-Mechanical-Draft Towers:** Also Chimney (hyperbolic) towers one of non-mechanical towers has been used primarily for large power installations, the heat transfer mode may be counter flow, cross-flow, or parallel flow. Air is induced through the tower by the air density differentials that exist between the lighter, heat-humidified chimney air and the outside atmosphere as shown in the figure (6).

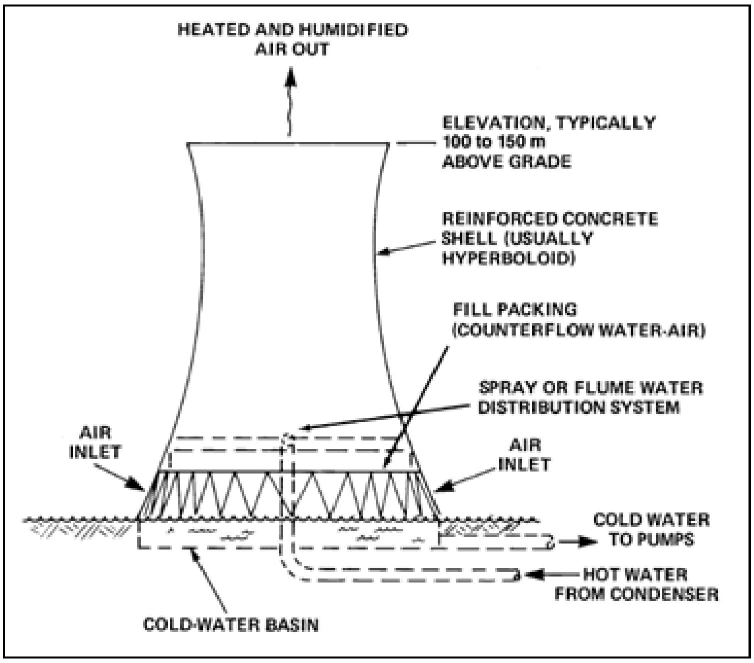


Fig (6). Natural draft cooling tower.

✓ **Mechanical draft cooling towers.**

- This type of cooling towers include two types called forced draft cooling towers , induced draft cooling towers , figure (7) indicates this two types and flow direction of air and water through cooling towers .

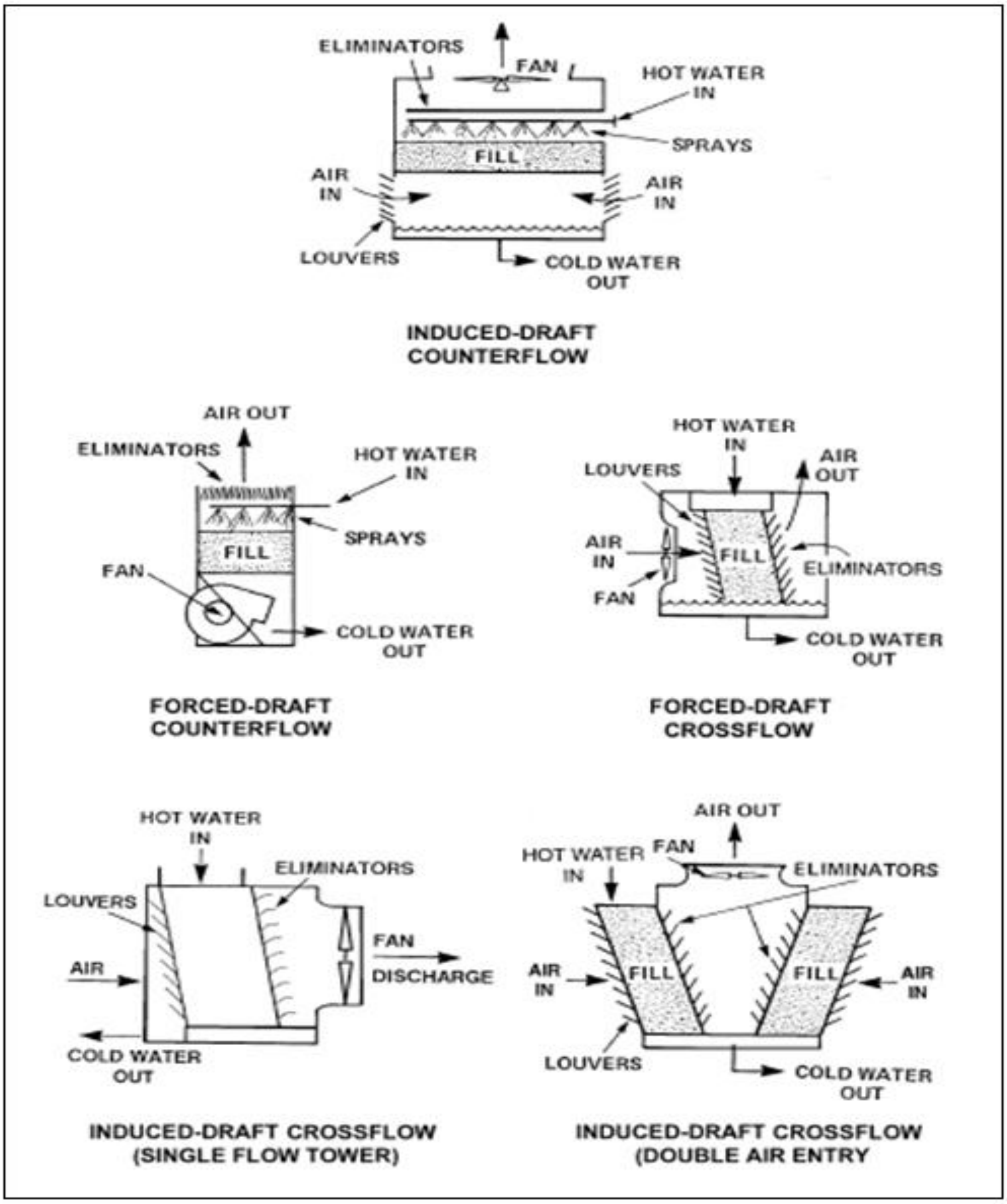


Fig (7).conventional mechanical draft cooling tower.

- In mechanical air-moving device the aspirating effect of the water spray, either vertical or a horizontal as shown in figure (8).

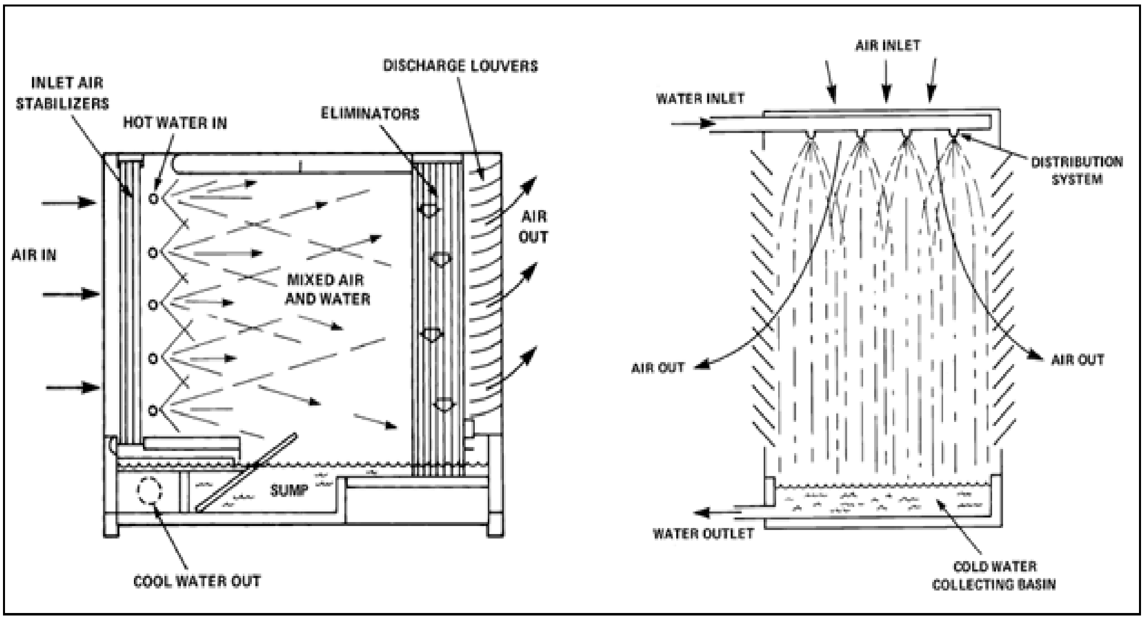


Fig (8). Horizontal and vertical spray tower.

- **Types of Indirect-Contact Towers**

  - ✓ Closed-Circuit Cooling Towers (Mechanical Draft). Both counter flow and cross-flow arrangements are used in forced and induced fan arrangements as shown in the previous figure of indirect contact or closed circuit evaporative cooling towers.

  - ✓ Coil Shed Towers (Mechanical Draft). Coil shed towers usually consist of isolated coil sections located beneath a conventional cooling tower. Counter flow and cross flow types are available with either forced or induced fan arrangements as shown in figure (9).

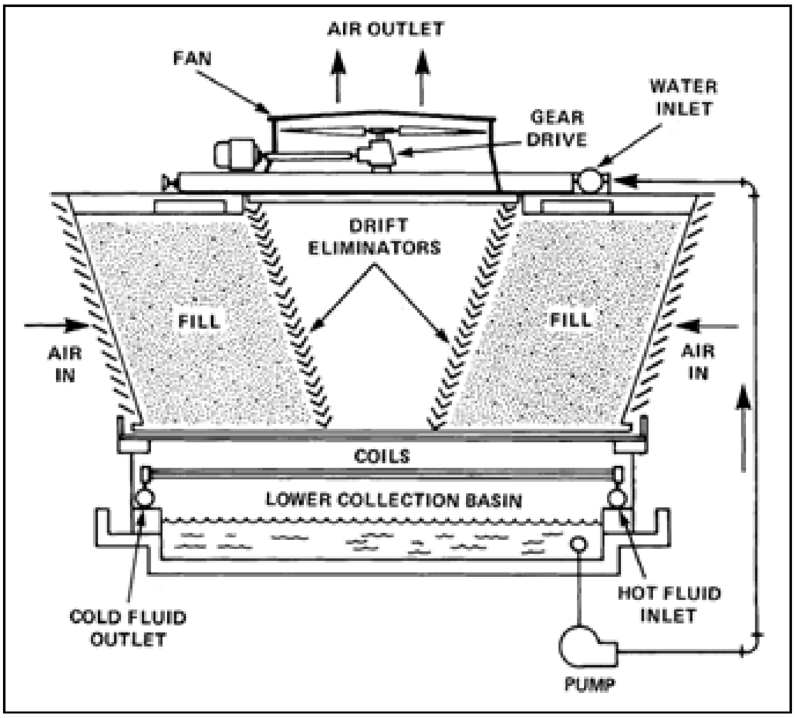


Fig (9).coil shed cooling tower.

- **Towers are typically classified:** as either factory-assembled, where the entire tower or a few large components are factory assembled and shipped to the site for installation, or field-erected, where the tower is constructed completely on-site.

2.2.4 capacity control

- Most cooling towers encounter substantial changes in ambient wet-bulb temperature and load during the normal operating season. Accordingly, some form of capacity control may be required to maintain prescribed condensing temperatures or process conditions. Temperature is not essential; fan cycling is an adequate and inexpensive method of capacity control. However, motor burnout from too frequent cycling is a concern.

- Two-speed fan motors or additional lower power pony motors, in conjunction with fan cycling, can double the number of steps of capacity control compared to fan cycling alone. This is particularly useful on single-fan motor units, which would have only one step of capacity control by fan cycling. Two-speed fan motors are commonly used on cooling towers as the primary method of capacity control.

- Modulating dampers in the discharge of centrifugal blower fans are used for cooling tower capacity control, as well as for energy management. In many cases, modulating dampers are used in conjunction with two-speed motors. Frequency-modulating controls for fan motor speed can provide virtually infinite capacity control and energy management, as can automatic. Variable-frequency fan drives are economical and can save considerable energy as well as extend the life of the fan and drive (gearbox or V-belt) assembly compared to fan cycling with two speed control.

2.2.5 Cooling towers problems and their solutions.

2.2.5.1 Winter Operation

- **Open Circulating Water.**

  ✓ Direct-contact cooling towers that operate in freezing climates can be winterized by a suitable method of capacity control. This capacity control maintains the temperature of the water leaving the tower well above freezing.

  ✓ On induced-draft propeller fan towers, fans may be periodically operated in reverse to deice the air intake areas. Forced-draft centrifugal fan towers should be equipped with capacity control dampers or variable-frequency drives to minimize the possibility of icing .Recirculation of moist discharge air on forced-draft equipment can cause ice formation on the inlet air screens and fans. Installation of vibration cut-out switches can minimize the risk of damage due to ice formation on rotating equipment.

- **Closed Circulating Water Precautions**

  ✓ Precautions in addition to those mentioned for open circulating water must be taken to protect the fluid inside the heat exchanger of a closed-circuit fluid cooler. The best protection is to use an antifreeze solution. When this is not possible, supplemental heat must be provided to the heat exchanger.

- **Sump Water**

  ✓ A good method for protecting the sump water for open and closed cooling towers is to use an auxiliary sump tank located within a heated space as shown in figure (10). Auxiliary heat must be supplied to the tower sump to prevent freezing. Common sources are electric immersion heaters, steam and hot water coils. All exposed water lines susceptible to freezing should be protected by electric tape or cable and insulation.

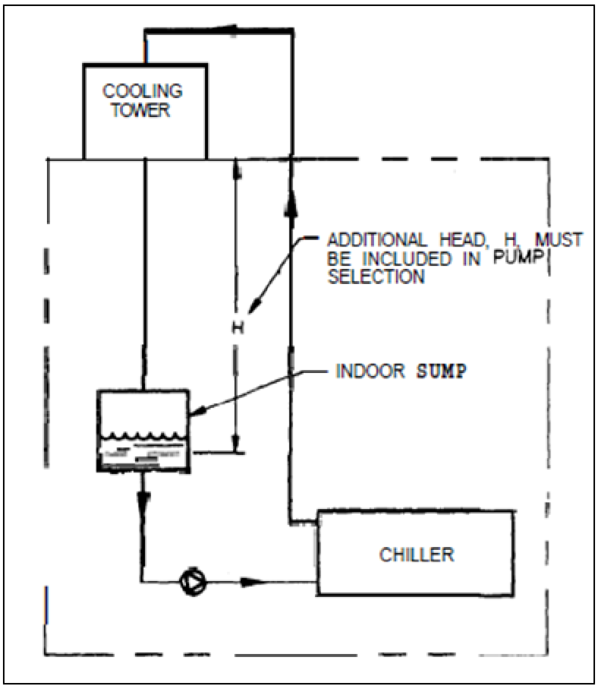


Fig (10).in door, remote sump.

- **sound**

  - ✓ To determine the acceptability of tower sound in a given environment, the first step is to establish a noise criterion for the area of concern. The second step is to estimate the sound levels generated by the tower at the critical area, taking into account the effects of geometry of the tower installation and the distance from the tower to the critical area. Lastly, the noise criterion is compared to the estimated tower sound levels to determine the acceptability of the installation.

  - ✓ In cases where the installation may present a sound problem, it is good practice to situate the tower as far as possible from any sound-sensitive areas. Two-speed fan motors should be considered to reduce tower sound levels (by a nominal 12 dB) during light load periods, such as at night.

  - ✓ In critical situations, effective solutions may include barrier walls between the tower and the sound-sensitive area or acoustical treatment of the tower.

- **Drift**

  - ✓ Generally, an efficient eliminator design reduces drift loss to between 0.002 and 0.2% of the water circulation rate. Because drift contains the minerals of the makeup water (which may be concentrated three to five times) and often contains water treatment chemicals, cooling towers should not be placed near parking areas, large windowed areas, or architectural surfaces sensitive To staining or scale deposits.

- **fogging (cooling tower plume)**

  ✓ The warm air discharged from a cooling tower is essentially saturated. Under certain operating conditions, the ambient air surrounding the tower cannot absorb all of the moisture in the tower discharge airstream, and the excess condenses as fog. Fogging can be presented on psychometric chart as figure (11).

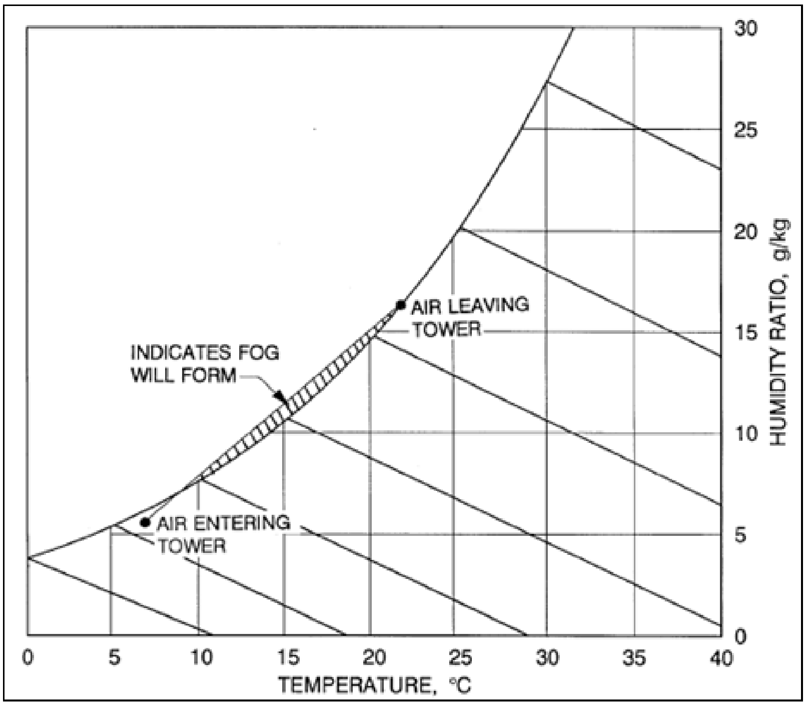


Fig (11).fog prediction using psychrometric chart

✓ The greater the area of intersection to the left of the saturation curve, the more intense the plume.

✓ Methods of reducing or preventing fogging have taken many forms, including heating the tower exhaust with natural gas burners or steam coils, installing precipitators, and spraying chemicals at the tower exhaust. However, such solutions are generally costly to operate and are not always effective.

✓ On larger, field-erected installations, combination wet-dry cooling towers, which combine the normal evaporative portion of a tower with a finned-tube dry surface heat exchanger section (in series or in parallel), afford a more practical means of plume control. In such units, the saturated discharge air leaving the evaporative section is mixed within the tower with the warm, relatively dry air off the finned-coil section to produce a sub saturated air mixture leaving the tower.

✓ When selecting cooling tower sites, the potential for fogging and its effect on tower surroundings, such as large windowed areas or traffic arteries, should be considered.

2.2.6 cooling towers performance curves

- The combination of flow rate and heat load dictates the range a cooling tower must accommodate .thus performance curves can be used to select the suitable cooling towers that meets the requirements.

- Cooling towers can accommodate a wide diversity of temperature levels, ranging as high as 65 to 70°C hot water temperature in the hydrocarbon processing industry. In the air-conditioning and refrigeration industry, towers are generally applied in the range of 32 to 46°C hot water temperature. A typical standard design condition for such cooling towers is 35°C hot water to 29.4°C cold water temperature, and 25.6°C wet-bulb temperature.

- Figure (12) indicates the cooling towers curves indicate typical performance of a cooling tower used for a typical air-conditioning system.

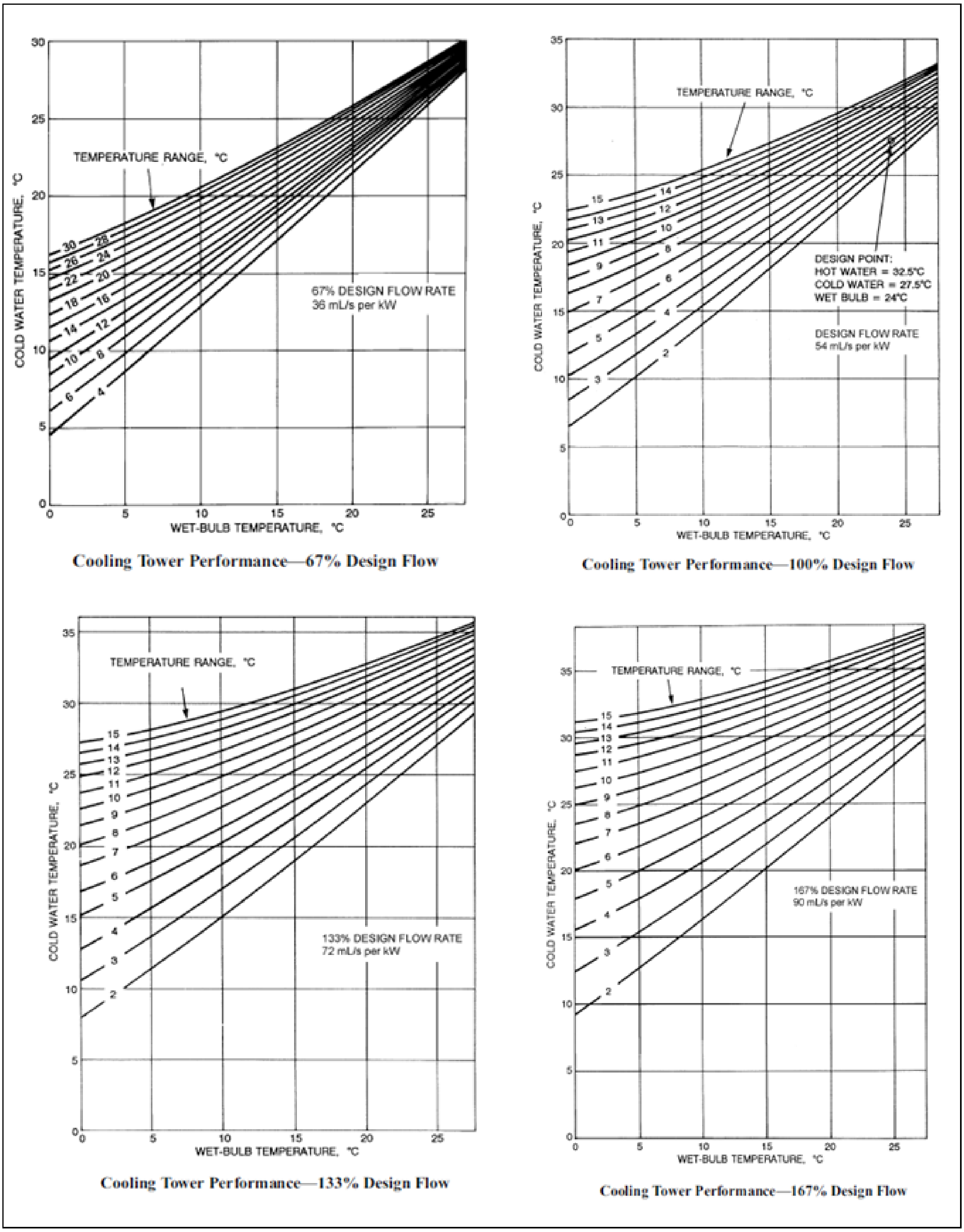


Fig (12).cooling towers curves.

2.2.7 Cooling towers selection

- Selecting cooling tower for air c conditioning done according to following conditions:

  - ✓ cooling capacity in l/s
  - ✓ Entering and leaving WATER TEMP (EWBT = 95 F, LWBT =85 F).
  - ✓ Entering wet bulb temperature (25.6 C = 78 F).

2.2.8 Cooling towers install considerations.

- Cooling towers has specific range of flow that provides acceptable performance this flow range can be only 80 to 100 percent of design flow on most cooling towers. It is responsibility of designers to ensure that the flow rate of the water system meets the needs of cooling towers.

- The first information must be obtained in piping a tower is to determine min and max flow rates required by using equipment such as electric chiller condenser or dx water cooled systems condenser. There were simple rule such as 3gpm per ton per cooling for electric chiller, and DX water cooled systems and this at specific design water temperature such as 95 F entering and 85 F leaving.

- The most conventional cooling tower pumping system as shown figure (13).

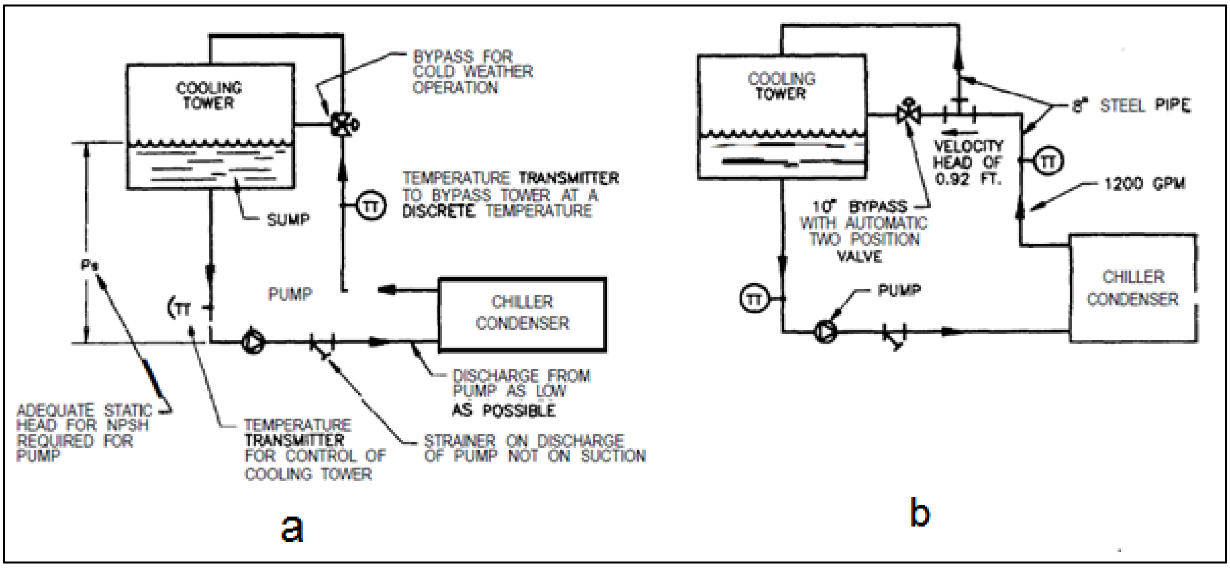


Fig (13).a-standard cooling tower-alternate cooling tower piping.

- Flow through the cooling tower is constant which constant speed pump is used for water circulation and equal to the needs of condenser chiller or condenser DX cooled water systems.

- The piping for cooling towers has been mostly steel .many designers now using thermoplastic piping this material available up to 12 in diameter.

- If multiple cooling towers are to be connected, it is recommended that piping be designed such that the loss from the tower to the pump suction is approximately equal for each tower. Figure (14) illustrate typical layouts for multiple cooling towers. Equalizing lines are used to maintain the same water level in each tower.

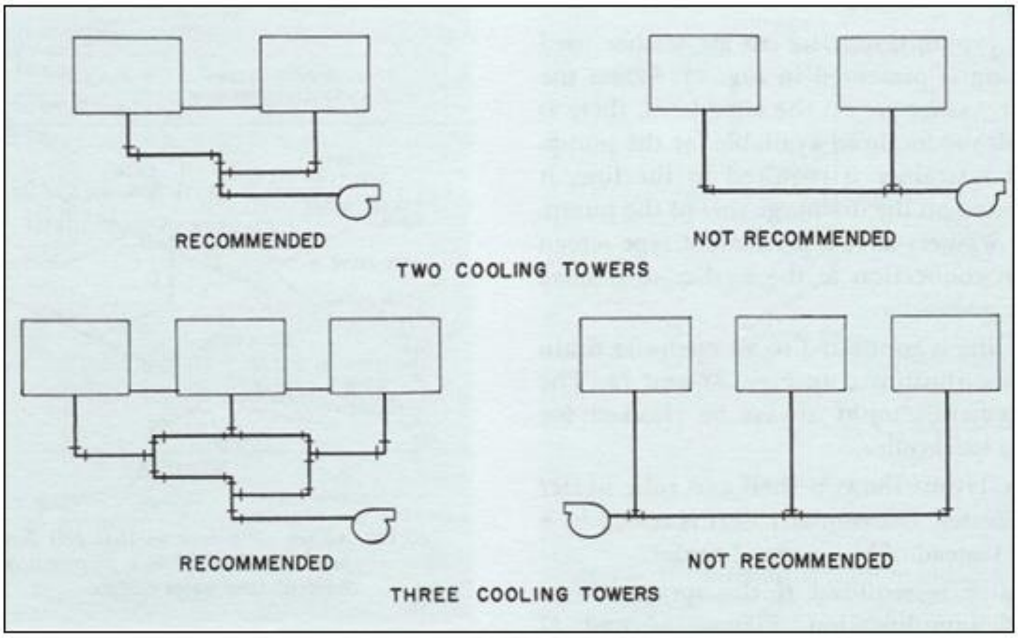


Fig (14).multiple cooling tower piping.

2.3 WATER PUMPS

- Centrifugal pumps provide the primary force to distribute and recirculate water in a variety of space conditioning systems. The pump provides a predetermined flow of water to the space load terminal units.

2.3.1 pumps operation

- In a centrifugal pump, an electric motor or other power source rotates the impeller at the motor's rated speed. Impeller rotation adds energy to the fluid after it is directed into the center or eye of the rotating impeller. The fluid is then acted upon by centrifugal force and rotational or tip speed force, as shown in vector diagram figure (15). These two forces result in an increase in the velocity of the fluid. The pump casing is designed for the maximum conversion of velocity energy of the fluid into pressure energy, either by the uniformly increasing area of the volute or by diffuser guide vanes.

- A centrifugal pump has either a volute or diffuser casing. Pumps With volute casings collect water from the impeller and discharge it Perpendicular to the pump shaft. Casings with diffusers discharge water parallel to the pump shaft as shown in figure (16).

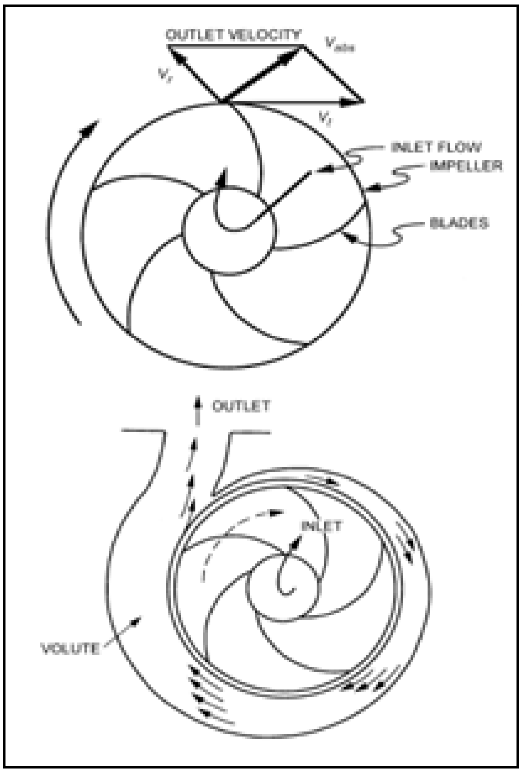


Fig (15).impeller and volute interaction.

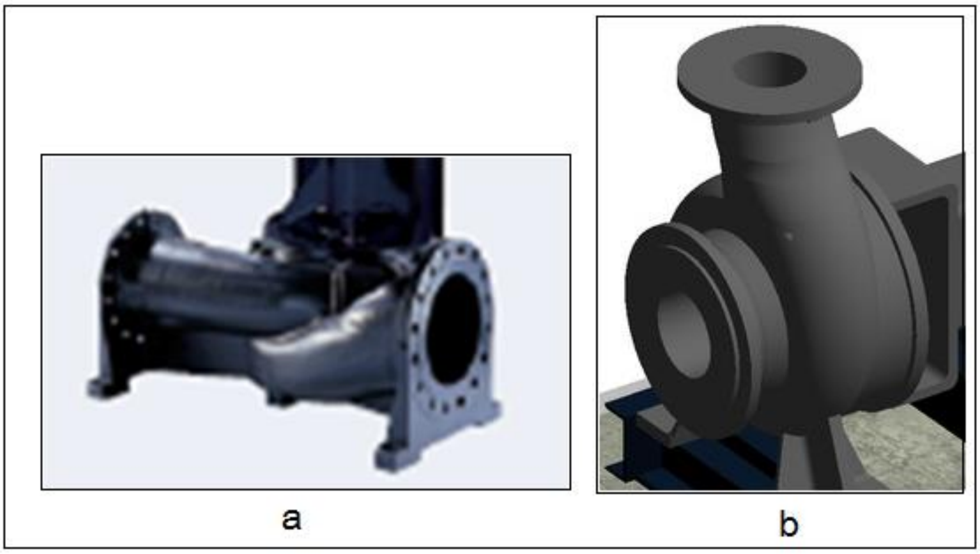


Fig (16).a-diffuser casing, b-volute casing.

2.3.2 pumps classifications

- **End suction pump:** the liquid runs directly into the impeller. Inlet and outlet have a 90° angle. The end suction pump may be classified as shown in figure (17).

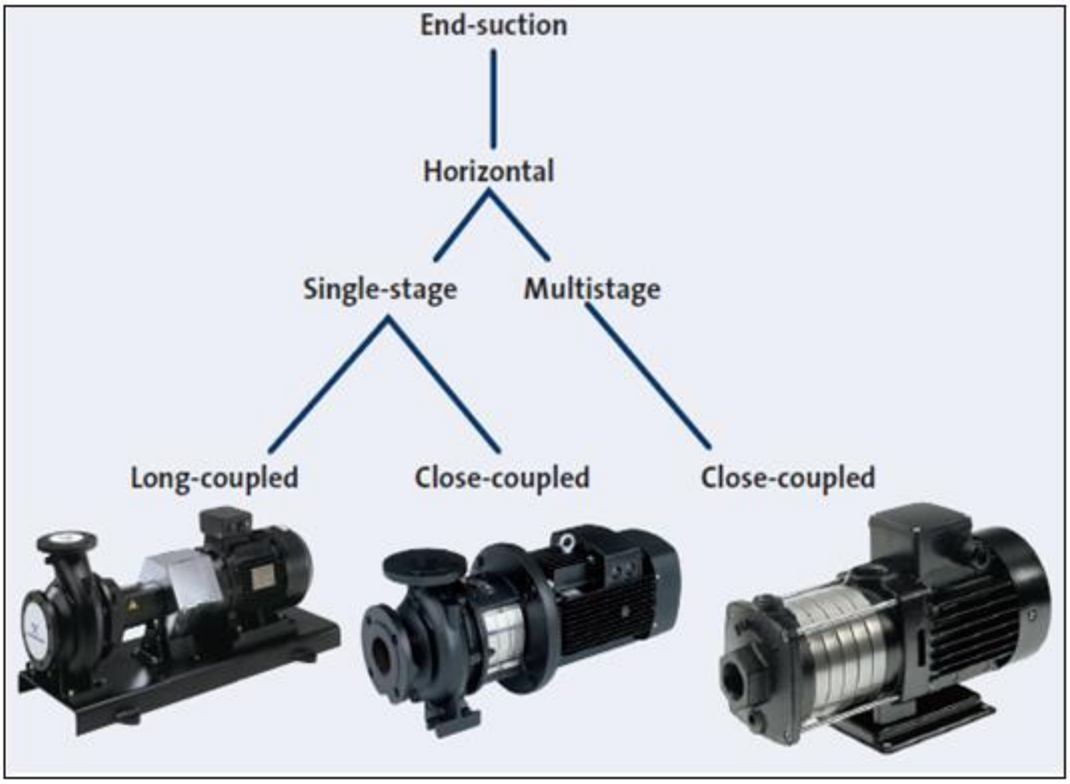


Fig (17).end suction pumps classification.

- **In line pumps:** the liquid runs directly through the pump in-line. The suction pipe and the discharge pipe are placed opposite one another and can be mounted directly in the piping system as shown in the figure (18).

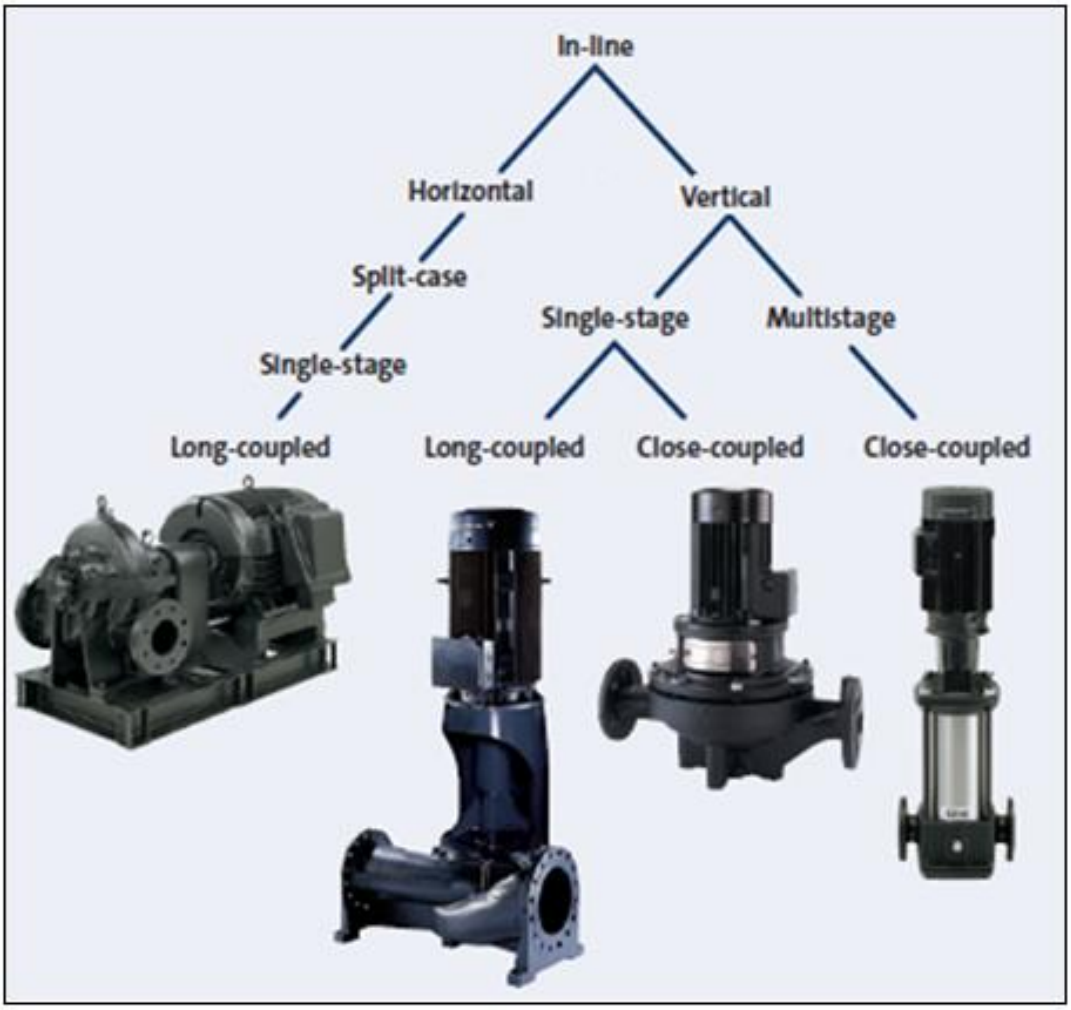


Fig (18).in line pumps classification.

2.3.2.1 terminology

- **Split-case pump:** Pump with an axially divided pump housing.
- **Horizontal pump:** Pump with a horizontal pump shaft.
- **Vertical pump: Pump** with a vertical pump shaft.
- **Single-stage pump:** Pump with a single impeller as shown in figure (19).

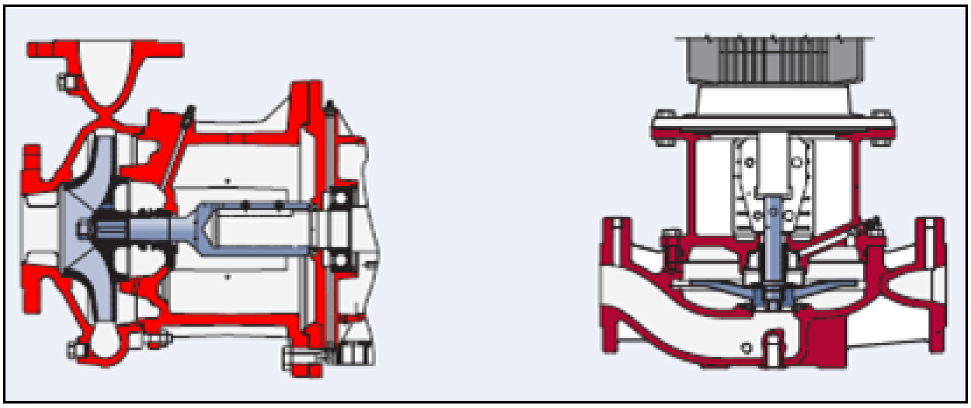


Fig (19).single stage end suction, and in line pump.

- **Multistage Pump:** Pump with several series-coupled stages as shown in figure (20).

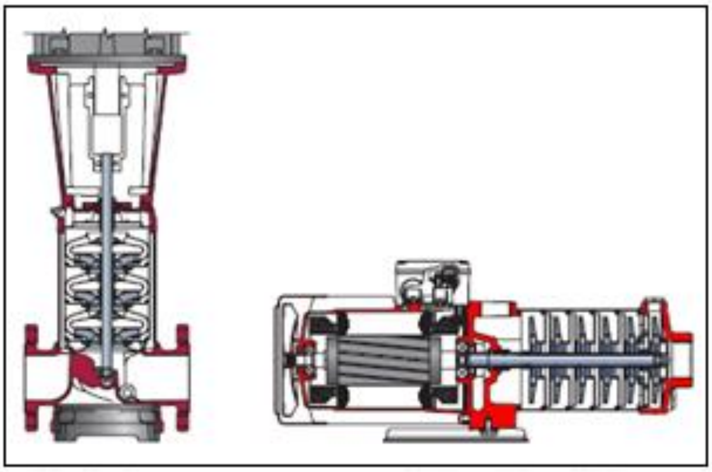


Fig (20).multi stage in line and end suction pump.

