Northern Technical University

Technical College of Engineering

Department of Refrigeration and Air Conditioning

Fourth Year

Air Conditioning System Design

Chapter Six

Psychometric Process and Advanced Applications

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Chapter Six
Psychometric Process and Advanced applications

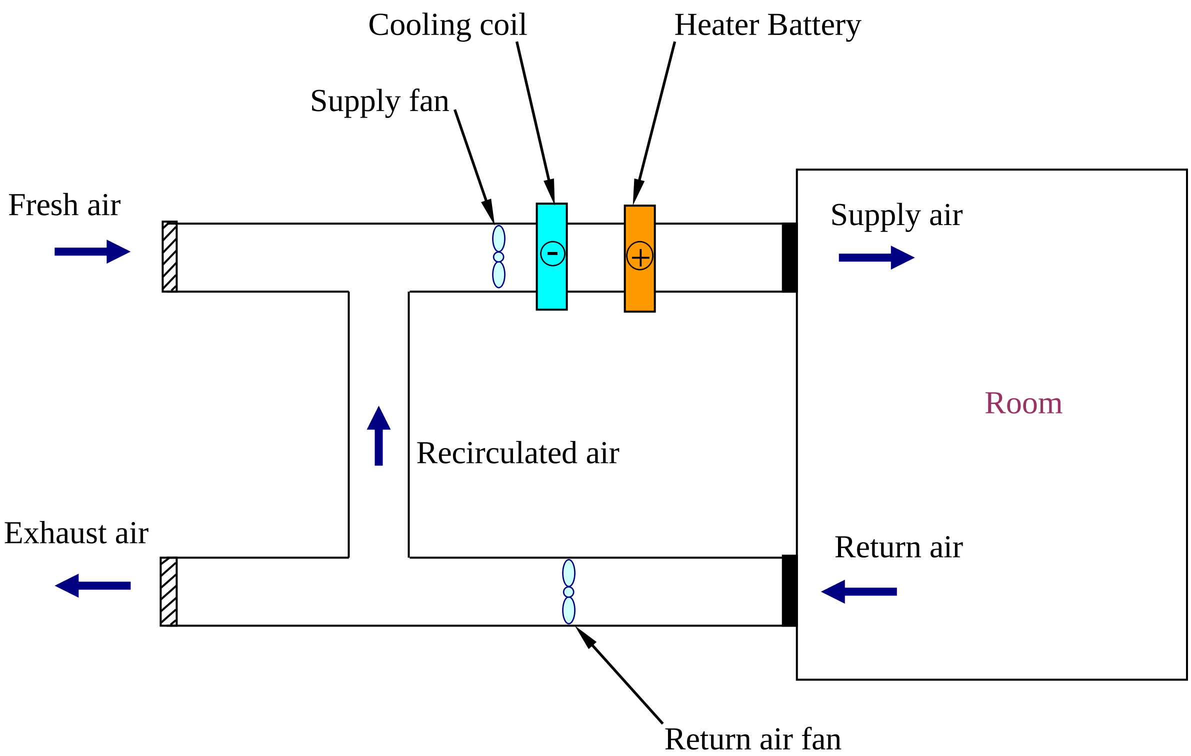
1. **Class** : Fourth Year
2. **Subject** : Psychometric Process and Advanced applications
3. **Number of weeks:** Five weeks

4. Central idea : Study the properties of air that entering to A/C system so as select correct A/C system to meeting the condition of human comfortable

5. **The Test & Problems** :

Central Plant Systems

A typical central plant air conditioning system is shown below.



Schematic Diagram of Central Plant Air Conditioning System

In winter the heater battery will be on and the cooling coil will probably be switched off for the majority of buildings.

In summer the heater battery will not need to have the same output and the cooling coil will be switched on.

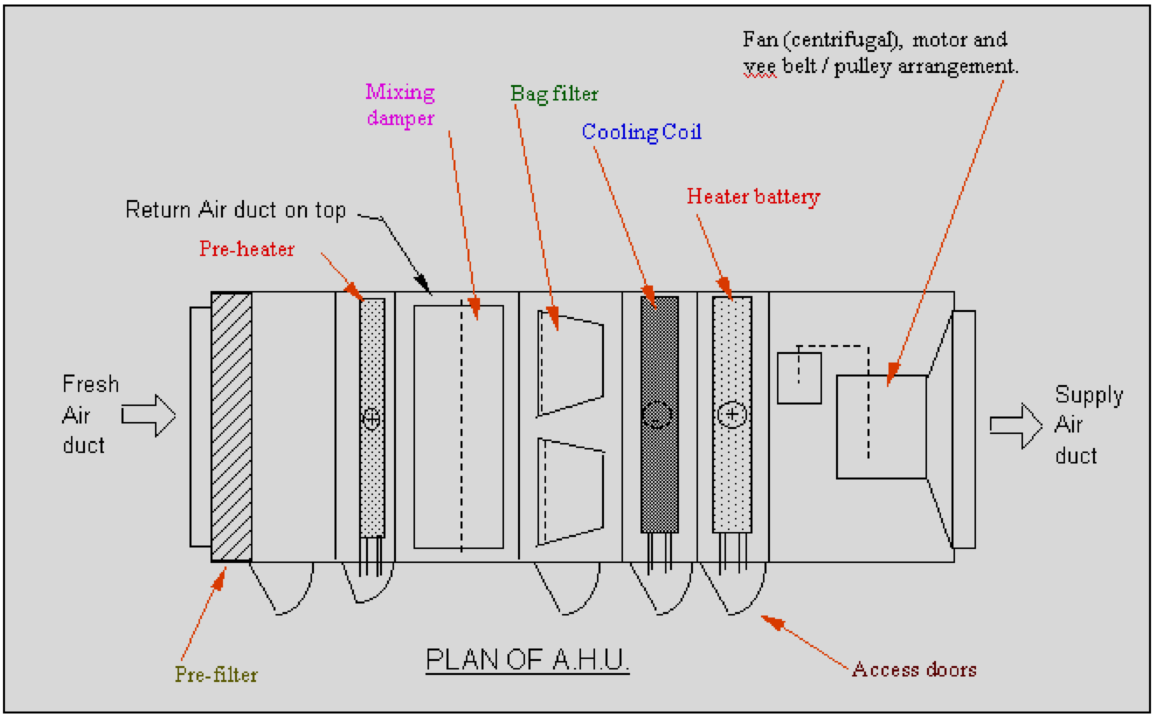
A **humidifier** may be required to add moisture to the air when it is 'dry'. This is when outdoor air has a low humidity of around **20% to 30%.**
In the U.K. low humidities are rare and therefore humidification is sometimes not used.
In dryer regions humidification is required through most of the year whereas in tropical air conditioning one of the main features of the system is the ability to remove moisture from warm moist air.

Dampers are used in air conditioning central plant systems to control the amount of air in each duct.
It is common to have 20% fresh air and 80% recirculate air to buildings.
In buildings with high occupancy the fresh air quantity should be calculated based on C.I.B.S.E. data., this may require a higher percentage of fresh air (i.e. more than 20%).
See Ventilation section for examples of fresh air rates.

Filters are required to remove particles of dust and general outdoor pollution.
This filter is sometimes called a coarse filter or pre-filter.
A removable fibreglass **dust** filter is positioned in the fresh air intake duct or in larger installation an oil filled viscous filter may be used.
The **secondary** filter, after the mix point, is used to remove fine dust particles or other contaminant picked up in the rooms and recirculate back into the plant. A removable **bag filter** is generally used for this where a series of woven fibre bags are secured to a framework which can be slid out of the ductwork or air handling unit (A.H.U.) for replacement.

Air Handling Units

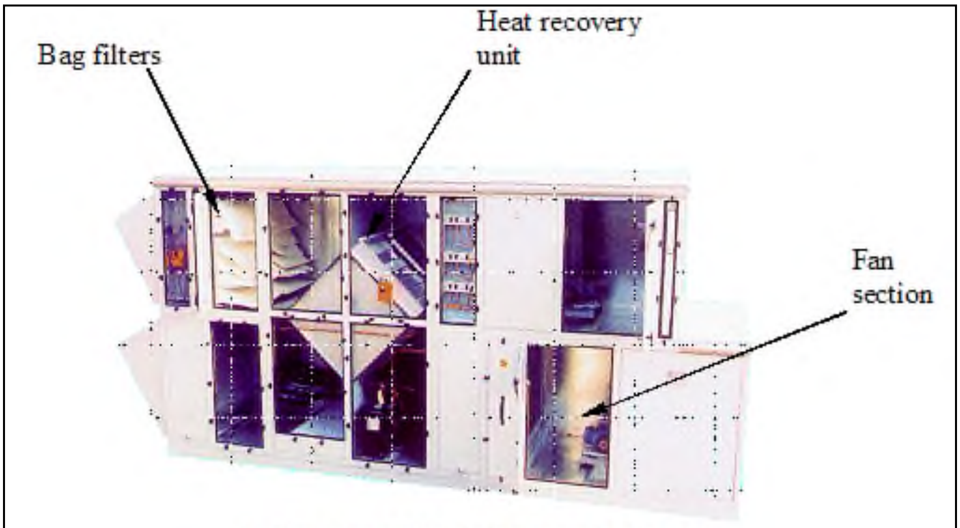
Air handling units (**A.H.U.**) are widely used as a package unit which incorporates all the main plant items as shown below.
Pipework, ductwork and electrical connections are made after the unit is set in place on site.
Since air conditioning plant rooms tend to be at roof level, the larger **A.H.U.'s** are lifted into place by crane before the roof is fixed.



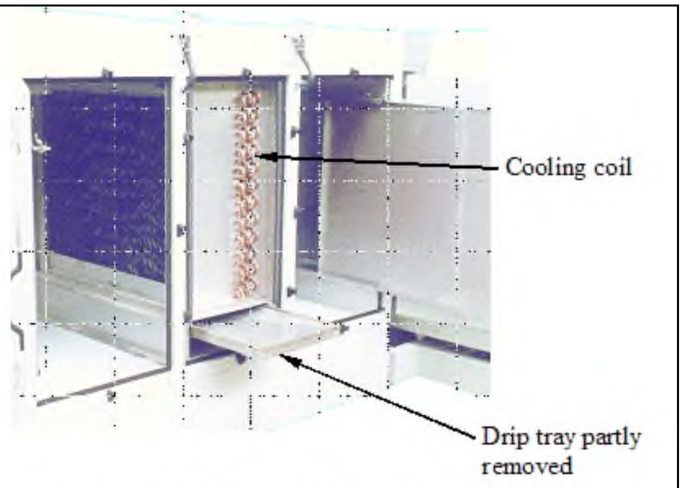
In some cases it is usual to place the fan in front of (that is upstream of) the heater battery and cooling coil.
This is because fans operate best if the system resistance is at the outlet rather than the inlet of the impeller.
This is shown on the schematic diagrams above.

The photograph below shows a typical air handling unit with handles on the doors for access to equipment.

TYPICAL AIR HANDLING UNIT



AIR HANDLING UNIT WITH DOORS REMOVED



AIR HANDLING UNIT COOLING COIL SECTION

Introduction

The aim of this section of the notes is to allow students to size air conditioning plant such as;

- **Cooling coil**
- **Heater battery**
- **Humidifier**

The first section deals with Psychrometry for air conditioning and discusses some properties of moist air. A simplified psychometric chart is shown for familiarization, and some examples of how to find air properties are provided.

A diagram of an air conditioning system is shown in schematic form in the section entitled AIR CONDITIONING PLANT FOR SUMMER & WINTER.

Before sizing takes place the student should also understand the processes that take place in air conditioning systems.
There are four basic processes for summer and winter air conditioning systems.
The following basic processes are explained:

1. Mixing
2. Sensible Cooling and Heating
3. Cooling with Dehumidification
4. Humidification

These are as detailed in the following sections of the notes.

Summer and Winter Cycles

1. Summer cycle psychrometrics
2. Summer cycle calculations
3. Winter cycle psychrometrics
4. Winter cycle calculations
5. Duct and Fan gains.

The final section is seven examples of plant sizing using psychometric charts.

Psychrometry for Air Conditioning

Psychrometry is the study of air and water vapour mixtures.
Air is made up of **five** main gases i.e.

Nitrogen 78.03%
Oxygen 20.99%
Argon 0.94%
Carbon Dioxide 0.03% and Hydrogen 0.01% by volume.
The Ideal Gas Laws are used to determine psychrometric data for air so that the engineer can carry out calculations. Air at any state point can be plotted on the psychrometric chart.

The information that can be obtained from a Psychrometric Chart is as follows:

1. Dry bulb temperature

2. Wet bulb temperature
3. Moisture content
4. Percentage saturation
5. Specific enthalpy
6. Specific volume.

The following is a brief description of each of the properties of air.

1. Dry bulb temperature

This is the air temperature measured by a mercury-in-glass thermometer.

2. Wet bulb temperature

This is the air temperature measured by a mercury-in-glass thermometer which has the mercury bulb wetted by gauze that is kept moist by a reservoir of water.

When exposed to the environment the moisture evaporates from the wetted gauze, which gives a lower reading on the thermometer.

This gives an indication of how 'dry' or how 'moist' the air is, since in 'dry' air the water will evaporate quickly from the gauze, which depresses the thermometer reading.