- **Long-coupled pump:** Pump connected to the motor by means of a flexible coupling. The motor and the pump have separate bearing constructions as shown in Figure (21).

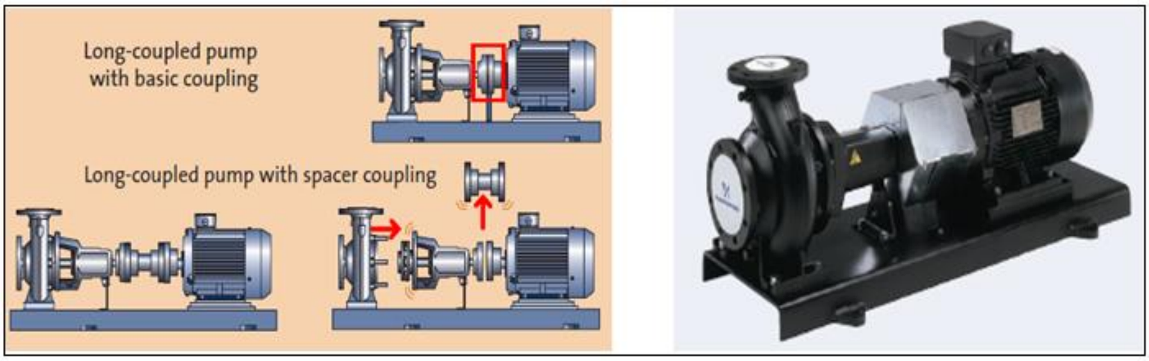


Fig (21).long coupled end suction pump.

- **Closed coupled pump:** A pump connected to the motor by means of a rigid coupling as shown in figure (22).

Fig (22).closed coupled end suction pump.

2.3.3 pump curve

- The discharge pressure decreases as the flow increases. Motors are often selected to be non-overloading at a specified impeller size and maximum flow to ensure safe motor operation at all flow requirements Pump curves represent the average results from testing several pumps of identical design under the same conditions as shown in the figure(23).

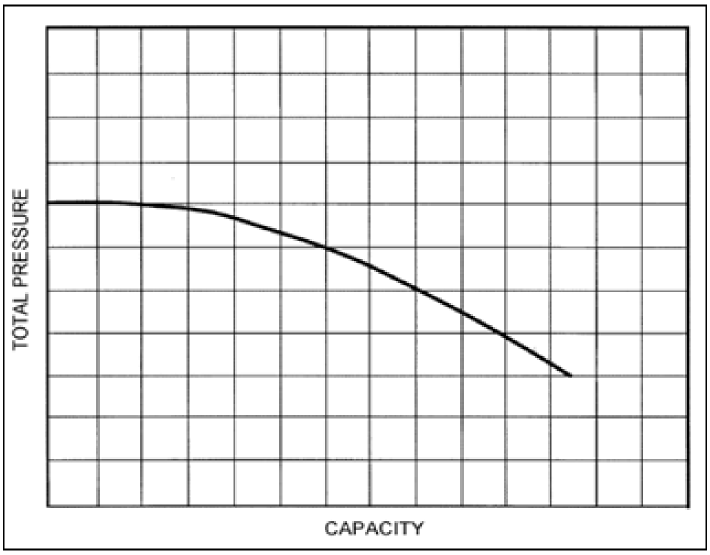


Fig (23).typical pump curve.

- The pump characteristic curve may be further described as flat or steep. Sometimes these curves are described as a normal rising curve (flat), a drooping curve (steep), or a steeply rising curve. The pump curve is considered flat if the pressure at shutoff is about 1.10 to 1.20 times the pressure at the best efficiency point.

- Flat characteristic pumps are usually installed in closed systems with modulating two-way control valves .also flat pumps have advantage that head varies slightly as flow changes.

- Steep characteristic pumps are usually installed in open systems, such as cooling towers, where higher pressure and constant flow are usually desired. Also pumps have advantage that head varies significantly as flow changes. Figure (24) described flat and steep pump curve.

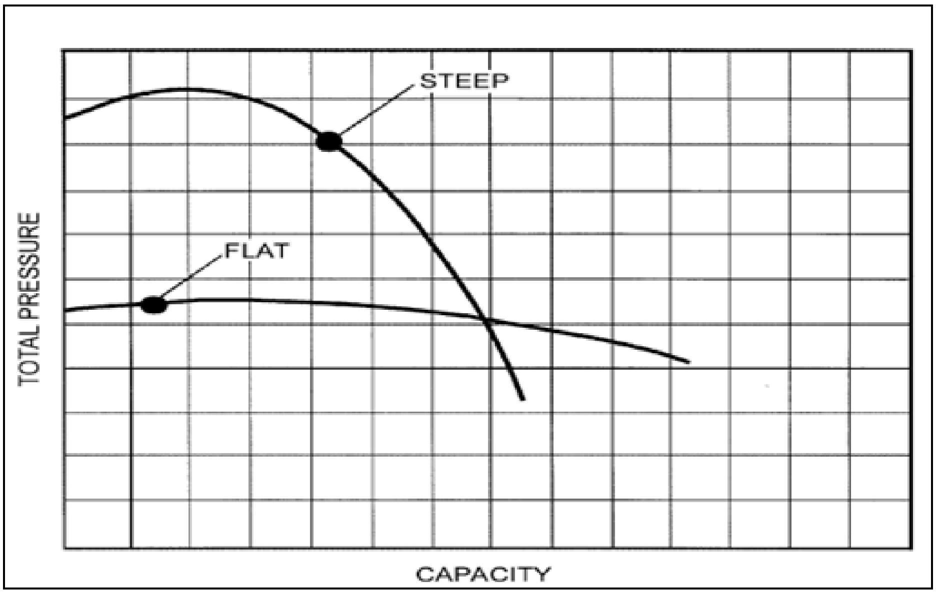


Fig (24).flat versus steep pump curve.

2.3.4 hydronic system curves

- Pressure drop caused by the friction of a fluid flowing in a pipemay be described by the Darcy-Weisbach equation.

$$\Delta p = f\frac{L}{D}\rho\frac{V^2}{2}$$

- Equation (1) shows that pressure drop in a hydronic system(Pipe, fittings, and equipment) is proportional to the square of the flow (V 2 or Q2 where Q is the flow).

$$\Delta h = \frac{\Delta p}{\rho g} = f \frac{L}{D} \frac{V^2}{2g}$$

Where

Δh = loss through friction, m (of fluid flowing)
Δp = pressure drop, Pa
ρ = fluid density, kg/m^3
f = friction factor, dimensionless
L = pipe length, m
D = inside diameter of pipe, m
V = fluid average velocity, m/s
g = gravitational acceleration, 9.8 m/s^2

- The system curves defines the pressure required to produce a given flow rate for a liquid and its characteristics in a piping system design as shown in figure (25).

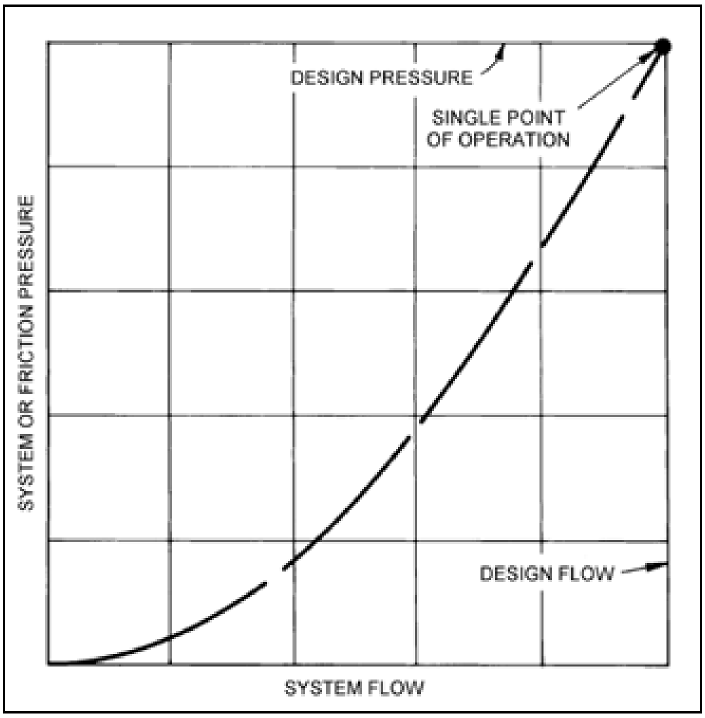


Fig (25).typical system curve.

- If static pressure is present due to the height of the liquid in the system or the pressure in a compression tank, this pressure is sometimes referred to as independent pressure and is added to the system curve as shown in figure (26).

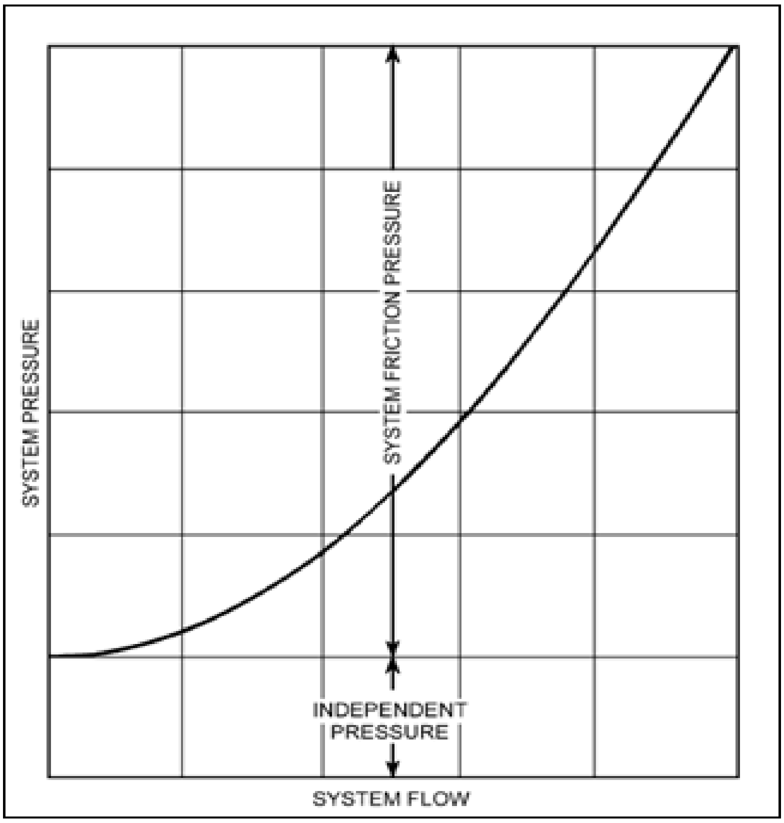


Fig (26).typical system curve with independent pressure.

2.3.5 pump and hydronic system curves

- The pump curve and the system curve can be plotted on the same graph. The intersection of the two curves is the system operating point, where the pump's developed pressure matches the system's pressure loss as shown in figure (27).

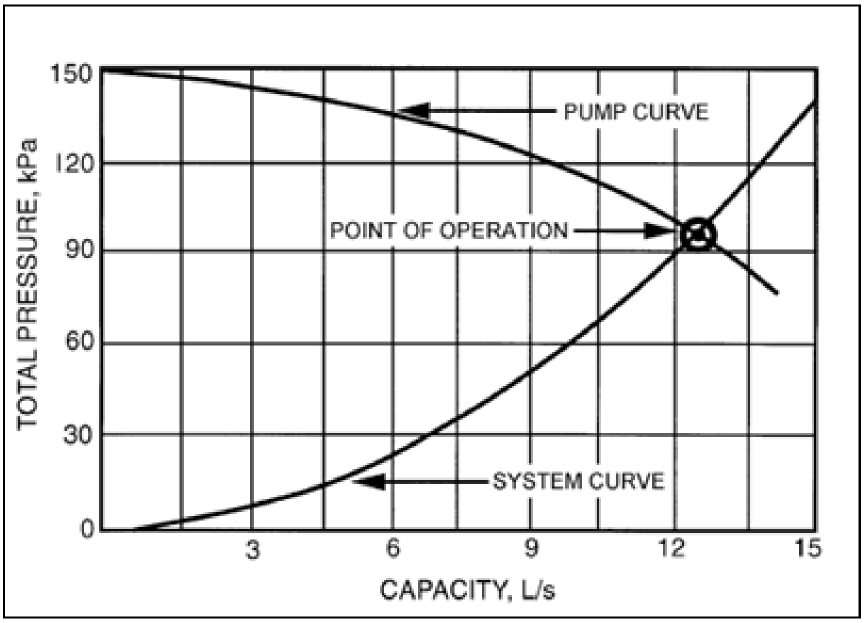


Fig (27).system and pump curve.

- Such a system might be an open piping circuit between a refrigerating plant condenser and its cooling tower. The elevation difference between the water level in the tower pan and the spray distribution pipe creates the fixed pressure loss. The fixed loss occurs at all flow rates and is, therefore, an independent pressure as shown in figure (28).

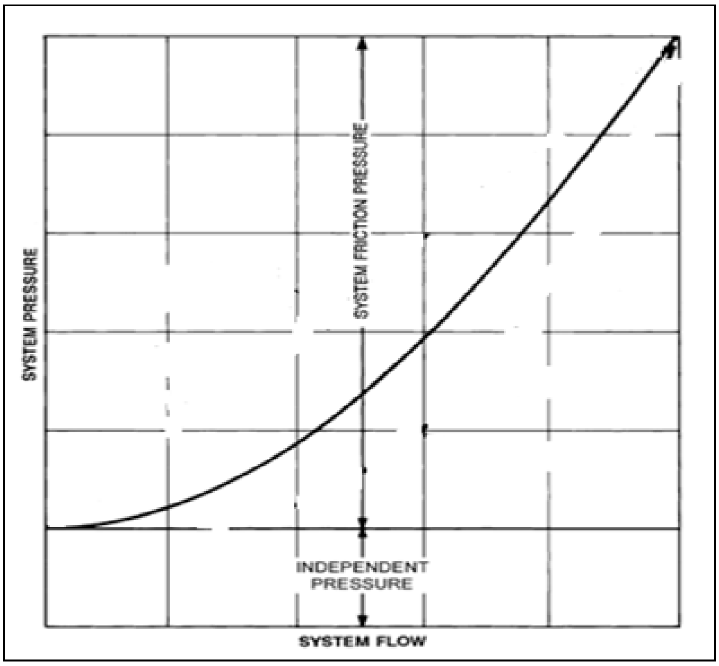


Fig (28).system curve with constant pressure loss.

2.3.6 pumps power

- The theoretical power to circulate water in a hydronic system is the water power Pw and is calculated as follows:

$$\boxed{P_w = \dot{m}\Delta p/\rho}$$

Where

$\dot{m}$ = mass flow of fluid, kg/s
Δp = pressure increase, Pa

- Figure (29) shows the relation between water power and flow.

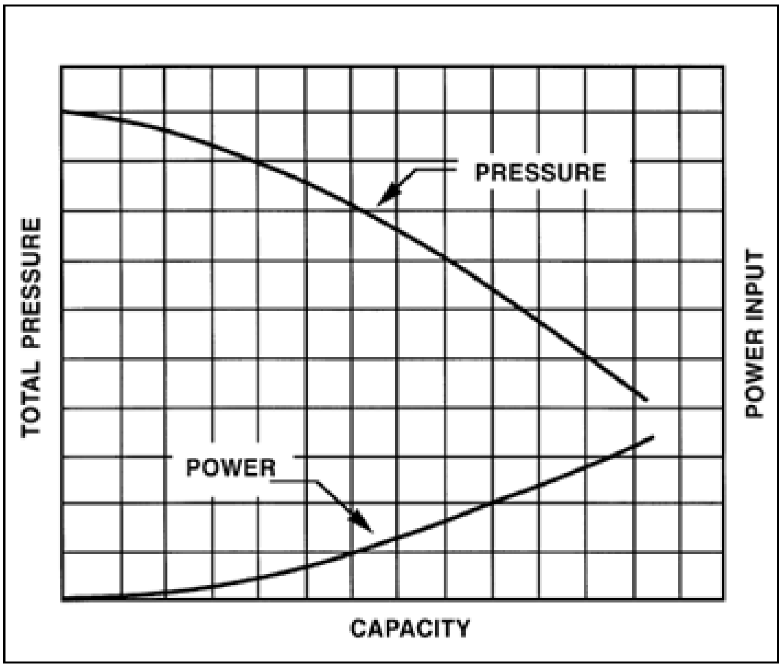


Fig (29).typical pump water power increase with flow.

2.3.7 pumps efficiency

- Pump efficiency is determined by comparing the output power to the input power.

$$\boxed{\text{Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{P_w}{P_t} \times 100\%}$$

- Figure (30) indicates relation between flow and efficiency.

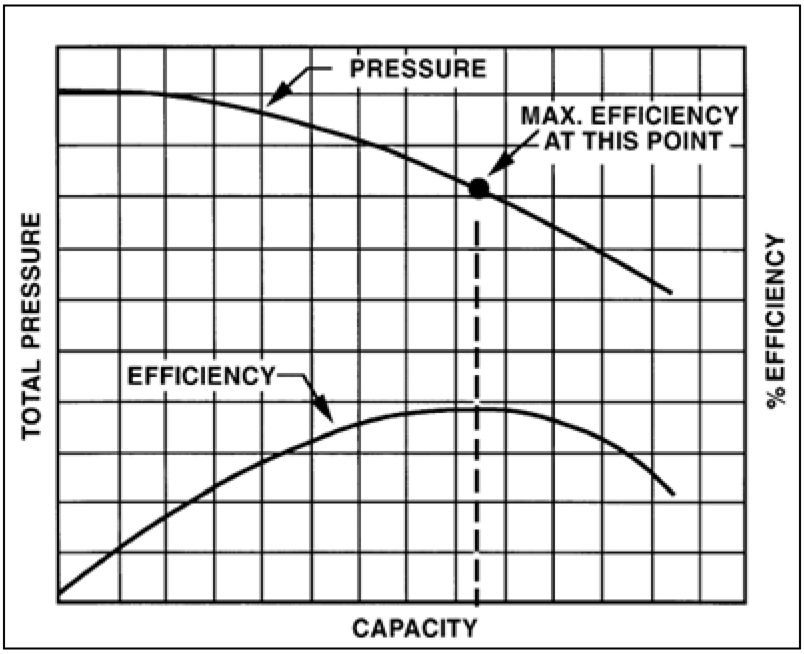


Fig (30).pump efficiency versus flow.

- The pump manufacturer usually plots the efficiencies for a given volute and impeller size on the pump curve to help the designer select the proper pump as shown in figure (31).

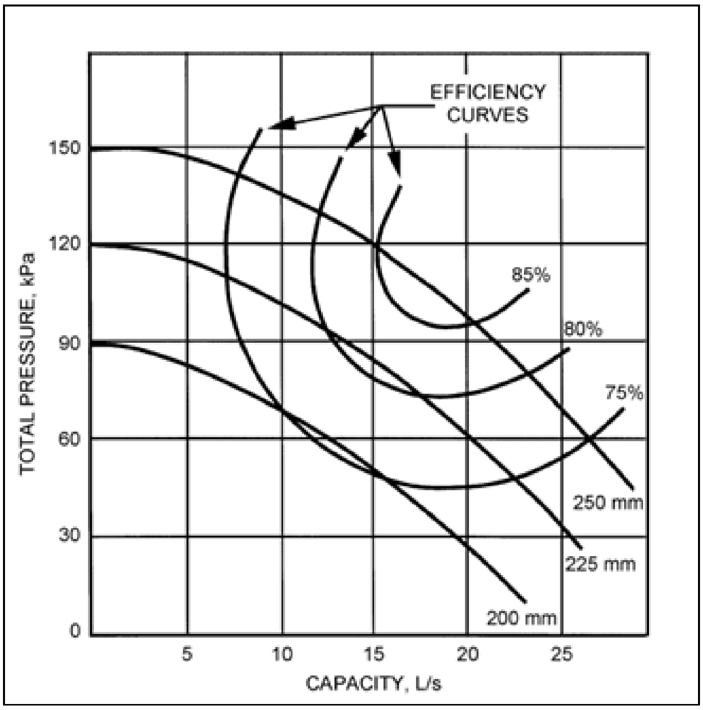


Fig (31). Pump efficiency curves.

- The best efficiency point (BEP) is the optimum efficiency for this pump operation above and below this point is less efficient. The locus of all the BEPs for each impeller size lies on a system curve that passes through the origin as shown in figure (32).

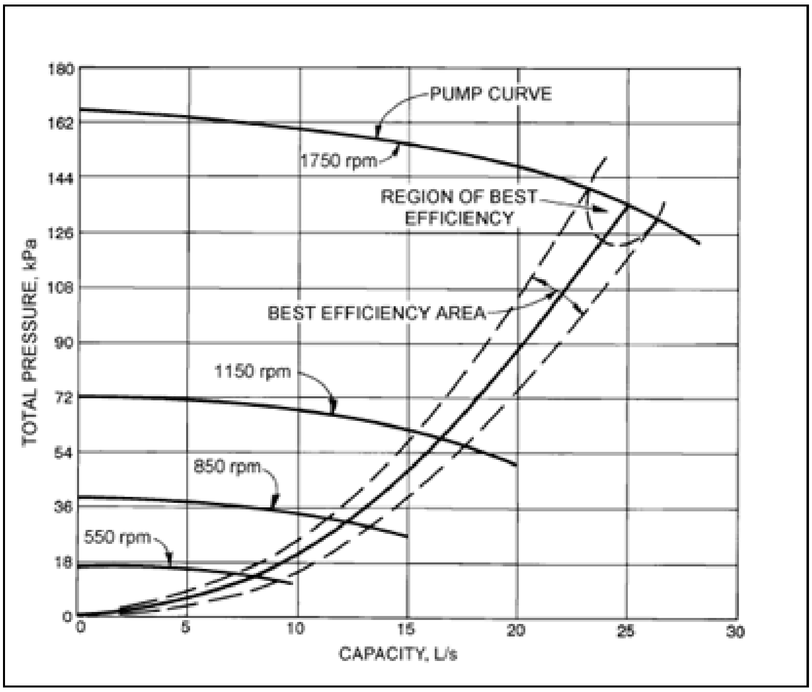


Fig (32).pump best efficiency curves.

2.3.8 affinity laws

- The centrifugal pump, which imparts a velocity to a fluid and converts the velocity energy to pressure energy, can be categorized by a set of relationships called affinity laws. See table (1).

Function	Speed Change	Impeller Diameter Change
Flow	$Q_2 = Q_1\left(\frac{N_2}{N_1}\right)$	$Q_2 = Q_1\left(\frac{D_2}{D_1}\right)$
Pressure	$p_2 = p_1\left(\frac{N_2}{N_1}\right)^2$	$p_2 = p_1\left(\frac{D_2}{D_1}\right)^2$
Power	$P_2 = P_1\left(\frac{N_2}{N_1}\right)^3$	$P_2 = P_1\left(\frac{D_2}{D_1}\right)^3$

Table (1).pump affinity law.

- The affinity laws are useful for estimating pump performance at different rotating speeds or impeller diameters D based on a pump with known characteristics. Figure (33) shows variation of pumping power, pressure, and flow versus speed.

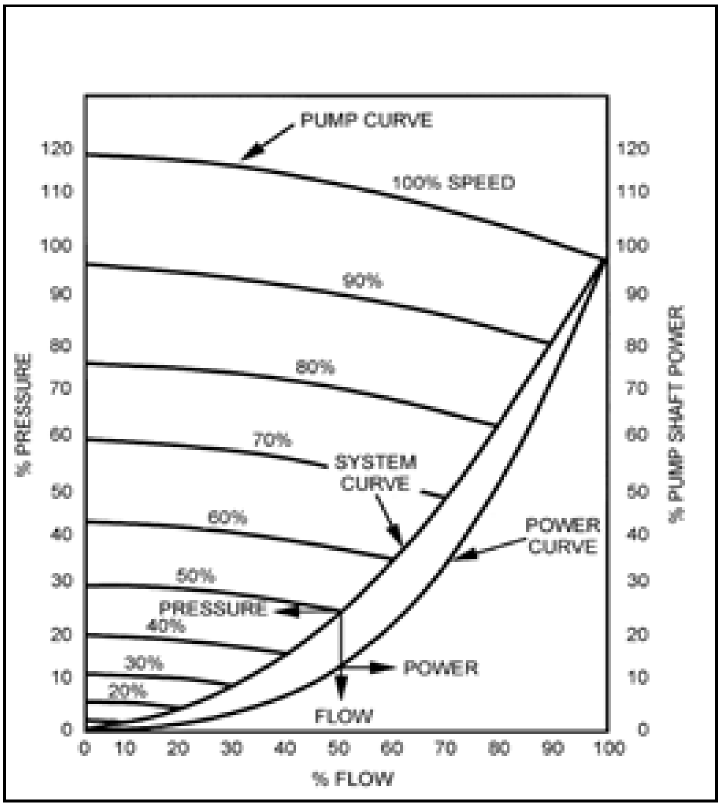


Fig (33).pumping power, pressure, and flow versus pump speed.

- Multiple-speed motors can be used to reduce system overpressure at reduced flow. Figure (34) shows an applicable example on affinity laws.

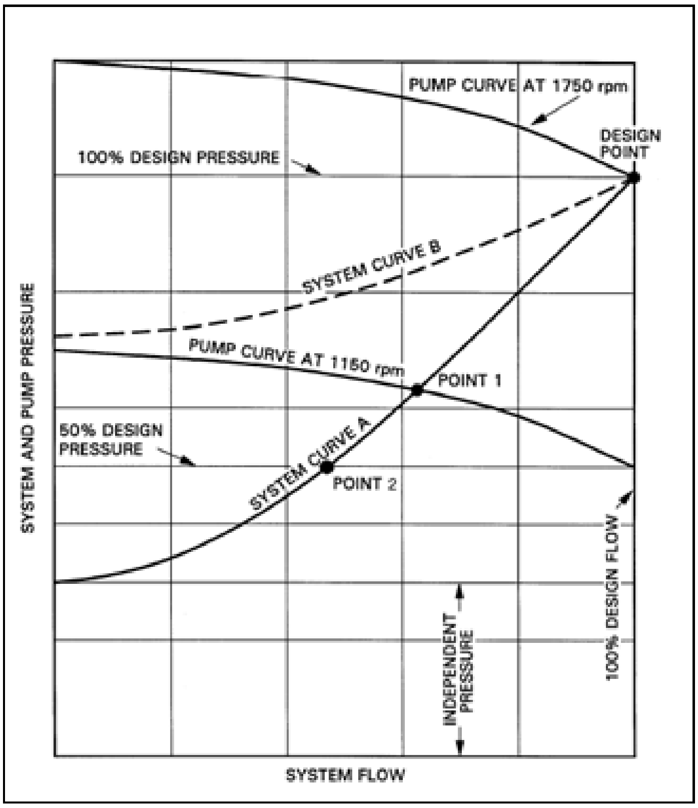


Fig (34).example application of affinity law.

2.3.9 pumps arrangement

- In a large system, a single pump may not be able to satisfy the full design flow and yet provide both economical operation at partial loads and a system backup. The designer may need to consider the following alternative pumping arrangements and control scenarios:

2.3.9.1 Multiple pumps in parallel.

- **Parallel pumping:** When pumps are applied in parallel, each pump operates at the same pressure and provides its share of the system flow at that pressure as shown in figure (35).

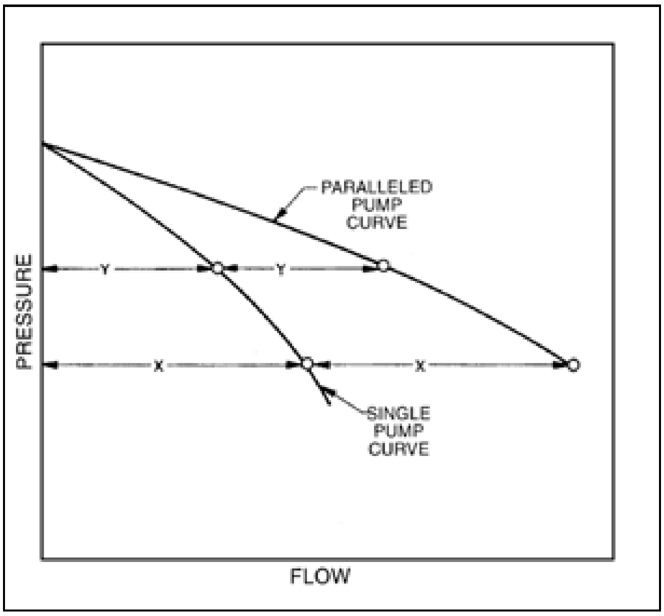


Fig (35).pump curve construction for parallel operation.

- Generally, pumps of equal size are recommended, and the parallel pump curve is established by doubling the flow of the single pump curve as shown in figure (36).

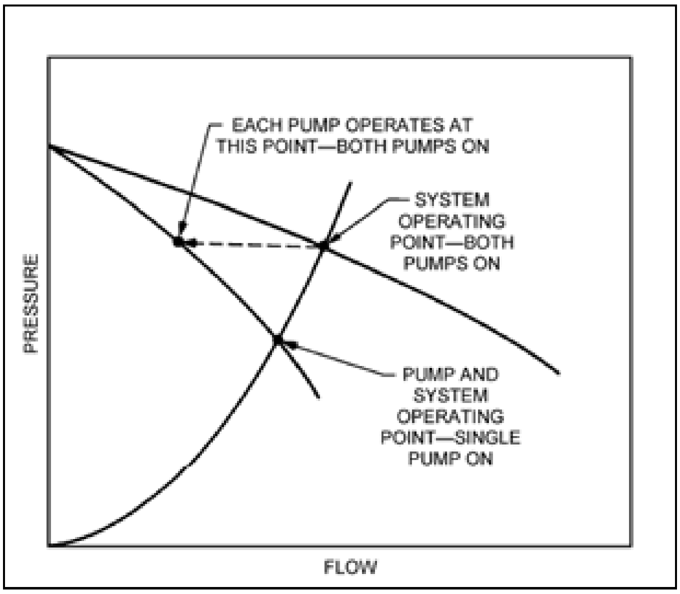


Fig (36).operating conditions for parallel operation.

- Figure (37) indicates Construction of the composite curve for two dissimilar parallel pumps.

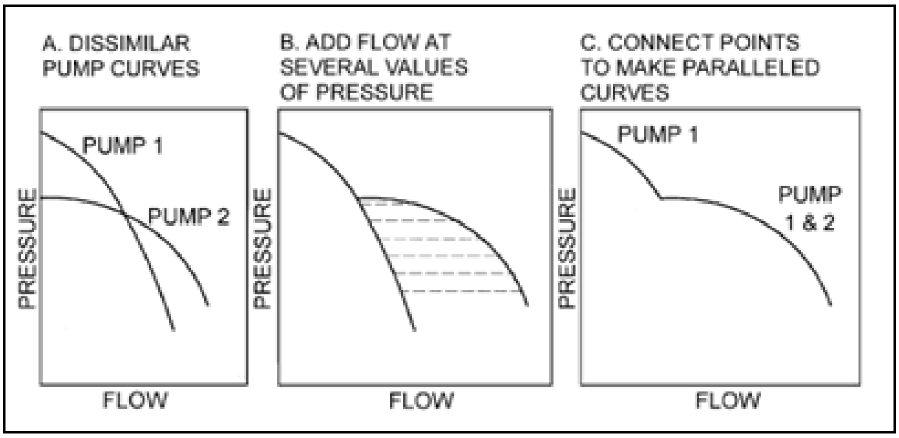


Fig (37).construction of curve for dissimilar parallel pumps.

- Figure (38) shows the typical piping for parallel pumps.

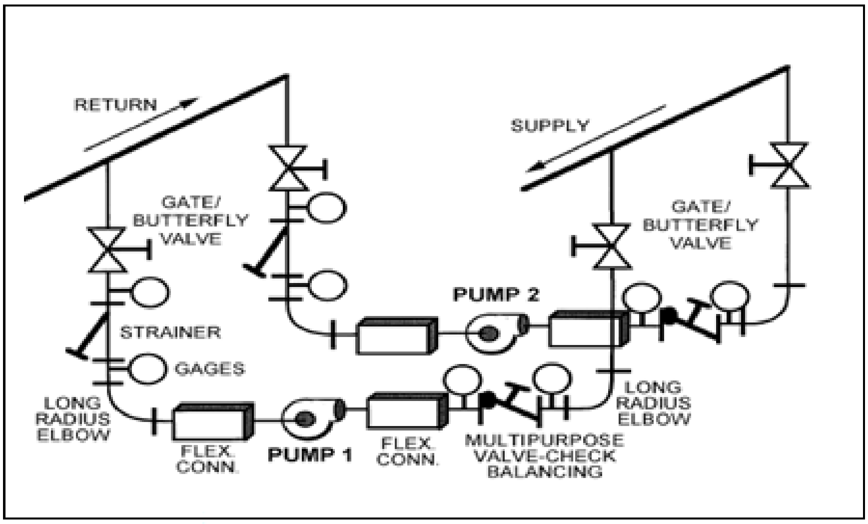


Fig (38).typical piping for parallel pumps.

- Flow can be determined (1) by measuring the pressure increase across the pump and using a factory pump curve to convert the pressure to flow, or (2) by use of a flow-measuring station or multipurpose valve.

2.3.9.2 Multiple pumps in series.

- **Series pumping:** When pumps are applied in series, each pump operates at the same flow rate and provides its share of the total pressure at that flow as shown in figure (39).

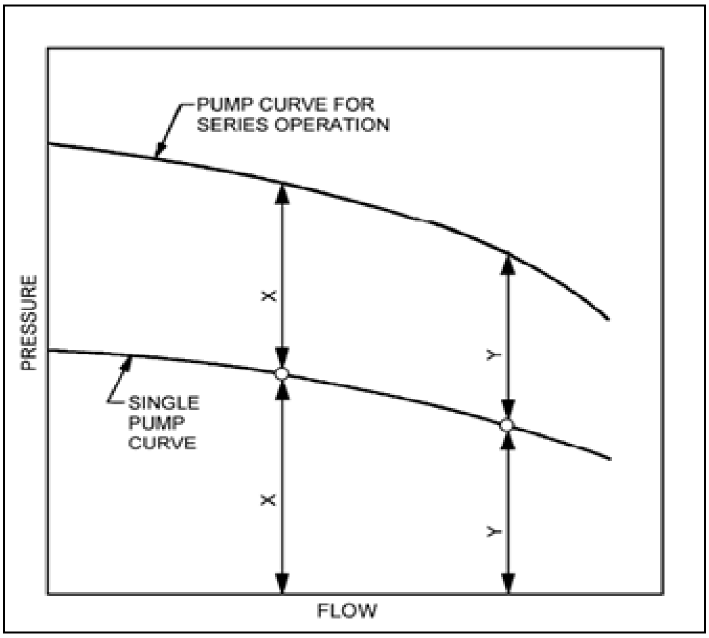


Fig (39).pump curve construction for series operation.

- A system curve plot shows the operating points for both single and series pump operation as shown in figure (40). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

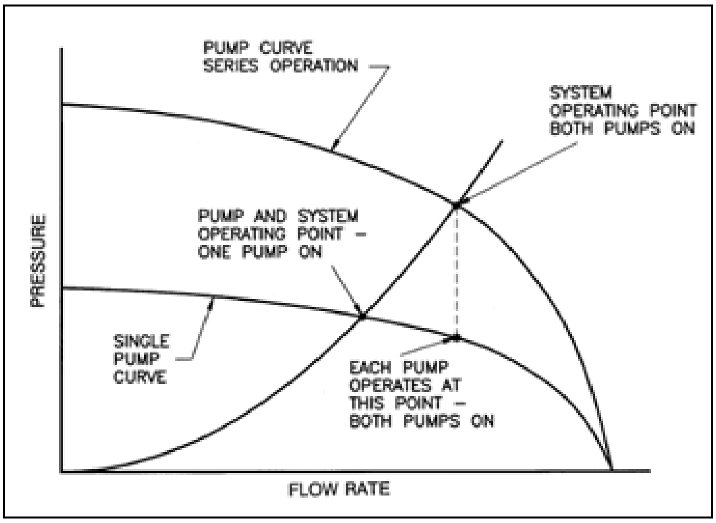


Fig (40).operating conditions for series operation.

- Figure (41) shows the typical piping for series pumps.

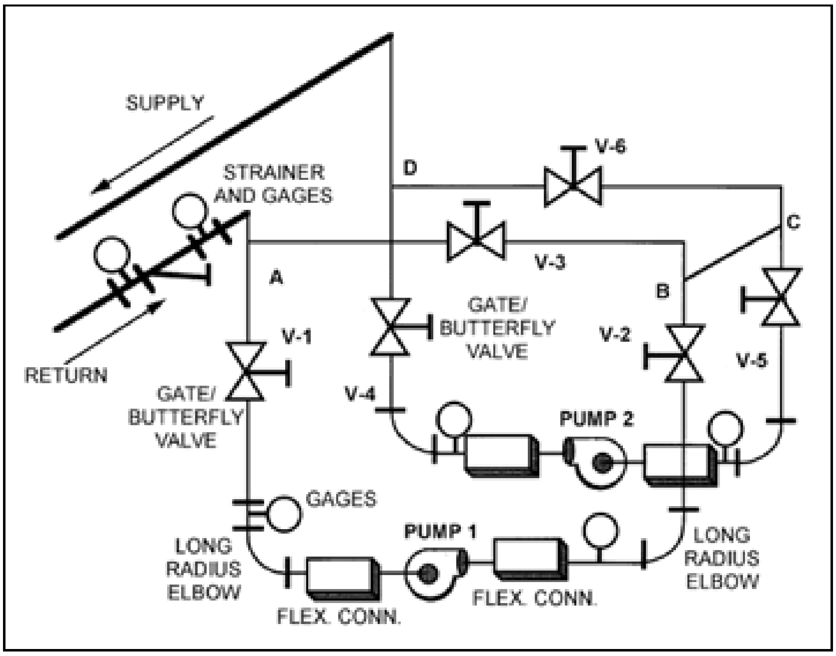


Fig (41).typical piping for series pumps.

2.3.10 Stand by pump

- A backup or standby pump of equal capacity and pressure installed in parallel to the main pump is recommended to operate during an emergency or to ensure continuous operation when a pump is taken out of operation for routine service.

2.3.11 Pump with two –speed motors.

- A pump with a two-speed motor provides a simple means of reducing capacity. Figure (42) shows an example with two parallel two-speed pumps.

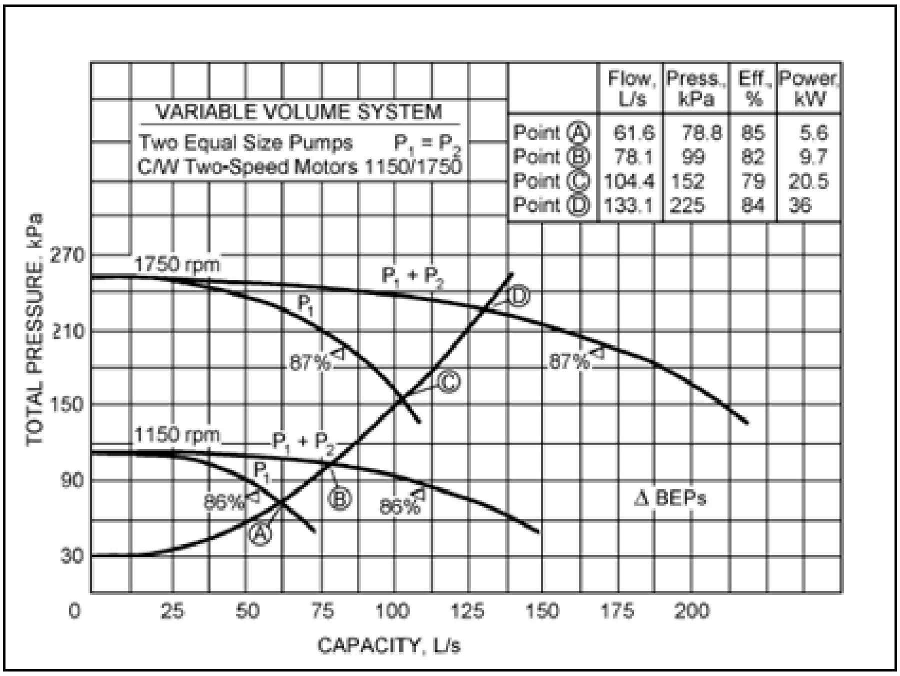


Fig (42).example of two parallel pumps with two speed motors

2.3.12 Pressure gradient along liquid path in pump suction.

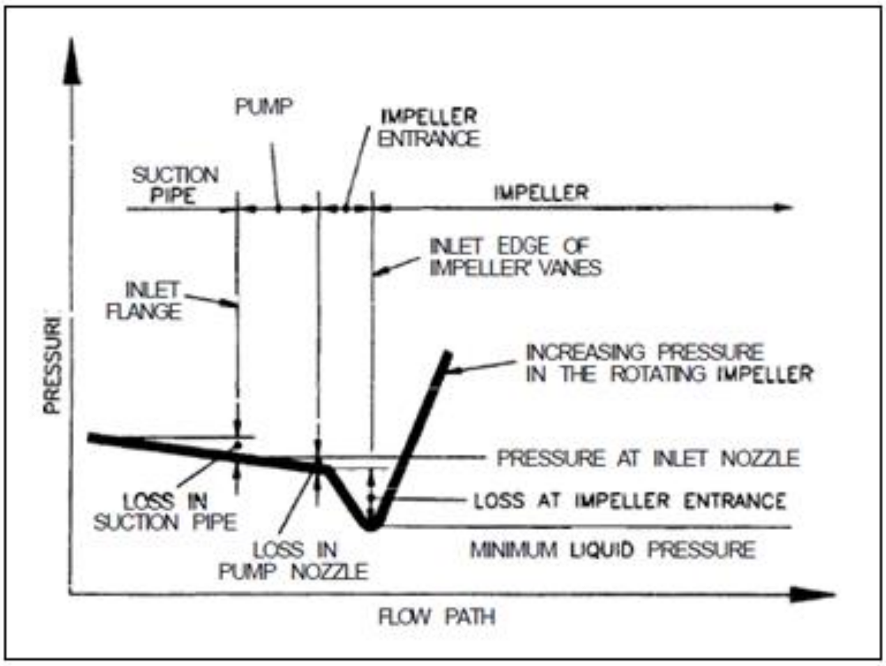


Fig (43).pressure gradient along liquid path in pump suction.