3. Moisture content

This is the amount of moisture in air given in kg of moisture per kg of dry air e.g. for room air **at 21ºC dry bulb and 15ºC wet bulb**, the moisture content is about 0.008 kg/kg d.a.

This is a small mass of moisture (0.008 kg = 8 grams) per kg of dry air or 9.5 grams per cubic metre of air.

4. Percentage saturation (Relative Humidity)

The Percentage saturation is another indication of the amount of moisture in air.

This is the ratio of the moisture content of moist air to the moisture content of saturated air at the same temperature.

When air is saturated it is at 100% saturation and cannot hold any more moisture.

5. Specific enthalpy

This is the amount of heat energy (kJ) in air per kg.

If heat is added to the air at a heater battery for example, then the amount to be added can be determined from Specific enthalpy change.

6. Specific volume

This is the volume of moist air (dry air + water vapor) per unit mass.

The units of measurement are m^3 per kg.

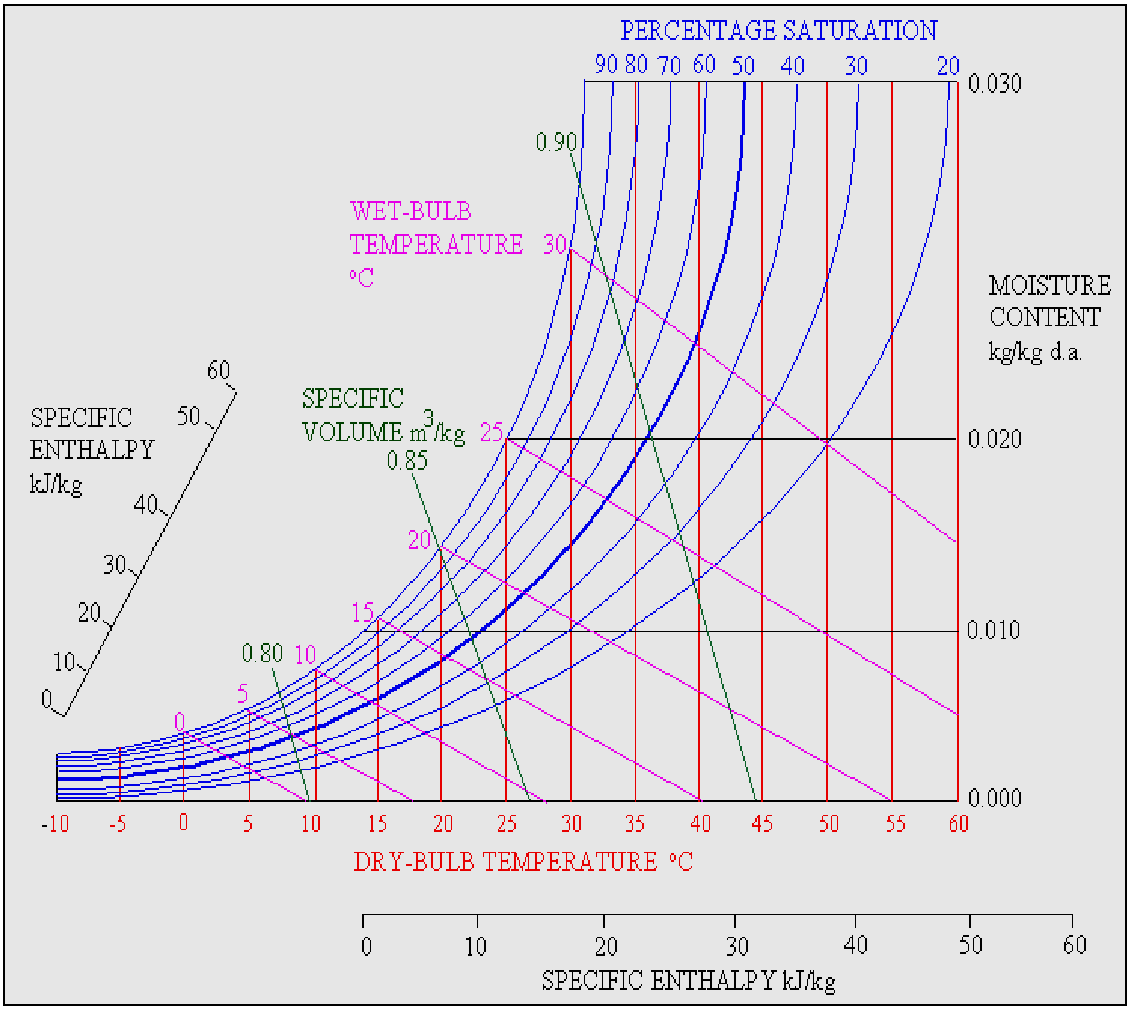
Also specific volume = 1 / density.

The Psychrometric Chart

The six properties of air previously discussed can be shown on one chart called a Psychrometric Chart.

One of the purposes of the Psychrometric Chart is to size heater batteries, cooling coils and humidifiers.

A simplified Psychrometric Chart is shown below.



This chart is only for demonstration purposes.

If accurate assessments are to be carried out use a C.I.B.S.E. chart.

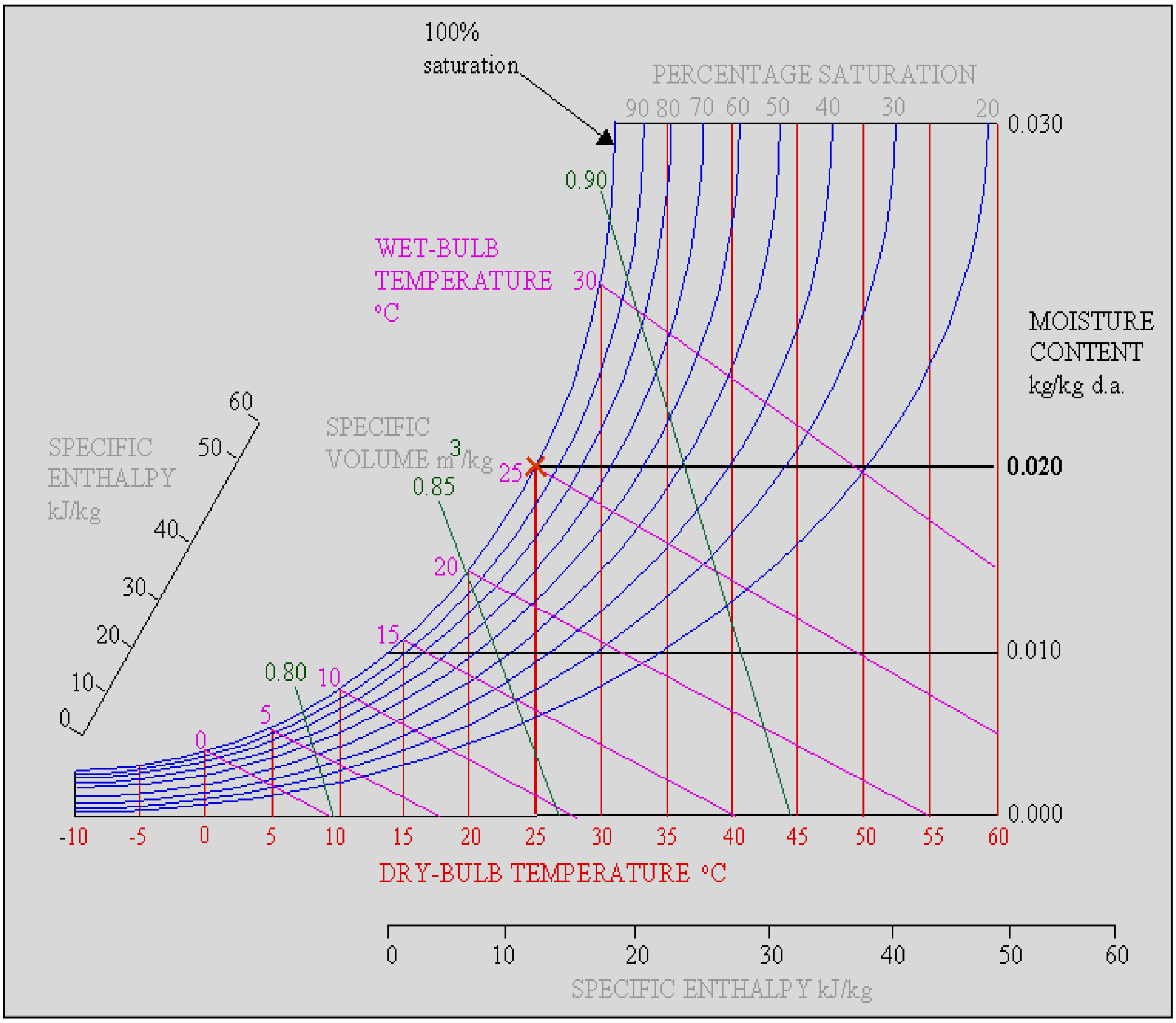
Using the Psychrometric Chart

If any two properties of air are known then the other four can be found from the psychrometric chart.

Examples of Psychrometric Properties

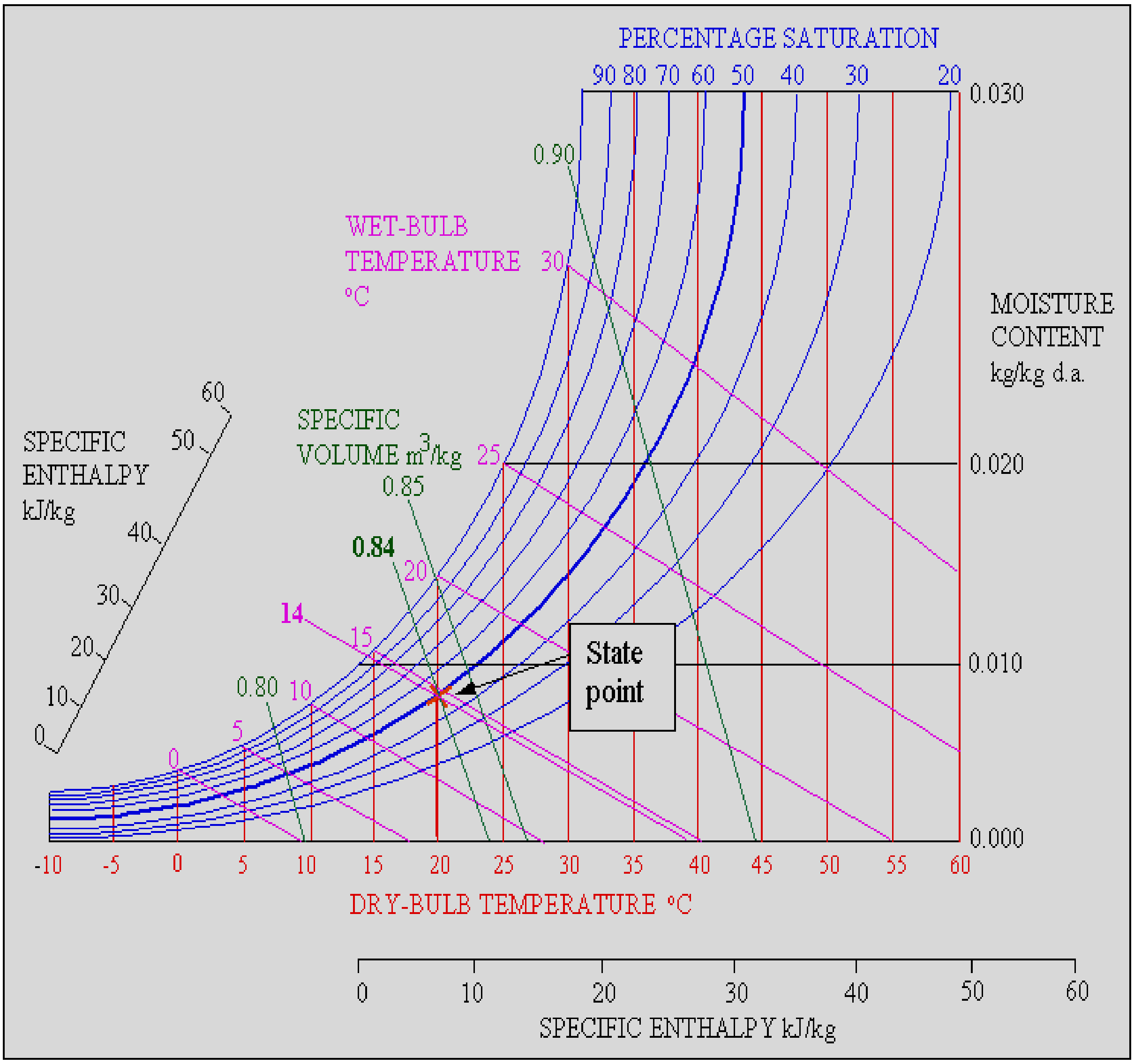
DIRECT EXAMPLE 1

Find the moisture content of air at 25ºC dry-bulb temperature and 25ºC wet-bulb temperature.
Referring to the chart below, a vertical line is drawn upwards from 25ºC dry-bulb temperature until it intersects at 25ºC wet-bulb temperature.
This intersection point happens to be on the 100% saturation line.
The intersection point is highlighted and a horizontal line is drawn to the right to find the corresponding moisture content.
The moisture content is therefore 0.020 kg/kg dry air.