2.3.13 Pump net positive suction head

- The important consideration for pump selection is determination pump net positive suction head to eliminate cavitation (cavitation is the result of change part of water stream from liquid to a gas it is occur when temperature of water reach to evaporation temperature for absolute temperature for that stream, the specific gravity of gas less than specific gravity of water so the result is hammering phenomena which causing damage internal parts of impeller or pump casing. So to avoid cavitation ensuring that pressure in every part of the water system is greater than the evaporation for any temperature of water in that system.)

- Net positive suction head on the pump suction must be always being equal or greater than the net positive suction head required by the pump.

$$\textbf{NPSHA} \geq \textbf{NPSHR}$$

- For calculating NPSH with open system (cooling towers) from following equation and this equation acceptable for HVAC operation up to temperature 85 F.

$$\text{NPSHA} = p_a + p_s - p_{vp} - p_f \text{ (in ft of head)}$$

Where

p_a= atmospheric pressure in feet at the installation altitude. (From table (2))

p_s = static head of water level above the pump impeller (this is negative if water level is below the impeller.

p_{vp}= vapor pressure of water, in feet, at operating temperature. (From table (3))

p_f= friction of suction pipe, fittings, and valves, in feet of head.

Temperature, °F	Absolute pressure, ftH$_2$O	Specific weight γ, lb/ft^3
32	0.20	62.42
40	0.28	62.42
45	0.34	62.42
50	0.41	62.38
55	0.49	62.38
60	0.59	62.34
65	0.71	62.34
70	0.84	62.26
75	1.00	62.23
80	1.17	62.19
85	1.38	62.15
90	1.62	62.11
95	1.89	62.03
100	2.20	62.00
105	2.56	61.92
110	2.97	61.84
115	3.43	61.80
120	3.95	61.73
130	5.20	61.54
140	6.78	61.39
150	8.75	61.20
160	11.19	61.01
170	14.19	60.79
180	17.85	60.57
190	22.28	60.35
200	27.60	60.13
210	33.96	59.88
212	35.38	59.81

Temperature, °F	Absolute pressure, psia	Specific weight γ, lb/ft^3
212	14.70	59.81
220	17.19	59.63
230	20.78	59.38
240	24.97	59.10
250	29.83	58.82
260	35.43	58.51
270	41.86	58.24
280	49.20	57.94
290	57.56	57.64
300	67.01	57.31
320	89.66	56.66
340	118.01	55.96
360	153.03	55.22
380	195.77	54.47
400	247.31	53.65
420	308.83	52.80
440	381.59	51.92
450	422.6	51.55

Table (2).vapor pressure and specific weight of water at various temperatures.

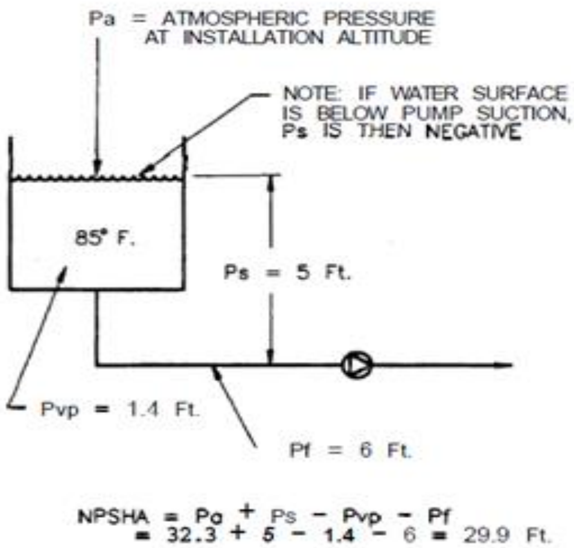
Altitude, ft	Average Pressure P, PSIA	Average Pressure P_a, ftH$_2$O, Up to 85°F
0	14.7	34.0
500	14.4	33.3
1,000	14.2	32.8
1,500	13.9	32.1
2,000	13.7	31.6
2,500	13.4	31.0
3,000	13.2	30.5
4,000	12.7	29.3
5,000	12.2	28.2
6,000	11.8	27.3
7,000	11.3	26.1
8,000	10.9	25.2
9,000	10.5	24.3
10,000	10.1	23.3
15,000	8.3	19.2
20,000	6.7	15.5

Table(3).variation of atmospheric pressure with altitude.

- for HVAC operation more than 85 F, NPSH calculated from the following equation

$$\text{NPSHA}= 144.\, p_{e/Y} + p_s - p_{vp} - p_f \text{ (in ft of head)}$$

- **NPSHA example**



2.3.14 Pump and install considerations.

- For every cooling tower installation should be checked to ensure that adequate net positive suction head available exits for the pumps selected for that cooling tower.

- The pump should be located near the cooling towers to avoid loop in suction pipe as shown in figure (44).

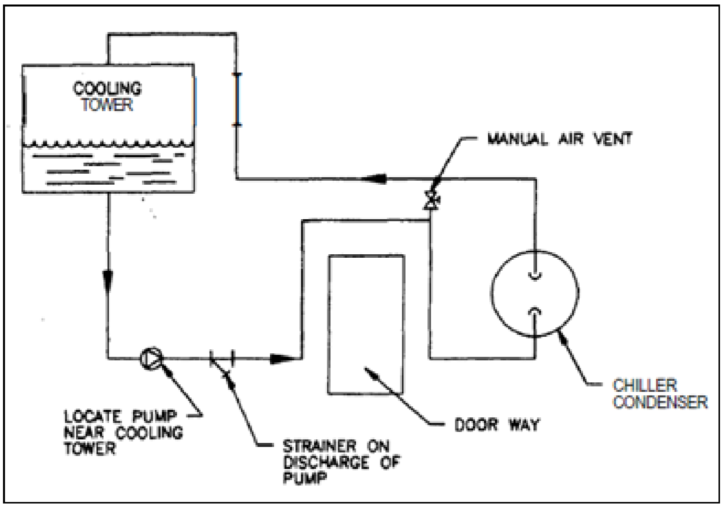


Fig (44).locate condenser pump near cooling tower on difficult installations.

- The strainer installed on pump discharge not on pump suction as shown in previous figure which used to protect tubes in chiller condenser or DX water system condenser. If the strainer installed in pump suction could be obliterated with algae then the pump overhead and be destroyed before it could be stopped but if the strainer installed on pump discharge protect it.

- Cooling towers pump should not be necessarily is located on the supply side of chiller condensers as shown in figure (45). On high rise buildings the condenser pumps can be located on the discharge of the condenser which reduce Pressure on the condenser water boxes and may eliminate the cost of higher pressure construction as shown in figure (46).

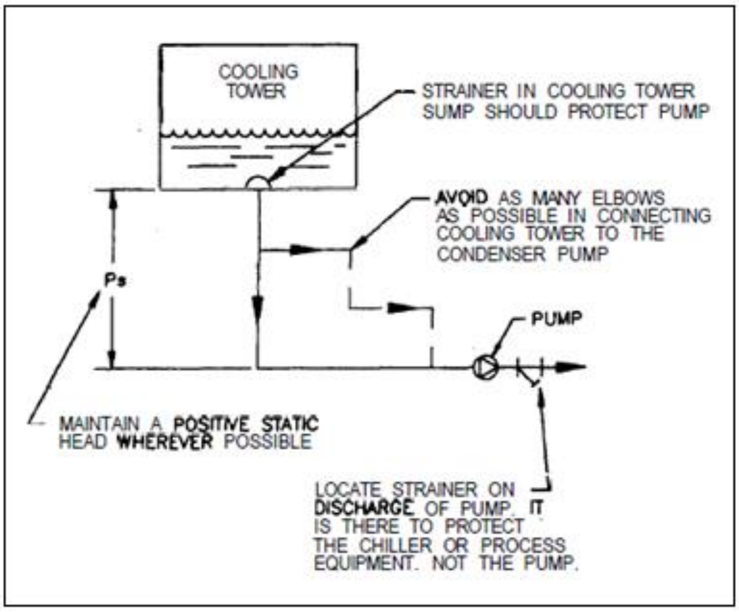


Fig (45).condenser pumps on supply side of chiller condenser.

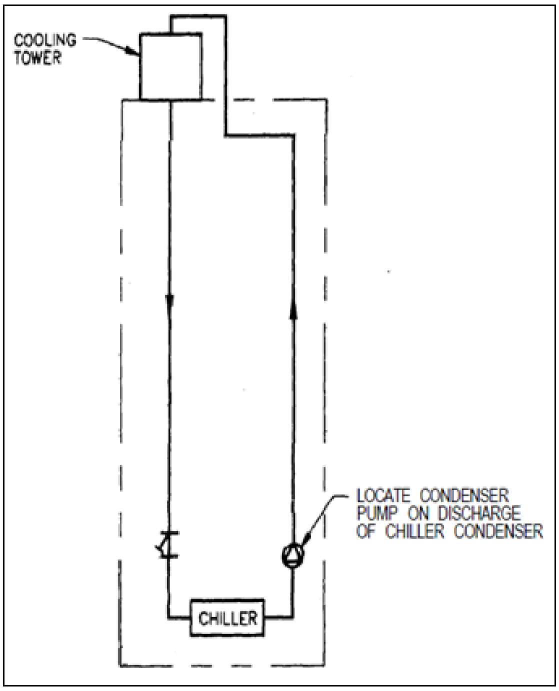


Fig (46).condenser pumps on discharge side of chiller condenser.

- The smaller chiller need only constant speed end suction pumps with a pump foe each chiller, and if stand by pump needed it can be added as shown in figure (47).

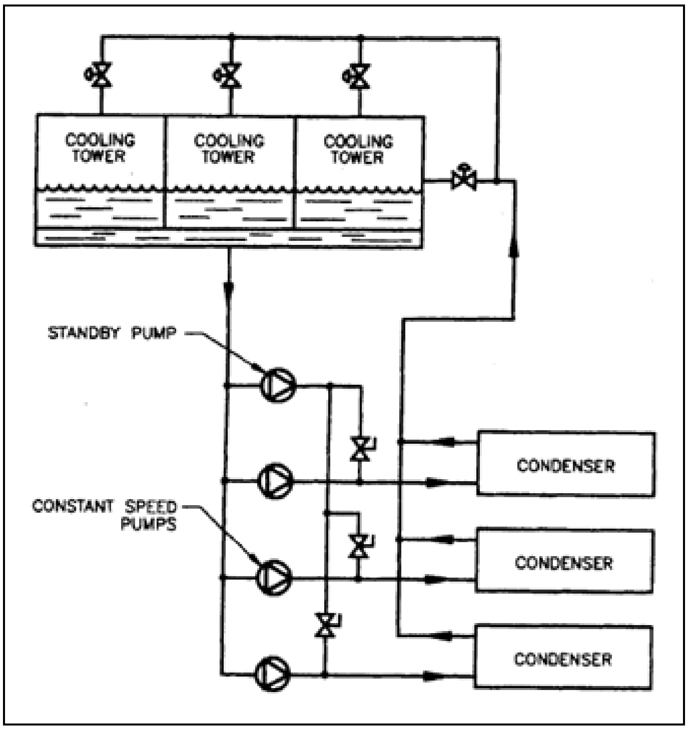


Fig (47).individual pumps for chiller condenser.

- Headering the condenser pump s is more expensive than stand by pump, its requires another pipe header, and each chiller must be equipped with a two- way automatic shutoff valve. As shown in figure (48).

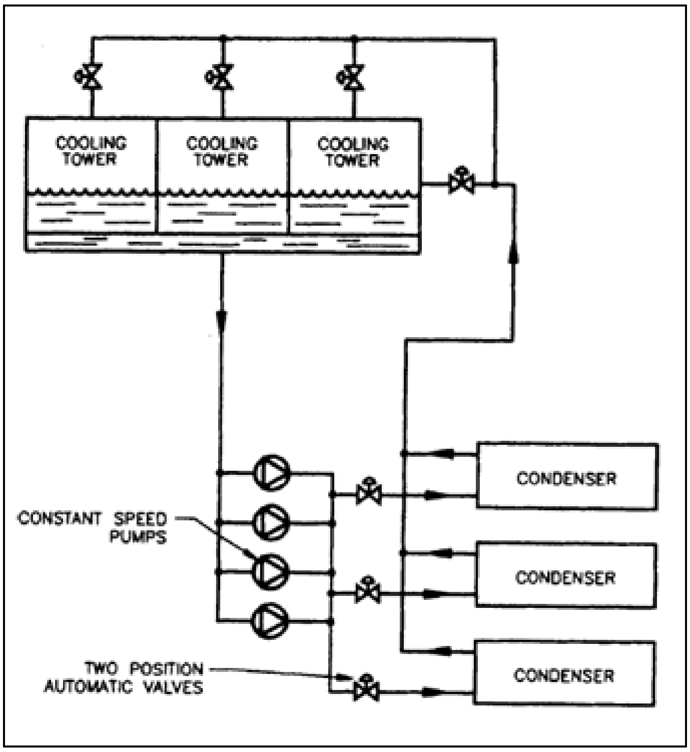


Fig (48).common discharge header for condenser pumps.

- the age-old problem of operating the chiller and cooling tower at their desired flow rates has eliminated by the use of so – called primary –secondary pumping ,as shown in figure (49) .the constant speed cooling tower pumps pressurize the supply to the variable speed condenser pumps ,so there is no NPSH problems with the condenser pumps. this pumping arrangement puts all static head of the cooling tower on the tower pumps, letting the condenser pumps operates on the total friction head Where variable speed pump is supplied for each chiller, the condenser pumps speed varied by refrigerant pressure difference between the high and low sides of chiller as shown in figure (50) or condenser pumps Can be controlled by pressure drop through the condensers.

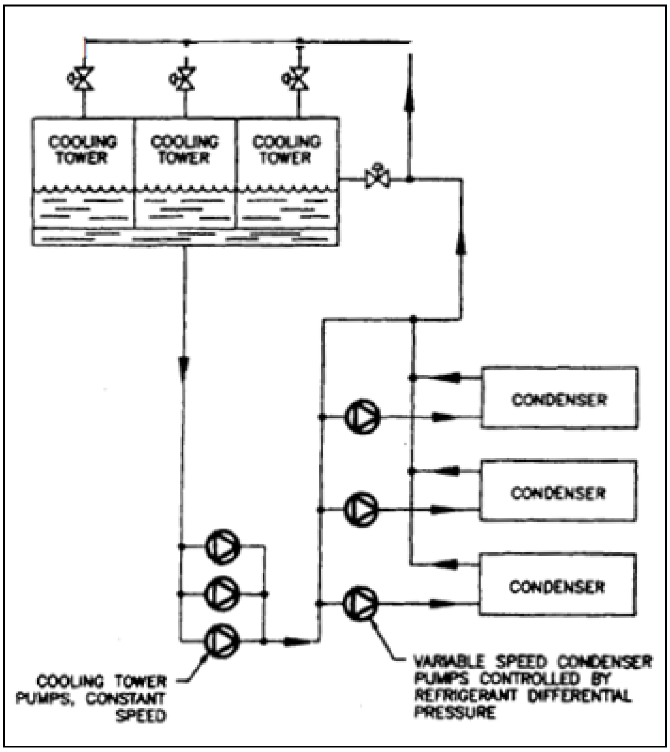


Fig (49).individual variable speed condenser pumps.

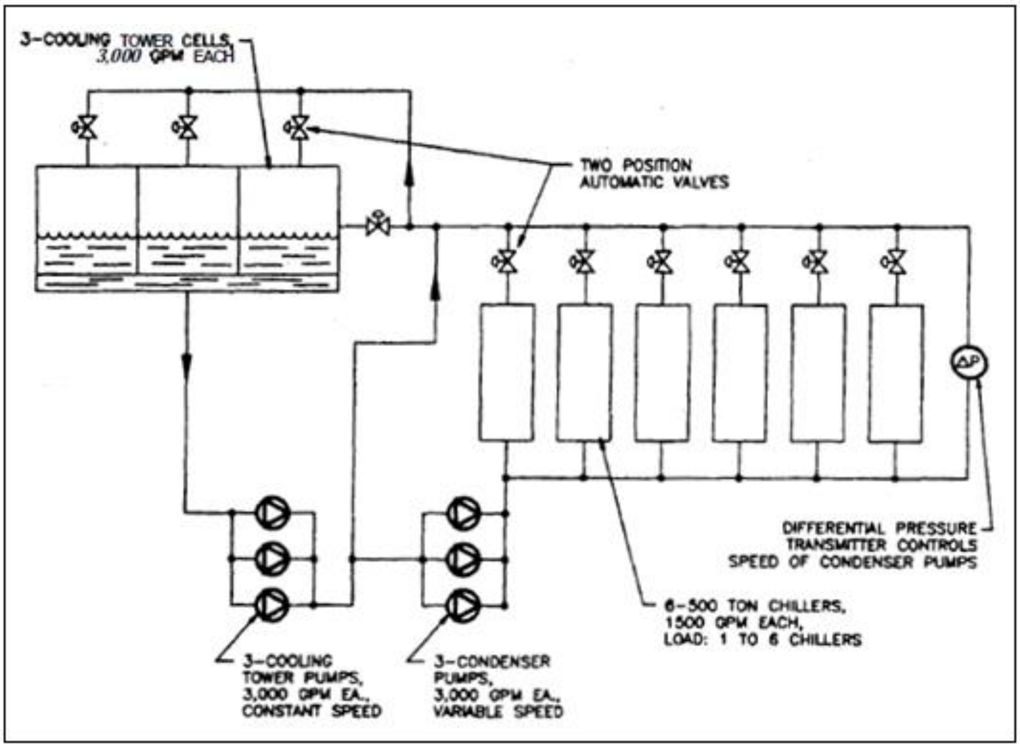


Fig (50).multiple chiller installation.

2.3.15 Condenser water Pumps selection.

- **for pump specification the following information must be available:**
- **pump type.**
- **capacity rating.**
- **pump RPM.**
- **pump flow rate.**
- **pump head:** for calculation the total head required for selecting proper pump for water circulation in open system the following information must be available .
  - ✓ GPM to be circulated .
  - ✓ total length of piping.
  - ✓ pressure drop across condenser.
  - ✓ tower hydrostatic lift.
  - ✓ number of valves ,fittings, and any other resistances in piping system.
  - ✓ type of pipe used.
- **pump head example**

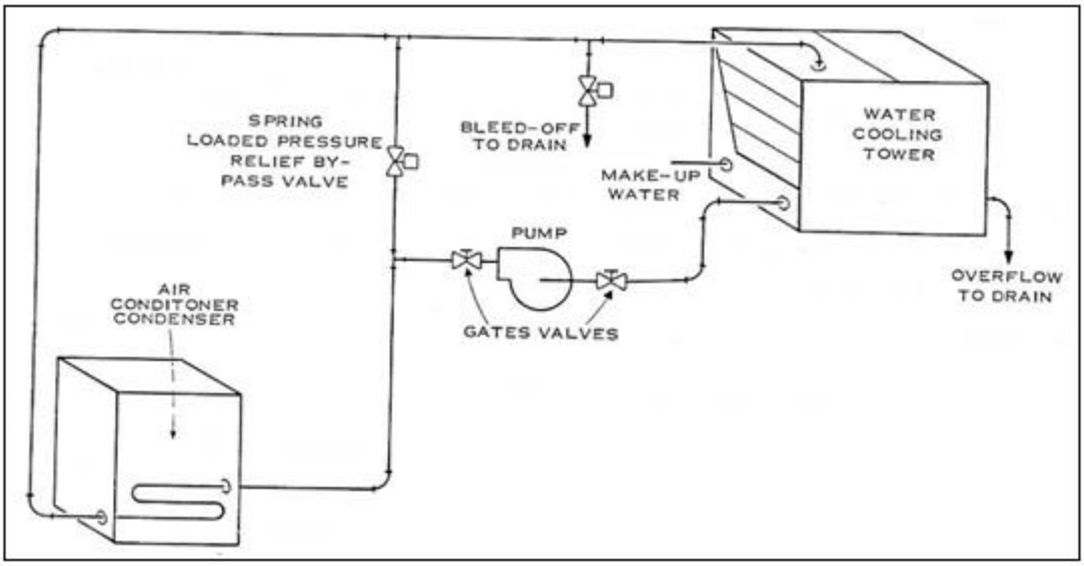


Fig (51). Typical condenser water piping arrangement.

- given information for the above system :
  - ✓ GPM to be circulated =30
  - ✓ 80 ft of total piping
  - ✓ 11.4 psi pressure drop across condenser at 30 GPM
  - ✓ 5 ft hydrostatic lift of tower
  - ✓ 6 standard 90 elbow
  - ✓ 2gate valve
  - ✓ 1 standard tee through side outlet.
  - ✓ 1 standard tee straight through.
  - ✓ sch40 pipe (assume to have" fairly rough" interior surface)

- **solution**

  - ✓ calculate total equivalent length
  - 80 ft of standard pipe = 80
  - 6 standard 90 elbow at 4.5 ft =27
  - 2gate valve at 0.9 ft =1.8
  - 1 standard tee through side outlet at 7.5 ft =7.5
  - 1 standard tee straight through at 3 ft 3
  - ✓ total equivalent length =119.3 feet
  - ✓ pipe size =1.5"
  - ✓ pipe pressure drop =119.3x13.4/100 = 16 ft of water.
  - ✓ **total pressure drop for pump selection=**pressure drop due to pipe and fittings pressure due to condenser pressure due to hydrostatic lift.
    =16+11.4x2.31+5 =47.3 feet.

- Pump manufacturers may compile performance curves for a particular set of pump volutes in a series this set of curves is known as **a family of curves** as shown in figure (52). A family of curves is useful in determining the approximate size and model required, but the particular pump performance curve must then be used to confirm an accurate selection as shown in figure (53).

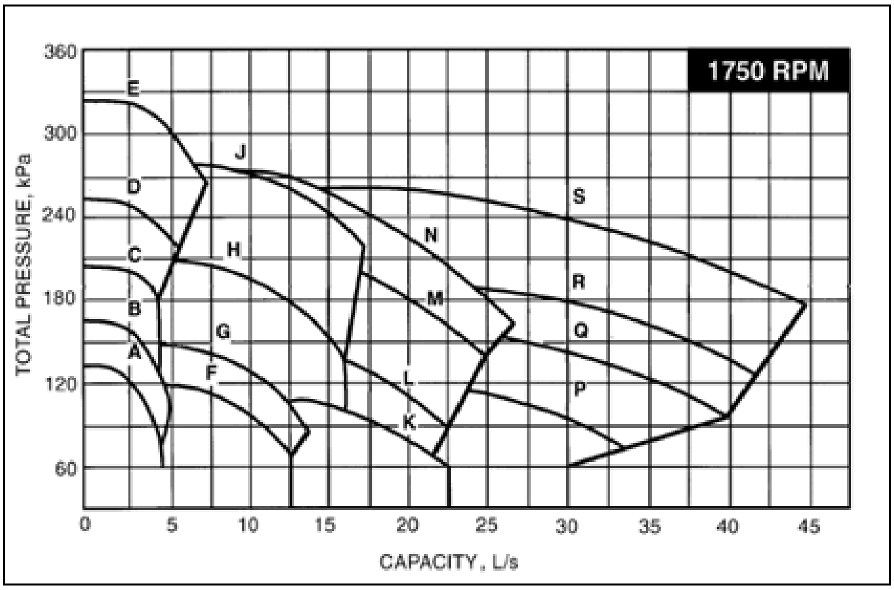


Fig (52).typical pump manufacture s performance curves.

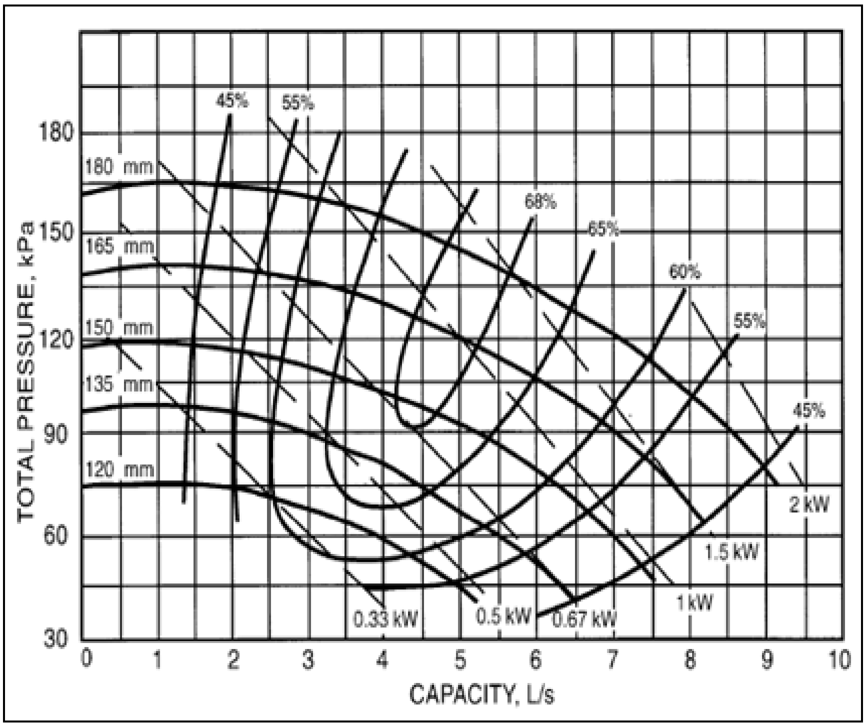


Fig (53).typical pump performance curve.

2.4 Condenser water piping system arrangement

2.4.1 Condenser water piping system arrangement

- Condenser water piping system can be also classified according to water return arrangements as follow:

- Reverse return piping.

  ✓ It's recommended if the units have the same or nearly the same pressure drop thru them. Reverse return piping as shown in figure (54) is recommended for most closed piping applications. It is often the most economical design on new construction. The length of the water circuit thru the supply and return piping is the same for all units. Since the water circuits are equal for each unit.

  ✓ The major Advantage of a reverse return system is that it seldom requires balancing. The following figure is a schematic sketch of this system with units piped horizontally and vertically.

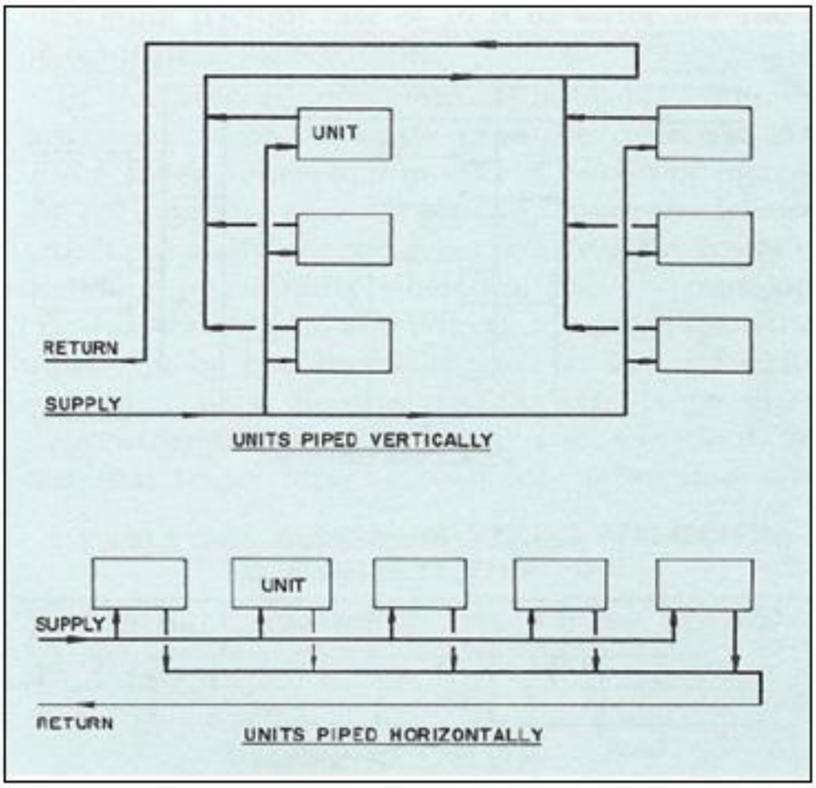


Fig (54).reverse return piping system.

- Reverse return header with direct return risers

  ✓ There are installations where it is both inconvenient and economically unsound to use a complete reverse return water piping system. This sometimes exists in a building where the first floor has previously been air conditioned. To avoid disturbing the first floor occupants, reverse return headers are located at the top of the building and direct return risers to the units are used as shown in figure (55).

  ✓ In this system the flow rate is not equal for all units on a direct return riser. Excessive unbalance in the direct supply and return portion of the piping system may dictate the need for balancing valves or orifices.

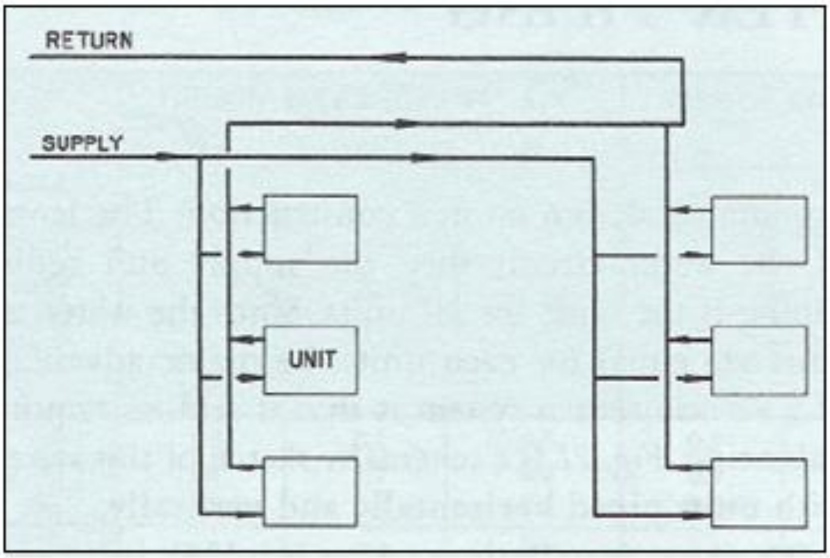


Fig (55).reverse return headers with direct return riser.

- Direct return piping.

  ✓ It's recommended if the all units have different pressure drops or require balancing valves, then it is usually more economical to use a direct return as shown in figure (56).

  ✓ Direct return piping is recommended for closed recirculating system where all the units require balancing valves and have different pressure drops. Such as Several fan-coil units piped together and requiring different water flow rates, capacities and pressure drops. The following figure show vertical and horizontal direct return piping system.

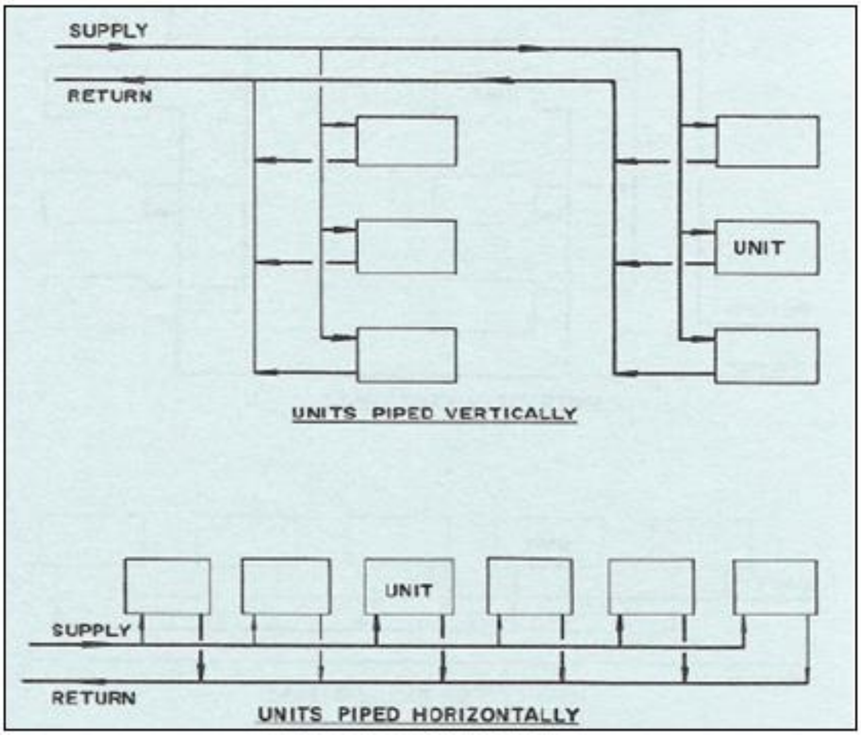


Fig (56).direct return piping system.

2.4.2 Condenser water system pipe sizing.

- There is a friction loss in any pipe thru which water is flowing. This loss depends on the following factors: **Water velocity, Interior surface roughness ,Pipe diameter, and Pipe length**

- The velocities recommended for water piping depend on two conditions:

  - ✓ The service for which the pipe is to be used.

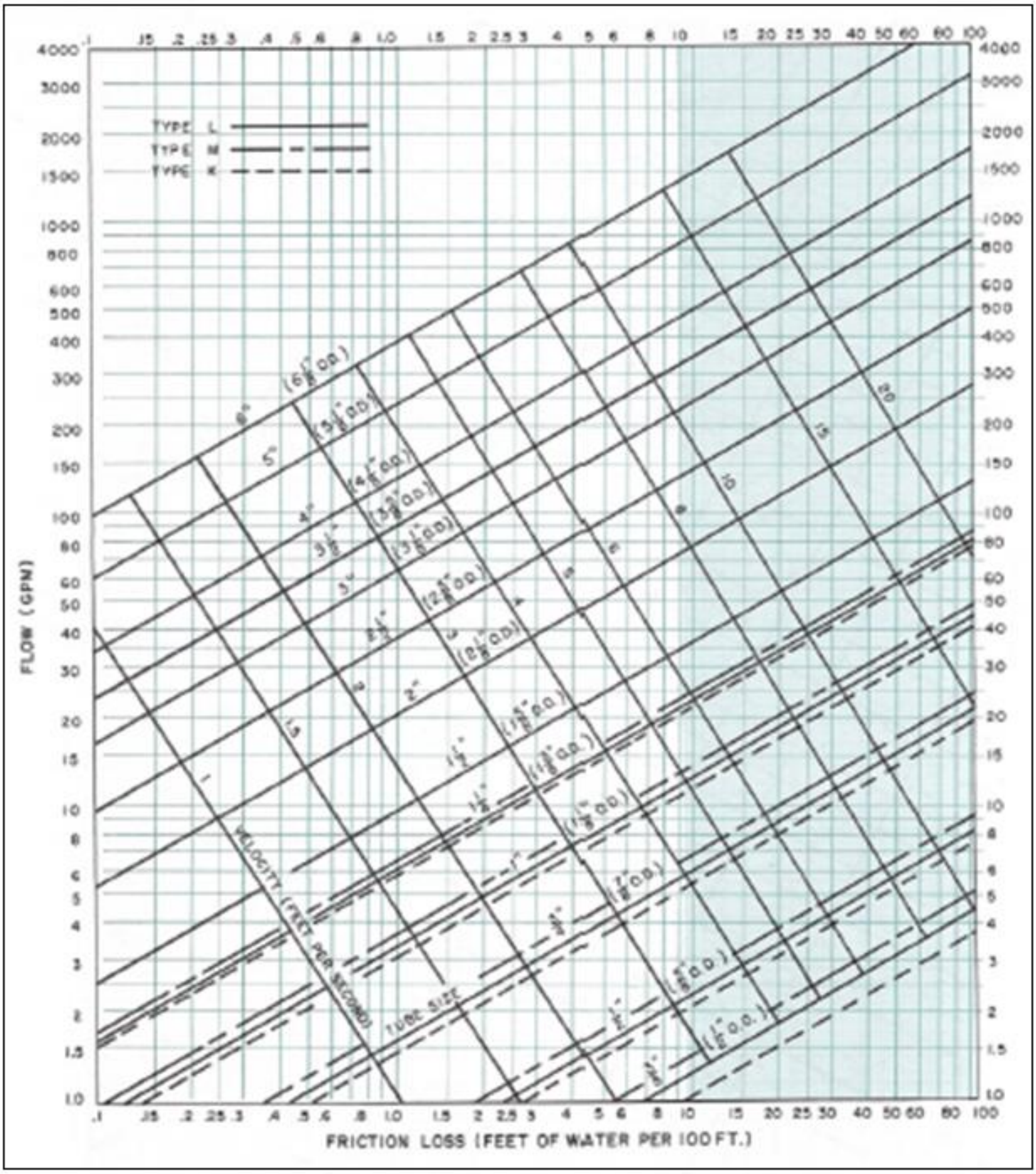
  - ✓ The effects of erosion: Erosion in water piping systems is the impingement on the inside surface of tube or pipe or rapidly moving water containing air bubbles, sand or other solid matter.

- Since erosion is a function of time, water velocity, and suspended materials in the water, the selection of a design water velocity is a matter of judgment. Table (4) indicates recommended velocity ranges for different services.

SERVICE	VELOCITY RANGE (fps)
Pump discharge	8 - 12
Pump suction	4 - 7
Drain line	4 - 7
Header	4 - 15
Riser	3 - 10
General service	5 - 10
City water	3 - 7

Table (4).recommended water velocity

- By knowing value of water quantity is determined from the air conditioning load and the water velocity by predetermined recommendations. These two factors are used to establish pipe size and friction rate ft of length. Figure (57), and (58) indicates the charts of FRICTION LOSS FOR OPEN PIPINGSYSTEMS Schedule 40 Pipe, and FRICTION LOSS FOR CLOSEDAND OPEN PIPING SYSTEMS Copper Tubing, and it is **normally good practice not to exceed friction loss above 10 feet per 100 feet.**



Fig(57).friction loss for closed and open piping system for copper tubing.

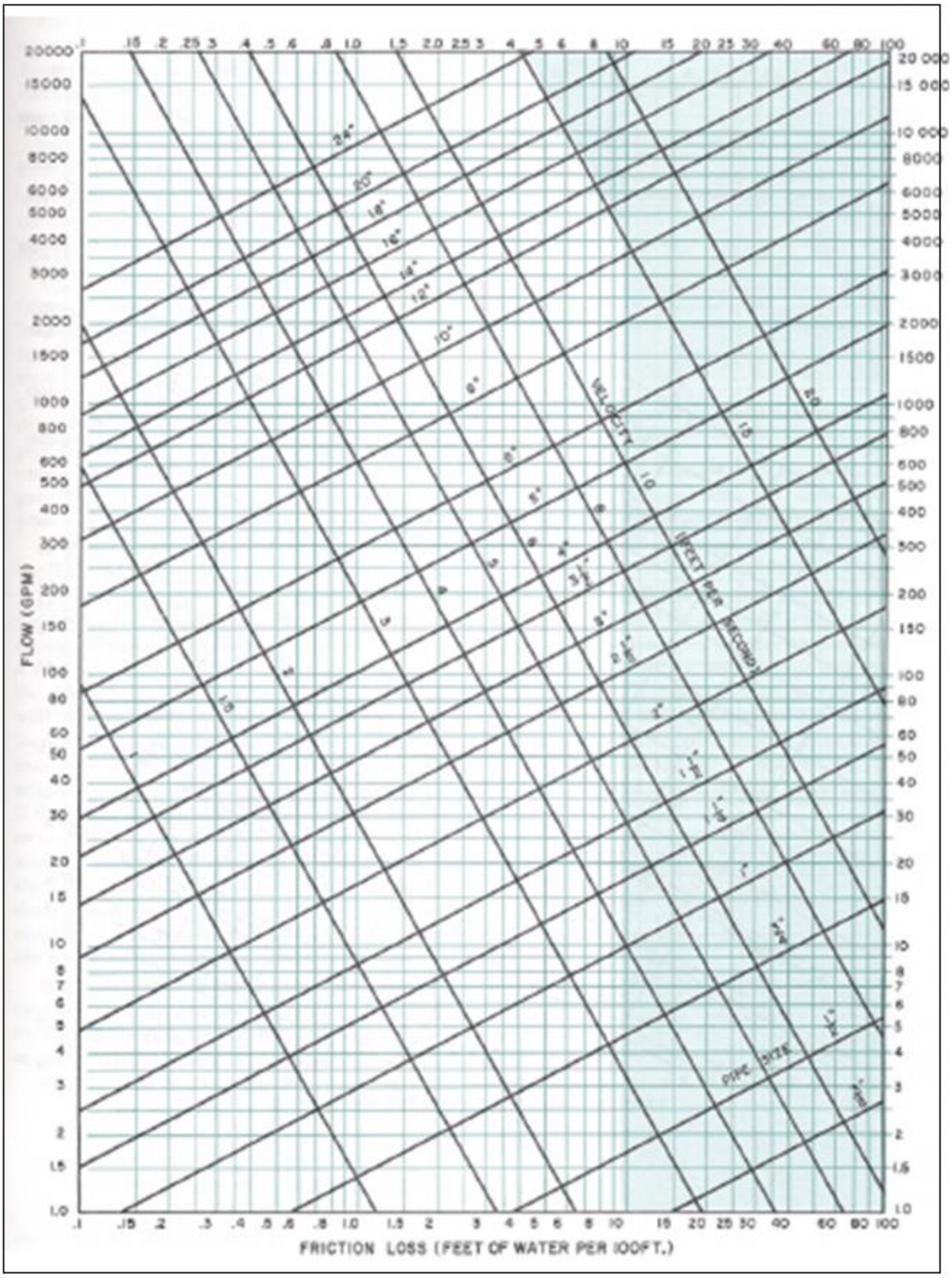


Fig (58).friction loss for open piping systems, schedule 40pipe.

- Condenser water system pipe sizing can do by using many of software programming like pipe sizer as shown in figure (59).