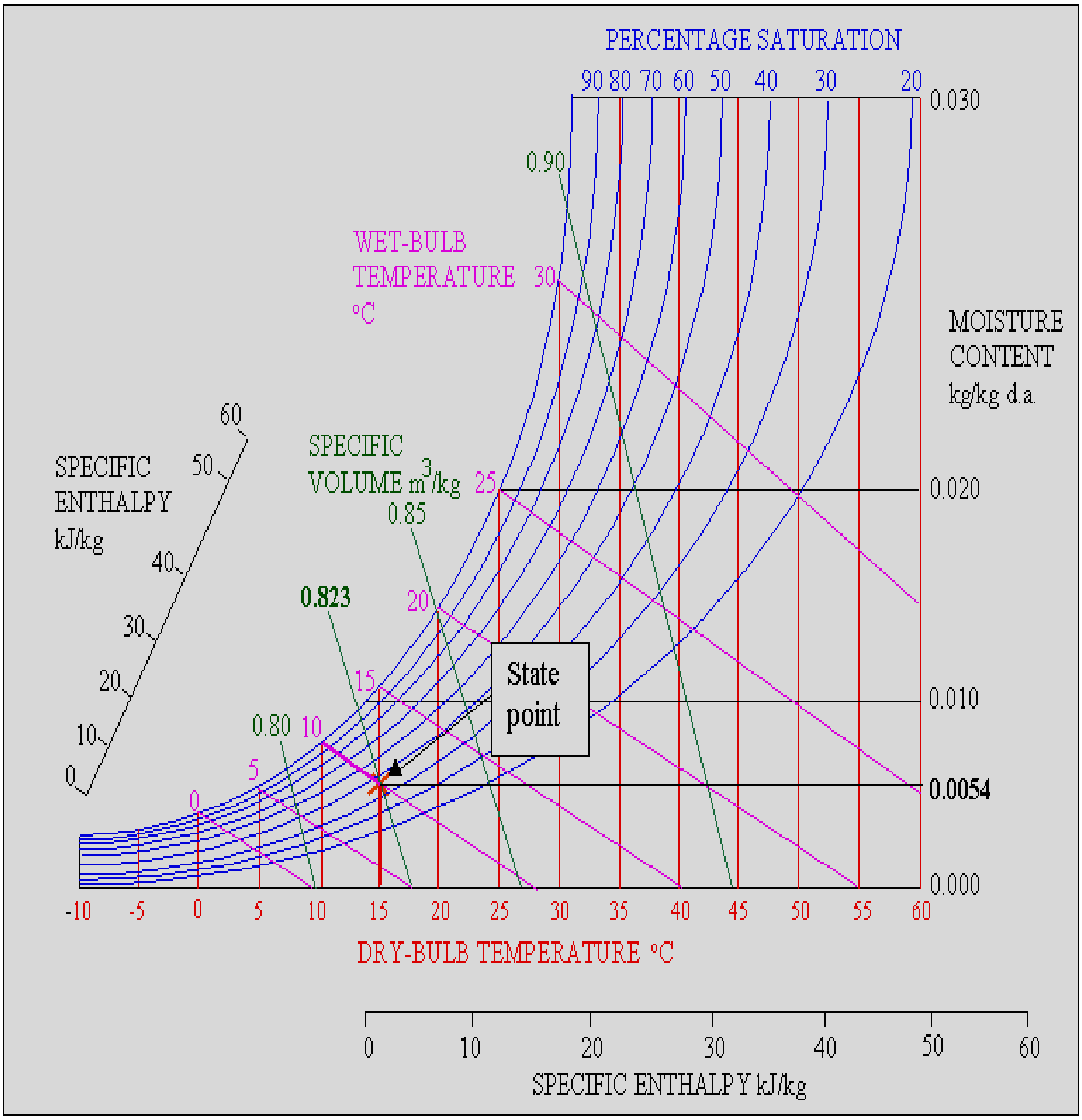
DIRECT EXAMPLE 2

Find the specific volume and wet-bulb temperature of air at 20ºC dry-bulb temperature and 50% saturation.
Referring to the chart below, a vertical line is drawn upwards from 20ºC dry-bulb temperature until it intersects with the 50% saturation curve.
The intersection point is sometimes referred to as the state point.
The specific volume is found to be 0.84 m^3/kg and the wet-bulb temperature is 14ºC



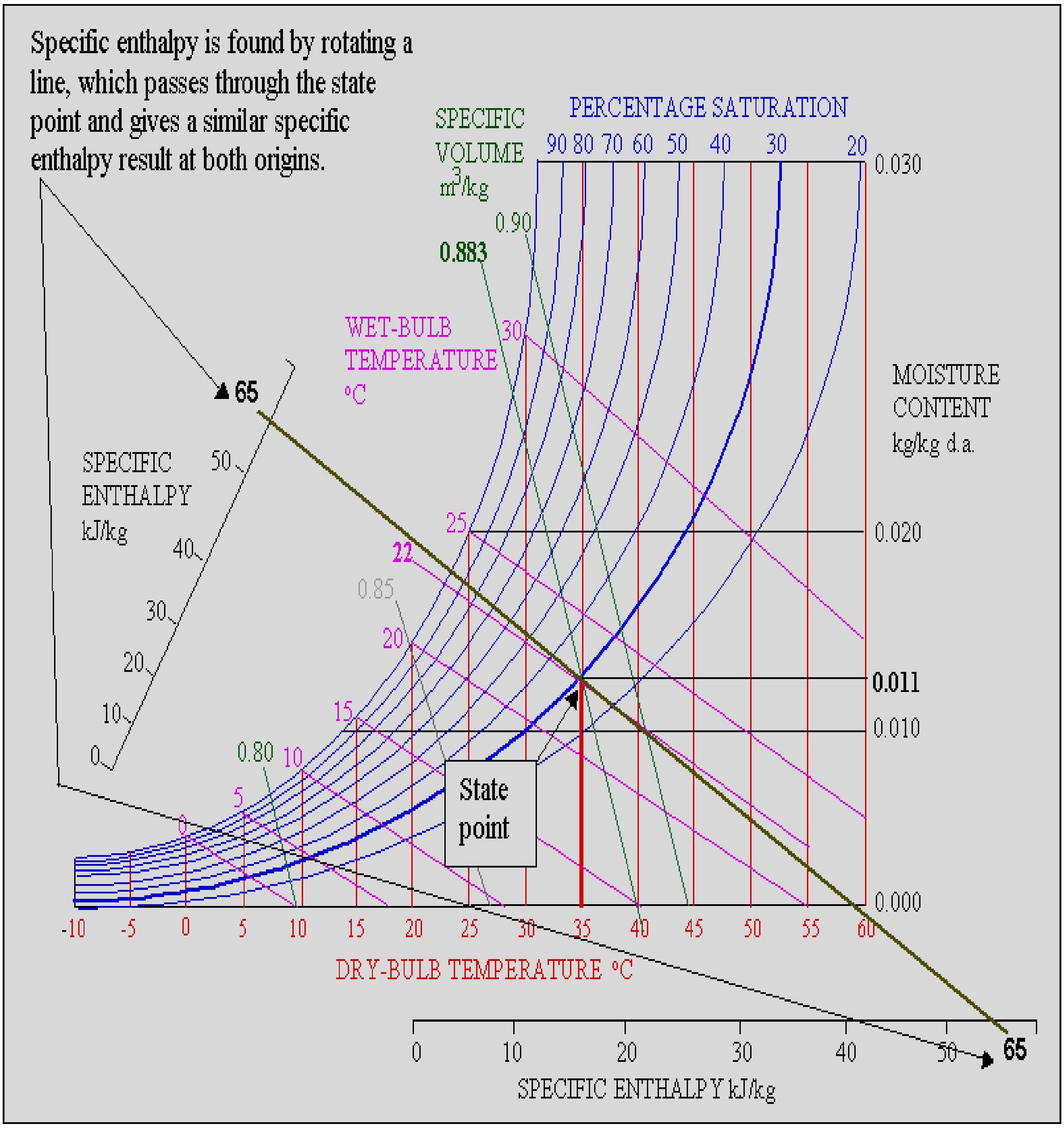
DIRECT EXAMPLE 3

Find the specific volume, percentage saturation and moisture content of air at 15°C dry-bulb temperature and 10°C wet-bulb temperature.
Referring to the chart below, a vertical line is drawn upwards from 15°C dry-bulb temperature until it intersects with the 10°C wet-bulb temperature line. This intersection is the state point.
The specific volume is found to be 0.823 m^3/kg, the percentage saturation 52% and the moisture content 0.0054 kg/kg d.a.



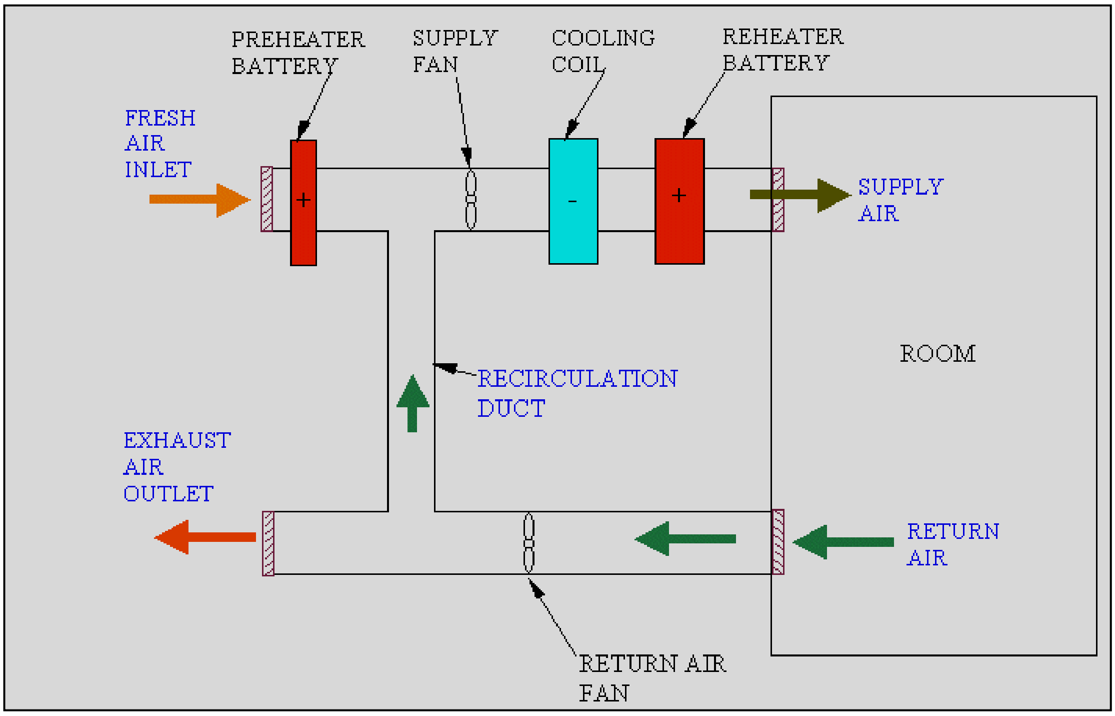
DIRECT EXAMPLE 4

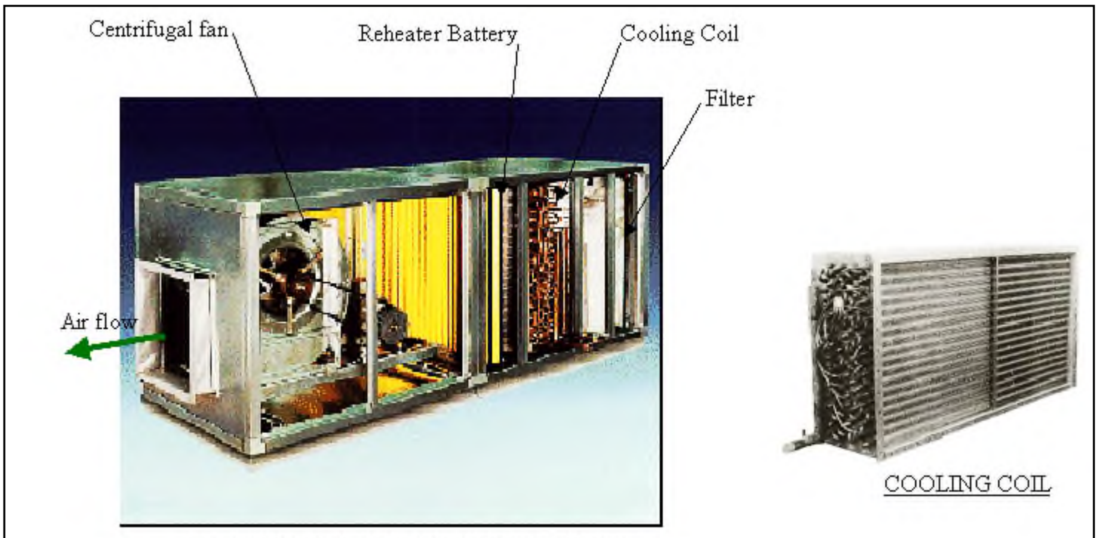
Find the specific volume, wet-bulb temperature, moisture content and specific enthalpy of air at 35ºC dry-bulb temperature and 30% saturation.
Referring to the chart below, a vertical line is drawn upwards from 35ºC dry-bulb temperature until it intersects with the 30% saturation curve.
This intersection is the state point.
The specific volume is found to be 0.883 m^3/kg, the wet-bulb temperature is 22ºC, the moisture content 0.011kg/kg d.a. and the specific enthalpy 65 kJ/kg.



Air Conditioning Plant for Summer & Winter:

In the summer time when cooling is required by the air conditioning plant it will be necessary to operate the cooling coil, reheater and possibly other plant as well.In winter time the preheater and reheater battery will probably be on to provide warm air to overcome heat losses. Other plant may be switched on as well. These plant items are shown in the diagram below.





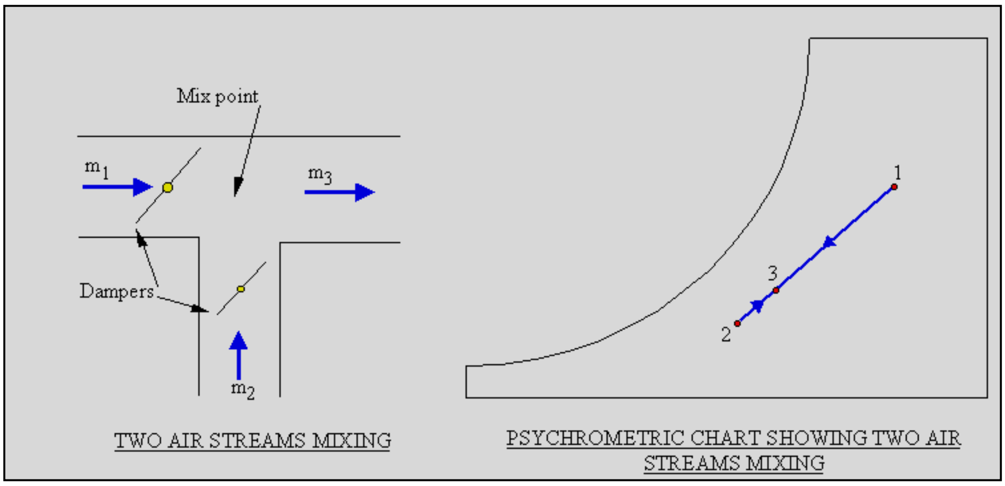
AIR HANDLING UNIT WITH FAN, FILTER, COOLING COIL AND REHEATER.

Basic Air Conditioning Processes

1. Mixing

Where two air streams are mixed the psychrometric process is shown as a straight line between two air conditions on the psychrometric chart, thus points 1 and 2 are joined and the mix point 3 will lie on this line.
Two air streams are mixed in air conditioning when fresh air (m_1) is brought in from outside and mixed with recirculated air (m_2).
The resulting air mixture is shown below as (m_3).

The mixing ratio is fixed by dampers.
Sometimes, in more sophisticated plant, modulating dampers are used which are driven by electric motors to control the mixture of air entering the system.The diagrams below show mixing of two air streams.



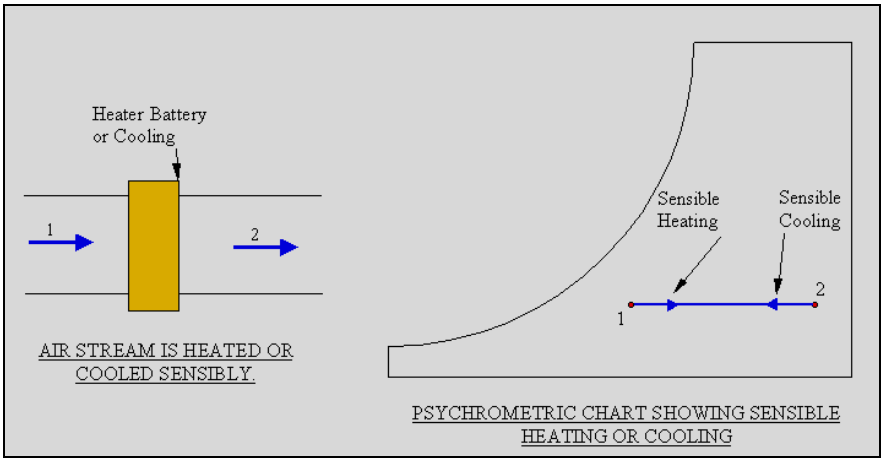
By the conservation of mass formula: $m_1 + m_2 = m_3$

By the conservation of energy formula: $m_1 h_1 + m_2 h_2 = m_3 h_3$

where: m = mass flow rate of air (kg/s)
h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

2. Sensible Cooling and Heating

When air is heated or cooled sensibly, that is, when no moisture is added or removed, this process is represented by a horizontal line on a psychrometric chart.



For sensible heating:

The amount of heating input to the air approximates to; $H_{1\text{-}2} = m \times Cp \times (t_2 - t_1)$

Or more accurately from psychrometric chart: $H_{1\text{-}2} = m \times (h_2 - h_1)$

For sensible cooling:

The amount of cooling input to the air approximates to; $H_{2\text{-}1} = m \times Cp \times (t_2 - t_1)$

Or more accurately from psychrometric chart: $H_{2\text{-}1} = m \times (h_2 - h_1)$

where: H = Heat or cooling energy (kW)
m = mass flow rate of air (kg/s)
Cp = Specific heat capacity of air, may be taken as 1.01 kJ/kg degC.
t = Dry bulb temperature of air (°C)
h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

3. Cooling with Dehumidification

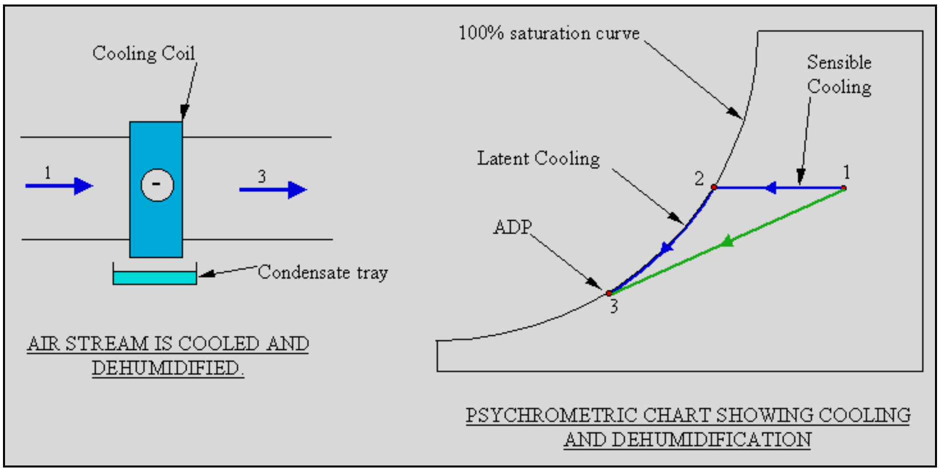
The most commonly used method of removing water vapour from air (Dehumidification) is to cool the air below its dew point.The dew point of air is when it is fully saturated i.e. at 100% saturation.
When air is fully saturated it cannot hold any more moisture in the form of water vapor.If the air is cooled to the dew point air and is still further cooled then moisture will drop out of the air in the form of condensate.

This can be shown on a psychrometric chart as air sensibly cooled until it becomes fully saturated (the dew point is reached) and then the air is cooled latently to a lower temperature.
This is apparent on the psychrometric chart as a horizontal line for sensible cooling to the 100% saturation curve and then the process follows the 100% saturation curve down to another point at a lower temperature.

This lower temperature is sometimes called the **Apparatus dew Point (ADP)** of the cooling coil. In reality the **ADP** of the cooling coil is close to the cooling liquid temperature inside the coil.

Chilled water or refrigerant may be the cooling liquid.

The psychrometric process from state point 1 to 2 to 3 may be shown as a straight line for simplicity as shown above with a yellow line.



The total amount of cooling input to the air approximates to;

$$H_{1\text{-}3} = m \times (h_1 - h_3)$$

The sensible heat removed is:

$$H_{1\text{-}2} = m \times (h_1 - h_2)$$

The latent heat removed is:

$$H_{2\text{-}3} = m \times (h_2 - h_3)$$

where: H = Cooling energy (kW)
m = mass flow rate of air (kg/s)
h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

In the absence of a suitable psychrometric chart the following formula may be used;

The sensible heat removed is: $H_{1\text{-}2} = m \times Cp \times (t_1 - t_2)$

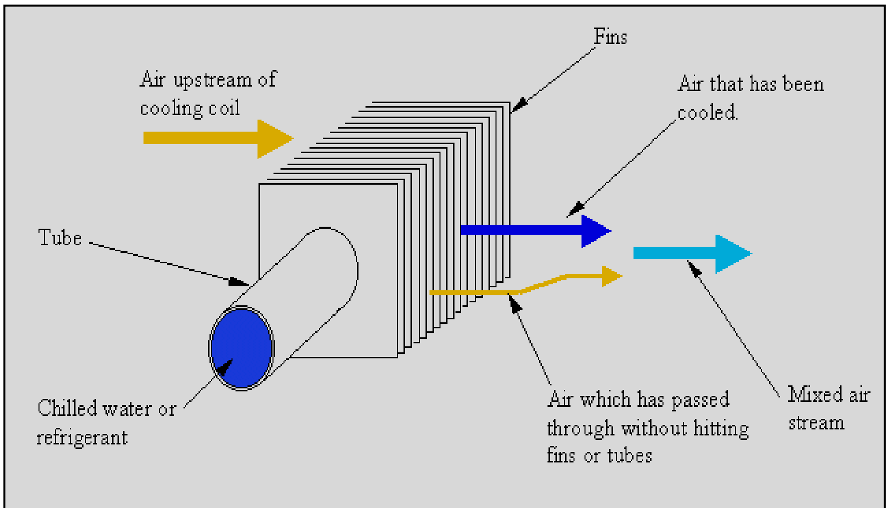
The latent heat removed is: $H_{2\text{-}3} = m \times hfg \times (g_2 - g_3)$

where: H = Cooling energy (kW)
m = mass flow rate of air (kg/s)
Cp = Specific heat capacity of air, may be taken as 1.01 kJ/kg degC.

t = Dry bulb temperature of air (°C)
hfg = latent heat of evaporisation, may be taken as 2454 kJ/kg @20°C.
g = moisture content of air from psychrometric chart (kg/kg dry air)

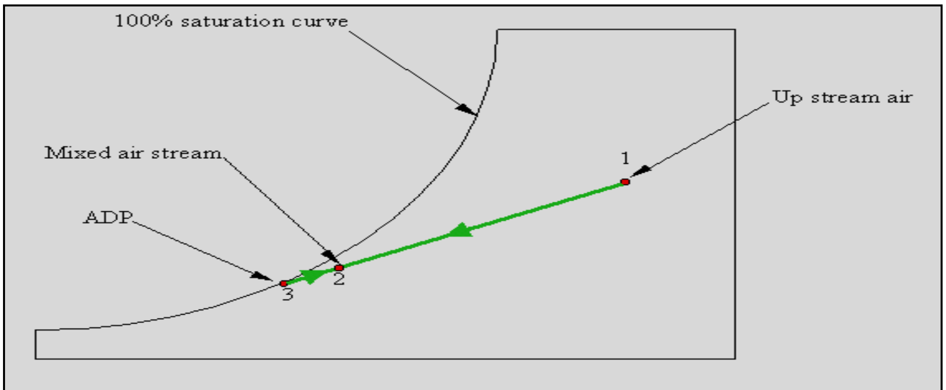
3.1 Cooling Coil Contact Factor:

Some of the air going through a cooling coil does not come into contact with the tubes or fins of the cooling coil and is therefore not cooled to the ADP temperature.A mixing process therefore takes place as two air streams mix downstream of the cooling coil as shown below.



A SECTION OF COOLING COIL SHOWING AIR STREAMS

One air stream is cooled down to the ADP and the other air stream by-passes the coil surfaces to give an off-coil air temperature (mixed air stream) a little higher than the ADP.This may be looked upon as an inefficiency of the coil and is usually given as the cooling coil contact factor.The process is shown on the psychrometric chart below.



PSYCHROMETRIC CHART SHOWING COOLING COIL CONTACT FACTOR

The contact factor of a cooling coil may be found from;

$$\text{Contact Factor} = \frac{(h_1 - h_2)}{(h_1 - h_3)}$$

Another expression for contact factor is;

$$\text{Contact Factor} = \frac{\text{Distance 1 to 2 (mm)}}{\text{Distance 1 to 3 (mm)}}$$

4. Humidification

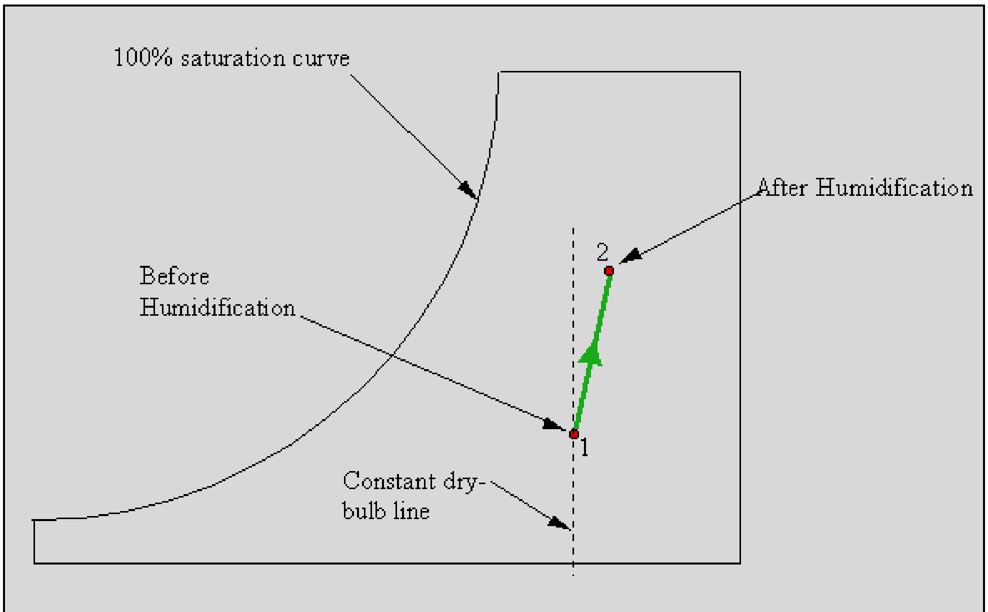
If is it necessary to add some moisture to the supply air then this is best done by injecting steam into the air stream.