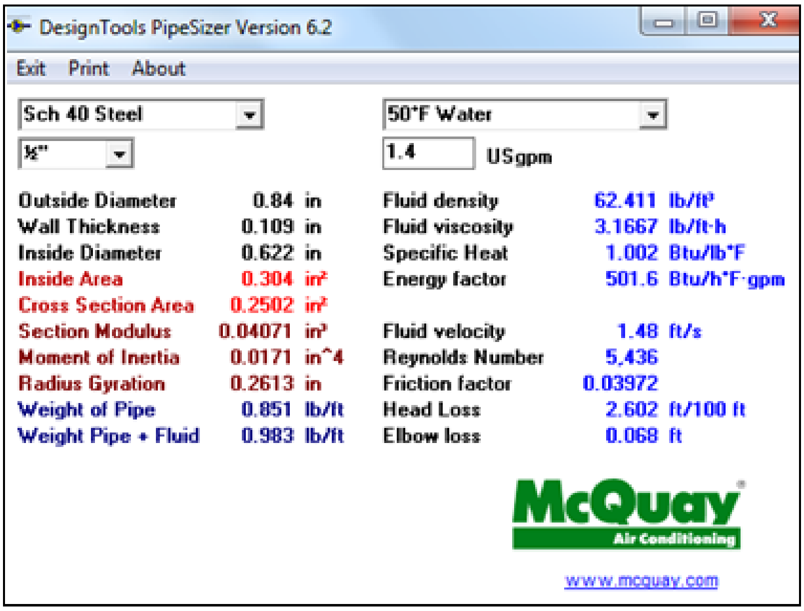


Fig (59).design tools pipe sizer.


SESSION END

Chapter 7 –chilled water systems Design Concepts

Contents

Chapter 7 –chilled water systems Design Concepts

1 Water distribution systems.

- There are 4 types of water distribution systems; they are defined by number of pipes used in the system as the following.

1.1 One Pipe systems.

- 1 pipe system water distribution has one pipe looping around the building and then returning as shown in the figure (1), And typical used for heating systems and it seldom to use for distribution of chilled water due to the quantity of water used with chilled water higher than with heating systems, unit coils water work on smaller temperature differentials with cooling mode than heating mode.

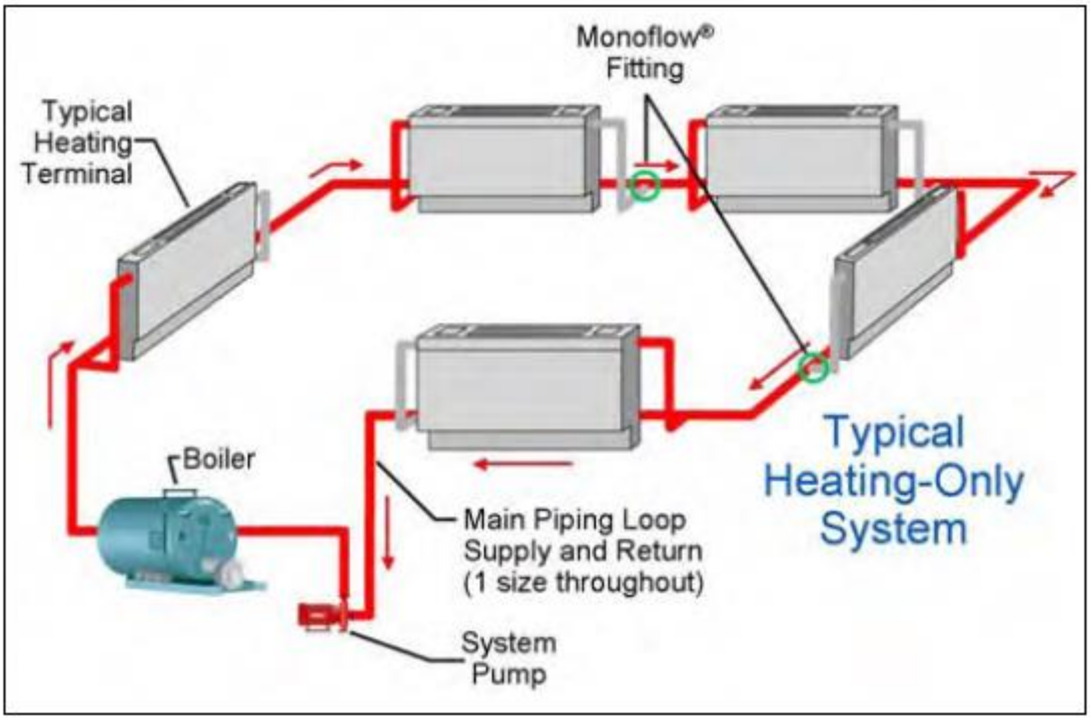


Fig (1).one piping system schematic.

- This pipe for both supply and return main it is size is constant throughout some advantages of this system: simple in design which leads to easy to install, and low cost.
- This system also has several disadvantages: pumping head is generally higher than in the other systems because of the resistance occurring in series. The change in temperature as the water moves through the system (water get colder after each successive terminal because of mixing) create the possible need for large units at the end of main. In order to keep the pressure drop through the unit coil low the water velocity must be kept low this required increasing tube diameter, and greater number of tubes in parallel there for physical space and terminal cost increased with using 1-pipe systems.

1.2 Two Pipe systems.

- 2 pipe systems are used with both cooling, and heating equipment. It useful to use with fan coil units, and medium, large central air handler using combination hot and chilled water coils.
- 2 pipe systems can be used to distribute either cold water or hot water alternate between two as shown in figure (2). So must be defined outdoor temperature (change over temperature) or other indicator building load at which point the hot water in the piping replaced by the chilled water and vice versa. There are 2 forms of 2 pipe water distribution systems in common use ,2 pipe direct return ,and 2 pipe reverse return.

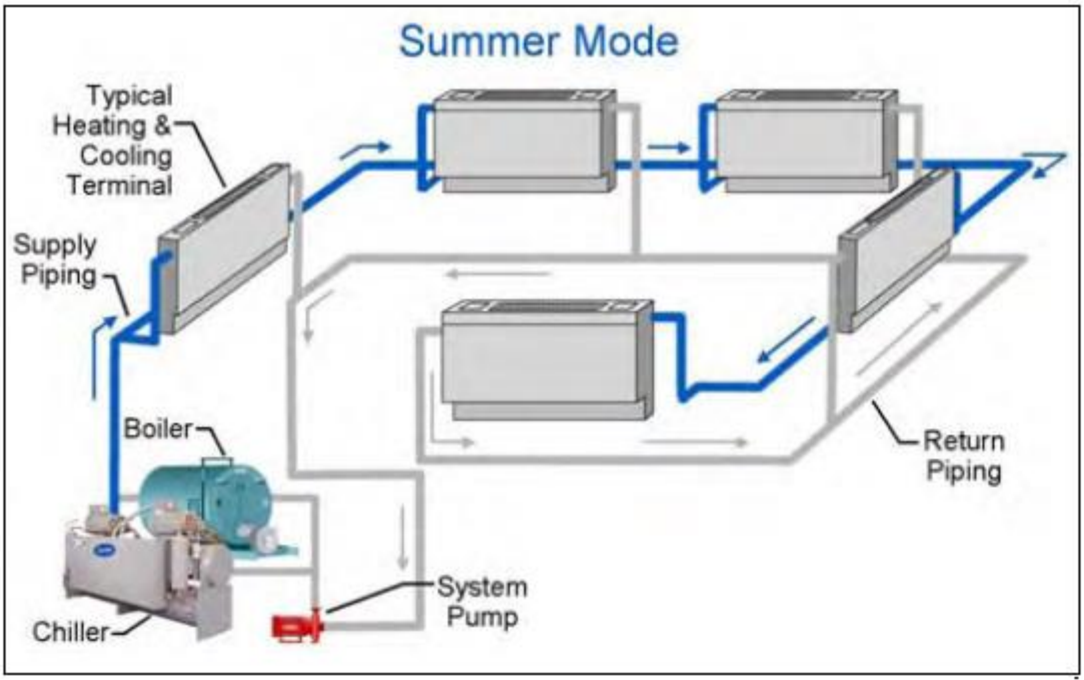


Fig (2). Two piping system schematic.

- In 2 pipe system pipe diameter, amount of water flow rate not constant through the supply and return piping loop.

.2.1 Advantages:

- Pumping head lower than 1-pipe systems due to zone terminals are in parallel not series, it easier to balance the flow to each zone terminals than 1-pipe systems, and temperature entering each zone terminals is same because return water not mix with supply water in the main supply.

.2.2 Disadvantages.

- Installed cost of this system is greater than 1-pipe system even though average pipe diameter in 2 pipe system small pipe diameter in 1- pipe system the extra pipe, and greater number of fitting causes increasing first cost, change over from cooling to heating mode or vice versa takes time (it is not practical to plan to change over frequently, and seasonal changeover is the most common method used).

1.3 Three Pipe systems.

- The 3 pipe water distribution system has one pipe for chilled water supply, one for hot water supply, and one common return main as shown in the figure (3).

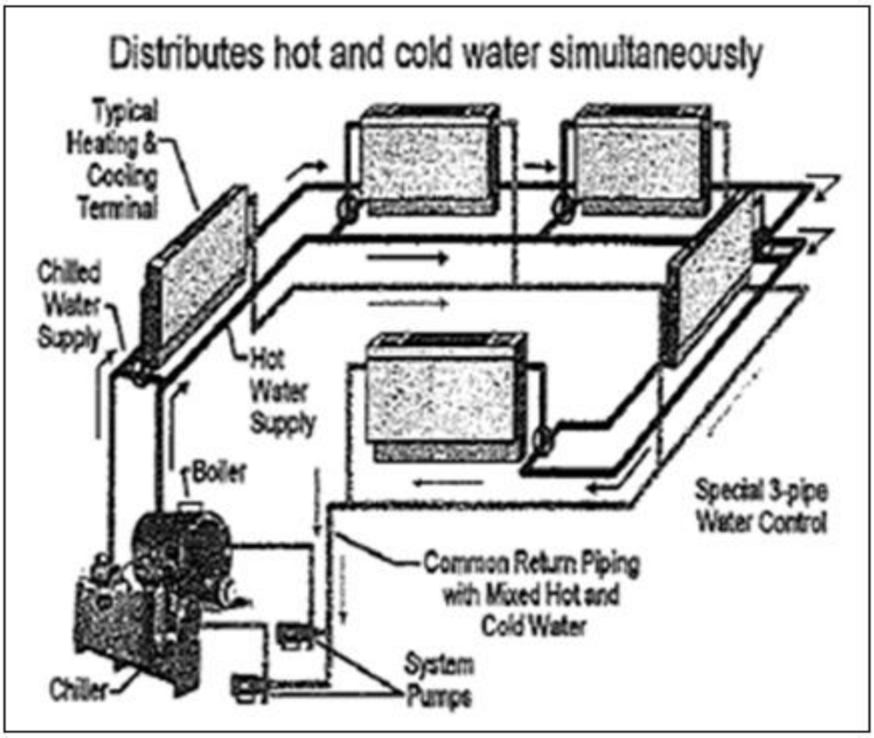


Fig (3). Three piping system schematic.

2 The chilled water supply and hot water supply are sized according to normal standard. But the return is sized to handle with max flow rate (which is cooling flow rate).

3 Return main can be either direct or reverse configuration.

4 ability to change from hot mode to cold mode at any time because 2 supply mains to each zone terminal ,there is always hot and cold water present at the entrance of zone coil ready to be use when needed.

- Note: ASHRAE does not allow for the use of 3 pipe system because the mixing between cold and hot water in the return pipe uses excess energy.

1.4 Four Pipe systems.

- This system actually 2 pipe systems in parallel each system consisting of it is own supply and return main. One system for chilled water distribution and the other system for hot water distribution, the heating and cooling systems are separately No point of connections between2systems as indicated in figure (4).

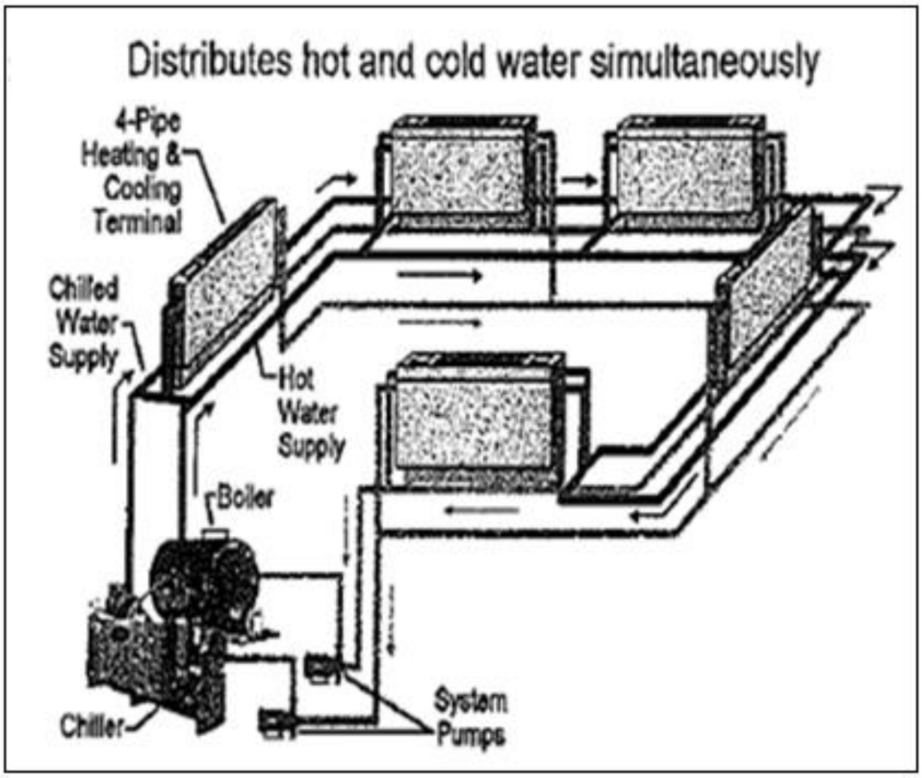


Fig (4). Four piping system schematic.

- Both hot and cold water are available to all units at one time.
- The extra pipe and valves at zone terminals make 4 pipe systems most costly in initial cost .also with 4 pipe systems required terminal units with 2 circuit coils which cost more.
- For commercial building , the choice comes down to 2 pipe system, 4pipe system, the comfort, and control advantage of 4pipe must weighted against the higher installed cost of 4pipe system ,where the building configuration and lay out of spaces may require long period of heating ,and cooling ,and occupant comfort is required ,4 pipe is match your need .

2 chilled water system components

2-1 cooling coil 2-2 chillers 2-3 cooling tower 2-4 pumps

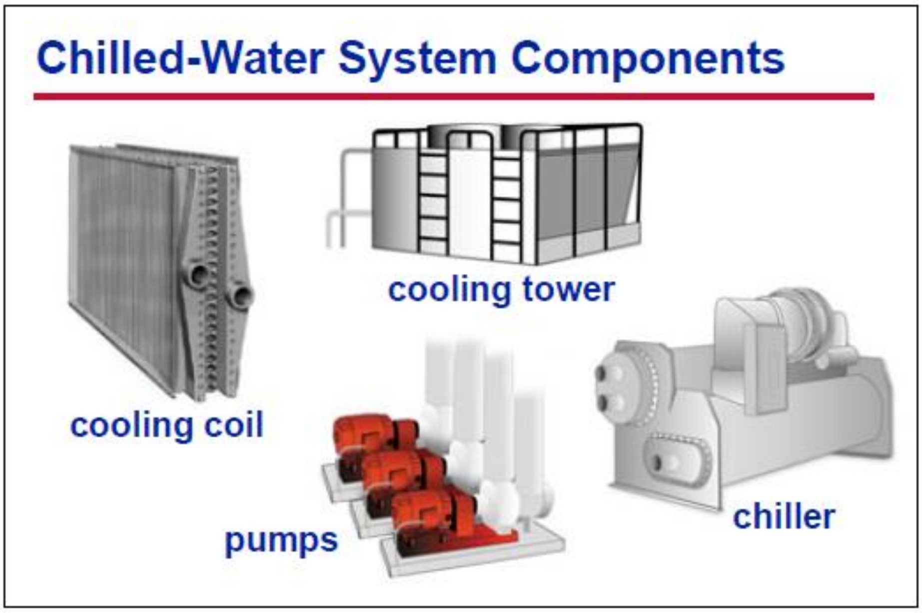


Fig (5). Chilled water system components.

.1 cooling coil

.1.1 Fan coil

- Fan coil unit is terminal unit with
  - ✓ Heat exchanger coil.
  - ✓ Fan motor.
  - ✓ Filter.
  - ✓ Condensate drain pan.
  - ✓ Fan coil can be in cased or uncased configuration, vertical or horizontal, and with or without fresh air inlet. Figure (6) shows FCU components.

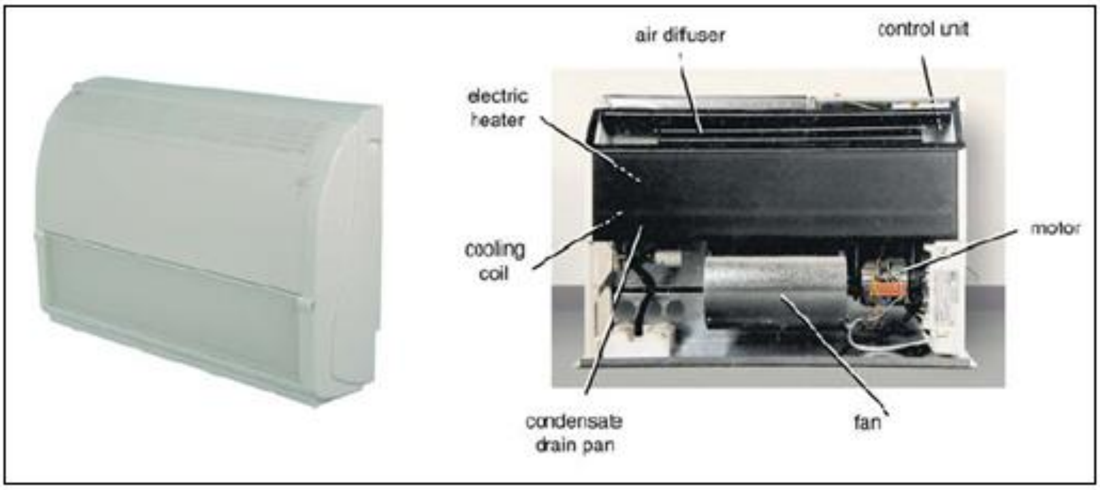


Fig (6). FCU components.

- **For positioning:** the fan coil can be finding in different position as shown in figure (7).

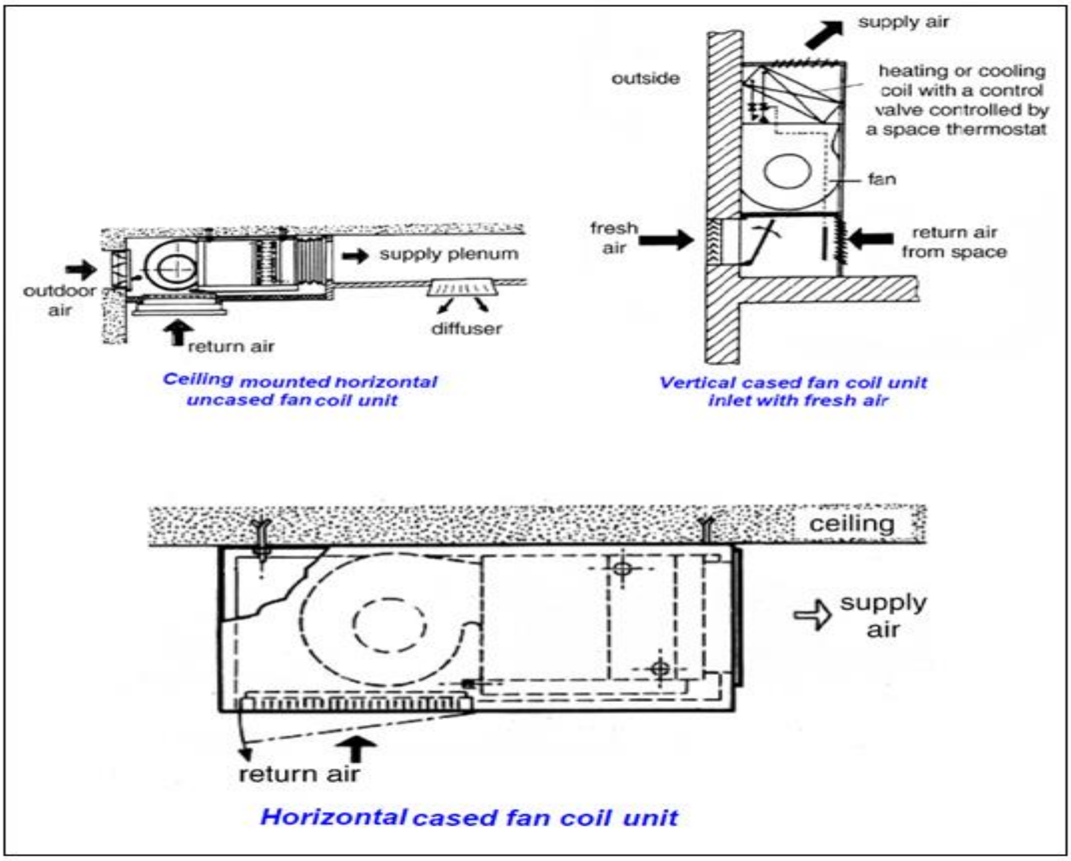


Fig (7). FCU positions.

- **For piping**: the fan coil can be divided into
- **Two piping fan coil:** as shown in figure (8).

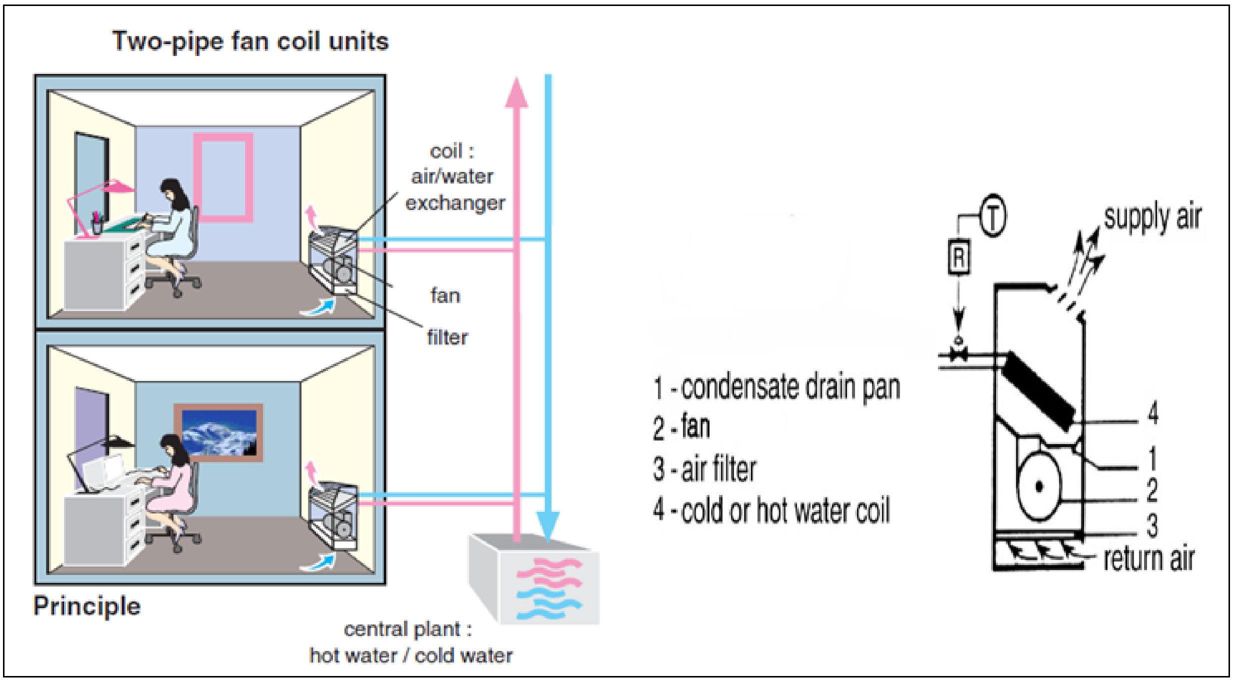


Fig (8).two piping fan coil.

- Field of application:

  ✓ This system used for divided spaces such as hotels, offices, where needs fluctuate but are the same type. It is not satisfied simultaneous heating, and cooling needs: heating is provided in the winter, and cooling the summer.

- **Four pipe fan coil unit**: as shown in figure (9).

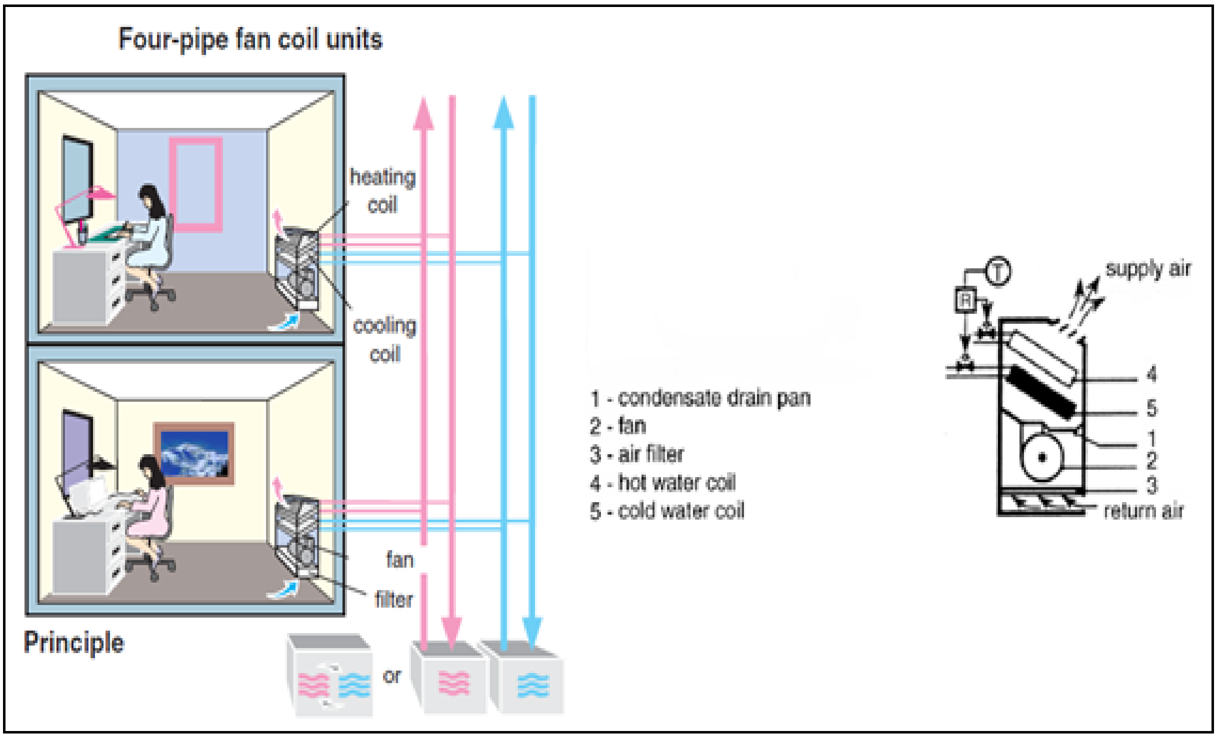


Fig (9). Four piping fan coil

- Field of application:

✓ This system used for hotels ,offices and other spaces with significant heating and cooling needs that fluctuate and are not of the same type. It delivers heating and cooling in the same time.

- **Fan coil unit with two pipes, two wire fan coil unit**: as shown in figure (10).

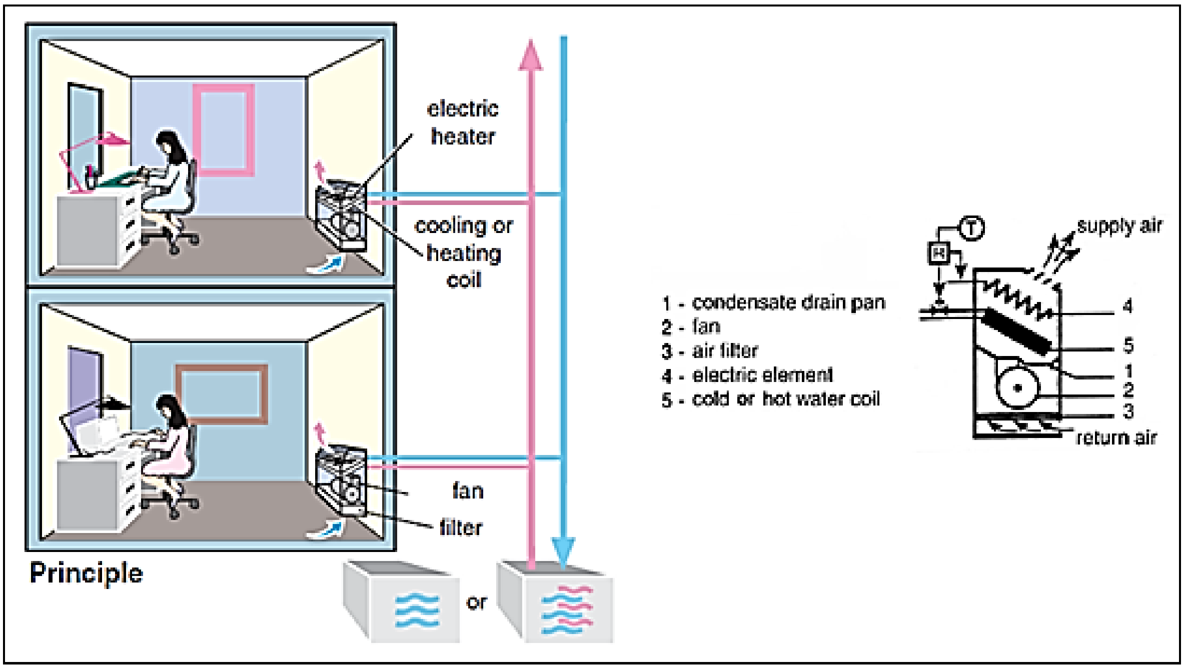


Fig (10). Fan coil unit with two pipes, two wire fan coil unit

- Fields of application
  - ✓ This system used for heavily insulated modern buildings, such as offers, hotels where heating needs are reduced.

- **Non self-contained air handing terminal units**

  - ✓ Have the same components as conventional fan coil units but are designed especially for installation in ceiling, suspended ceiling, and raised floors. There are two types of such terminal units, fan coil units, and mini air handling unit Fan coil units: are designed for installation in suspended ceilings. They both supply air to and recirculate the air directly. They have no available pressure and cannot be connected to duct work.

- There are two model of **cassette type** fan coil unit.
  - ✓ **A - Coadis cassette type unit:** This type of unit can be installed in the corner of a space.as shown in figure (11).

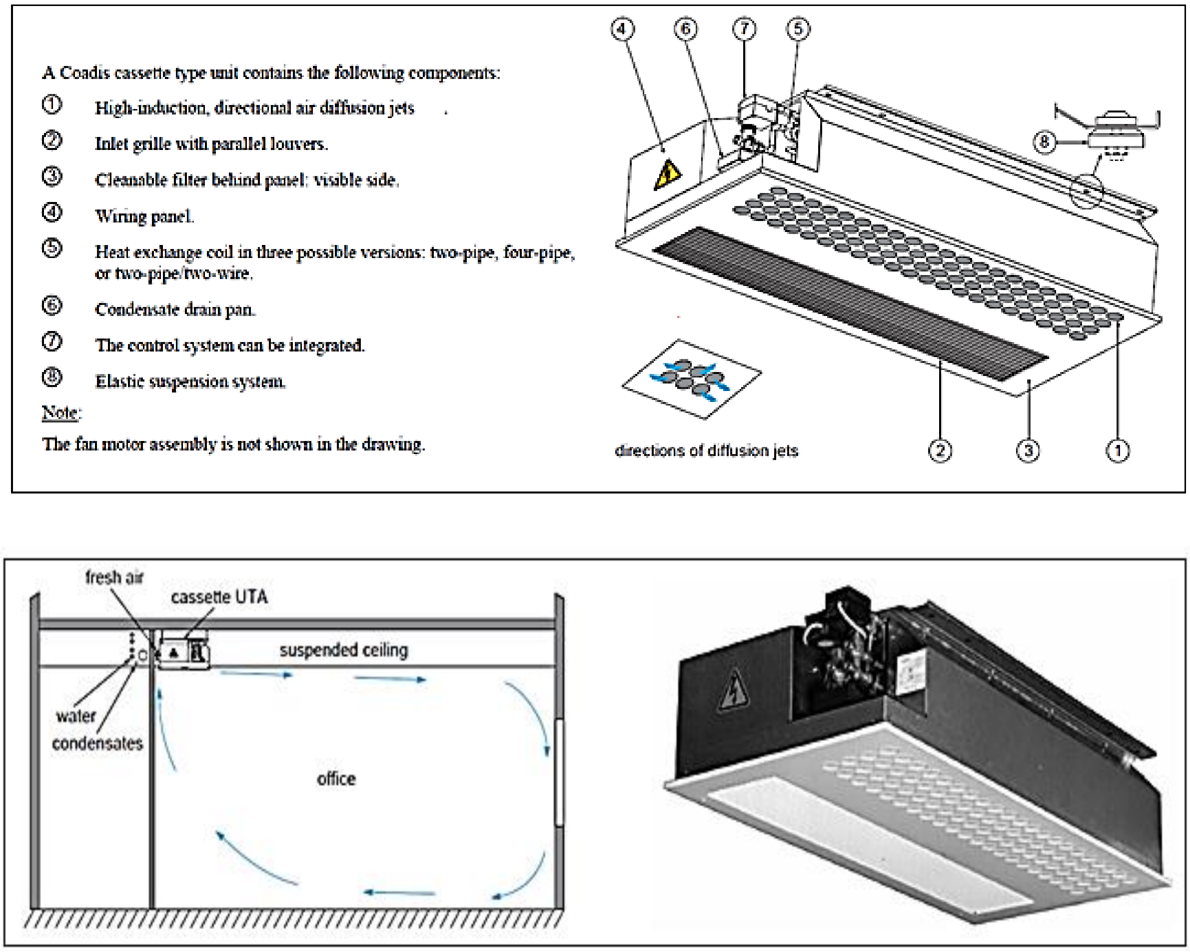


Fig (11). Coadis cassette type unit.

✓ **Melody cassette type fan coil:** Melody fan coil units designed especially for installation in the center of space far from walls.as shown in figure (12).

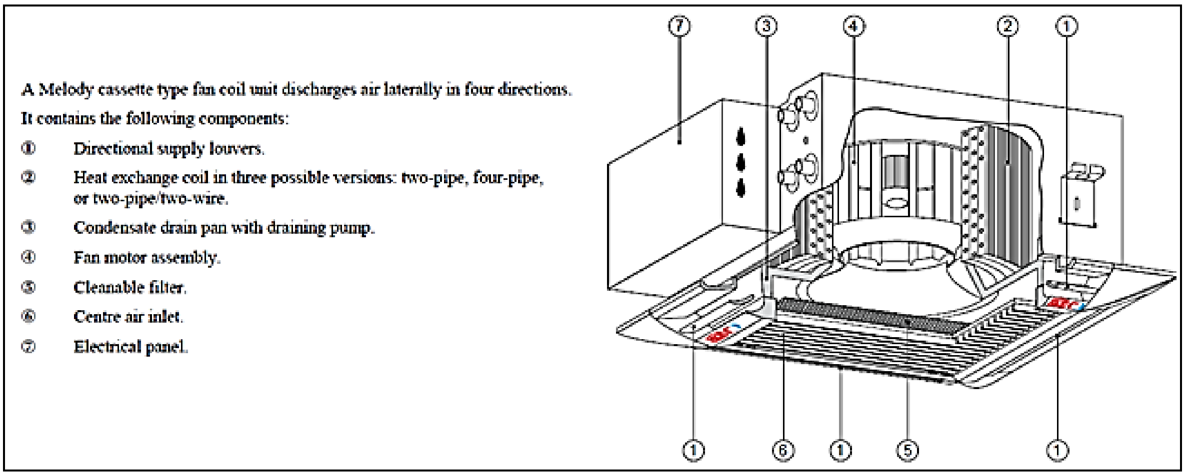


Fig (12). Melody cassette type fan coil

Field of application:

- ✓ Divided spaces (offices, small businesses in shopping centers)
- ✓ Large spaces as reception halls, airports and meeting rooms.

.1.2 Air handling unit (AHU)

- The air handling contains the following components as shown in figure (13).

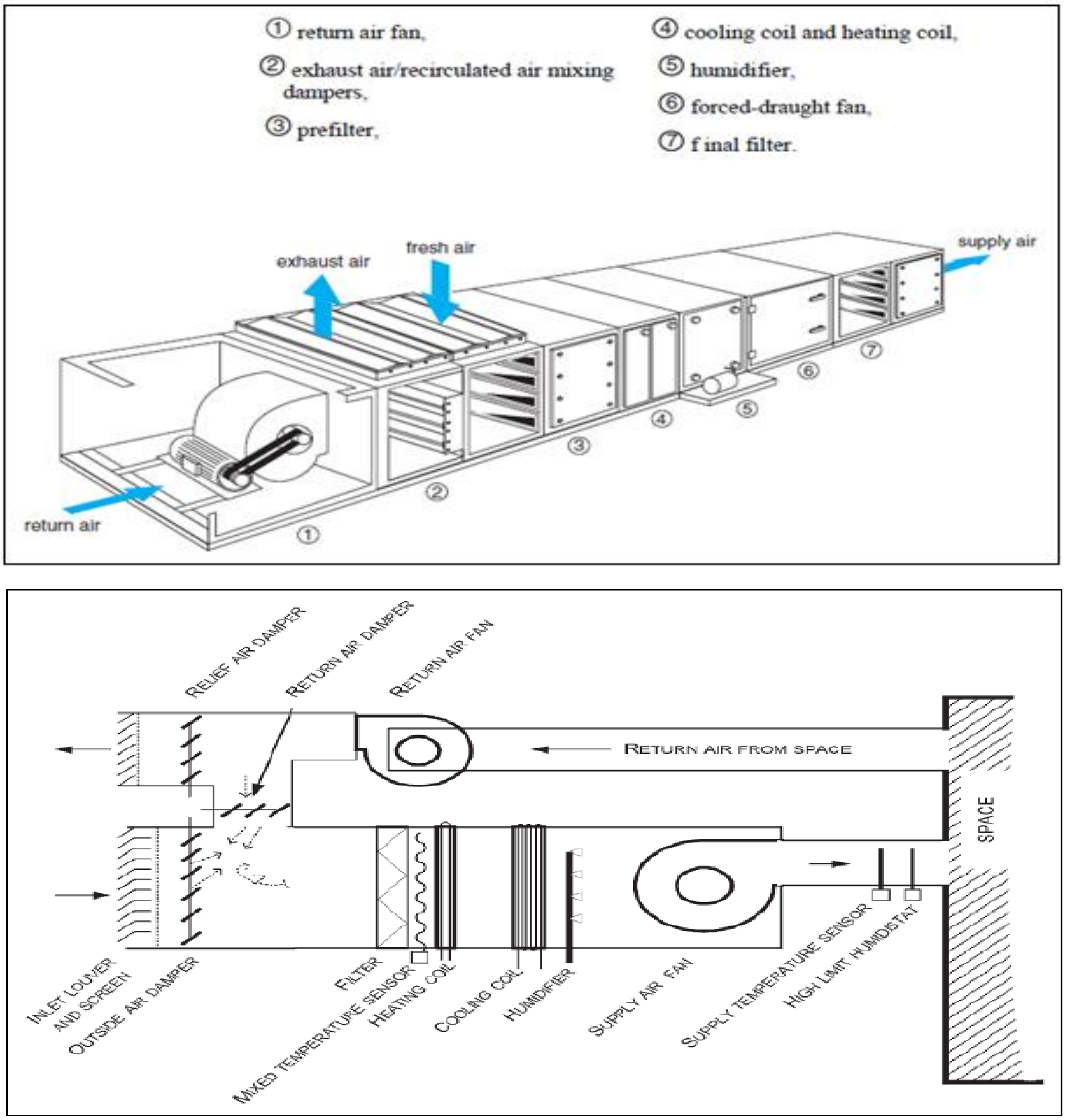


Fig (13) .AHU components.

- **Air Inlet and Mixing Section**

  - ✓ The inlet louver and screen restrict entry into the system. The inlet louver is designed to minimize the entry of rain and snow. A very simple design for the inlet louver is shown in the diagram. Maintaining slow air-speed through the louver avoids drawing rain into the system. A parallel blade damper is shown for both the outside air damper and the relief air damper.

//////

  - ✓ These dampers direct the air streams toward each other, causing turbulence and mixing. Mixing the air streams is extremely important in very cold climates, since the outside air could freeze coils that contain water as the heating medium It is also possible to install opposed blade dampers:

/\/\/\

  - ✓ These do a better job of accurately controlling the flow, but a rather poorer job of promoting mixing. Some air will be exhausted directly to the outside from washrooms and other specific sources. The remainder will be drawn back through the return air duct by the return air fan and either used as return air, or exhausted to outside through the relief air damper. This exhausted air is called the **relief air**. It is common, therefore, to link the outside-air damper, the return-air damper and the relief air dampers and use a single device, called an **actuator**, to move the dampers in unison.

- **Mixed Temperature Sensor.**

  - ✓ Generally, the control system needs to know the temperature of the mixed air for temperature control. A mixed-temperature sensor can be strung across the air stream to obtain an average temperature. If mixing is poor, then the average temperature will be incorrect. To maximize mixing before the temperature is measured, the mixed temperature sensor is usually installed downstream of the filter.

  - ✓ When the plant starts up, the return air flows through the return damper and over the mixed temperature sensor. Because there is no outside air in the flow, the mixed-air temperature is equal to the return-air temperature. The dampers open and outside air is brought into the system, upstream of the mixed-air sensor. If the outside temperature is higher than the return temperature, as the proportion of outside air is increased, the mixed-air temperature will rise. Conversely, if it is cold outside, as the proportion of outside air is increased; the mixed-air temperature will drop. In this situation, it is common to set the control system to provide a mixed-air temperature somewhere between 55 and 60°F. The control system can simply adjust the position of the dampers to maintain the set mixed temperature.

- ✓ For example, consider a system with a required mixed temperature of 55°F and return temperature of 73°F. When the outside temperature is 55°F, 100% outside air will provide the required 55°F. When the outside air temperature is below 55°F, the required mixed temperature of 55°F can be achieved by mixing outside air and return air. As the outside temperature drops, the percentage required to maintain 55°F will decrease. If the return temperature is 73°F, at 37°F there will be 50% outside air, and at 1°F, 20% outside air.

- ✓ If the building's ventilation requirements are for a minimum of 20% outside air, then any outside temperature below 1°F will cause the mixed temperature to drop below 55°F. In this situation, the mixed air will be cooler than 55°F and will have to be heated to maintain 55°F.The mixed-air temperature-sensor will register a temperature below 55°F. The heating coil will then turn "on" to provide enough heat to raise the supply-air temperature to 55°F.Now let us consider what happens when the outside-air temperature rises above 55°F. Up to 73°F, the temperature of the outside air will be lower than the return air, so it would seem best to use 100% outside air until the outside temperature reaches 73°F. In practice, this is not always true, because the moisture content of the outside air will influence the decision.

- **Filter:** All packaged units include at least minimal filters. Often it is beneficial to specify better filters.

- **Heating Coil:** Some systems require very high proportions, or even 100% outside air. In most climates this will necessitate installing a heating coil to raise the mixed air temperature. The heat for the heating coil can be provided by electricity, gas, water or steam.

- **The electric coil** is the simplest choice, but the cost of electricity often makes it an uneconomic one.

- **A gas-fired heater: often** has the advantage of lower fuel cost, but control can be an issue. Inexpensive gas heaters are "on-off" or "high-low-off" rather than fully modulating. As a result, the output temperature has step changes.

- **Hot water coils:** are the most controllable, but there is a possibility that they will freeze in cold weather. If below-freezing temperatures are common, then it is wise to take precautions against coil freezing.

- **Cooling Coil:** Cooling is usually achieved with a coil cooled by cold water, or a refrigerant, the cold water is normally between 42°F and 48°F. There are numerous refrigerants that can be used, whether using chilled water or a refrigerant, the coil will normally be cooler than the dew point of the air and thus condensation will occur on the coil. This condensation will run down the coil fins to drain away.

- **Humidifier:** A humidifier is a device for adding moisture to the air. The humidifier can either inject a **water-spray** or **steam** into the air. **The water-spray** consists of very fine droplets, which evaporate into the air; the supply of water must be from a **potable** source, fit for human consumption. The alternative is to inject **Steam** into the air stream. Again, the steam must be potable. The humidifier will normally be controlled by a humidistat, which is mounted in the space or in the return airflow from the space.

- **The unit control logic will then be:**

  ✓ Humidifier off when unit off.
  ✓ Humidifier off when cooling in operation.
  ✓ Humidifier controlled by space humidistat when unit in operation.
  ✓ Humidifier to shut down until manually reset if high limit humidity sensor Operates.

- **Fan** The fan provides the energy to drive the air through the system. There are two basic types of fan, the **centrifugal**, and the **axial**.

  ✓ **A- Centrifugal fan**: air enters a cylindrical set of rotating blades and is centrifuged, thrust radially outwards, into a scroll casing. This fan is a very popular choice due to its ability to generate substantial pressure without excessive noise.

  ✓ **B -Axial fan**: where the air passes through a rotating set of blades, like an aircraft propeller, which pushes the air along. This is a simpler, straight-through design that works really well in situations that require high volumes at a low pressure-drop.

  ✓ **C- Return Fan:** A return fan is usually included on larger systems, unless there is some other exhaust system to control building pressure. If there is no return fan, the building will have a pressure that is a bit above ambient (outside).

- Air handling unit can be integrated with heat recovery unit installed on the fresh air/exhaust air circuit as shown in figure (14), will illustrate it @next course level.

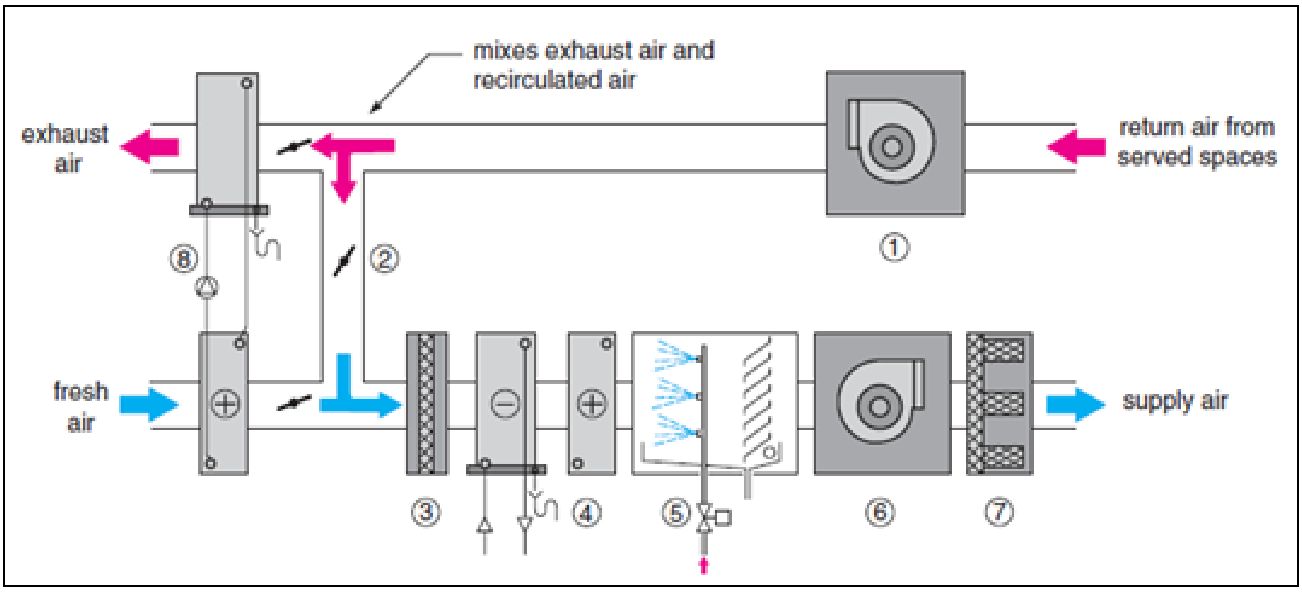


Fig (14). Heat recovery AHU.

.1.2.1 AHU can be classification

- **Local air handling unit.**

a- Figure (15) has been Indicated each space is supplied with conditioned air by it is own specific unit with the following features.

- ✓ Constant air flow.
- ✓ Fresh air (suction) is usually condition in the AHU which also extracts the stale air.
- ✓ The components (filter, cooling, heating coils, and humidifier) are determined based on the requirements of each space.
- ✓ Heating and refrigeration equipment is with unit in the mechanical room or outside.

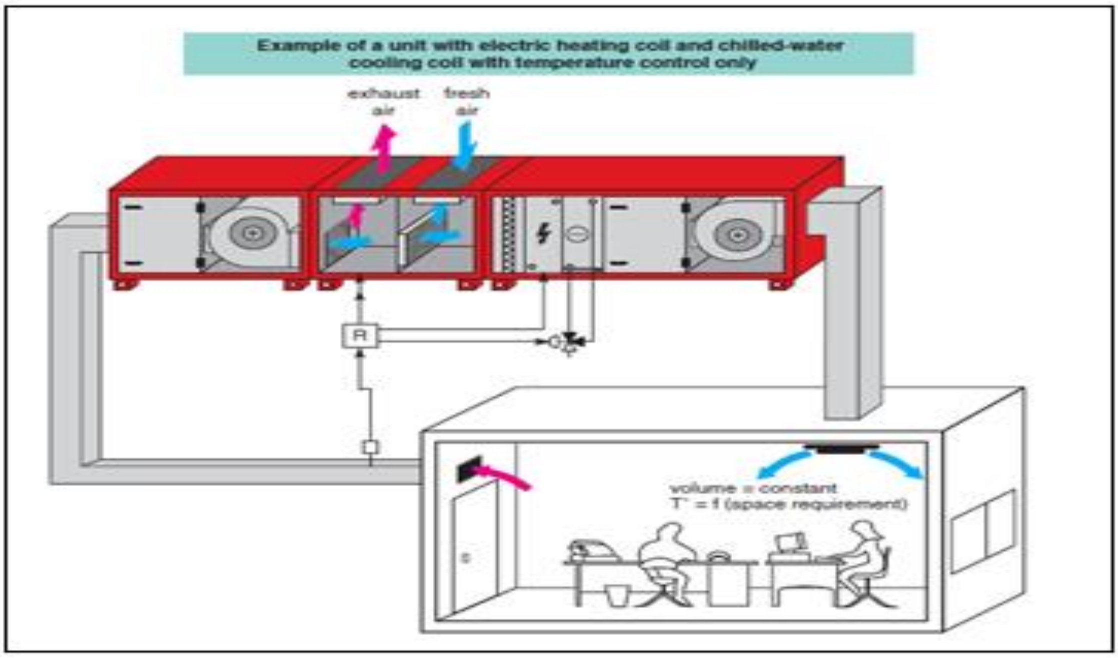


Fig (15). Local air handling unit.

b- Operation

- ✓ The unit adjusts the condition base on space loads.
- ✓ Regulates temp (summer, and winter).
- ✓ Regulates humidity (winter).
- ✓ In some case dehumidifies the air (summer).
- ✓ Regulations sequence occurs as shown in the following table. Figure (19) below shows an installed of single zone air handling unit control.

Sensors	Effect on
Temperature	cooling coil (chilled water) (1) heating coil (hot water) (2)
Relative humidity	cooling coil (chilled water) (1) humidifier (3)
Occupancy	fresh air damper
Outdoor temperature	free cooling (energy savings)

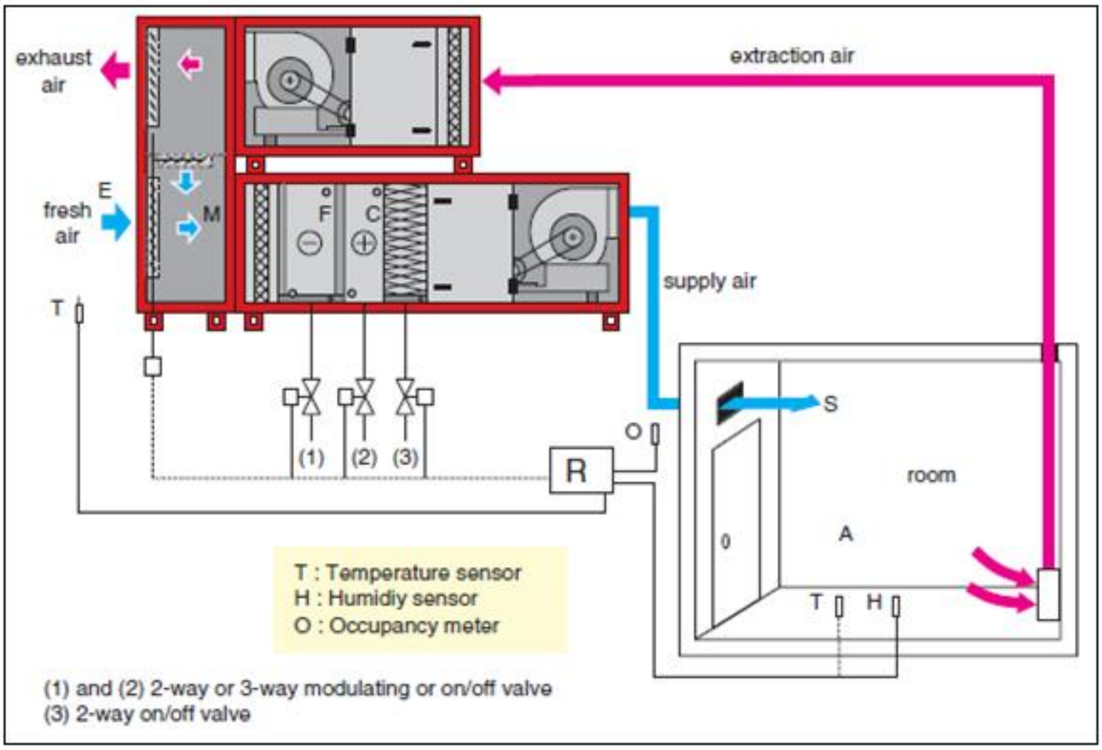


Fig (16). Local AHU control.

c- Mollier chart: The changes in temp and humidity are plotted in the following manner on mollier chart.

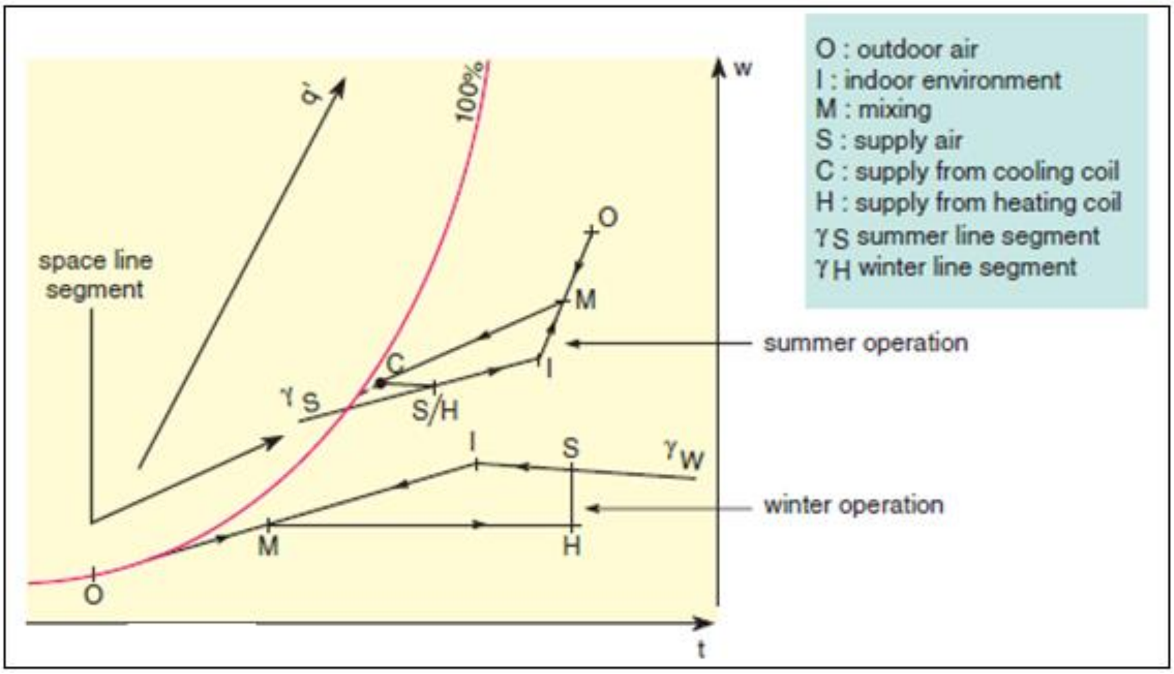


Fig (17).local AHU operation in mollier chart.

Field of application. This type of systems used for:

- ✓ Large volume spaces.
- ✓ Meeting room, theatres concert, and halls
- ✓ Spaces with heavy load conditions that vary.
- ✓ Low tolerance environmental controls.

- **Single zone air handling unit**

a- concept.

- ✓ Air supplied to several spaces by single air handling unit.
- ✓ The air flow for each individual space is constant and calculated based on it is max heat load.
- ✓ Fresh air is generally conditioned in the AHU before being supplied to all the spaces.
- ✓ The components (filter, cooling, heating coils, and humidifier) are determined based on the requirements of each space.
- ✓ Heating and refrigeration equipment is with unit in the mechanical room or outside.
- ✓ Detection devices (temp and relative humidity sensors) are arranged in the control room.

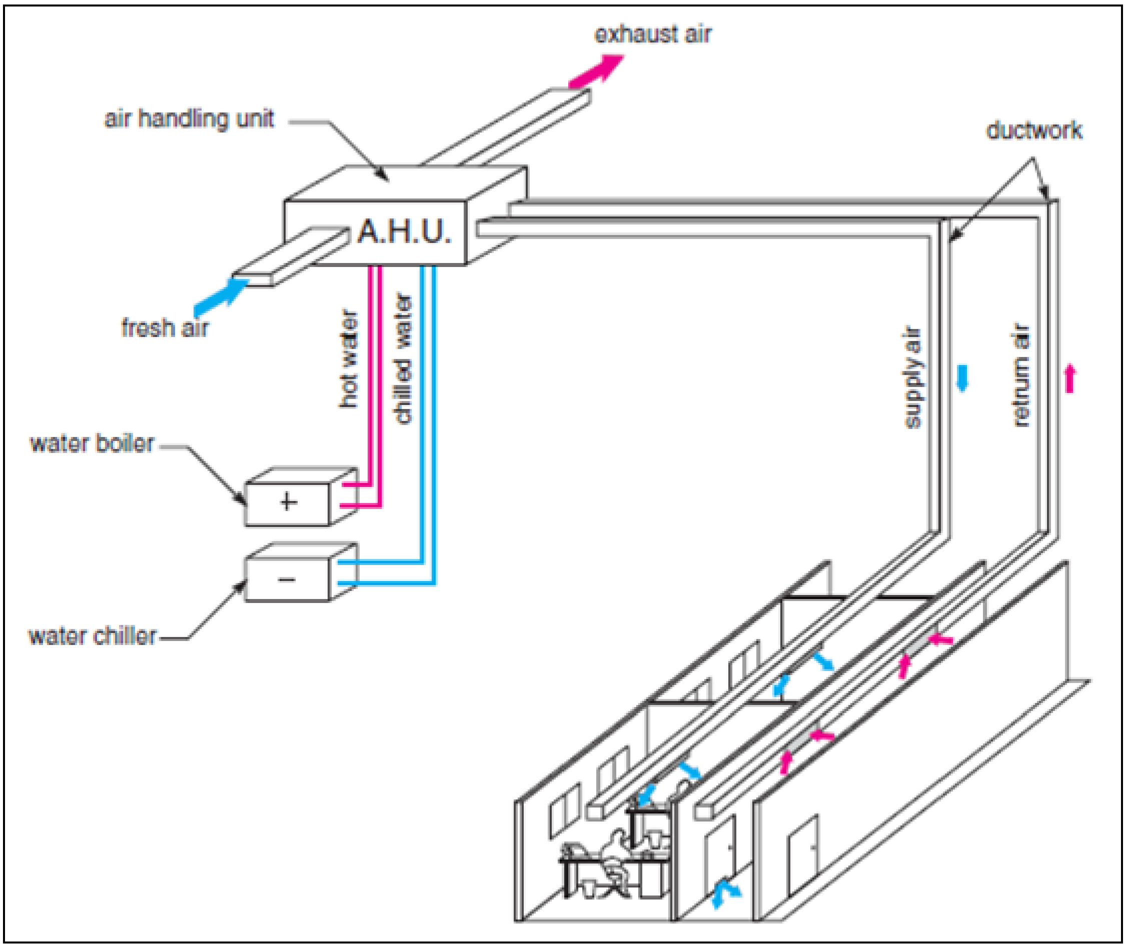


Fig (18).single zone AHU.

- Operation.

  - ✓ This system is used for spaces where the following loads are identical.
  - ✓ Distribution of sensible, and latent heat Changes and variation in the same direction and proportions.
  - ✓ Air properties supplied to each space are the same.
  - ✓ Fresh air is distributed based on the total load not on the density of occupants in the space.
  - ✓ The control devices and temp, and humidity sensors may be installed.

In a control room (e.g. A_1)	The other spaces (A_2, A_3 etc.) are governed by the A_1 law.
In the main return air section	All the spaces (A_1, A_2, A_3 etc.) are governed by the law of averages.

  - ✓ Space conditions cannot be adjusted to personal preference.
  - ✓ As a matter of fact spaces rarely experience the same changes in heat .deviations occur in environmental parameters on account of the fact that the air supply conditions are the same for all spaces .the system would not be an adequate choice for precision air conditioning .figure (19) below shows an installed of single zone air handling unit control.

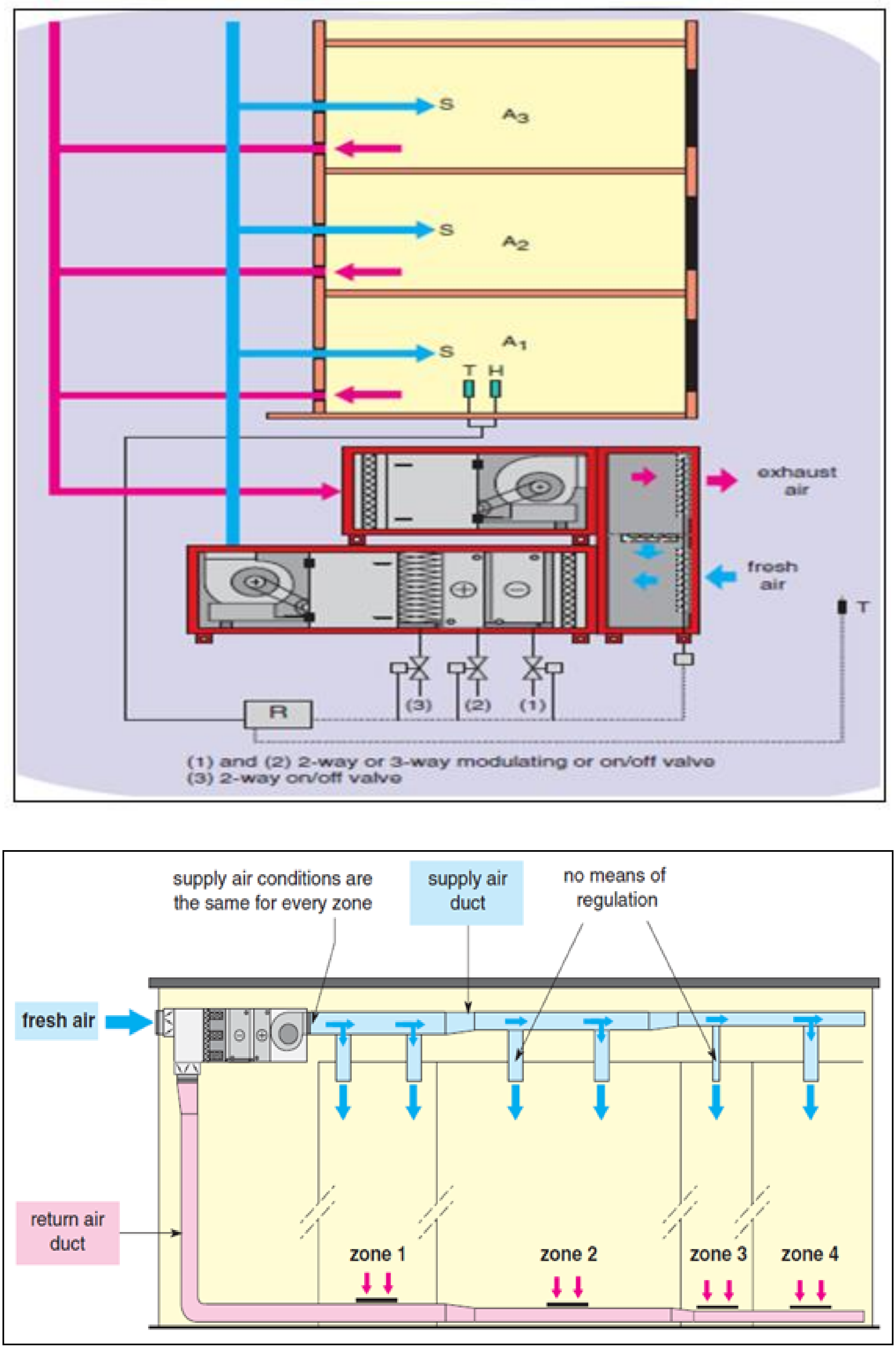


Fig (19).single zone AHU control.

- Mollier chart:

Changes in the air supplied to the space are plotted on the mollier chart. This diagram does not show the various handling /conditioning processes the air under goes in the unit .it shows the changes in the air in each space based on sensible and latent heat. A is space set point when the same air S is distributed to all spaces it is clear that the space conditions cannot be controlled when their loads are different. The segment line is rarely completely identical all pass by the same supply air point the indoor environmental should have the value but as the loads are not really identical the value of A controlled only for the space where the sensors are installed.

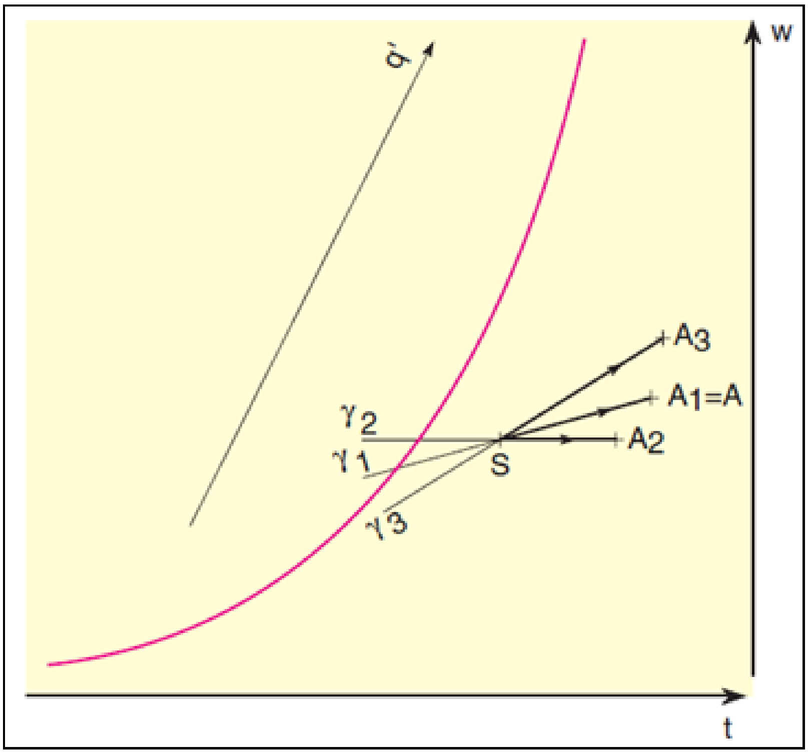


Fig (20).single zone AHU operation in mollier chart.

- Field of application

  - ✓ Because it is economic the system may be used for spaces with identical thermal loads and very in the same direction and proportions.

- AHU can also classify into dual duct air handling unit, Multi zone unit, and AHU with variable volume diffusion boxes for each space this types will in next course.

Chillers.

- The vapor-compression and absorption refrigeration cycles are the two most common cycles used in commercial air conditioning as shown in figure (21). At this part we will concern with chiller using vapor-compression.

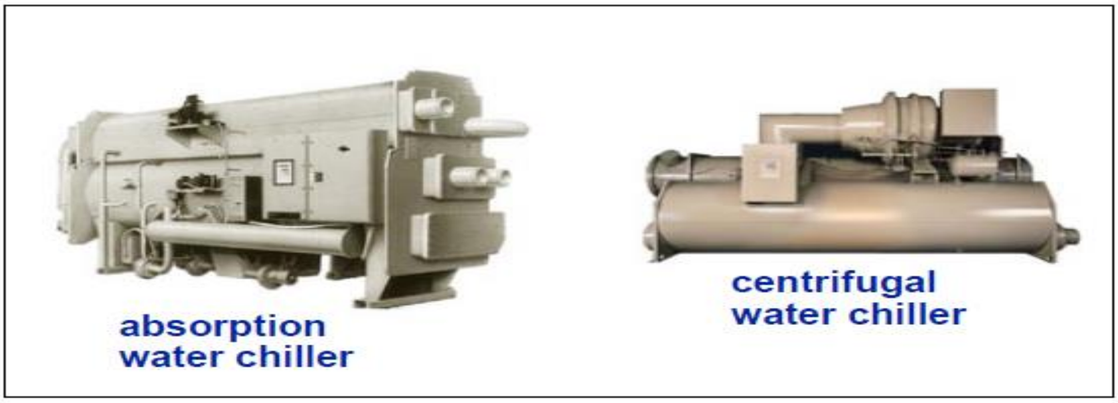


Fig (21).chiller types.

- Water chillers using the vapor-compression refrigeration cycle components as shown in the following figure.

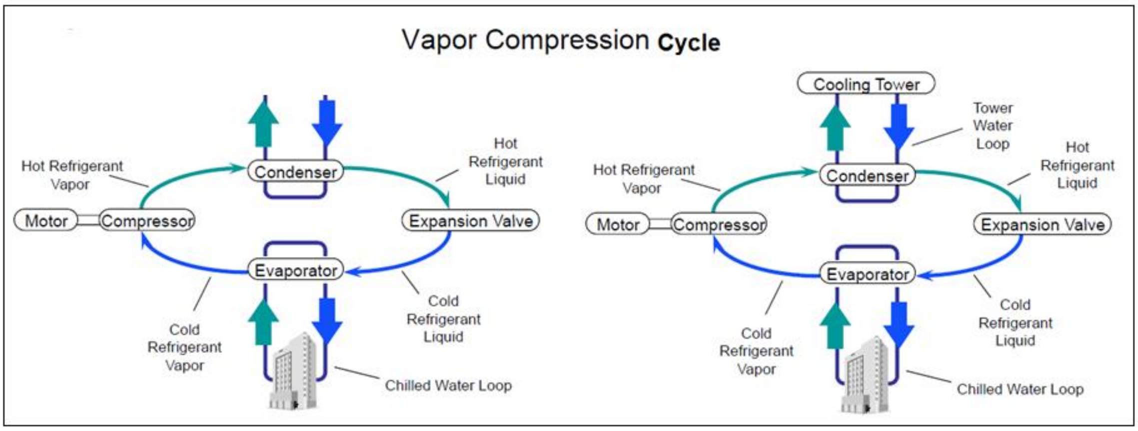
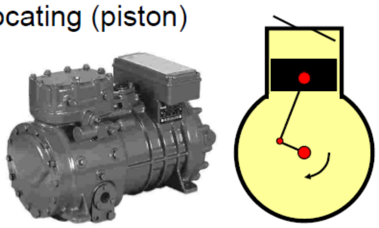


Fig (22) .vapor compression refrigeration cycle.

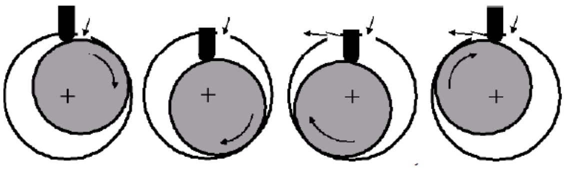
.1.3 **Water chillers using the vapor-compression refrigeration cycle** vary by the type of compressor used.

- Reciprocating, scroll, helical-rotary, and centrifugal compressors are common types of compressors used in vapor-compression water chillers as shown in figure (23).

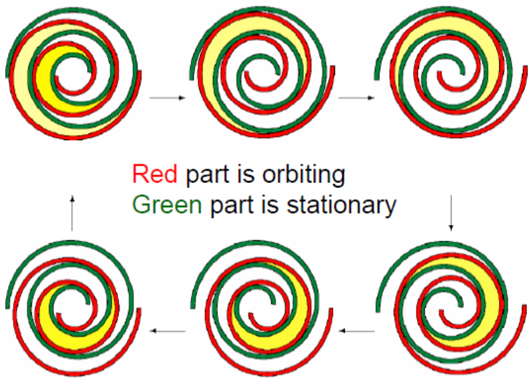
- Reciprocating (piston)



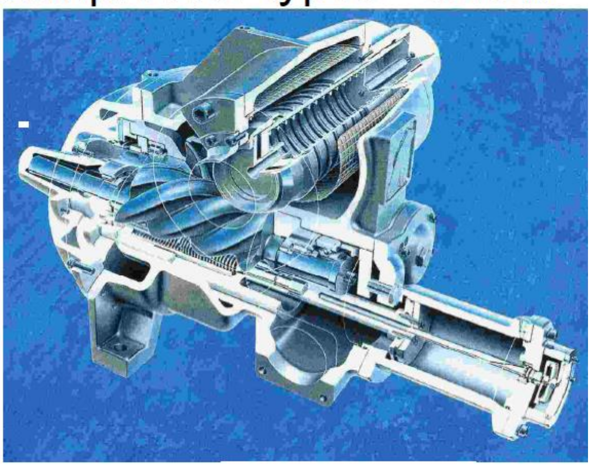
- Rotary



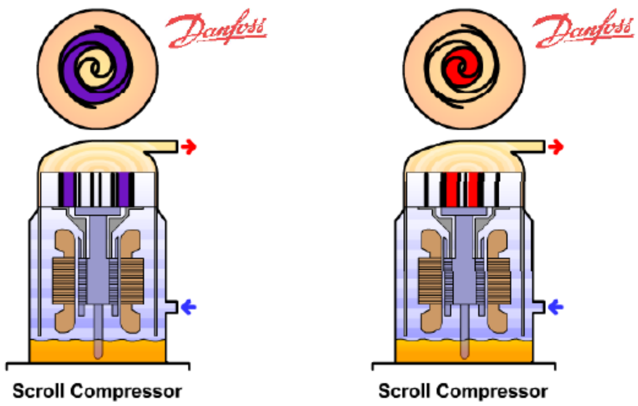
Scroll compressor



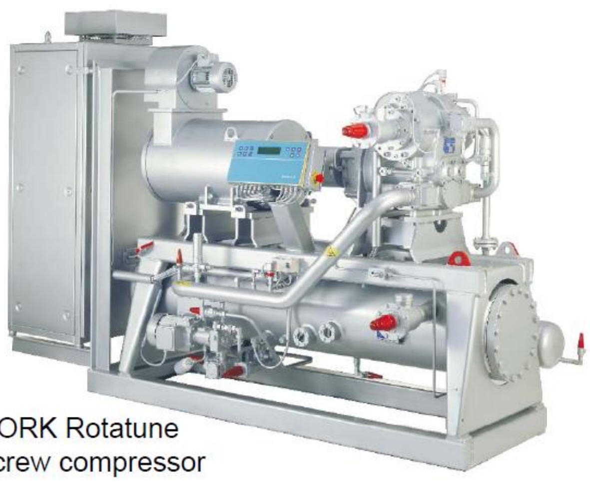
Compressor types - screw



Scroll compressor

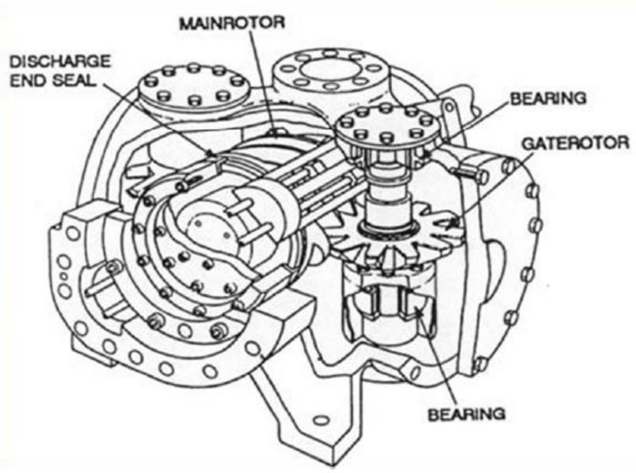


Screw compressor

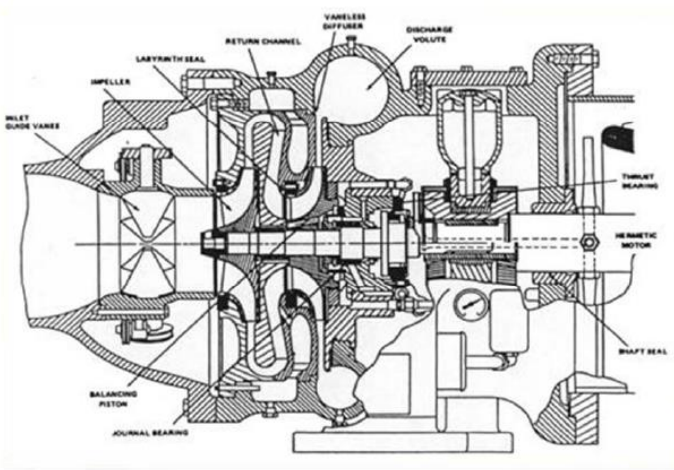


YORK Rotatune
screw compressor

Screw Compressor



Centrifugal Compressor



Compressor types - centrifugal

Also called
turbo compressor

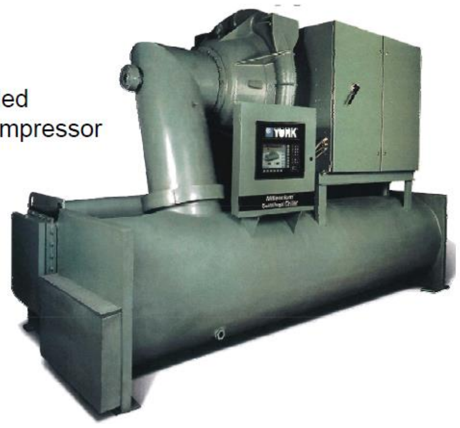


Fig (23).compressor types.

- Compressor according to maintenance operation can be divided into.

Open compressor	semi hermetic compressor	hermetic compressor
Reciprocating	Reciprocating	Reciprocating
Centrifugal	Centrifugal	Centrifugal
screw	screw	screw
		Scroll
		Rotary

Open
- Motor outside shell
- Shell can be opened

Semi-hermetic
- Motor inside shell
- Shell can be opened

Hermetic
- Motor inside shell
- Shell can't be opened

Farouk E

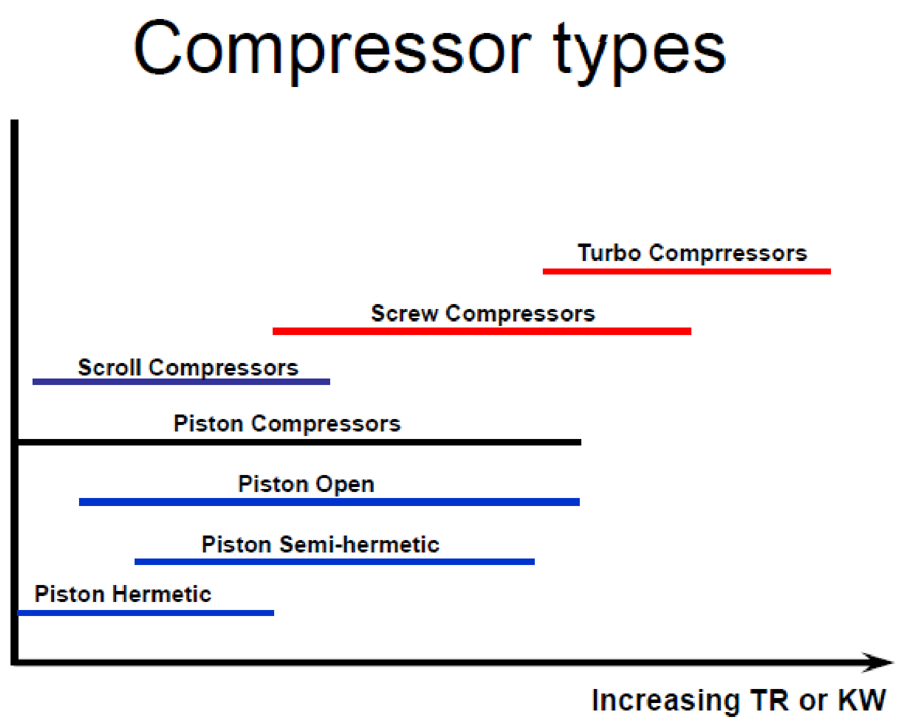
- Available chiller capacities with different compressor type

Type of chiller	Nominal capacity range (kw)	Refrigerant used in systems	Range in full load efficiency(kw/ton)
Reciprocating	50 To 1750	R22	0.8 TO 1.00
screw	160 to 2350	R 134a OR R-22	0.6 & 0.75
scroll	30 to 200	R-22	0.81&0.92
centrifugal	500 to 18,800	R134a or R-123	0.5 to 0.7

- Available chiller capacities:

  - ✓ Reciprocating &scroll → up to 25tons.
  - ✓ Screw & reciprocating →25 to 200 tons.
  - ✓ Screw or centrifugal →200 to 800 tons.
  - ✓ Centrifugal → above 800 tons.

- Compressor type vs.TR.



2-2-2 Chillers classified according to medium used to cooling the refrigerant in the condenser:

2.2.2.1 Air cooled chiller.

- Many small to medium chiller plants use air cooled chillers with air-cooled screw chillers being common in the 150 to 400- ton range.

- Air cooled chiller offer substantial capital savings which with using air cooled chiller avoid the need for cooling towers, condenser pumps and condenser piping, and do not require mechanical room space also an air-cooled chiller avoids the need to operate a cooling tower in cold (freezing) weather.

- They do not consume water like water cooled chillers. A 400-ton chiller will consume over 700 gallons per hour to offset cooling tower makeup. Where water is scarce, this can be a significant cost. In addition, condenser water treatment is avoided.

- Air-cooled chillers have lower performance (consume more power) than water or evaporative Cooled chillers because of the increased lift. Refrigeration work is proportional to lift; doubling the lift will approximately double the work required. (For this purpose, consider lift to be the difference between chilled water supply and either cooling tower supply or ambient air dry bulb) Since air cooled chillers must raise the refrigerant temperature above ambient dry bulb, they consume more power. They are designed to work well around the ARI 550/560 design conditions (54°F EWT, 44°F EWT). The design temperature range should stay within 20% of these operating conditions.

- Air-cooled chiller is an excellent solution for applications that require a small amount of chilled water during the winter.

- The air-cooled chiller will offer equal performance to a water-cooled chiller at low ambient conditions.in winter operation there are two issues to deal with.

  - ✓ The first is the necessary changes to the chiller to operate in cold temperatures. The condensing fans are staged off, or slowed down to maintain the correct condensing temperature.
  - ✓ The second issue is protecting the chilled water from freezing. Here are some possible solutions as shown in figure (24).

- Heat traces the piping and evaporator.