Humidification can be carried out by spraying a fine mist of water droplets into the air but this is not recommended in rooms occupied by people due to the risk of bacteria carry over.

Dry steam may be injected from a steam supply pipe or generated in a local packaged unit as shown in the photograph below. A disadvantage of using an existing steam supply is smells may be carried over into the air.

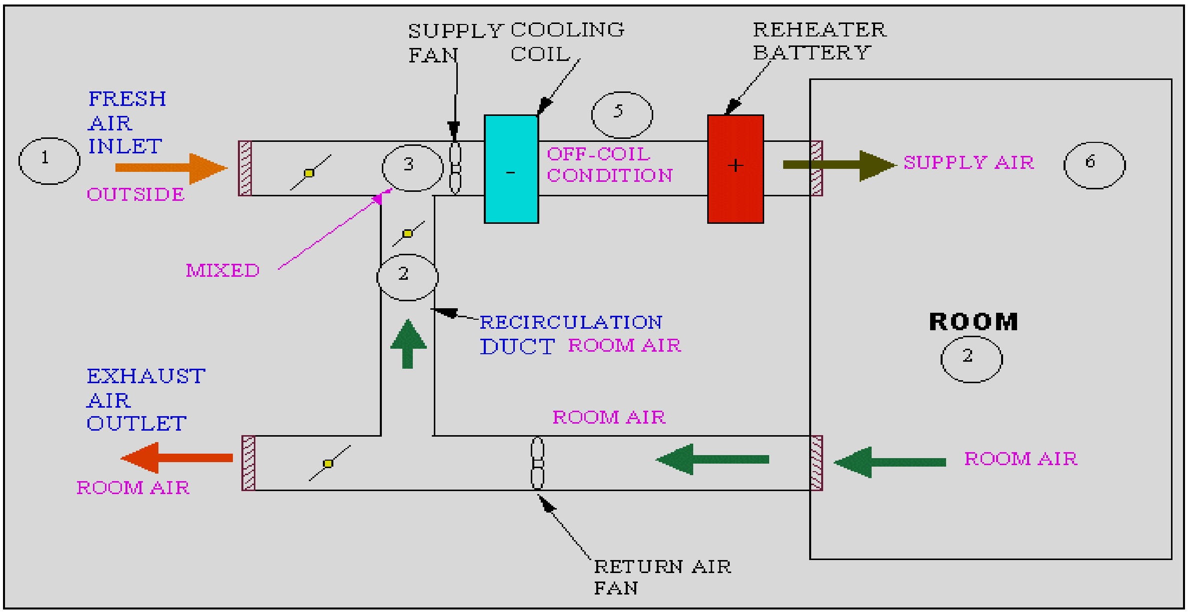
The steam package unit is situated close to the air duct and is sized to meet the maximum requirements; this is usually in winter in the U.K.A steam pipe (sometimes hoses are used) passes from the packaged unit to the air duct and steam at 100°C is injected into the air stream via. a sparge pipe. The un-used steam is drained from the system via a condensate tundish and drain. It is important to layout the steam pipework so that any condensate will drain back to the unit.

The psychrometric process is shown below.

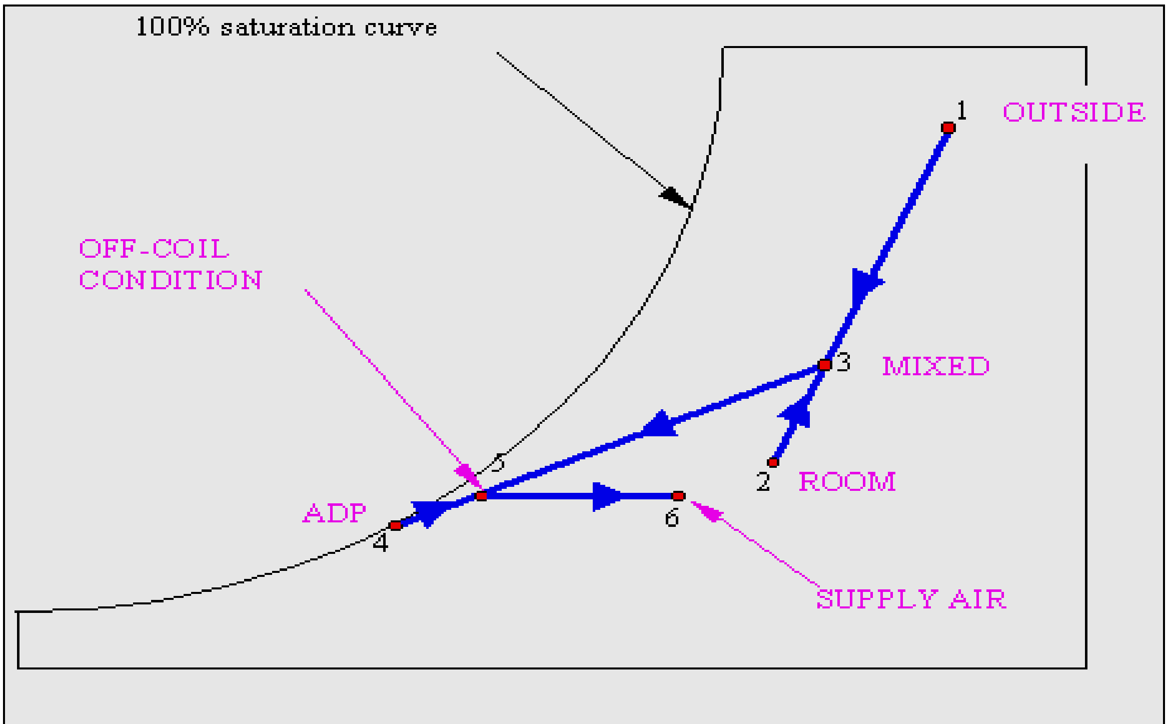


PSYCHROMETRIC CHART SHOWING HUMIDIFICATION

The schematic diagram below shows a typical plant system for **summer air** conditioning.

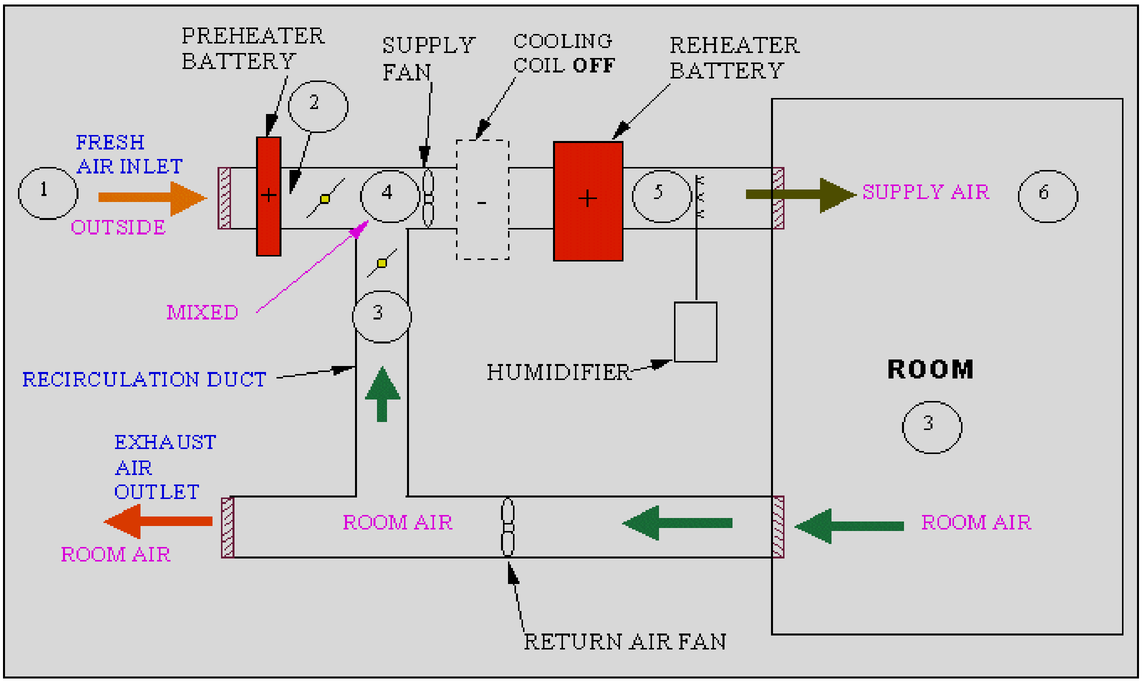


The psychrometric diagram below shows a typical **summer cycle.**

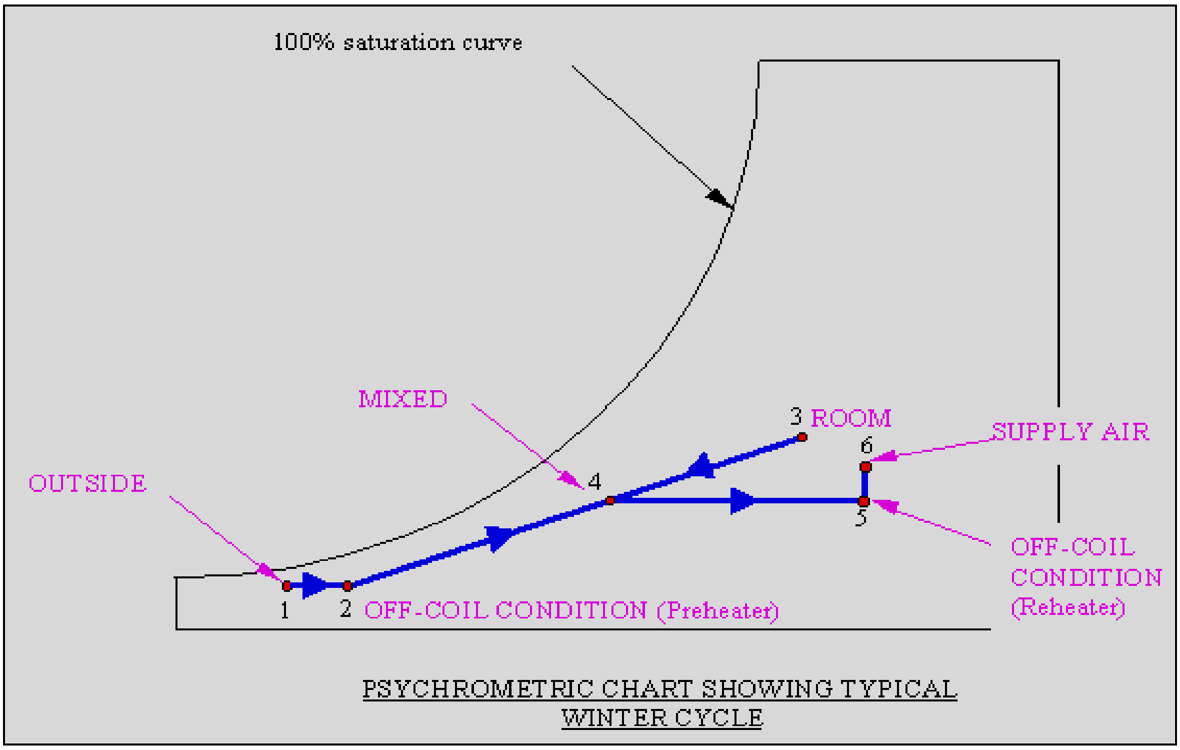


PSYCHROMETRIC CHART SHOWING TYPICAL SUMMER CYCLE

The schematic diagram below shows a typical plant system for **winter air** conditioning.



The psychrometric diagram below shows a typical **winter cycle.**



Annotation

The state points on a psychrometric chart may be given numbers or symbols to identify them. If symbols are used the following system may be adopted:

Air State point	Letter
Outside	O
Room	R
Mixed	M
Apparatus Dew Point	ADP
Off cooling coil condition	W
Room Ratio Line	RRL
Preheater off coil condition	P
Upstream of Humidifier	H
Supply	S
Duct, fan gain allowance	D

Room Ratio

This is the ratio of sensible to total heat in the room for summer or winter. The total heat gain (summer) or loss (winter) will be determined by adding the Latent and Sensible heat in a room or rooms, i.e.

(SUMMER) Total heat gain= Sensible heat gain + Latent heat gain

(WINTER) Total heat =Sensible heat loss + Latent heat gain

The room ratio is used on a psychrometric chart to determine the supply air state point.

A room ratio line is superimposed from the protractor on the psychrometric chart onto the main body of the chart by a line passing through the room state point R.