- Add antifreeze. A common solution is to add either propylene or ethylene glycol to the chilled water. A loss of antifreeze in the system due to flushing or a leak and subsequent water make-up can allow the chilled water loop to become vulnerable to freezing.

- Relocate the evaporator barrel inside the building envelope. Relocating the evaporator avoids antifreeze but will require field refrigerant piping. There are also limitations on piping distances and elevation changes.

- Use an indoor chiller (condenser less chiller). The primary advantage of this configuration is that the compressors are located indoors, which makes maintenance easier during inclement weather and virtually eliminates the concern of refrigerant migrating to the compressors during cold weather.

- The final configuration includes a packaged compressor-and-evaporator unit that is located in an indoor equipment room and connected to an indoor, air-cooled condenser. The air used for condensing is ducted from outdoors, through the condenser coil, and rejected either outdoors or inside the building as a means for heat recovery. Indoor condensers typically use a centrifugal fan to overcome the duct static-pressure losses, rather than the propeller fans used in conventional outdoor air-cooled condensers. Again, the components are connected with field-installed refrigerant piping.

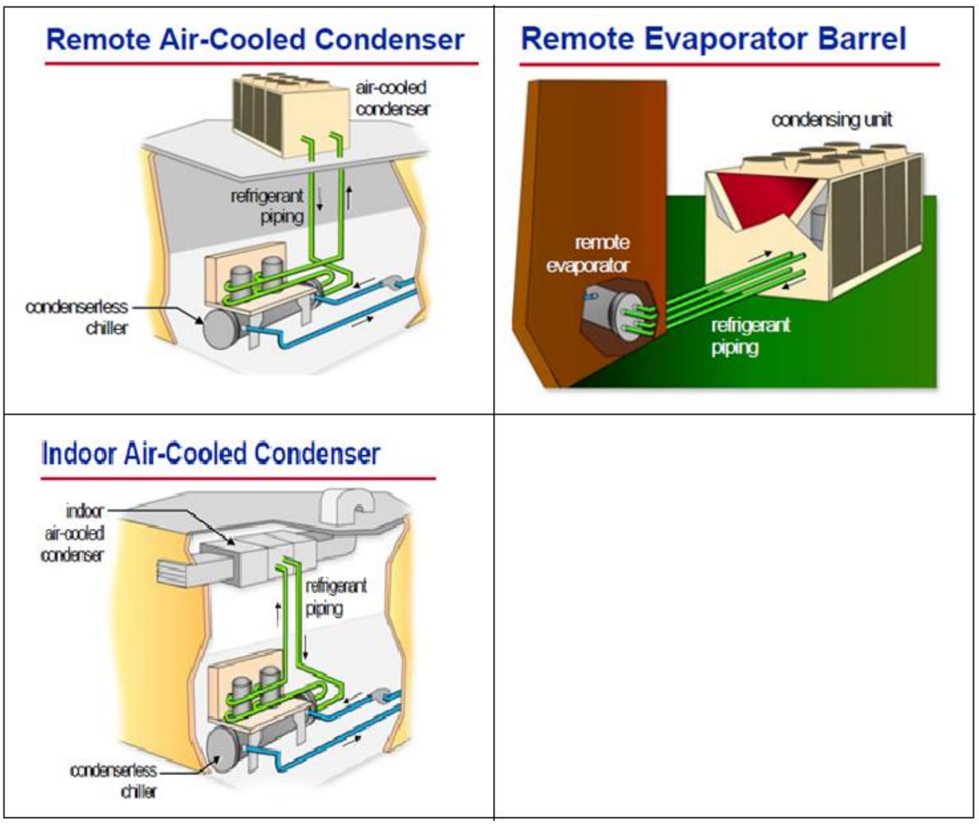


Fig (24). Anti-freezing solutions

2.2.2.2 Water or evaporative Cooled Chillers.

- This ties the condensing temperature to ambient wet bulb like a water-cooled chiller.
- Evaporative-cooled chillers are often associated with hot, dry climates.
- Whereas a water cooled chiller will require a cooling tower, condenser pump and field erected piping.

2.2.2.3 Air-Cooled or Water-Cooled Condensing comparison

- Configuration

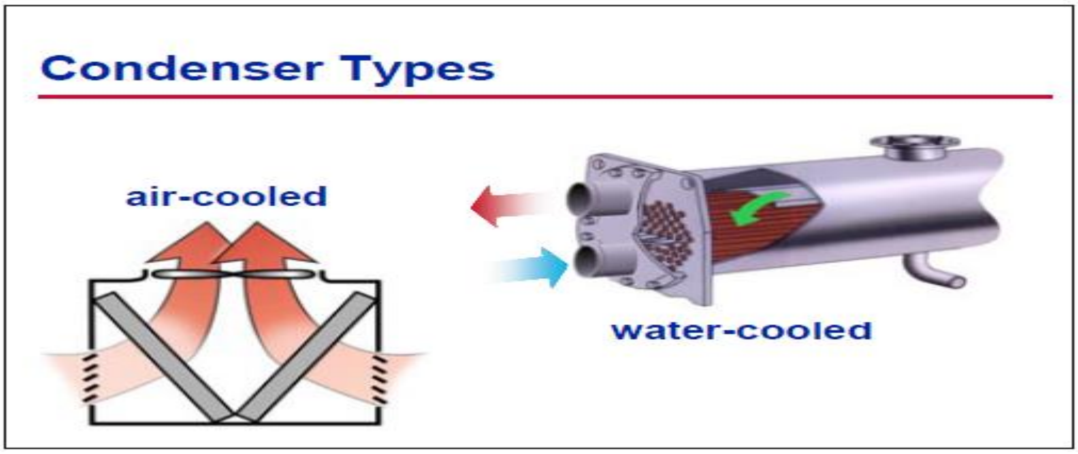


Fig (25). Condenser types

- available capacity:
  - ✓ Air-cooled chillers are typically available in ranging from 7.5 to 500 tons.
  - ✓ Water-cooled chillers are typically available from 10 to 3,000 tons.

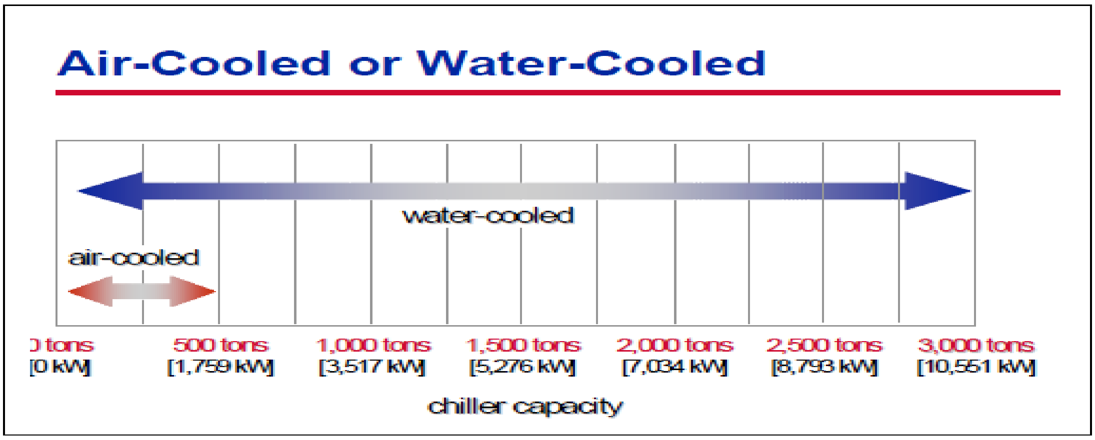


Fig (26).air and water cooled available capacity.

- Maintenance

  ✓ Air cooled chiller offer substantial capital savings due to eliminate of using cooling tower, condenser water pump, pump condenser and associated maintenance requirements.

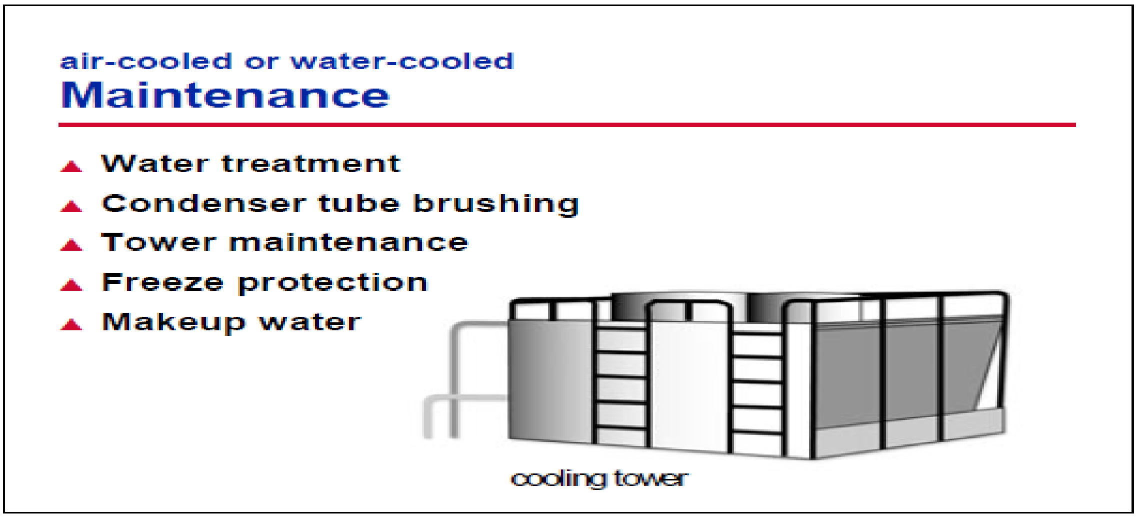


Fig (27). Air cooled or water cooled maintenance.

- For low ambient Operation

  ✓ Air-cooled condensers have the ability to operate in below-freezing weather, and can do so without the problems associated with operating the cooling tower in these conditions. Cooling towers may require special control sequences (see cooling tower section).

Fig (28). Air cooled or water cooled low ambient operation.

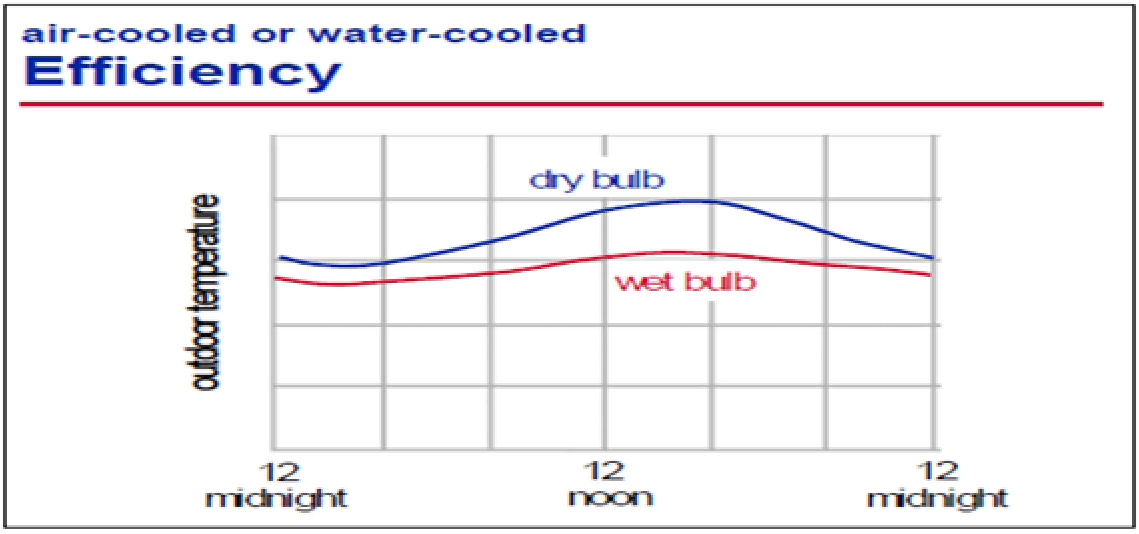
- Efficiency

  ✓ Water-cooled chillers are typically more energy efficient than air-cooled chillers. The refrigerant condensing temperature in an air-cooled chiller is dependent on the ambient dry-bulb temperature. The condensing temperature in a water-cooled chiller is dependent on the condenser-water temperature, which is dependent on the ambient wet-bulb temperature. Since the wet-bulb temperature is often significantly lower than the dry-bulb temperature, the refrigerant condensing temperature (and pressure) in a water-cooled chiller can be lower than in an air-cooled chiller. A lower condensing temperature, and therefore a lower condensing pressure, means that the compressor needs to do less work and consume less energy.

  ✓ This efficiency advantage may lessen at part-load conditions because the dry-bulb temperature tends to drop faster than the wet-bulb temperature. As a result, the air-cooled chiller may benefit from greater condenser relief.

  ✓ The efficiency advantage of a water-cooled chiller is much less when the additional cooling tower and condenser pump energy costs are considered.

- Water-cooled chillers typically last longer than air-cooled chillers. This difference is due to the fact that the air-cooled chiller is installed outdoors, whereas the water-cooled chiller is installed indoors. Also, using water as the condensing fluid allows the water-cooled chiller to operate at lower pressures than the air-cooled chiller. In general, air-cooled chillers last 15 to 20 years, while water-cooled chillers last 20 to 30 years.



Fig(29).air cooled or water cooled efficiency.

- **To summarize the comparison of air-cooled and water-cooled chillers**

- Include lower maintenance costs, a prepackaged system for easier design and installation, and better low-ambient operation. Water-cooled chiller advantages include greater energy efficiency (at least at design conditions) and longer equipment life.

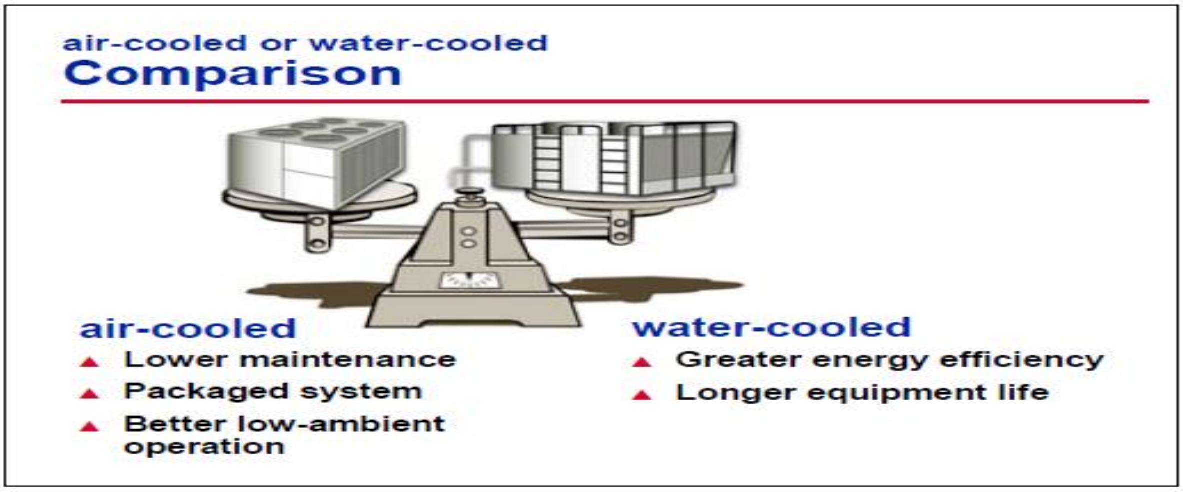


Fig (30). Air cooled or water cooled comparison.

2.2.2.4 Chiller arrangements:

- Single-Chiller System.
  - ✓ Single chillers are sometimes used in small systems (less than100 tons [35 kW]), while larger or critical systems typically use multiple chillers.
  - ✓ This system uses a single pump to move water through the chiller and load terminals. The load terminals are controlled using three-way modulating valves. The pump delivers a constant flow of water throughout the entire system. Figure (31) shows single chiller piping system.

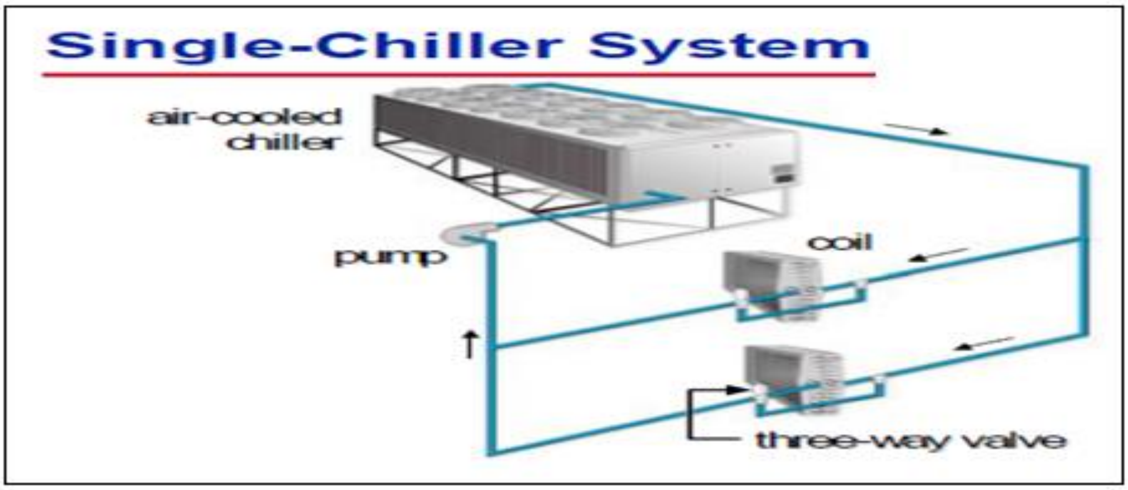


Fig (31).single chiller system

- **Multiple-Chiller Systems:** multiple-chiller systems are more common than single-chiller systems for the Redundancy, and Part-load efficiency. There are several configurations used to connect multiple chillers in these Systems.

- Chillers piped in parallel, and Single Pump.

  - ✓ Uses a single pump to deliver chilled water both to chillers and to the system load terminals.

  - ✓ This configuration can be used in systems that use constant-flow methods of terminal control (three-way valves or face-and-bypass dampers), or in systems that use variable-flow methods of terminal control (two-way valves).

  - ✓ at partial load one of chiller cannot be operated this causing increasing value of supply temp due to mixing between water supplied from operating and non-operating chiller may result in problems with building comfort or humidity control A chiller-plant controller may be used to reset the set point of The operating chiller down ward, in an attempt To compensate for this condition.

  - ✓ Reducing the set point of the operating chiller has its limits, however, depending on the operating characteristics and evaporator freeze limits of the specific chiller.

  - ✓ Additionally, ASHRAE/IESNA Standard 90.1–1999 (Section 6.3.4.2) prohibits this type of system when the pump is larger than 10 hp [7.5 kW]. The standard requires that, in systems that contain more than one chiller piped in parallel, system water flow must be reduced when a chiller is not operating. Figure (32) shows chillers piped in parallel and single pump.

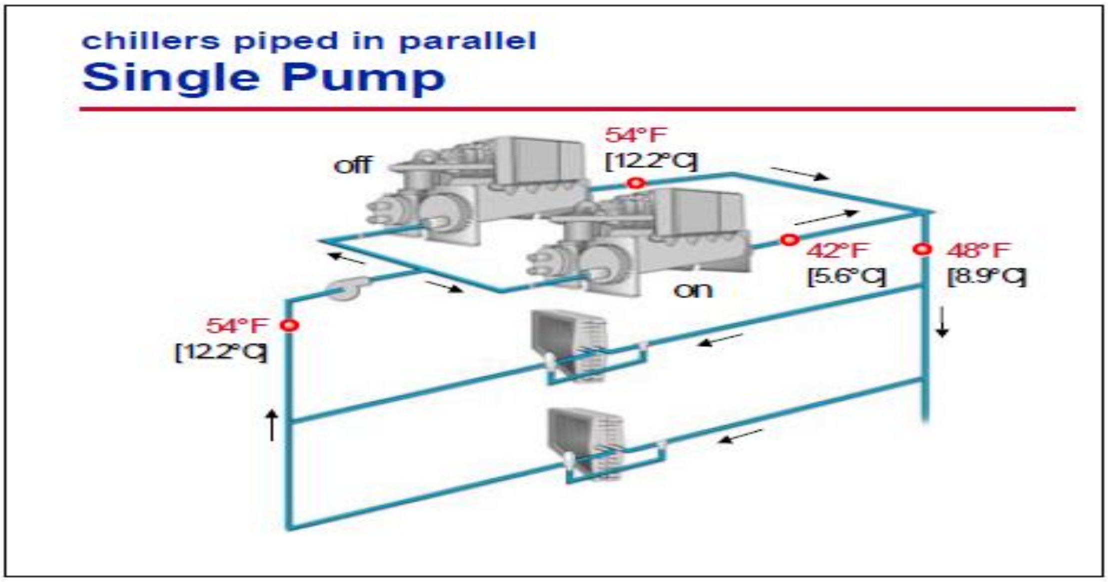


Fig (32).chillers piped in parallel and single pump.

- **chillers piped in parallel, and dedicated Pumps**

  ✓ This solves the temperature mixing problem that occurred in the previous, single-pump configuration, but it presents a new problem in a system that uses a constant flow method of terminal control.

  ✓ Below 50-percent load, only one chiller and one pump are operating. The total water flow in the system decreases significantly, typically 60 to 70 percent of full system flow.

  ✓ This configuration also presents problems to chiller operation. The starting or stopping of a pump for one chiller affects the flow through the other chiller.

  ✓ If one chiller is operating and a second chiller and pump are started, the total water flow in the system does not double. The system and pump performance curves will "rebalance," resulting in an increase in system flow of only 35 percent of total flow. The new total flow rate, however, is now divided equally between the two chillers. This results in a rapid reduction in water flow through the original operating chiller, from 65 percent of total system flow to 50 percent. This rapid decrease in flow often results in a loss of temperature control and may cause the chiller to shut off on a safety.

  ✓ In order to overcome this problem, the chiller-plant control system should anticipate the starting of additional pumps and unload operating chillers prior to the start of an additional chiller. Again, this configuration is sometimes acceptable for two-chiller systems, but is not often used in larger systems because the part-load system flow problems are further multiplied. Figure (33) Chillers piped in parallel, and dedicated Pumps.

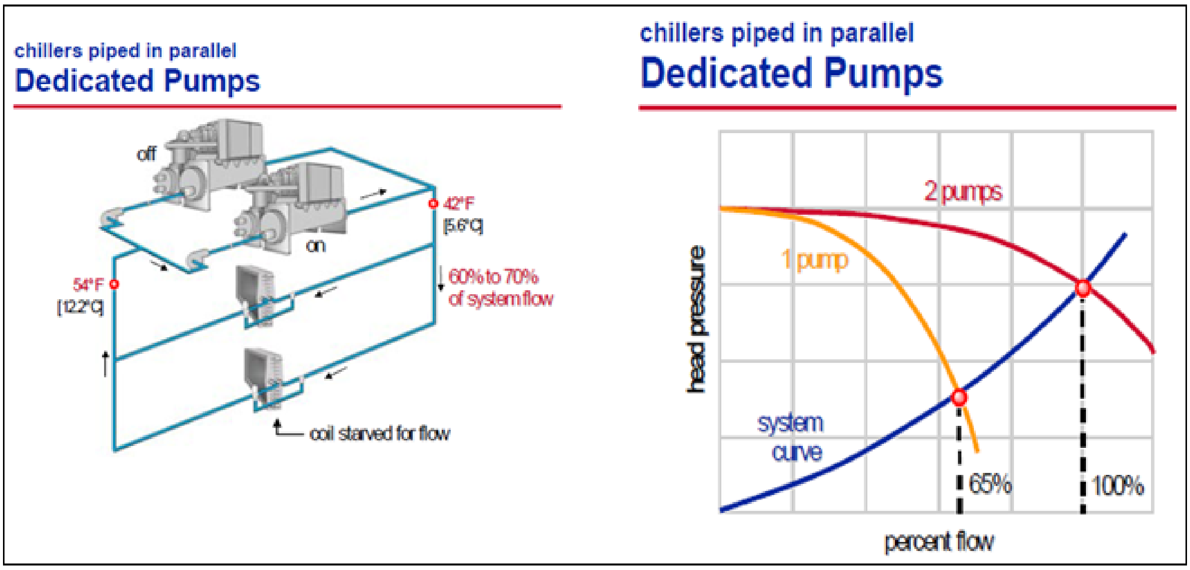


Fig (33). Chillers piped in parallel, and dedicated Pumps.

- **Chillers Piped in Series.**

✓ Used for Large temp drop (greater than 18F) is desiredthe chiller-tube pass arrangement must accommodate double the water quantity within acceptable velocity and pressure drop limits. This typically requires a reduced number of passes in the evaporator and may impact chiller Efficiency.

✓ System pressure drop also increases because the pressure drops through the chillers are additive. This can result in increased pump size and energy costs.

✓ Because of the pressure drop limitations, it is difficult to apply more than two chillers in series. Systems involving three or more chillers typically use either the primary-secondary configuration or parallel sets of two chillers in series. Figure (34) shows chillers piped in series.

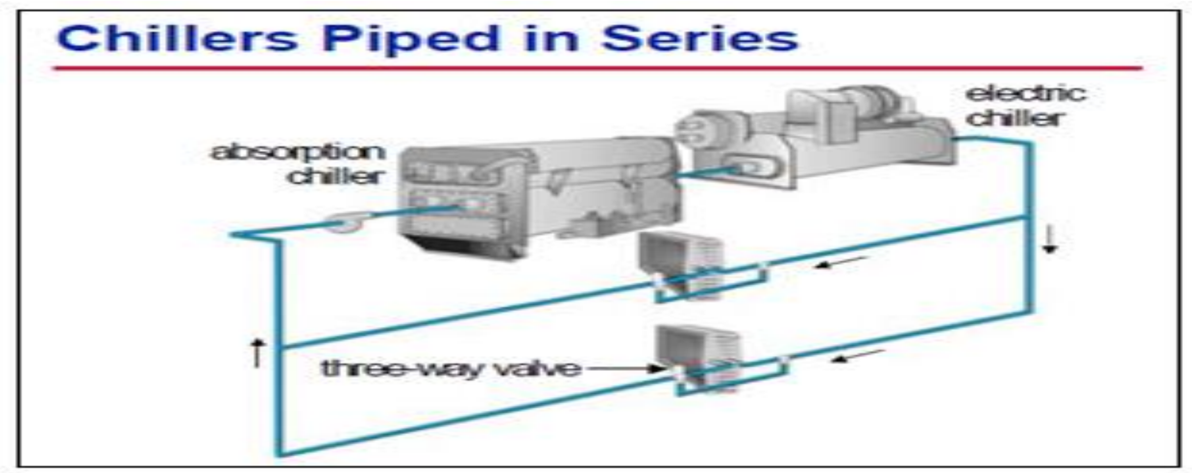


Fig (34). Chillers Piped in Series.

✓ Temperature control in a series system as shown in figure (34) can be accomplished by upstream chiller is Operated at full capacity and any portion of the load that remains is handled by the downstream chiller. The set point for the downstream chiller is set equal to the desired system supply temperature, and the set point for the upstream chiller is then dynamically reset to maintain equal loading on both chillers.

✓ Other way for Temperature control in a series system can be accomplished by an alternative method of controlling chillers in series involves staggering the set points of the two chillers. These results in the *downstream* chiller operating first and being preferentially loaded. Any portion of the load that the downstream chiller cannot meet is handled by the upstream chiller.

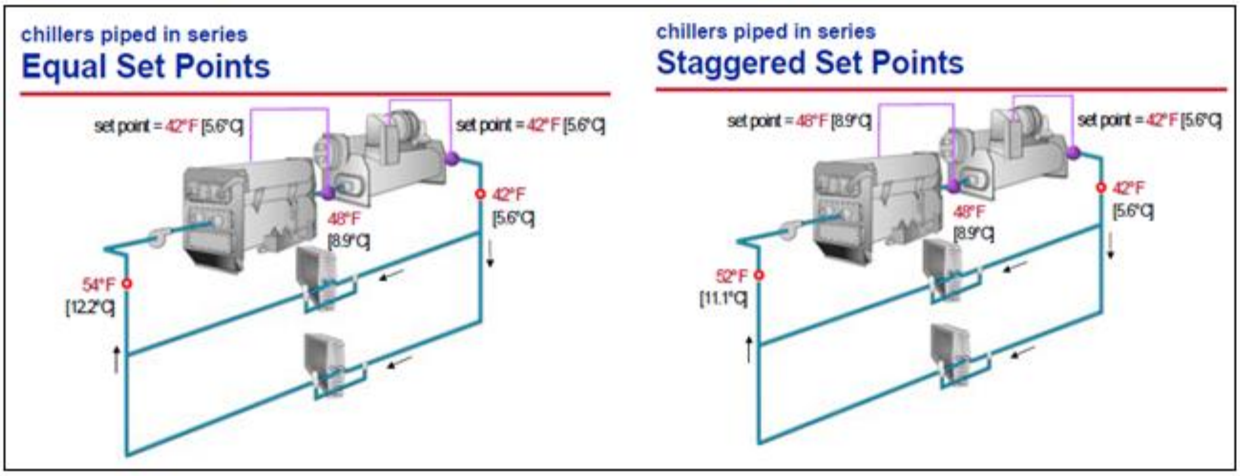


Fig (34). Temperature control in a series system.

- **Mani folded Production Pumps**: Alternatively, the production loop can be configured with Mani folded pumps and automatic, two-position isolation valves at each chiller. When turning on a chiller, a pump is turned on and the isolation valve is opened. Figure (35) shows manifolded production pumps.

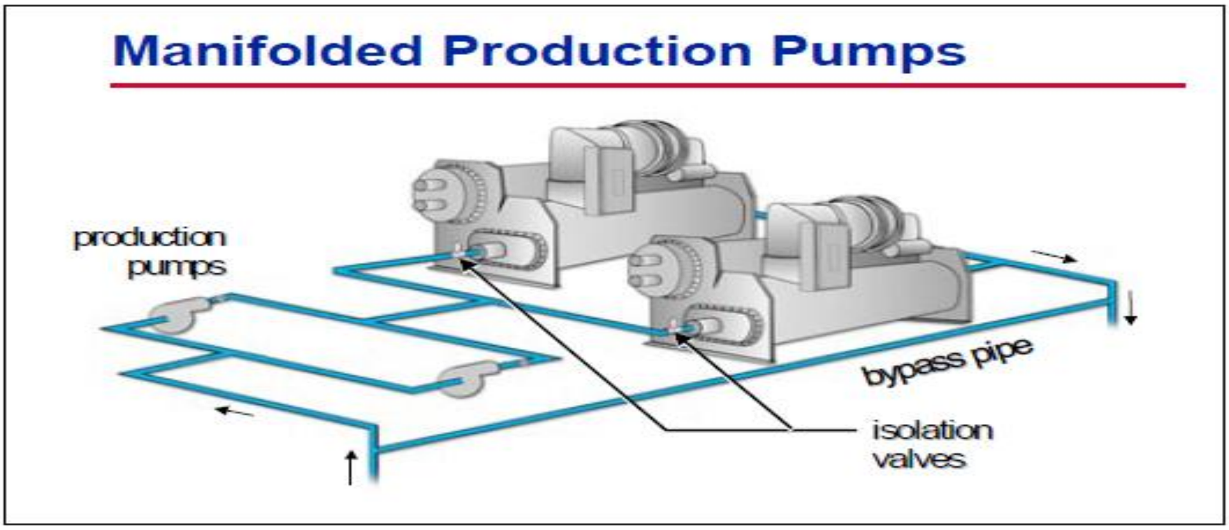


Fig (35). Shows manifolded production pumps.

2.3 Cooling tower. (See chapter 5).

2.3.1 Chiller and cooling tower connection.

- Single water cooled chiller loop as shown in figure (36).

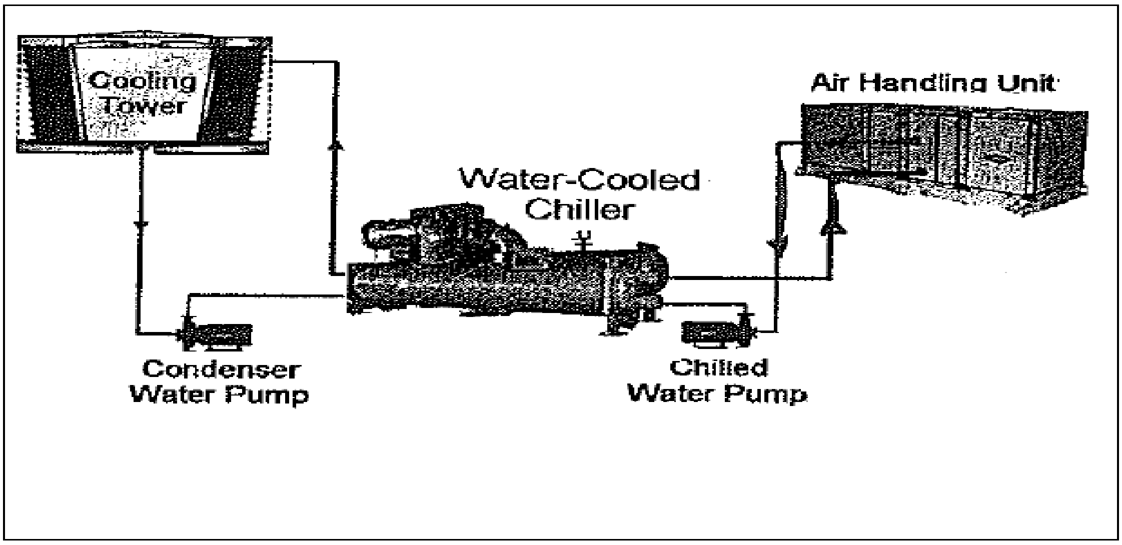


Fig (36).single water cooled chiller loop

- Multiple water cooled chiller loop with dedicated pumps as shown in figure (37).

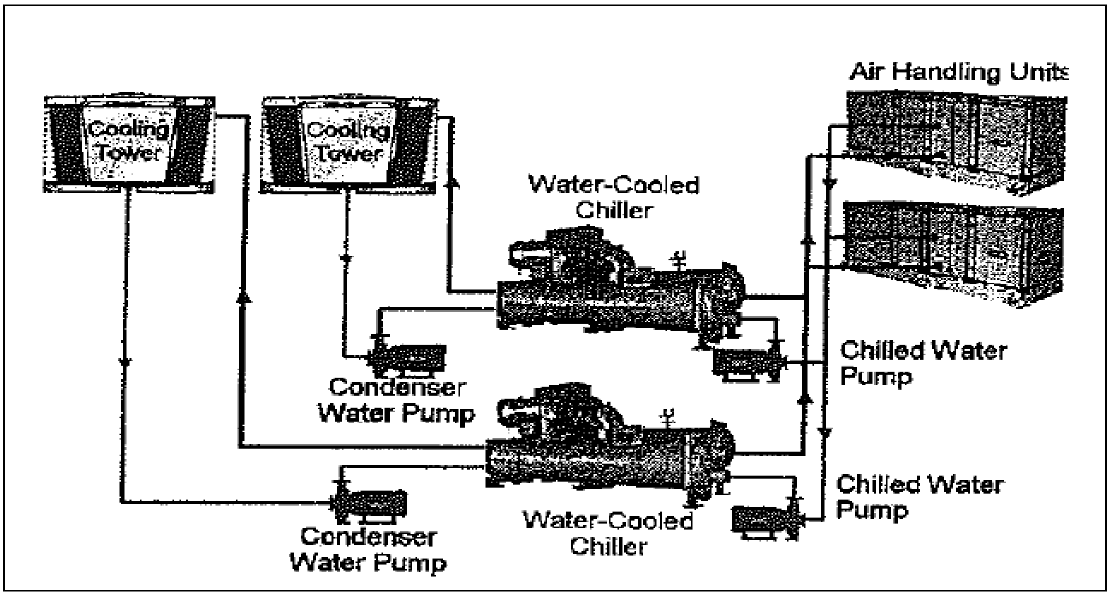


Fig (37). Multiple water cooled chiller loop.

- Multiple water cooled chiller with manifold pumps as shown in figure (38).

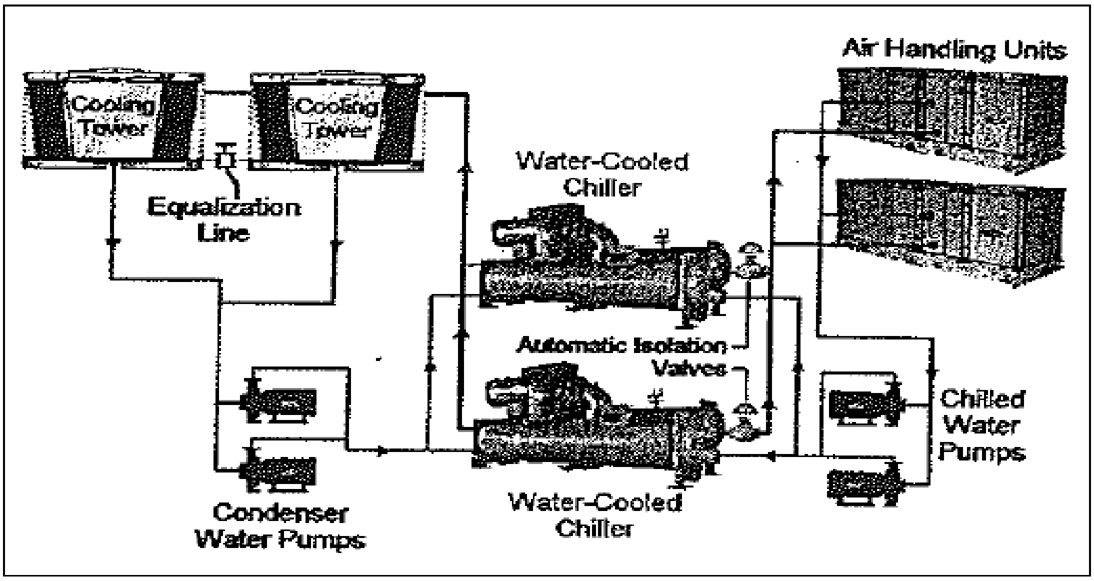


Fig (38). Multiple water cooled chiller with manifold pumps.

2.2.3 Head pressure control piping method.

- Diverting valve

✓ Locate condenser water pump outside the bypass line such that flow rate will be constant through the cooling tower.

✓ Locate condenser water pump inside the bypass line such that flow rate will be constant through the condenser chiller .The manufacture recommends this scheme, because full flow is maintained in the condenser.

- With using VFD or modulating valve

  ✓ With using modulating valve vary the flow in condenser .reduced flow through the condenser result in less heat transfer, and saturated condensing temperature. The flow through condenser water pump and cooling tower varies as the system head varies due to change in valve position .the valve will modulate with variation in water temperature.
  ✓ VFD used to control flow directly from head pressure control signal from chiller .this method varies flow through both chiller condenser and cooling tower.
  ✓ VFD or valve modulating it is not common as the other methods using diverting valve. Figure (39) show Head pressure control piping method.

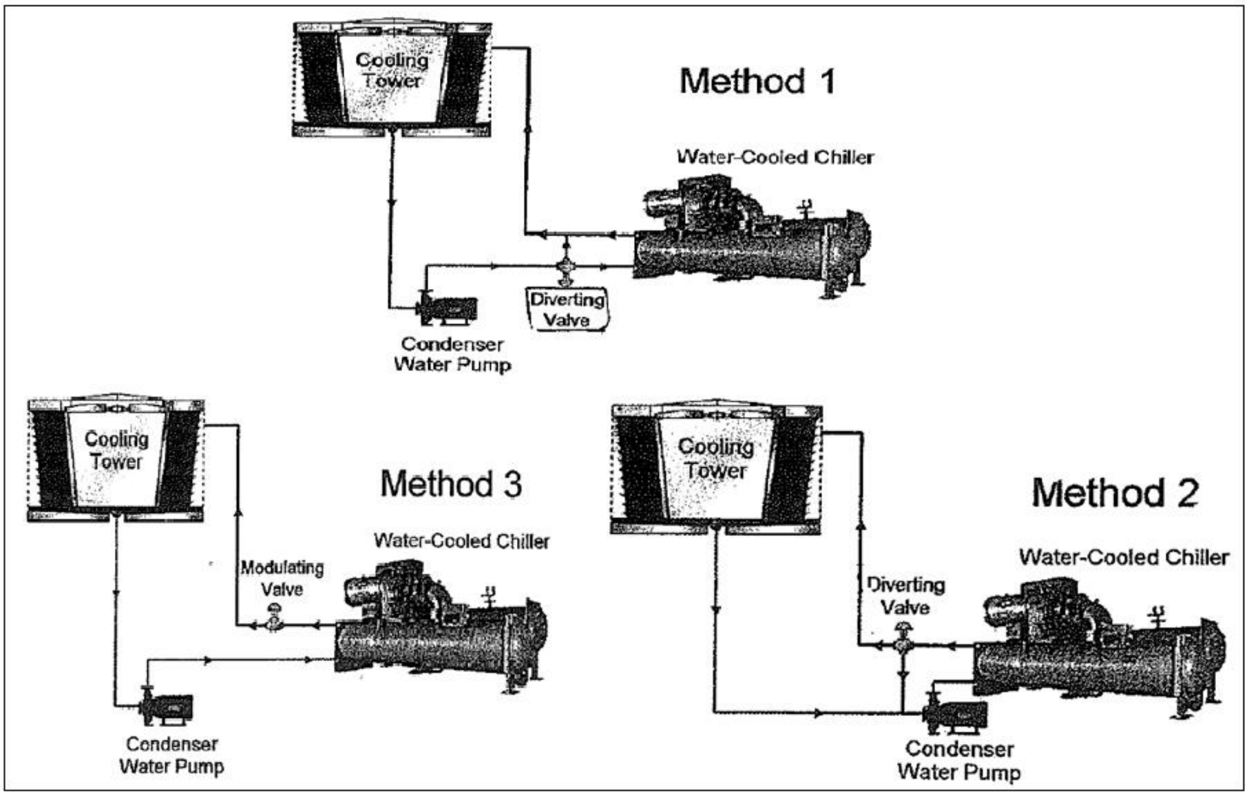


Fig (39). Head pressure control piping method.