An example calculation is as follows:

Sensible heat gain =9.0 kW

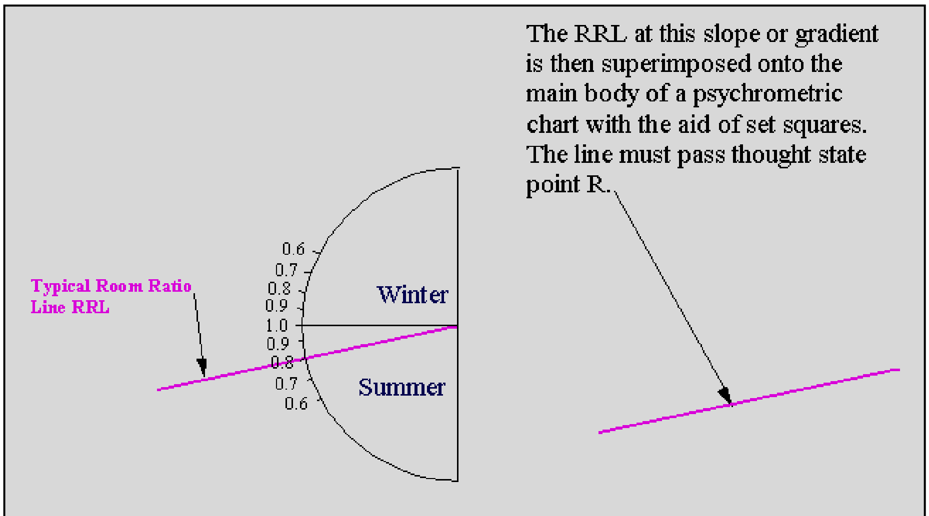
Latent heat gain=2.25 kW

Total heat gain=9.0 kW + 2.25 kW=11.25 kW.

Room ratio =Sensible / Total heat

Room ratio =9 / 11.25= 0.8

The supply air state point must also be somewhere on this room ratio line to meet the room heat gain requirements i.e. the room ratio line always passes through points R and S.



HEAT RATIO PROTRACTOR ON PSYCHROMETRIC CHART

Summer and Winter Cycles

1. SUMMER CYCLE PSYCHROMETRICS

1. Draw schematic diagram of air-conditioning plant.
2. Plot room condition R on psychrometric chart.
3. Plot outside condition O on psychrometric chart.
4. Join points O and R.
5. Find the mix point M by measuring the length of the line O-R and multiply this by the mixing ratio.
   If there is more recirculated air than outside air at the mix point, then point M will be closer to point R than point O.

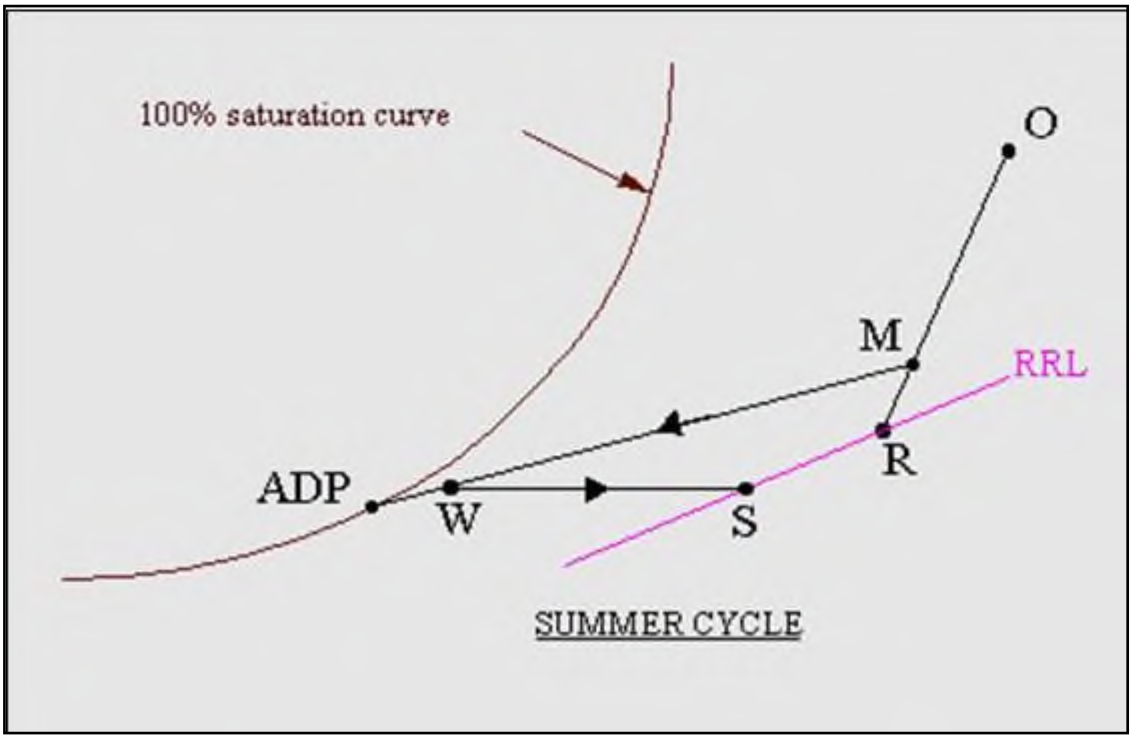
6. Find the room ratio.
   This is the sensible to total heat gain ratio.
   Plot this ratio on the protractor, bottom segment, on the psychrometric chart and transfer this line onto the chart so that it passes through point R.

7. Plot the Apparatus Dew Point ADP of the cooling coil.
   This is on the 100% saturation curve.
   The wet bulb and dry bulb temperatures at this point will be equal.

8. Join points M and ADP.

9. Find the off-coil condition W by measuring the length of the line M-ADP and multiply this by the cooling coil contact factor.
   Measure down from point M to the point W.
   The closer the contact factor is to unity, the closer point W will be to point ADP.

10. Plot the supply air condition S.
    The reheater process will be a horizontal line from point W to point S.
    Point S is on the room ratio line.



2. SUMMER CYCLE CALCULATIONS

2.1 MASS FLOW RATE

When the supply air temperature has been found from the psychrometric chart then the mass flow rate of air can be calculated from the following formula:

$$H_s = m_a \times Cp \; (t_r - t_s)$$

where:

H_s	=	Sensible heat gain to room (kW)
m_a	=	mass flow rate of air (kg/s)
Cp	=	Specific heat capacity of humid air (approx.1.01 kJ/kg degC)
t_r	=	room temperature (°C)
t_s	=	supply air temperature (°C)

2.2 COOLING COIL OUTPUT

The cooling coil output is as follows:

$$H_{cooling\ coil} = m_a \; (h_M - h_{ADP})$$

where:

$H_{cooling\ coil}$	=	Cooling coil output (kW)
m_a	=	mass flow rate of air (kg/s)
h_M	=	specific enthalpy at condition M (kJ/kg)
h_{ADP}	=	specific enthalpy at condition ADP (kJ/kg).

2.3 HEATER BATTERY OUTPUT

The heater battery or reheater output is as follows:

$$H_{heater\ battery} = m_a \; (h_S - h_W)$$

where:

$H_{heater\ battery}$	=	Heater battery output (kW)
m_a	=	mass flow rate of air (kg/s)
h_S	=	specific enthalpy at condition M (kJ/kg)
h_W	=	specific enthalpy at condition W (kJ/kg)

Summer and Winter Cycles

3. WINTER CYCLE PSYCHROMETRICS

1. Draw schematic diagram of air-conditioning plant.

2. Plot room condition R on psychrometric chart.

3. Plot outside condition O on psychrometric chart.

4. Plot the after Preheater condition P if there is one in the system.
   The Preheater process will be a horizontal line from O to P and will be only a few degrees dry bulb if it acts as a frost coil.

5. Join points P and R. If there is no frost coil read O-R for P-R.

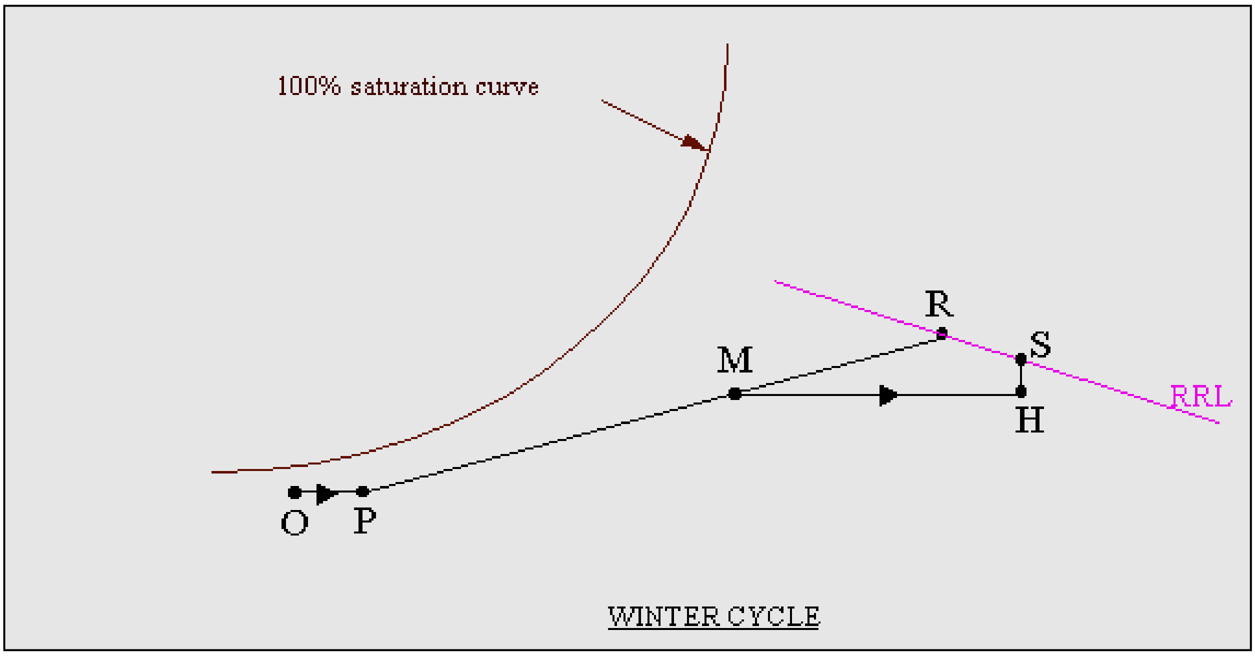
6. Find the mix point M by measuring the length of the line P-R and multiply this by the mixing ratio.
   If there is more recirculated air than outside air at the mix point, then point M will be closer to point R than point P.

7. Find the room ratio.
   This is the sensible to total heat ratio.
   Neglect signs ie. the total heat for the room will be Sensible loss plus Latent gain.
   Plot this ratio on the protractor, top segment, on the psychrometric chart and transfer this line onto the chart so that it passes through point R.

8. Find the supply air dry bulb temperature by calculation.
   The mass flow rate of air is the same as that for Summer for a Constant Volume system.

9. Plot the supply air condition S on the room ratio line.

10. Plot condition H on the psychrometric chart.
    This is vertically down from point S, and horizontally across from point M.
    This is because M-H is the reheater process and thus a horizontal line and H-S is the humidification process and is close to a vertical line if steam is used.



4. WINTER CYCLE CALCULATIONS

4.1 SUPPLY AIR DRY BULB TEMPERATURE

When the mass flow rate of air is calculated for the summer condition then the winter supply air dry bulb temperature can be calculated from the following formula:

$$H_s = m_a \times Cp \ (t_s - t_r)$$

where:

H_s = Sensible heat gain to room (kW)
m_a = mass flow rate of air (kg/s)
Cp = Specific heat capacity of humid air (approx.1.01 kJ/kg degC)
t_r = room temperature (°C)
t_s = supply air temperature (°C)

4.2 PREHEATER BATTERY OUTPUT (or frost coil)

The preheater battery output is as follows:

$$H_{preheater\ battery} = m_{af} \ (h_P - h_O)$$

where:

$H_{preheater\ battery}$ = Preheater battery output (kW)
m_{af} = mass flow rate of fresh air (kg/s)
h_P = specific enthalpy at condition P (kJ/kg)
h_O = specific enthalpy at condition O (kJ/kg)

4.3 REHEATER BATTERY OUTPUT

The reheater battery output is as follows:

$$H_{reheater\ battery} = m_a \ (h_H - h_M)$$

where:

$H_{reheater\ battery}$	=	Reheater battery output (kW)
m_a	=	mass flow rate of supply air (kg/s)
h_H	=	specific enthalpy at condition H (kJ/kg)
h_M	=	specific enthalpy at condition M (kJ/kg)

4.4 HUMIDIFIER OUTPUT

The amount of moisture added to the air may be calculated from the following formula:

$$m_{moisture\ added} = m_a \ (m_{sS} - m_{sH})$$

where:

$m_{moisture\ added}$	=	The amount of moisture or added or steam flow rate (kg/s)
m_a	=	mass flow rate of air (kg/s)
m_{sS}	=	moisture content at condition S (kg/kg d.a.)
m_{sH}	=	moisture content at condition H (kg/kg d.a.)

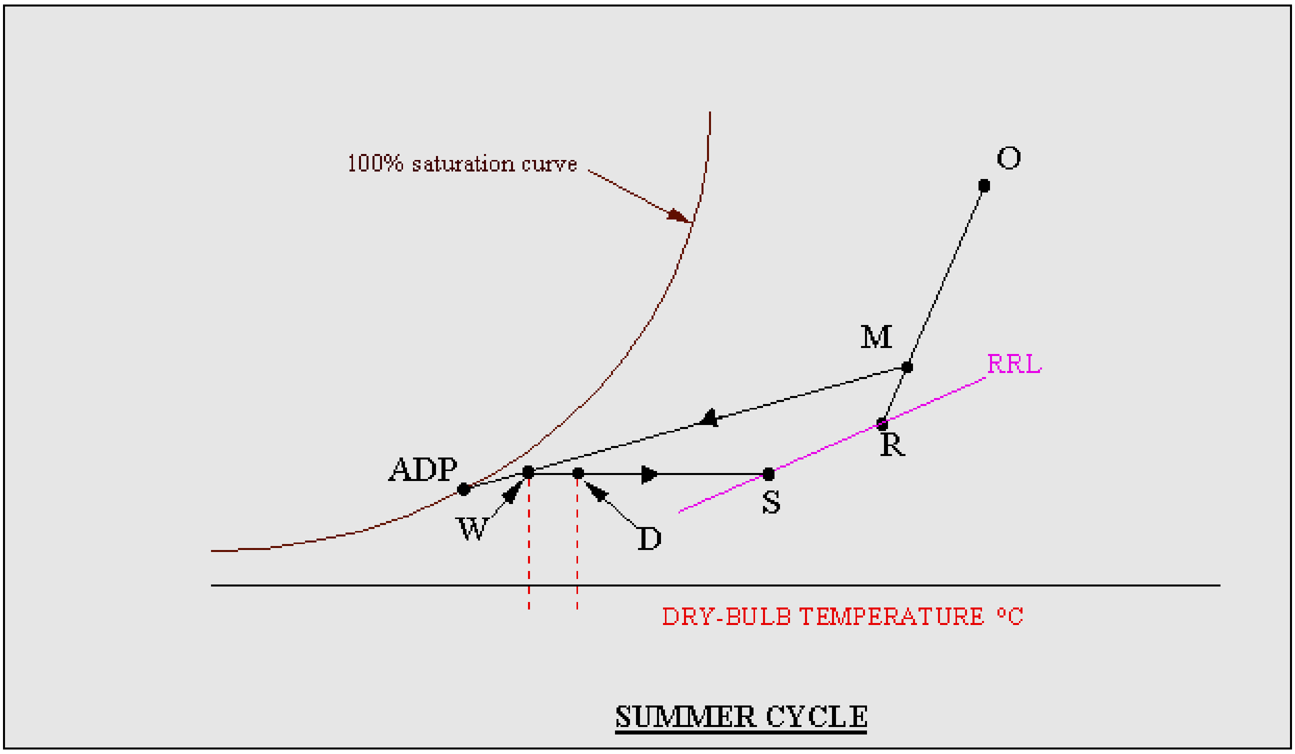
Summer and Winter Cycles

5. DUCT AND FAN GAINS

The air in a duct is slightly heated from the fan electric motor and heat is also transmitted through the duct wall from warm areas Into the air stream, for example;
a duct contains air at 15°C and passes through a roof space at 30°C in summer.
There will be heat transferred through the duct wall, which increases the air temperature slightly.

To allow for this in the summer psychrometric process an additional sensible heating state point D is added as shown below.

The air may be heated by several °C depending on the fan motor, length of duct and type of duct insulation used, if any. The distance from point W to point D may be typically 1°C to 3°C dry bulb temperature in the U.K.



If duct and fans gains are to be allowed for, the reheater battery output is as follows:

REHEATER BATTERY OUTPUT

$$H_{heater\ battery} = m_a \ (h_S - h_D)$$

where:

$H_{heater\ battery}$ = Heater battery output (kW)
m_a = mass flow rate of air (kg/s)
h_S = specific enthalpy at condition S (kJ/kg) determined from psychrometric chart.
h_D = specific enthalpy at condition D (kJ/kg) determined from psychrometric chart.

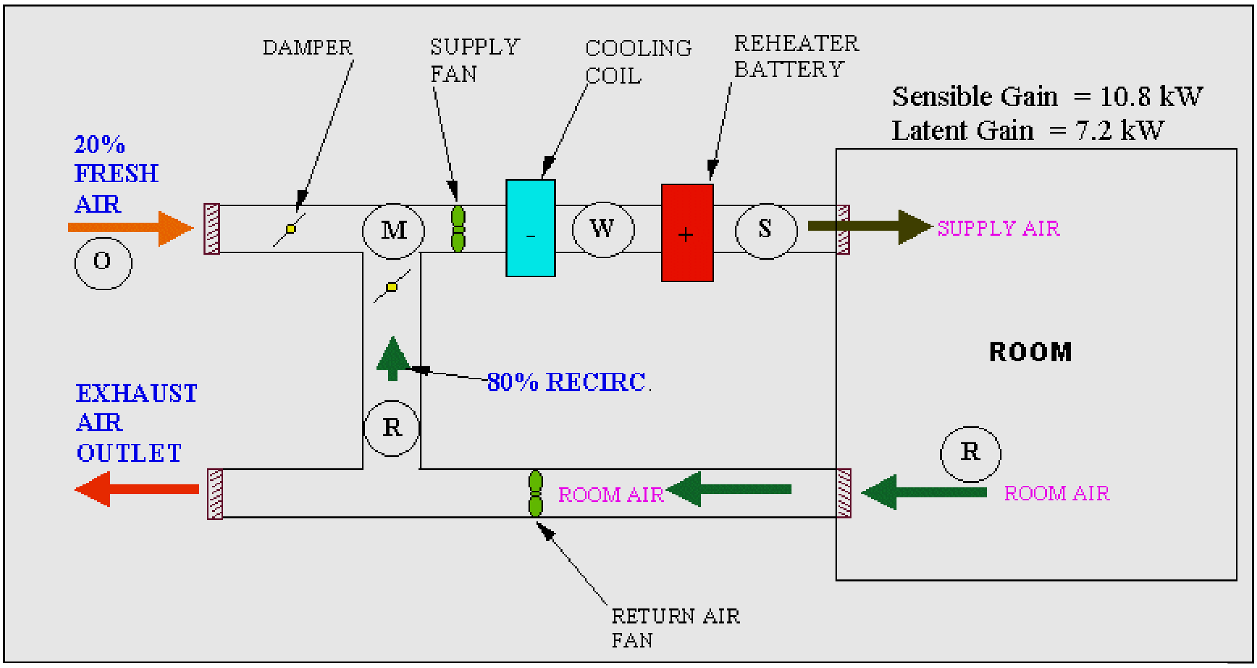
See example 7.

Examples of Psychrometric Calculations for Summer and Winter

Example 1. Summer Cycle

A room is to be maintained at 22°C dry-bulb temperature, 50% saturation, when the sensible heat gain is 10.8 kW in summer.
The latent heat gain is 7.2 kW.
Determine the cooling coil and reheater outputs required by using a psychrometric chart if the plant schematic is as shown below.

DATA:
Outdoor condition is 28°C, 80% saturation.The outdoor air and recirculated air ratio is 20%/80%. The Apparatus Dew Point ADP is 8°C Neglect the cooling coil contact
factor.



Procedure (Summer Cycle)

1. Draw schematic diagram of air-conditioning plant (see above)
2. Plot room condition R on psychrometric chart.
3. Plot outside condition O on psychrometric chart.
4. Join points O and R.

5. Find the mix point M by measuring the length of the line O-R and multiply this by the mixing ratio.

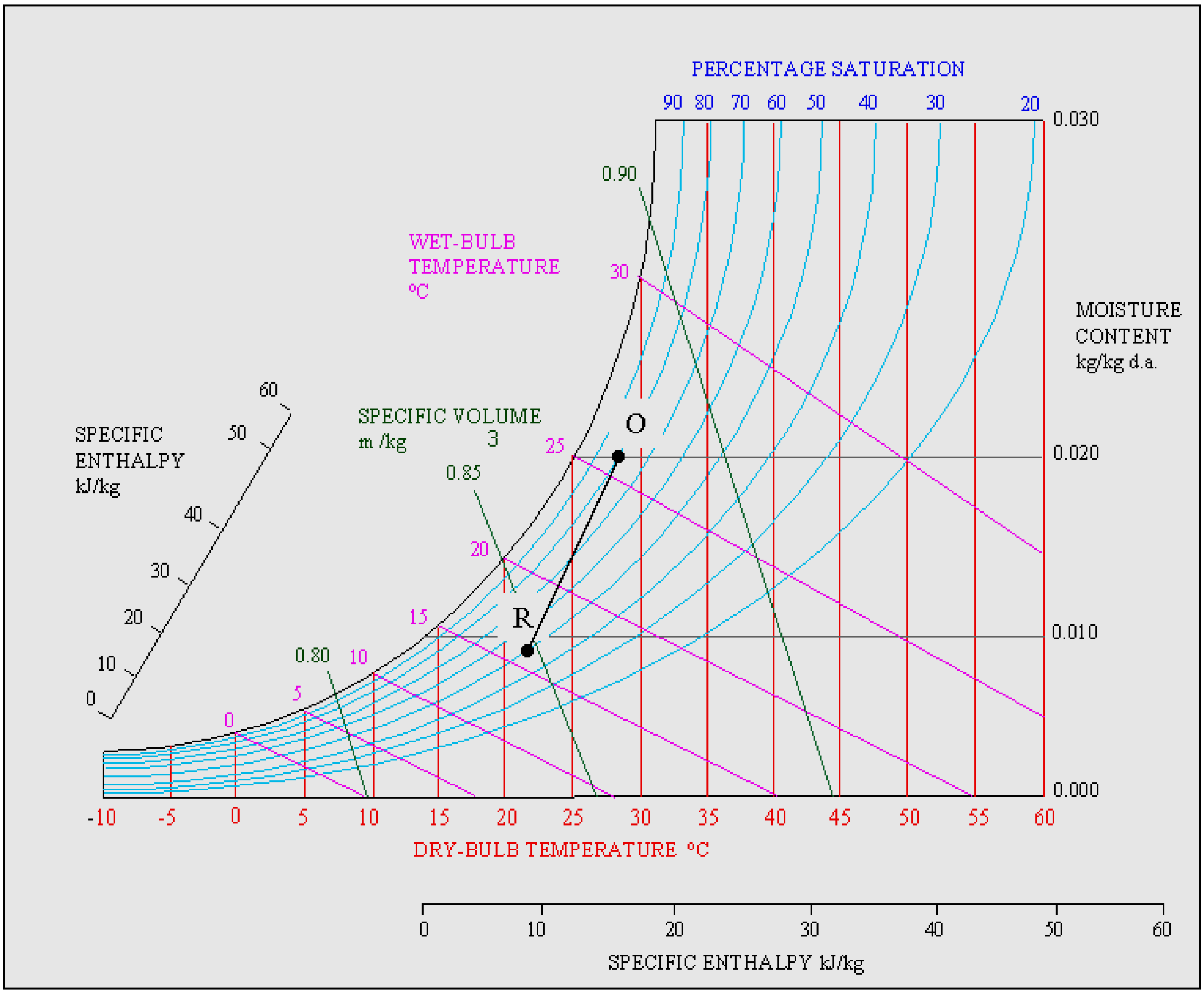
On a full size CIBSE psychrometric chart this measures 85mm.
The ratio of recirculated air is 0.8.
85mm x 0.8 = 68mm
Measure down the O-R line from point O by 68mm.
This determines point M .

If there is more recirculated air than outside air at the mix point, then point M will be closer to point R than point O.

6. Find the room ratio.
This is the sensible to total heat gain ratio.

Total heat = 10.8 kW sensible + 7.2 kW latent = 18 kW total.
Heat ratio = 10.8 / 18.0 = 0.6

Plot this ratio on the protractor, bottom segment, on the psychrometric chart and transfer this line onto the chart so that it passes through point R.