2-4 Pump

For pump details see previous section (DX water cooled concept of design).

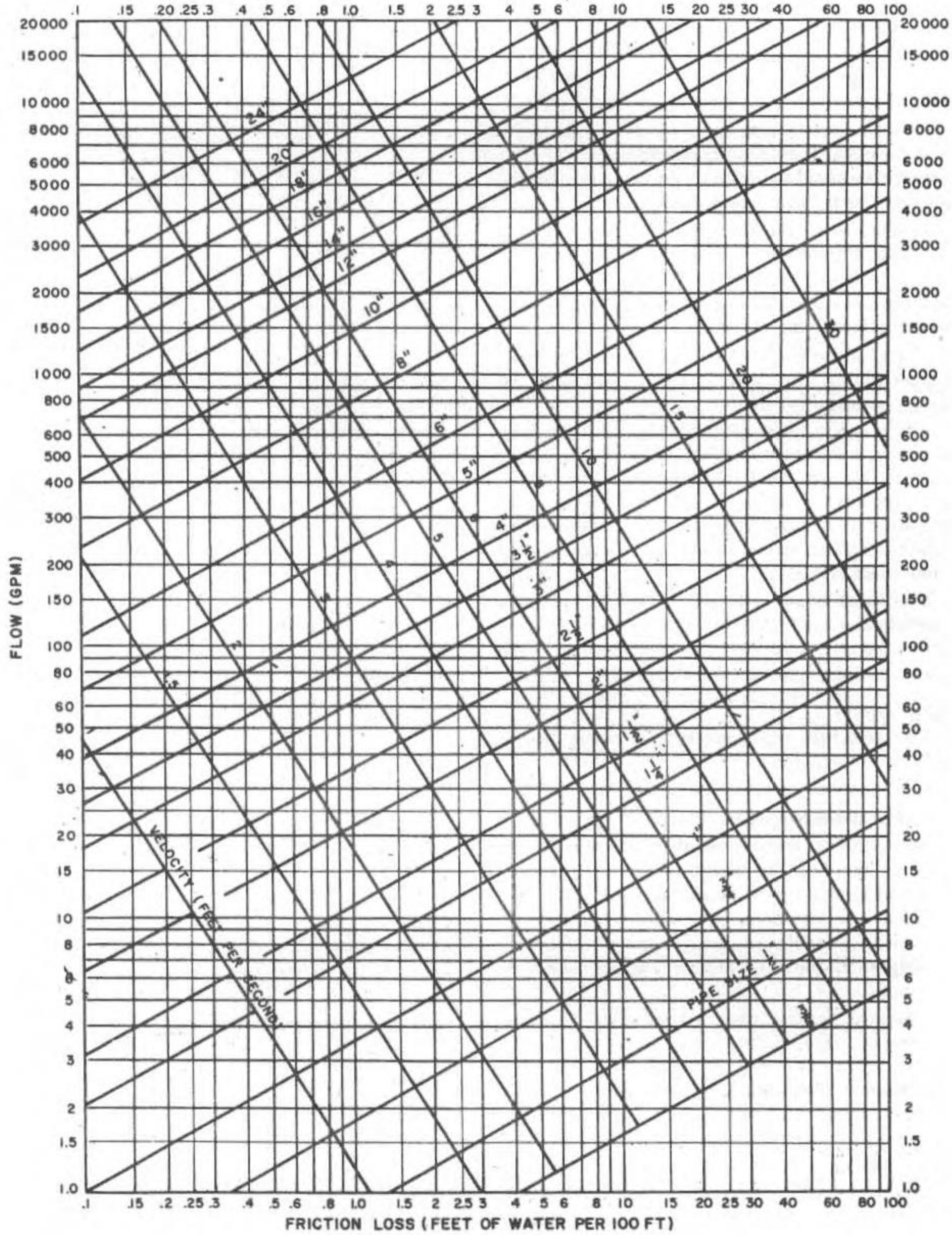
2-4-1 Pump head calcultion. For pump head calcultion the following information must be avaialble :

- Gpm of water to be circulated in piping circuit.
  - Calculation amount of water required for each equipment.
  - Chiller & equipment capacity, gal/min = rated tons * 24/ $\Delta T\ (f)$.
  - For ΔT = 10 → Chiller& equipment capacity, gal/min =2.4 * rated tons.
  - For ΔT = 12 → Chiller & equipment capacity, gal/min =2 * rated tons.

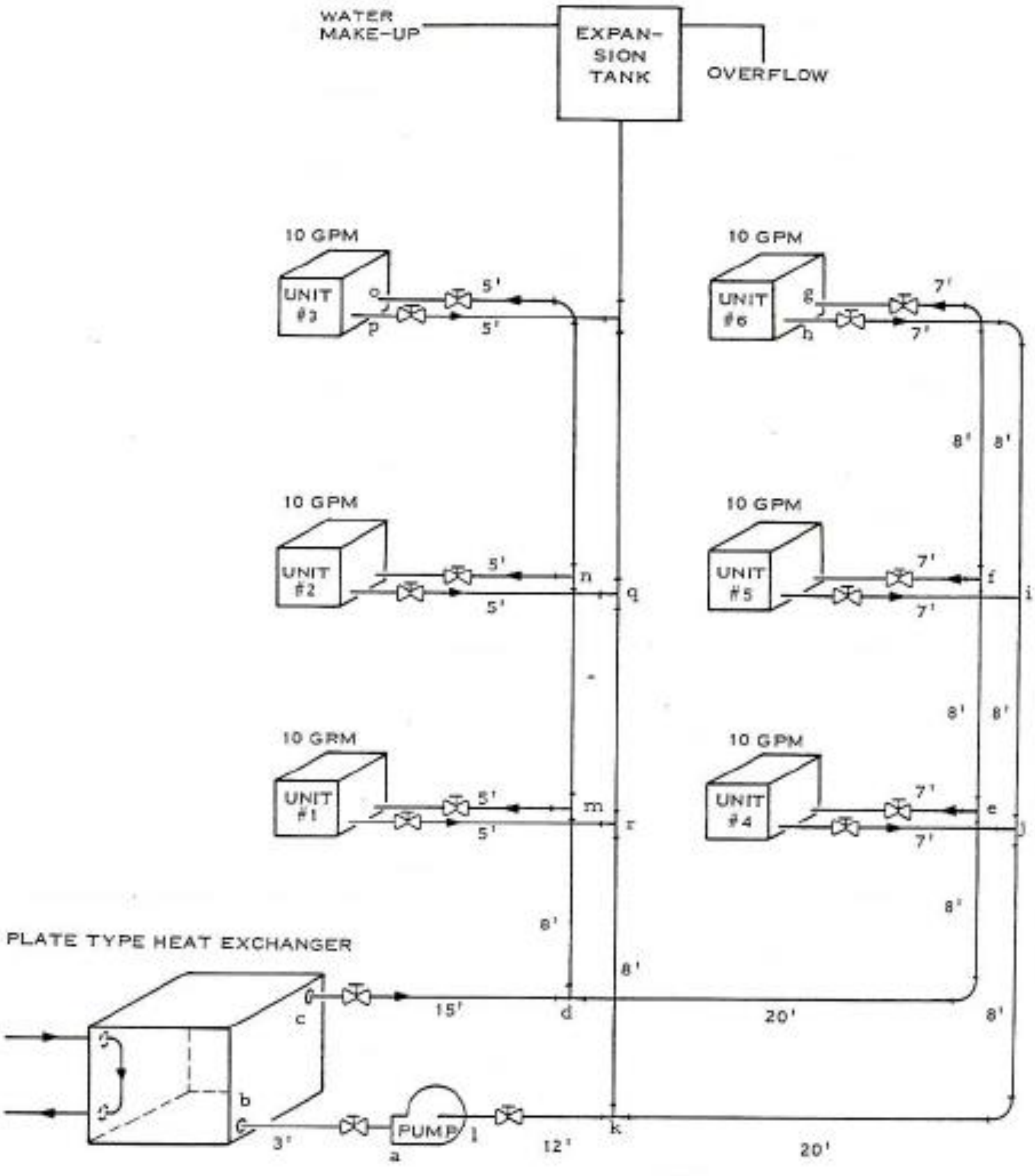
- System lay out showing length of all runs with units location ,valves,and fitting .
- Pipe sizing can be done based on fluid flow rate, and fluid velocity as previous mentioned in cooling towers chapter.
- Sizing can be also done with using the following chart for sizing closed loop of black steel sch-40.also the sizing can be done with using software called pipe sizer as shown in previous chapter.
- Pressure drop through condensers,heate exchange .
- Type of pipe to be used.

CHART 3—FRICTION LOSS FOR CLOSED PIPING SYSTEMS

Schedule 40 Pipe



- **Pump head Sample calculation.**



Typical Closed Water Re-circulation System

- 60 gpm is total circuit flow rate.
- system layout,valves ,and fiting as shown in the previous figure.
- condenser pressure drop 11.3 ft.
- heat exchanger pressure drop =15ft of water .

- schudle 40 pipe

- **for first path pressure drop calculation.**

- (From a-b)

Pipe size =2"

Flow rate =60gpm

Lenght of pipe =3ft

1gate valve =1.2 ft

Total equivelant leght =4.2 ft

Pressure drop =12ft/100ft

Thus pressure drop through this part = 4.2x12/100 =0.5ft of water

- (from b-c)

Pressure drop through heat exhanger = 15 ft of water

- (from c-d)

Pipe size =2"

Flow rate =60gpm

Lenght of pipe =15ft

1gate valve =1.2 ft

Total equivelant leght =16.2 ft

Pressure drop =12ft/100ft

Thus pressure drop through this part = 16.2x12/100 =1.9ft of water

- (from d-e)

Pipe size =1.5"

Flow rate =30gpm

Lenght of pipe =28ft

St elbow 90 =4.5ft

Total equivelant leght =35.5 ft

Pressure drop =13.4ft/100ft

Thus pressure drop through this part = 35.5x13.4/100 =4.8ft of water

- (from e-f)

Pipe size =1.25"

Flow rate =20gpm

Lenght of pipe =8ft

St T trough =2.5ft

Total equivelant leght =10.5 ft

Pressure drop =15.5ft/100ft

Thus pressure drop through this part = 15.5x10.5/100 =1.6ft of water

- ✓ (from f-g)

Pipe size =1"

Flow rate =10gpm

Lenght of pipe =15ft

1gate valve = 0.6 ft

St elbow 90 =2.5ft

St T through =2 ft

Total equivelant leght =20.1 ft

Pressure drop =11.8ft/100ft

Thus pressure drop through this part = 20.1x11.8/100 =2.4ft of water

- ✓ (from g-h)

Condenser pressure =11.3ft

- ✓ (from h-i)

Pipe size =1"

Flow rate =10gpm

Lenght of pipe =15ft

1gate valve = 0.6 ft

St elbow 90 =2.5ft

Total equivelant leght =18.1 ft

Pressure drop =11.8ft/100ft

Thus pressure drop through this part = 18.1x11.8/100 =2.1ft of water

- ✓ (from i-j)

Pipe size =1.25"

Flow rate =20gpm

Lenght of pipe =8ft

St elbow 90 =2.5ft

Total equivelant leght =10.5 ft

Pressure drop =15.5ft/100ft

Thus pressure drop through this part = 15.5x10.5/100 =2.1ft of water

- ✓ (from j-k)

Pipe size =1.5"

Flow rate =30gpm

Lenght of pipe =28ft

St elbow 90 =2.5ft

St T through =3 ft

Total equivelant leght =35.5 ft

Pressure drop =13.4ft/100ft

Thus pressure drop through this part = 13.4x35.5/100 =4.8ft of water

- ✓ (from k-i)

Pipe size =2"

Flow rate =60gpm

Lenght of pipe =12ft

Gate valve =1.5ft

St T through =3.5 ft

Total equivelant leght =16.7 ft

Pressure drop =12ft/100ft

Thus pressure drop through this part = 12x16.7/100 =2.ft of water

TOTAL PRESSURE DROP = 48 FT

- ✓ in the same way calculated prseeure drop for the othre path
- ✓ (from d-m)

Thus pressure drop through this part =2.3ft of water

- ✓ (from m-n)

Thus pressure drop through this part =1.6ft of water

- ✓ (from n-o)

Thus pressure drop through this part =2.1ft of water

- ✓ (from o-p)

Thus pressure drop through this part =11.3ft of water

- ✓ (from p-q)

Thus pressure drop through this part =2.3ft of water

- ✓ (from q-r)

Thus pressure drop through this part =1.6ft of water

- ✓ (from r-k)

Thus pressure drop through this part =6.ft of water

TOTAL PRESSURE DROP = 27.2FT

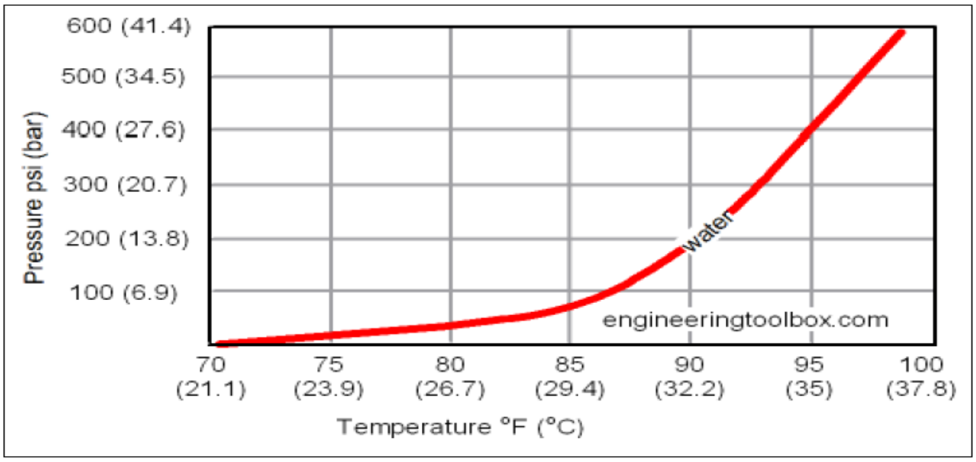
The following table contain the taabulated data for the closed water system sample problem .

Runs	Gallons per Minute	Nominal Pipe Size, Inches	Length of Pipe Feet	Fittings, Equivalent Feet of Pipe				Total Equivalent Feet	Press. Drop Ft. Water Per 100 Lin. Ft.	Total Press. Drop Feet Water	Actual Press. at Various Points in Feet Water
				Gate Valve	Std. Tee Thru Run	Std. Tee Thru Branch	Std. 90° Elbow				
a -b	60	2	3	1.2				4.2	12.0	.5	a 48.0
b -c	60									15.0	b 47.5
c -d	60	2	15	1.2				16.2	12.0	1.9	c 32.5
d -e	30	1-1/2	28		3.0		4.5	35.5	13.4	4.8	d 30.6
e -f	20	1-1/4	8		2.5			10.5	15.5	1.6	e 25.8
f -g	10	1	15	.6	2.0		2.5	20.1	11.8	2.4	f 24.2
g -h	10					Unit #6				11.3	g 21.8
h -i	10	1	15	.6			2.5	18.1	11.8	2.1	h 10.5
i -j	20	1-1/4	8		2.5			10.5	15.5	1.6	i 8.4
j -k	30	1-1/2	28		3.0		4.5	35.5	13.4	4.8	j 6.8
k -l	60	2	12	1.2	3.5			16.7	12.0	2.0	k 2.0
a -l	Total Pressure Drop (Ft. of water) in Longest Run									48.0	l 0
d -m	30	1-1/2	8			9.0		17.0	13.4	2.3	d 30.6
m-n	20	1-1/4	8		2.5			10.5	15.5	1.6	m 28.3
n -o	10	1	13	.6	2.0		2.5	18.1	11.8	2.1	n 26.7
o -p	10					Unit #3				11.3	0 24.6
p -q	10	1	13	.6		5.5		19.1	11.8	2.3	p 13.3
q -r	20	1-1/4	8			2.5		10.5	15.5	1.6	q 11.0
r -k	30	1-1/4	8		2.5	7.5		18.0	33.5	6.0	r 9.4
d -k	Total Pressure Drop (Ft. of water) in Shorter Run									27.2	k 3.4 *

3 hydronic system accessories.

3.1 Expansion tank.

- Handle with excess water those results from temperature change.
- Provide make up location for automatic replacement the system that loss due to leakage.
- To avoid unacceptable increase of system pressure during heat up. Keep the pressure inside the circuit at constant value.



- Assist in providing a pressure higher than net positive suction head at the pump suction to prevent cavitation.

- To maintain pressure higher than the minimum water pressure required to vent air.

  ✓ The pressure equals the pressure of air inside the tank plus or minus any fluid pressure due to the elevation difference between the tank liquid surface and the pipe as shown in figure (40).

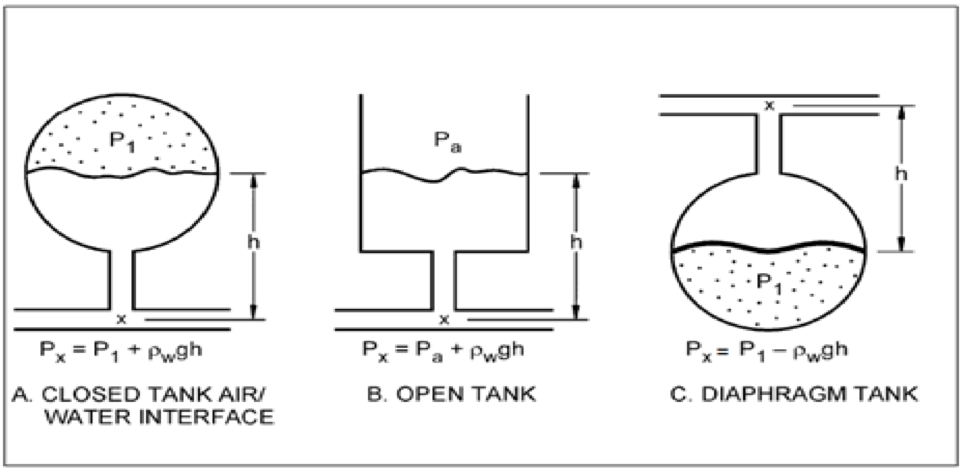
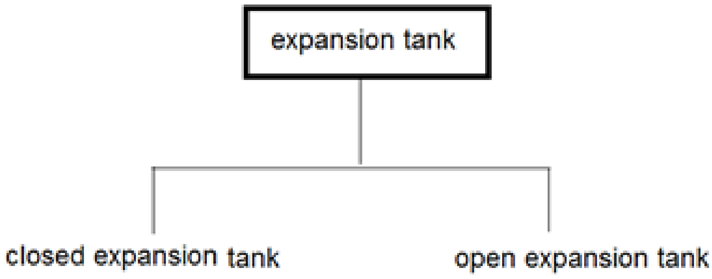


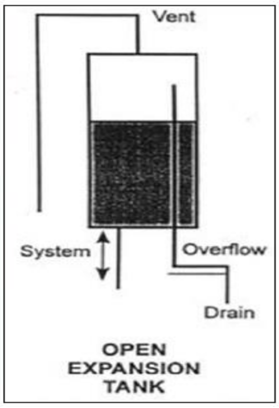
Fig (40). Tank pressure related to system pressure.

3-1-1 expansion tank types



3.1.1.1 Open expansion tank

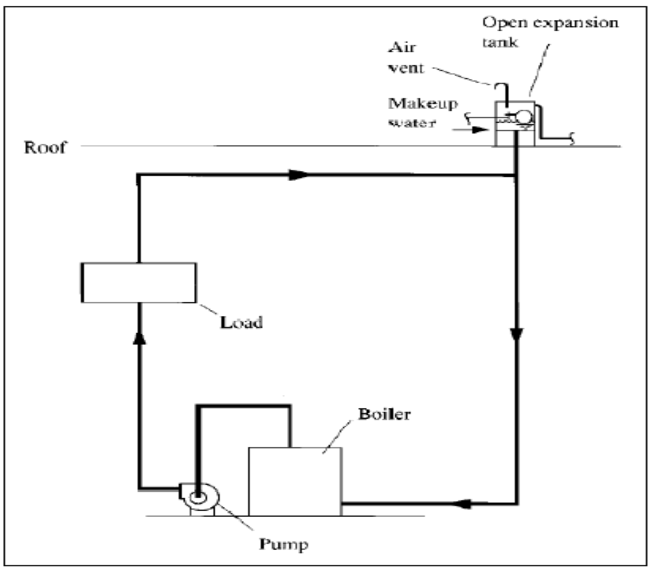
- **concepts .**



- ✓ Required large space due to it contain large amount of water to make the required pressure .
- ✓ Due to this tanke open to atmospheric the water inside will be exposed to infect by dist and sand ,so water inside the tank required fillteration .
- ✓ This tank must be put at the highest point in the system to insure achieveing the required circuit pressure.
- ✓ Open expansion tank has disadvantage of allowing the air to enter the system via absorption in the water .in general it must be located in the top of the buildingwhere it also may be exposed to freezing.

- **Method of tank installion .**

- ✓ tank installed at the highest point in the system at least 3m above. the highest point in the water system.
- ✓ Tank installed at pump suction line .As shown in the following sketch.



- **Tank sizing method**

  - ✓ Calculate the total amount of water inside the water piping circuit from the following table (2,3 CARRIER HAND BOOK PIPING DESIGN CHAPTER).
  - ✓ Calculate the volume of water in the coils and heat exchangers.
- ✓ Determine the percent increase in the volume of water due to operating at increased temperatures from following table.

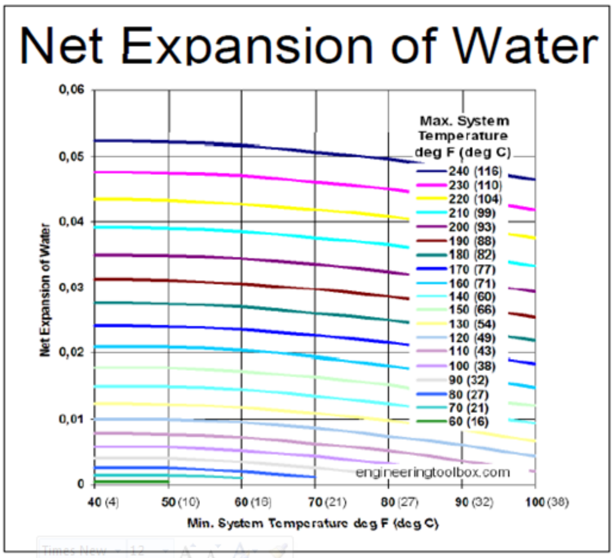
$$V_{net} = (v_1 / v_0) - 1$$

- V_{net} = *necessary expansion volume of water (gallon, liter)*
- v_0 = *specific volume of water at initial (cold) temperature (ft3/lb, m3/kg)*
- v_1 = *specific volume of water at operating (hot) temperature (ft3/lb, m3/kg)*

EXPANSION OF WATER

(Above 40 F)

TEMP (F)	VOLUME INCREASE (%)	TEMP (F)	VOLUME INCREASE (%)
100	.6	275	6.8
125	1.2	300	8.3
150	1.8	325	9.8
175	2.8	350	11.5
200	3.5	375	13.0
225	4.5	400	15.0
250	5.6		



Tank volume = E x VS

E = the percent increase in the volume of water due to operating at increased temperatures (HIGHEST WATER TEMP IN COOLING PROCESS =40C OR 105F)

VS= total amount of water inside the circuit.

- **According ASHARE code :**

$$V_T = 2\,VS\,\{[(\nu 2/\nu 1) - 1] - 3\alpha\Delta t\}$$

3-1-1-2 Closed expansion tank

- **Concept.**

✓ This type of expansion tank common to use due to the following reason.
✓ Not required large space.
✓ Not need to put at the highest point in the circuit.
✓ Water in this tank not subject to atmospheric air.

- **Installation method**

  ✓ Tank installed at pumps suction line to prevent the formation of air bubbles inside the pumps, and achieve the pressure required for pumps drawn.as shown in figure (41).

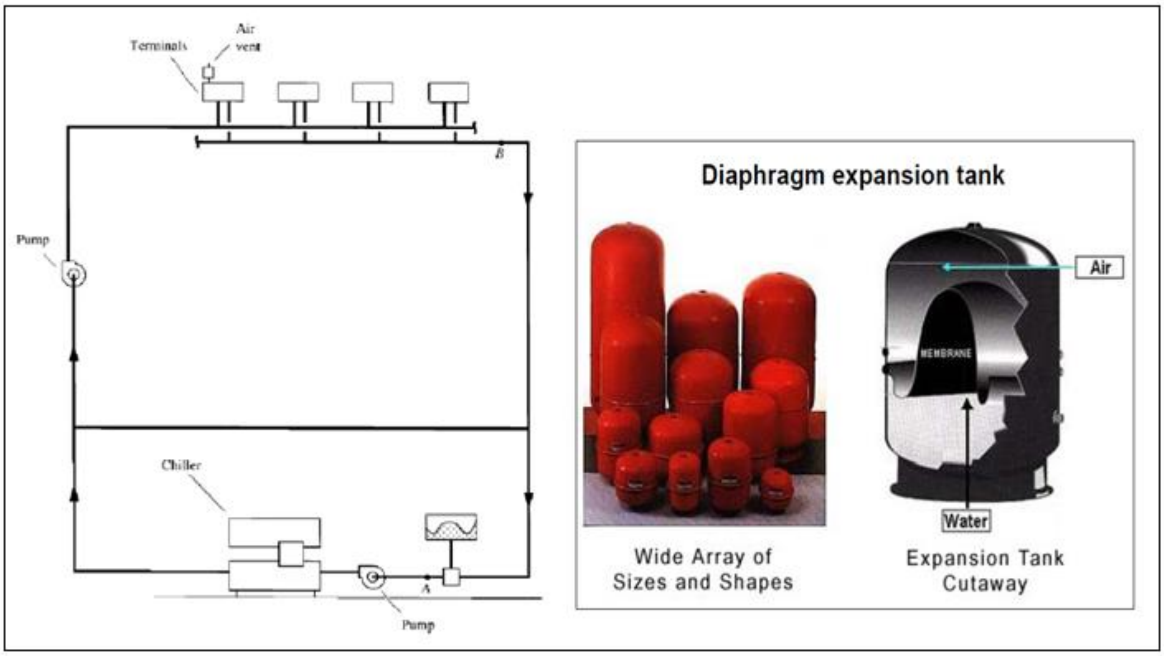


Fig (41). Closed expansion tank (balder type) position at closed circuit system

- **Calculation method**

- **With diaphragm /bladder tank**

  - ✓ The second, bladder type is usually applied with chilled water systems because cold water tends to absorb the air in the free surface type of tank and release it elsewhere in the system, where it is removed.

  - ✓ The location of pump in water system that uses diaphragm expansion tank should be arranged so that the pressure at any point in the water system is greater than atmospheric pressure.

  - ✓ In such arrangement, air does not leak into the system, and the required NPSH can be maintained at the suction inlet of the water pump. As shown in following figure.

$$V_T = V_S \frac{[(\nu 2/\nu 1) - 1] - 3\alpha\Delta t}{1-(P_1/P_2)}$$

- **With Closed tank with air water interface**, calculated from the following equation: Is usually used in hot water systems because it provides a convenient place for air to collect when released from the heated water in the boiler.

$$V_T = V_S \frac{[(\nu 2/\nu 1) - 1] - 3\alpha\Delta t}{(P_a/P_1) - (Pa/P2)}$$

<u>**Where:**</u>

V_T=volume of expansion tank (gal).

V_S= volume of water in system (gal).

T_1= lower temp (supply water temperature) (F°).

T_2=higher temp (ambient temperature "design weather data") (F°).

p_a = atmospheric pressure, (14.7 psia).

p_i =initial fill or min pressure at tank (the system working pressure or hot/chilled circulating pump head) (Psig).

p_1=absolute pressure @lower temp (Pi+Pa) (Psia).

p_{max}= max system allowable pressure (it is from 1.5 to 2 working pressure) (Psig).

p_2 =absolute pressure @higher temp (pmax+pa).

$\vartheta_1$1 = specific volume of water @lower temp (FT^3/LB).

ϑ_2= specific volume of water @higher temp (FT^3/LB).

α =linear coff of thermal expansion, (for steel pipes =6.5x10-6 IN/INF° ,FOR copper pipes 9.5X10-6 IN/IN F°).

- **Foot note:**

Water specific volume @ different temperature.

Temp °F	Specific Volume v (ft³/lb)	Temp °F	Specific Volume v (ft³/lb)	Temp °F	Specific Volume v (ft³/lb)
10	0.01744	83	0.01608	140	0.01629
20	0.01746	84	0.01608	150	0.01634
30	0.01747	85	0.01609	160	0.01639
40	0.01602	86	0.01609	170	0.01645
42	0.01602	87	0.01609	180	0.01651
44	0.01602	88	0.01609	190	0.01657
50	0.01602	89	0.01610	200	0.01663
60	0.01604	90	0.01610	210	0.01670
70	0.01605	91	0.01610	220	0.01677
71	0.01605	92	0.01611	230	0.01684
72	0.01606	93	0.01611	240	0.01692
73	0.01606	94	0.01611	250	0.01700
74	0.01606	95	0.01612	260	0.01708
75	0.01606	96	0.01612	270	0.01717
76	0.01606	97	0.01612	280	0.01726
77	0.01607	98	0.01612	290	0.01735
78	0.01607	99	0.01613	300	0.01745
79	0.01607	100	0.01613		
80	0.01607	110	0.01617		
81	0.01608	120	0.01620		
82	0.01608	130	0.01625		

3-2 Strainer.

- Used to prevent construction debris that may from entering the equipment during initial startup, and to catch any small debris that may be circulate through the system during normal operation.
- Strainers normally installed at the inlet side of a chiller as well as the suction side of a pump.
- Systems with large quantities of debris may use a basket strainer in lieu of Y strainer.
- Basket strainer is expensive and is used some times for extra straining of condenser water. Figure (42) shows the strainer types.

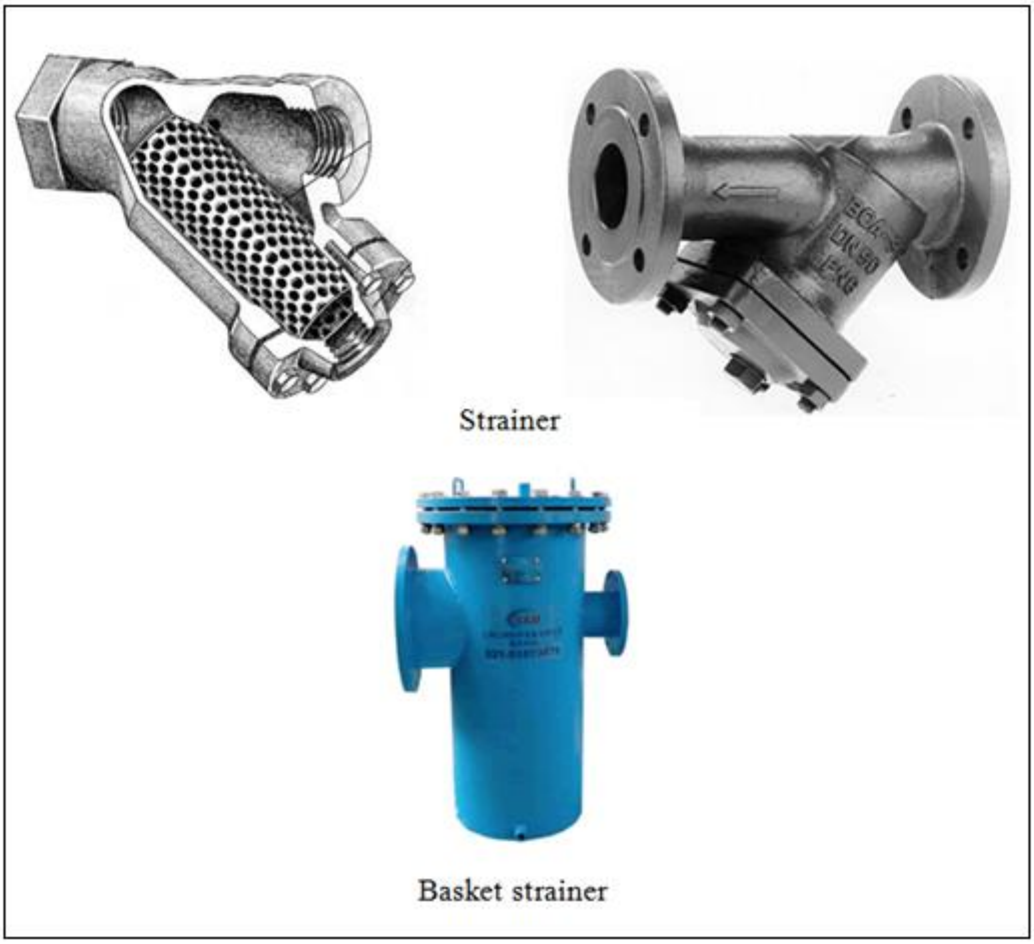


Fig (42). Strainer type

- **<u>Foot note</u>**

- Strainer is very important in open loop system and once through systems since debris can enter the system at any time.

3-3 Air elimination.

3-3-1 Air in water system concept.

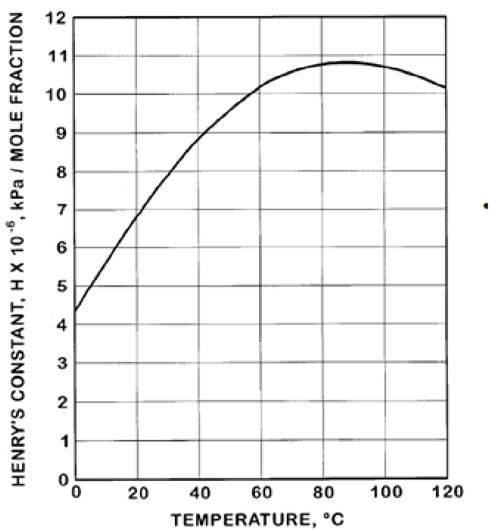
- The are two sources of air and gas in water system one is air water interface expansion tank ,open tank ,and air dissolved in city water supply.
- In closed recirculated water system air and nitrogen are present in following forms:
  - ✓ Dissolved in water.
  - ✓ Gas bublles.
  - ✓ Pockets of air or gas.
  - ✓ The behavior of gas or air dissolved in liqued governed and described by henry equation (henry stae the amount of gas dissolved in water at constant temperature is directly proportional to the absoulte pressure of that gas acting on the liquid).

- **The presene of air or gas in water system causes** the following effets for closed water systen with plain closed expansion tank.

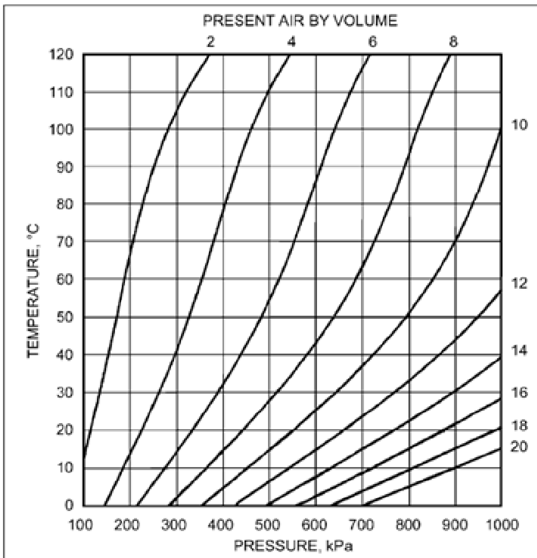
  - ✓ Prsence of air in termianl or heat exchanger which reduce heat transfer.
  - ✓ Corrosion due to oxgen react with pipes.
  - ✓ Water logging in plain expansion tank.
  - ✓ Unstable system pressure.
  - ✓ Poor pump performace due to air bubbles
  - ✓ Noise problem.
- Before tank sizing the control or elimination of air must be must be considered,and The amount of air that will be absorbed and can be held in solution with the water is expressed by henry equation.

$$X = \frac{p}{H}$$

X = amount of dissolved gas in solution by volume.
P = partial pressure of that gas, Pisa.
H = Henry constant change with temperature.



Henry's Constant Versus Temperature for Air and Water



solubility versus temperature and pressure for air /water solutions

3-3-2 air elimination techniques

Air elimination can be done with using:

- **air siprators**
  - ✓ are used to reduce entrained air from the system which remove large percantage of air dissolved with the water lead to increase overal heat transfere efficiency of the system (air is insulator).Reduce corrosion affect caused by disolved oxgen .
  - ✓ air siprator located @ pump suction.figure (43) indicates the air siprator configuration.

Fig (43). Air sipratorconfiguration.

- **Air vent.**
  - ✓ Used to remove unwated air from the a closed loop system.they should be located at the high point in the system where air my be present .current air vent deisgns allow air to be vented withot loss of fluid from the sysyem.
  - ✓ Both manual or automatic air vent can be used in havc industry .manual air vents require aserviceperson for bleed the air from the system periodically .but automatic vent bleed tha air based on the pressure setting .figure (44) indicates air vent details

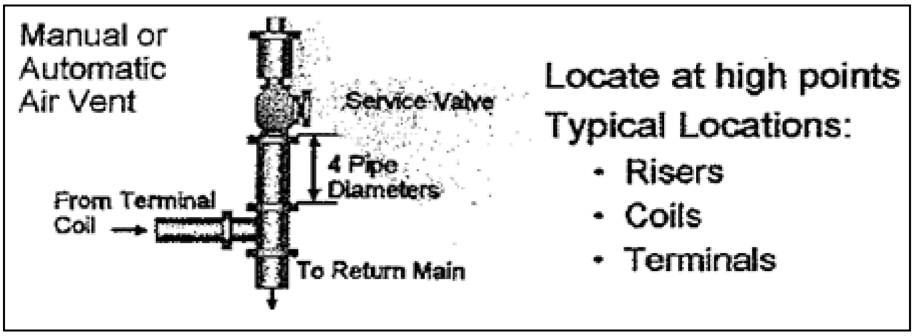


Fig (44). Air vent details

4 Hook up& valves.

4-1 Pump hook up.

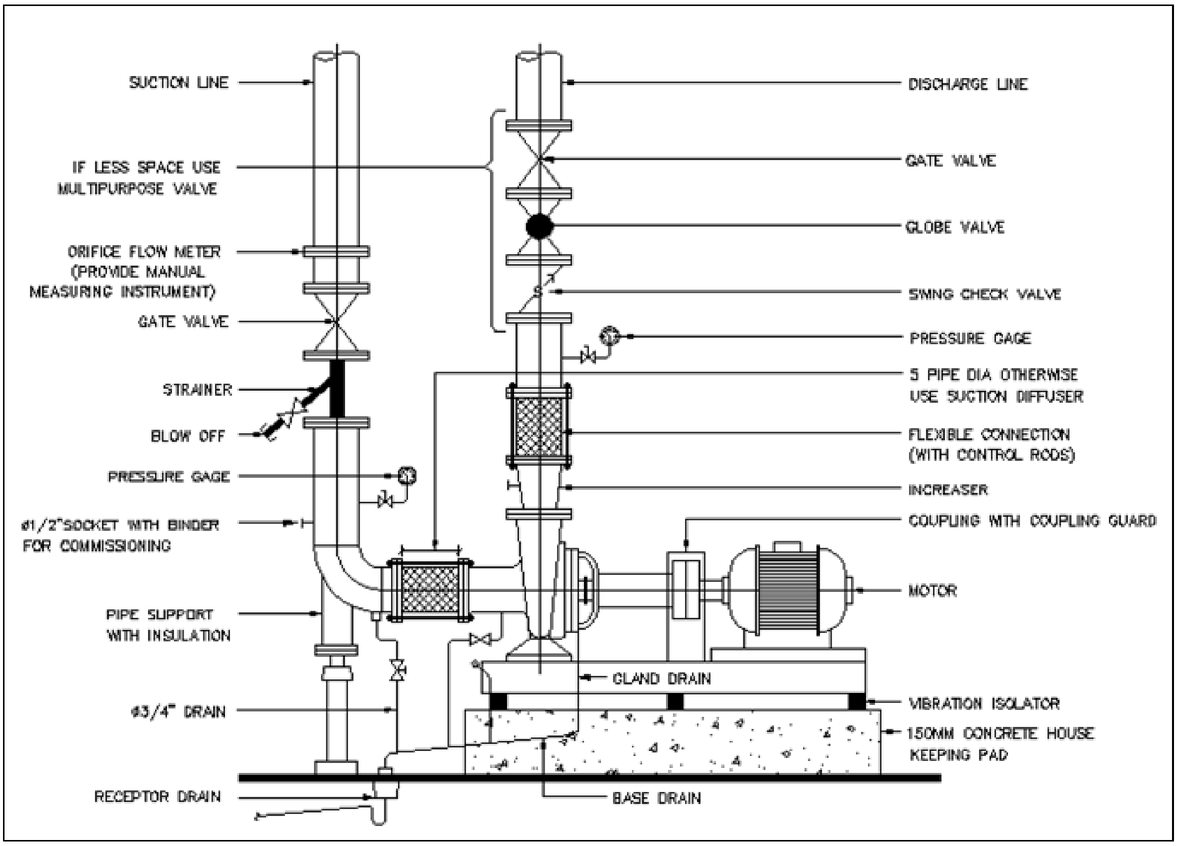


Fig (45).End suction pump hook up details

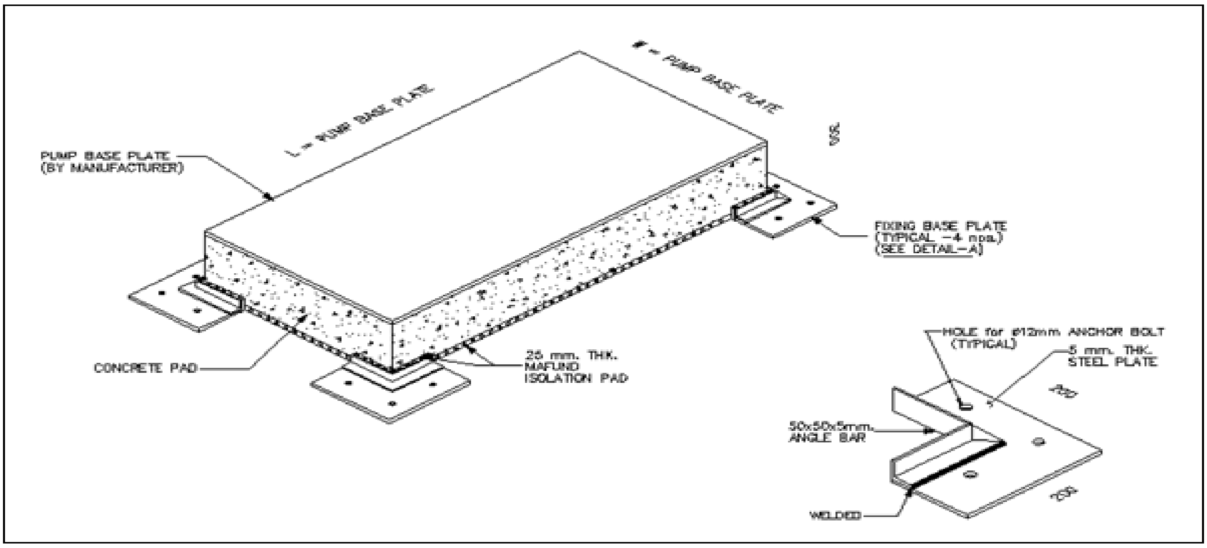


Fig (46). Pump concrete pad.

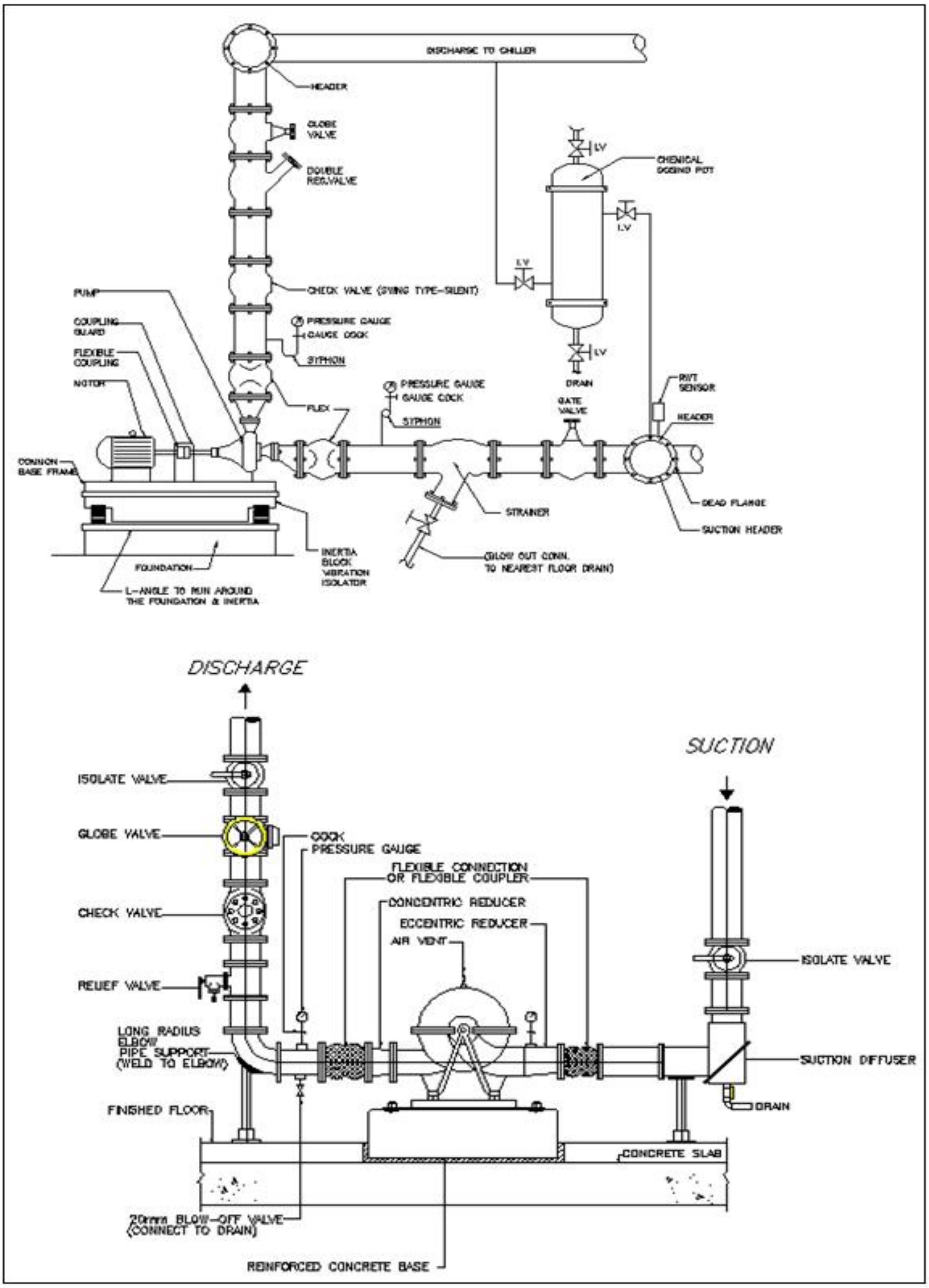


Fig (47). Horizontal split case pump hook up details

4-2 FCU hook up details.

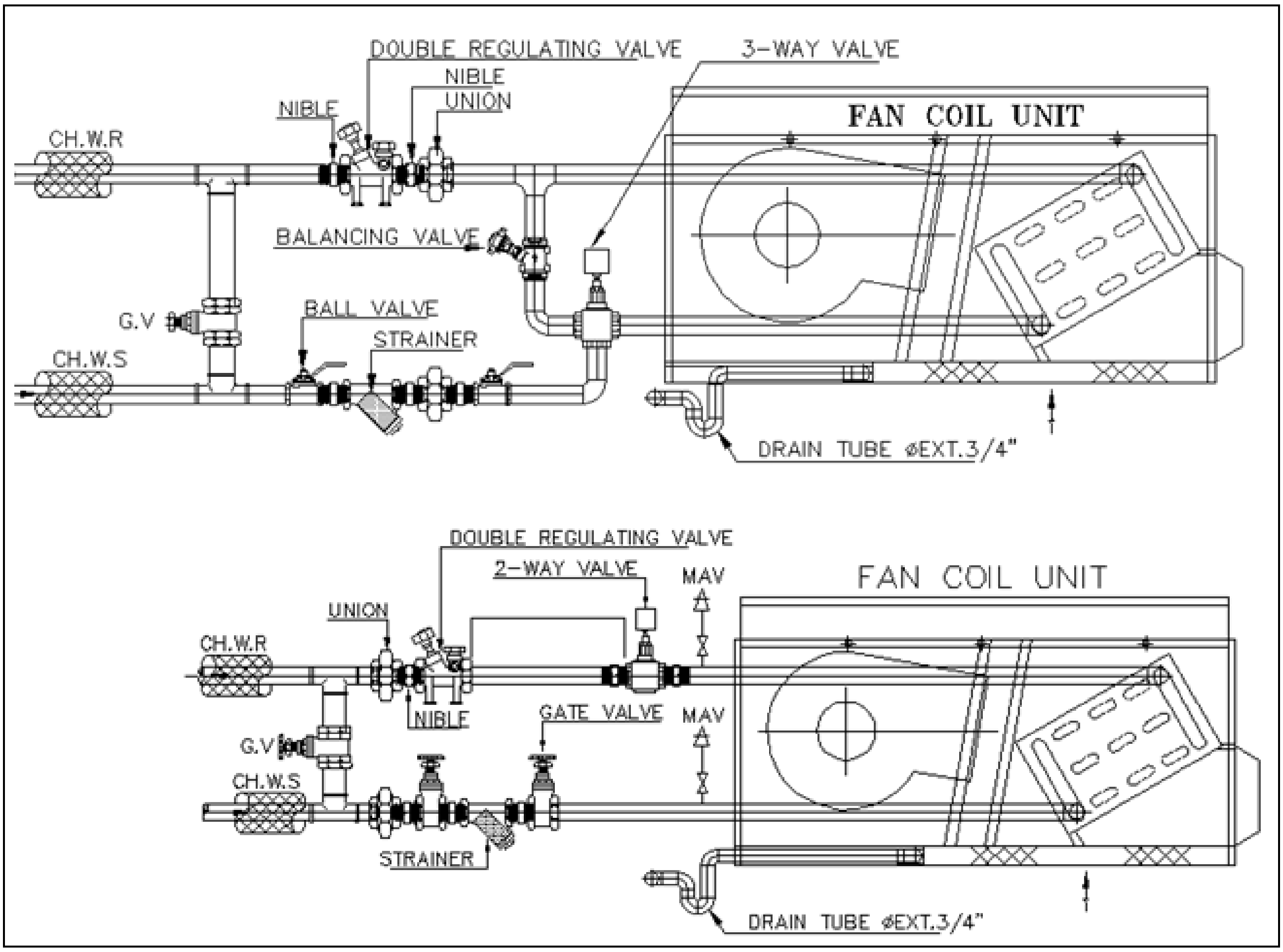


Fig (48). Fcu hook up details

4-3 AHU hook up details.

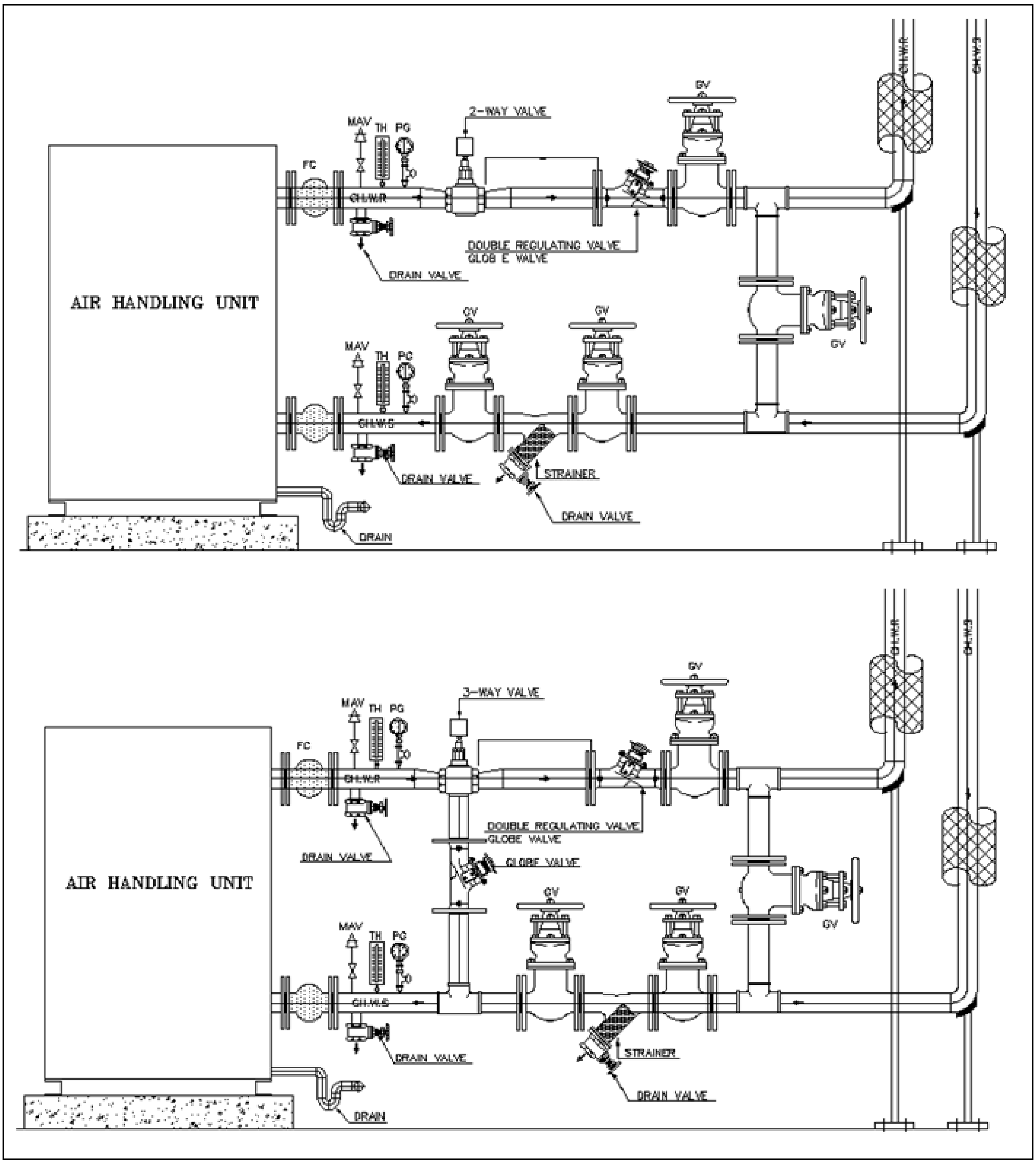


Fig (49). AHU hook up details.

4-4 Chiller hook up details

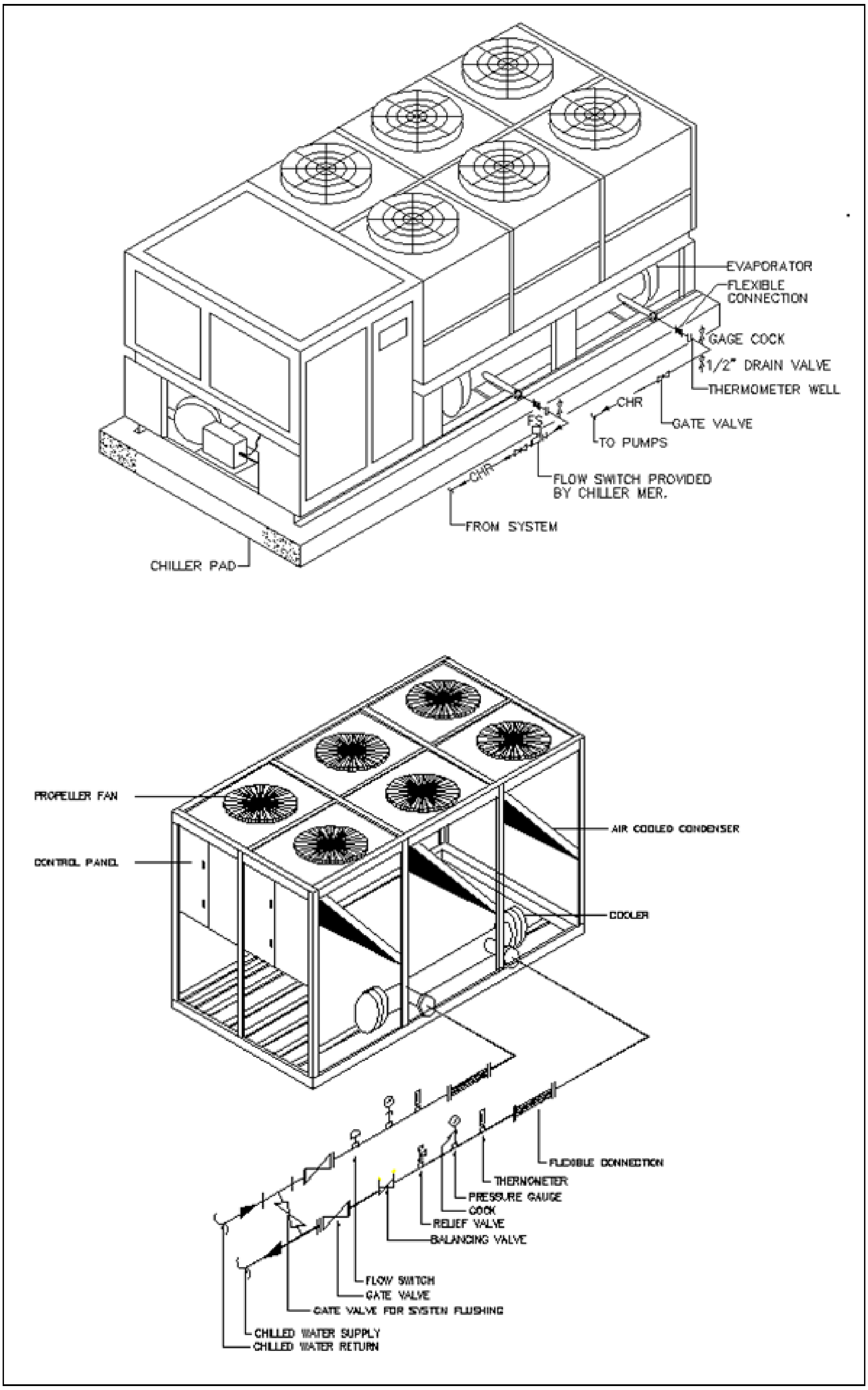


Fig (50). Chiller hook up details.

4-5 Valves

Many types of valves are available in HVAC industry each type of valve has certain characteristic that make it better for certain application such as shutoff, balancing control. Also some valves are suitable for multiple application .a brief description of different type valves and their application s are listed below.

- **Check valves (non return valves)**

- Check valves prevent the flow of water in reverse direction .there are two basic designs of check valves, firstly swing check valves may be used in a horizontal or in vertical line if the flow is up ward .they are used in combination with gate valve, as shown in the figure (51).

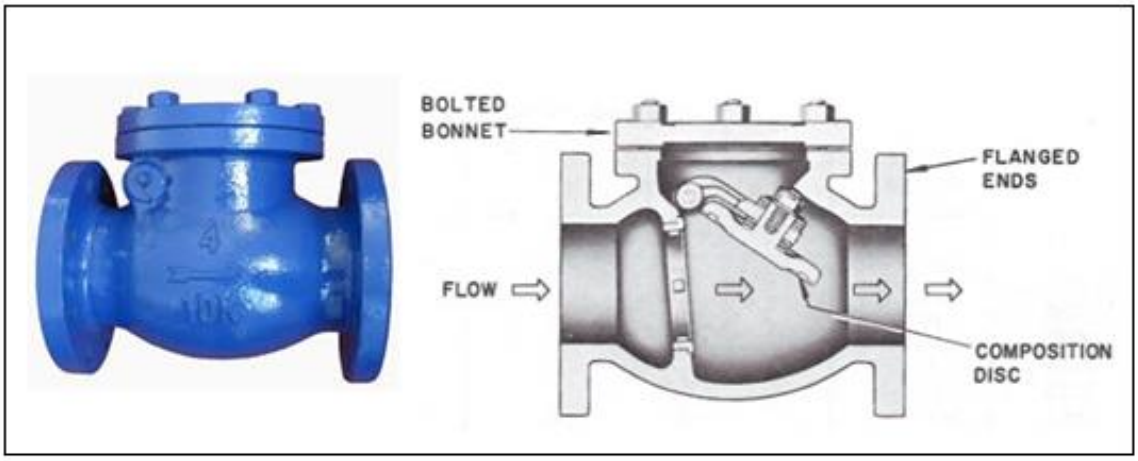


Fig (51). Swing check valve.

- Secondly lift check valve. It operates in a manner similar to that of globe valve .lift check valves should only be installed in horizontal piping and usually is used in combination with globe, angle, and Y Valves with flanged or threaded end as shown in figure (52).

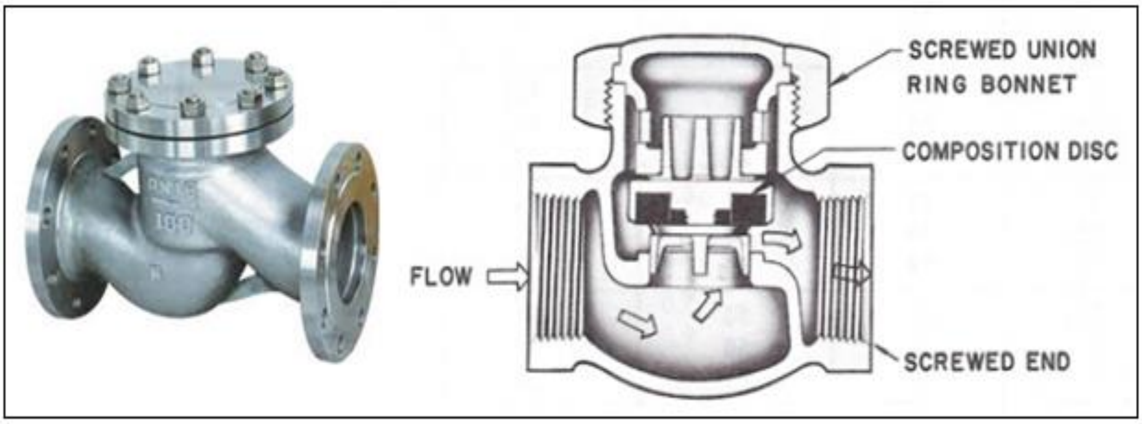


Fig (52). Lift check valve

- **Stop valves**
- **Ball valves:**used for shutt off duty service with limited requirement for precise ontrol as shown in figure (53).

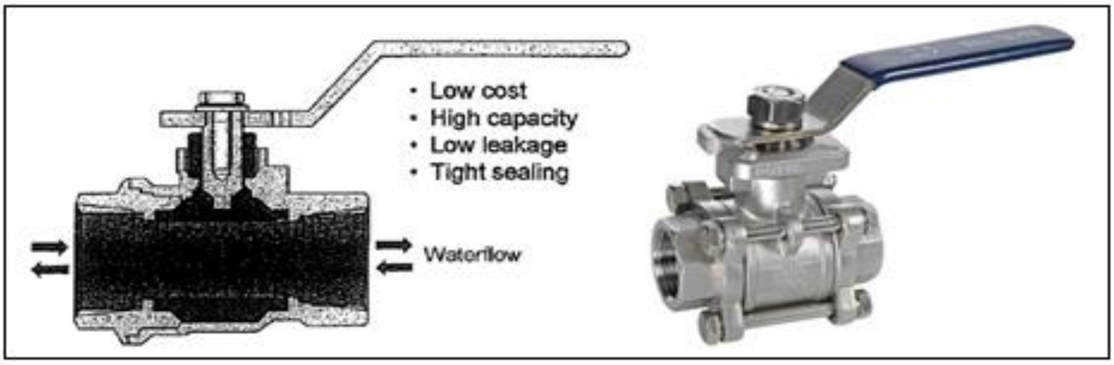


Fig (53). Ball valves

- **Gate valves (stop valves)**
  - Are designed for shutoff duty when the valve is in the wide open position, the gate completely out of fluid stream, thus providing straight through flow and a very low pressure drop.
  - Gate valve not be used for throttling .they not designed for this type service and consequently it is difficult to control fluid flow with any degree of accuracy .vibration ,and chattering of the disc occurs when the valve is used for throttling ,resulting in damage to the seating surface .
  - Gate valve can be installed regard the direction of flow within the pipe. They can be seat in either direction as shown in figure (54).

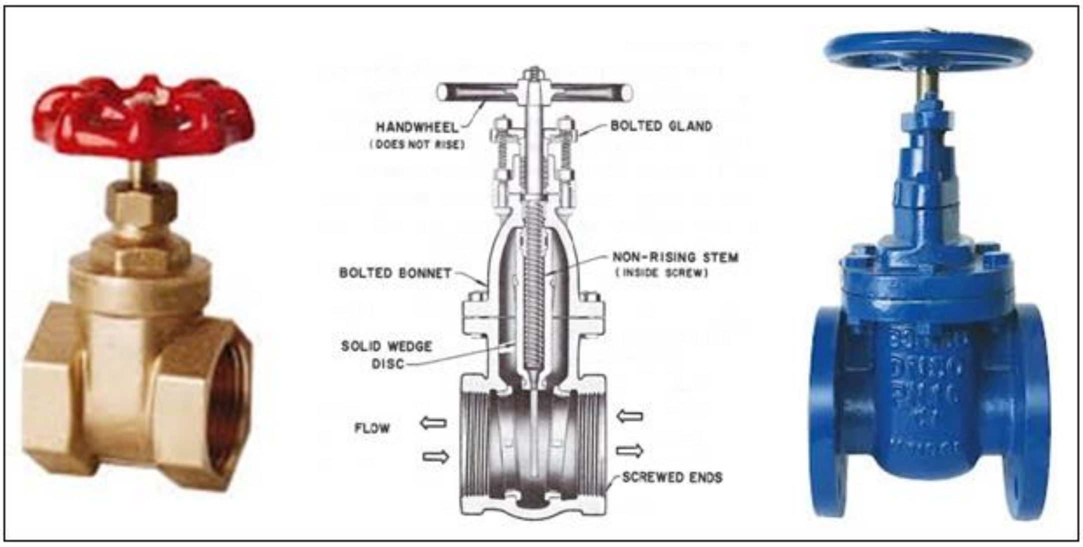


Fig (54). Gate valve configuration

- **Balancing, and stop valves.**
- **Butter fly valve**
  - Are generally found on large systems for shutoff, throttling duty, and where is frequent operation.
  - They have good flow control (linear relation between percent open and percent of full flow through the valve).
  - Low cost, high capacity and low pressure drop. they typically have bigger valves, and are used on pipe size 2.5" and larger .lug pattern will either through bolt between two flanges or be secured at the end of pipe section ,while a wafer pattern is a more economical style that just sits between the bolted flanges without it is own lugs .figure (55) indicates butter fly valve different configurations.

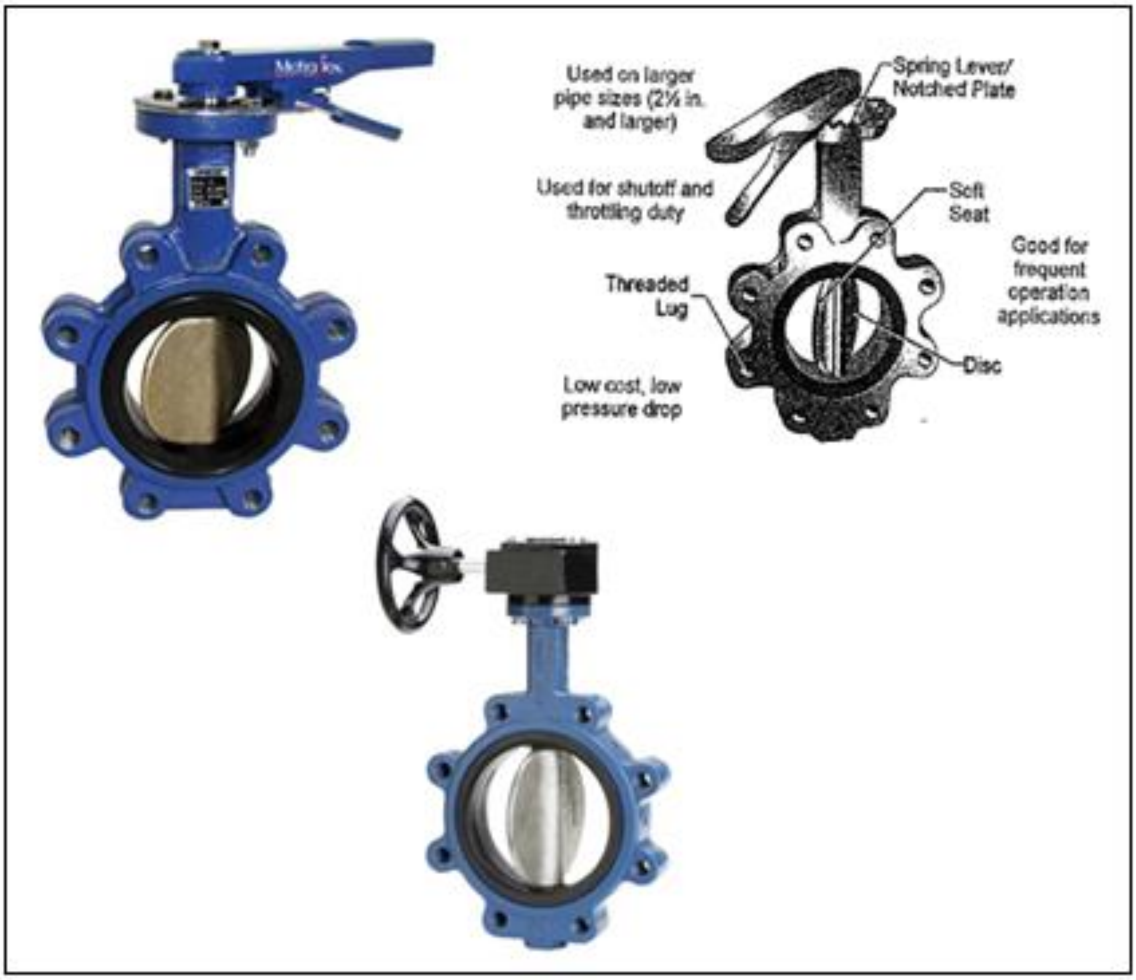


Fig (55). Butter fly valve different configurations

- **Globe valve, angle, and Y valve.**

  - Are designed primarily for throttling (balancing) duty .the angle or Y pattern valve is recommended for full flow service since it has a substantially lower pressure at this condition than the globe valve .also the angle valve cab be located to replace an elbow thus eliminating one fitting .
  - They can be opened faster than the gate valve due to shorter lift time of the disc, when they are to be operated frequently they provide the conventional operation.
  - The seating of these valves are subject to less wear and the discs and seats are easy to replace compared to gate valve as shown in figure (56).

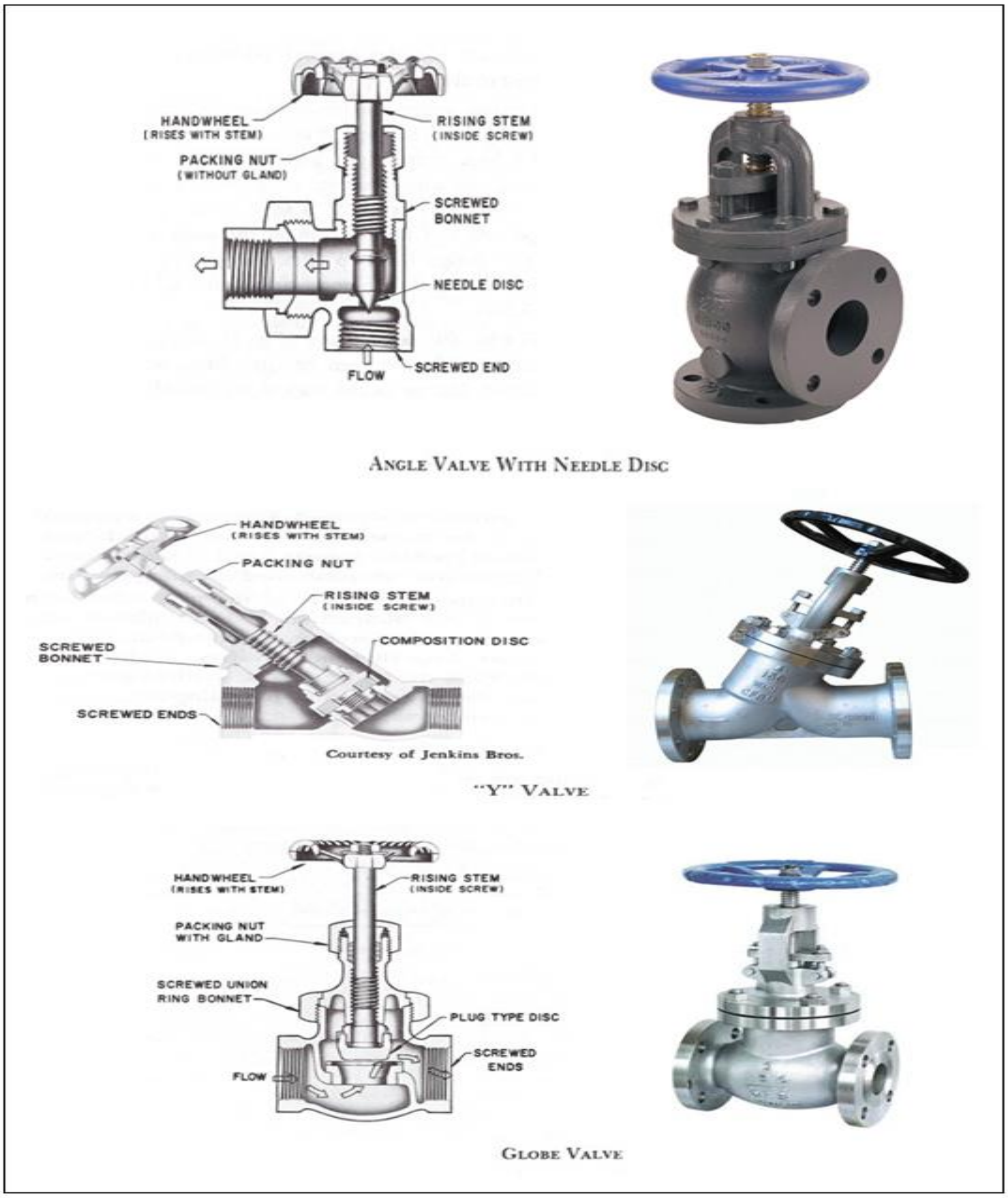


Fig (56). Globe valve, angle, and Y valve configuration.

· Plug valves.

- Plug cocks are primarily used for balancing duty on smaller flow applications because of cost.
- Plug cocks have approximately the same line loss as a gate valve when in the fully open position. When partially closed for balancing, this line loss increases substantially.
- They are normally less expensive than globe type valves and the setting cannot be tampered with as easily as a globe valve.
- For large flow rate applications, a globe or butterfly will be used instead of plug valve. Figure (57) shows plug valve configuration.

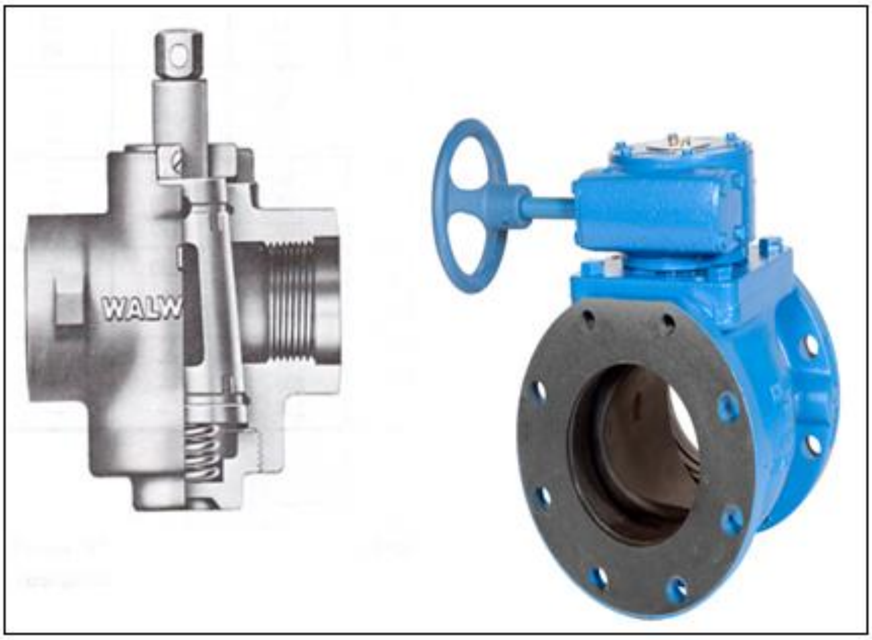


Fig (57).plug valve.

- **Flow control valves:** At partial load need to change the amount of flow to the equipment (AHU&FCU) to accommodate this change in load .this can do by 2 or 3ways valve.

✓ **Three way valve.**

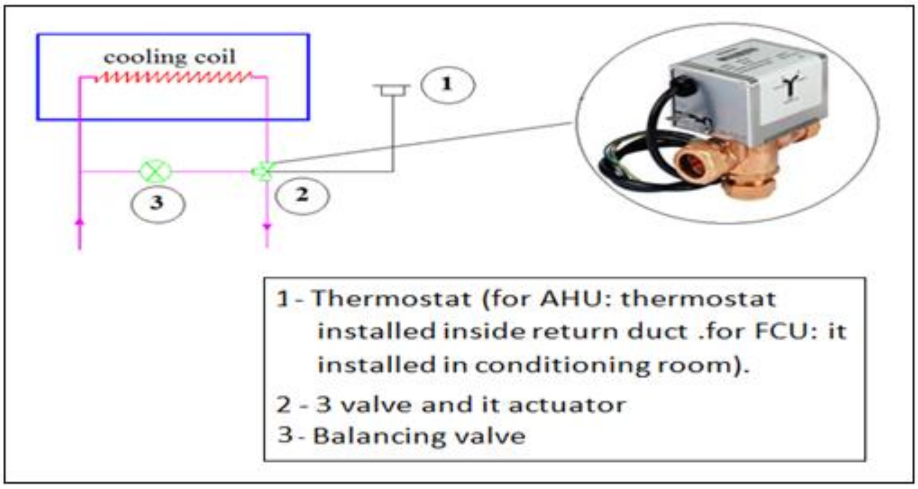


Fig (58). Three way valve

✓ **Two way valve.**

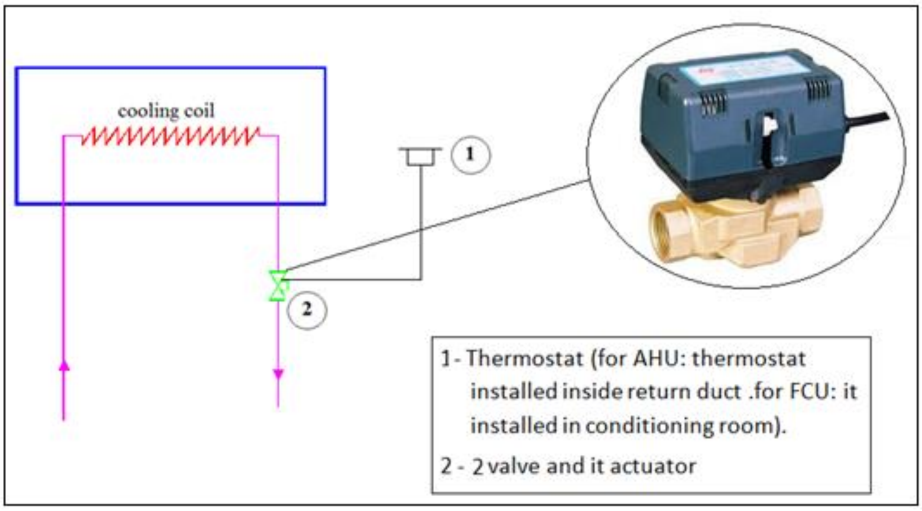


Fig (59).two way valve.

- 2 way valve make resistance for flowing flow to the equipment .its allow the required amount of flow to path through the cooling coil according to the change in thermal load ,but this causes problem which the extra flow return again to the pipes network ,and lead to increase the pressure of network .this problems can solve two methods.

- Using variable speed pumps: The variable speed pumps regulate amount of flow pumped to the network by regulating its revolution per minute according to according to the change in thermal load of each zone which work to decrease the network pressure as shown in the following figure. But the variable speed pumps expensive, and need periodic maintenance (**FLOW RATE**$\propto$ $\boldsymbol{RPM}$) as shown in figure (60).

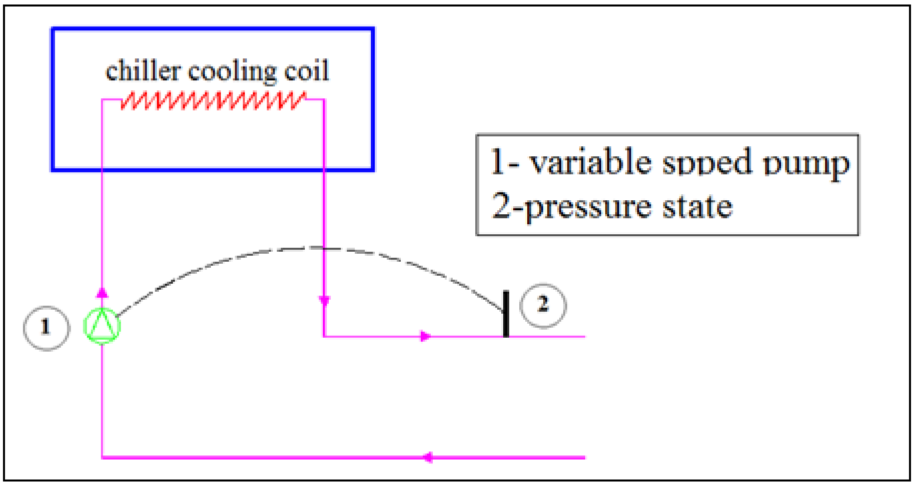


Fig (60).first solution for overpressure network

- Using constant speed pumps as shown in figure (61).

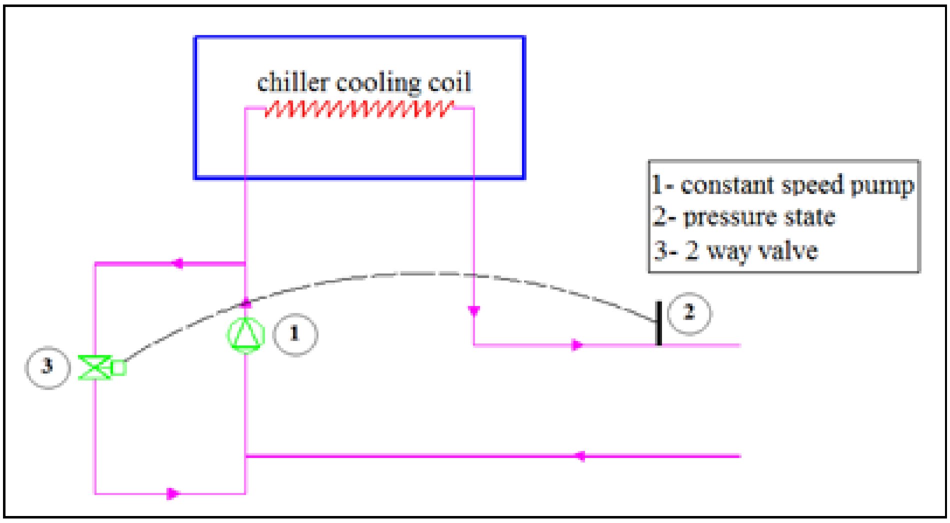


Fig (61).second solution for overpressure network


SESSION END

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FROM SCIENCE TO SOLUTION