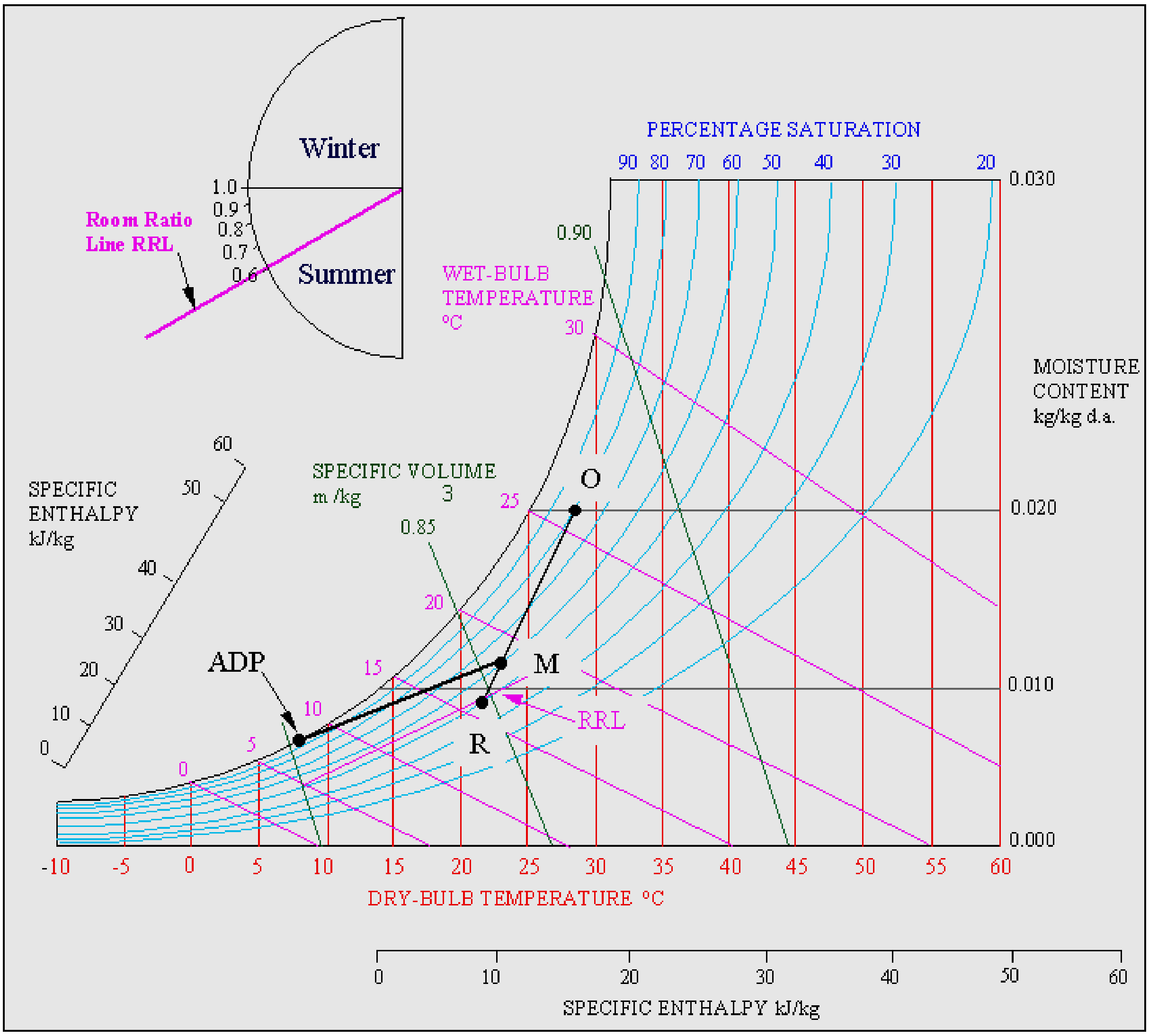


7. Plot the Apparatus Dew Point ADP of the cooling coil.
This is on the 100% saturation curve.

The ADP is 8^0C.

The wet bulb and dry bulb temperatures at this point will be equal.

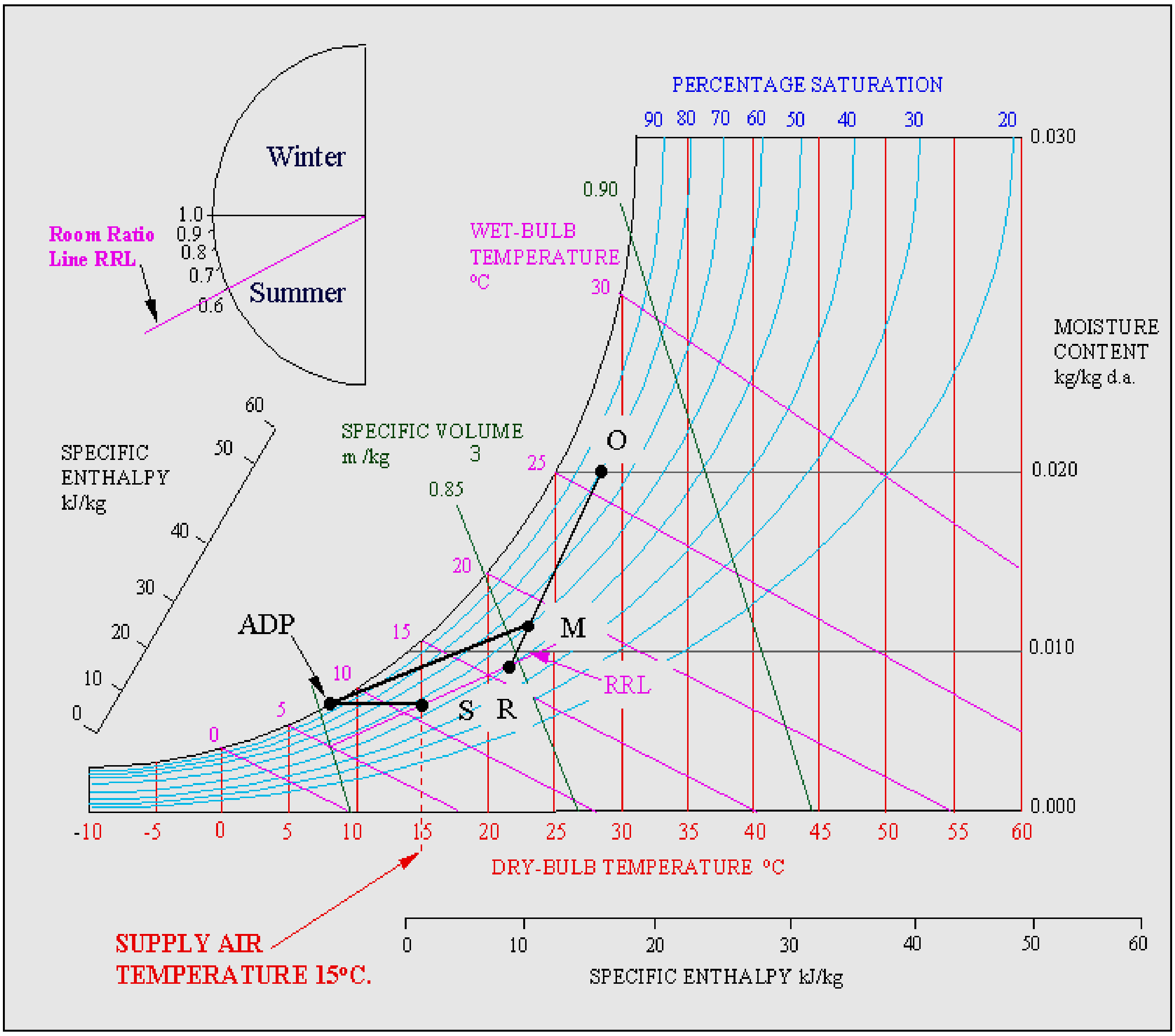
8. Join points M and ADP.



9. Since there is no cooling coil contact factor we can proceed with reheating from point ADP.

10. Plot the supply air condition S.
The reheater process will be a horizontal line from point ADP to point S.
Point S is on the room ratio line.

The supply air temperature is 15°C.


Winter
Summer
Room Ratio Line RRL
1.0
0.9
0.8
0.7
0.6
PERCENTAGE SATURATION
90 80 70 60 50 40 30 20
0.030
0.90
WET-BULB TEMPERATURE °C
MOISTURE CONTENT kg/kg d.a.
SPECIFIC ENTHALPY kJ/kg
SPECIFIC VOLUME m /kg 3
0.85
O
M
ADP
S
R
RRL
0.020
0.010
0.000
-10 -5 0 5 10 15 20 25 30 35 40 45 50 55 60
DRY-BULB TEMPERATURE °C
SUPPLY AIR TEMPERATURE 15°C.
0 10 20 30 40 50 60
SPECIFIC ENTHALPY kJ/kg

SUMMER CYCLE CALCULATIONS

MASS FLOW RATE

When the supply air temperature has been found from the psychrometric chart then the mass flow rate of air can be calculated from the following formula:

$$H_s = m_a \times Cp \ (t_r - t_s)$$

where:

H_s = Sensible heat gain to room (kW)
m_a = mass flow rate of air (kg/s)
Cp = Specific heat capacity of humid air (approx.1.01 kJ/kg degC)
t_r = room temperature (°C)
t_s = supply air temperature (°C)

The supply air temperature is 15°C.
Rearranging the above formula gives:

$$m_a = H_s \ / \ (Cp \ (t_r - t_s))$$

$$m_a = 10.8 \ / \ (1.01 \ (22 - 15))$$

$$m_a = 1.528 \text{ kg/s}$$

COOLING COIL OUTPUT

The cooling coil output is as follows:

$$H_{cooling\ coil} = m_a \ (h_M - h_{ADP})$$

where:

$H_{cooling\ coil}$ = Cooling coil output (kW)

m_a = mass flow rate of air (kg/s)

h_M = specific enthalpy at condition M (kJ/kg) determined from psychrometric chart.

h_{ADP} = specific enthalpy at condition ADP (kJ/kg) determined from psychrometric chart

The specific enthalpies at points M and ADP are shown on the psychrometric Chart below.

$$H_{cooling\ coil} = 1.528 \ (50 - 25)$$

$$H_{cooling\ coil} = \underline{38.2\ kW}$$

Note:

The cooling coil output of 38.2 kW is a much higher value than the sensible heat gain of 10.8 kW.
It should be remembered that the difference is these two values is mostly from the fresh air cooling load. It takes quite a lot of energy in summer to cool fresh air coming into air handling units .This can be minimised by bringing in minimum fresh air but not too little otherwise the building will suffer from lack of oxygen and feel stuffy.
Sometimes mistakes are made when sizing cooling apparatus .If a cooling coil or indoor cooling unit is sized on the sensible heat gain only without allowing for fresh air load then it will be grossly undersized. That is why psychrometric charts are required to calculate cooling coil output including fresh air loads.
So, don’t size cooling coil and indoor cooling units on sensible heat gain only if there is fresh air coming into the plant.
Size these items of plant using a psychrometric chart.

HEATER BATTERY OUTPUT

The heater battery or reheater output is as follows:

$$H_{heater\ battery} = m_a \ (h_S - h_{ADP})$$

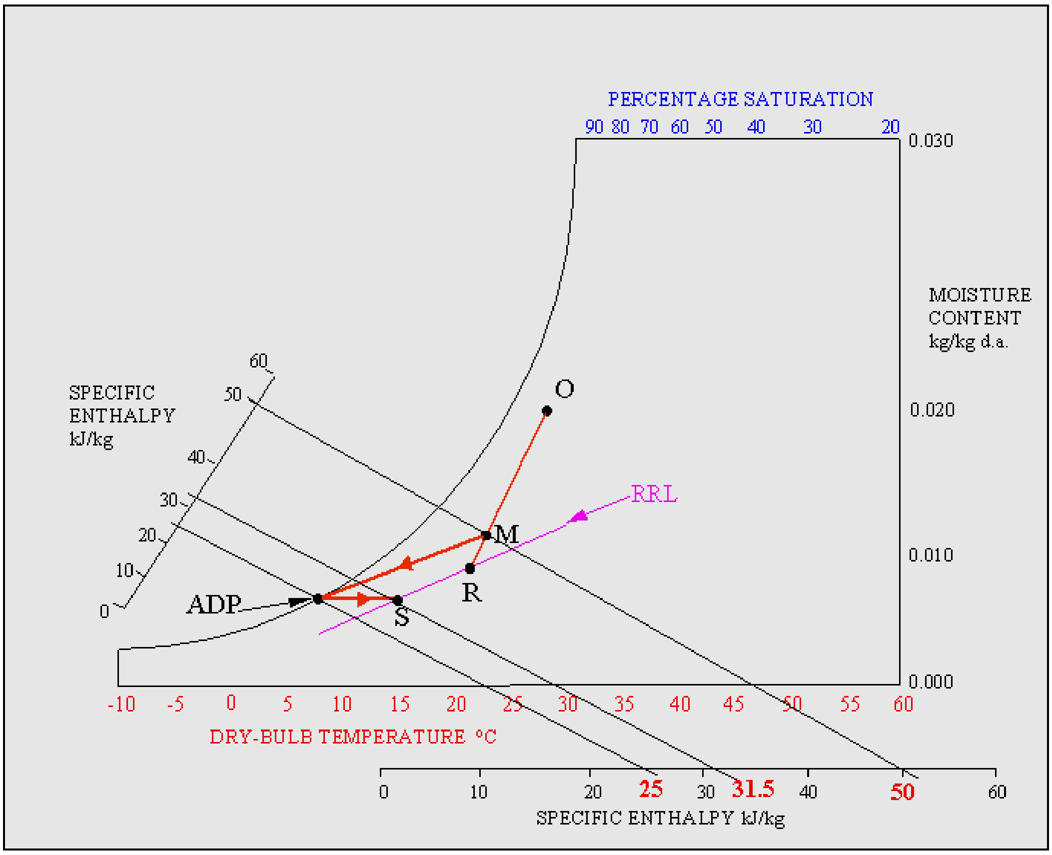
where:

$H_{heater\ battery}$ = Heater battery output (kW)
m_a = mass flow rate of air (kg/s)
h_S = specific enthalpy at condition S (kJ/kg) determined from psychrometric chart.
h_{ADP} = specific enthalpy at condition ADP (kJ/kg) determined from psychrometric chart.

The specific enthalpies at points ADP and S are shown on the psychrometric Chart below.

$$H_{heater\ battery} = 1.528 \ (31.5 - 25)$$

$$H_{heater\ battery} = \underline{9.932 \text{ kW}}$$

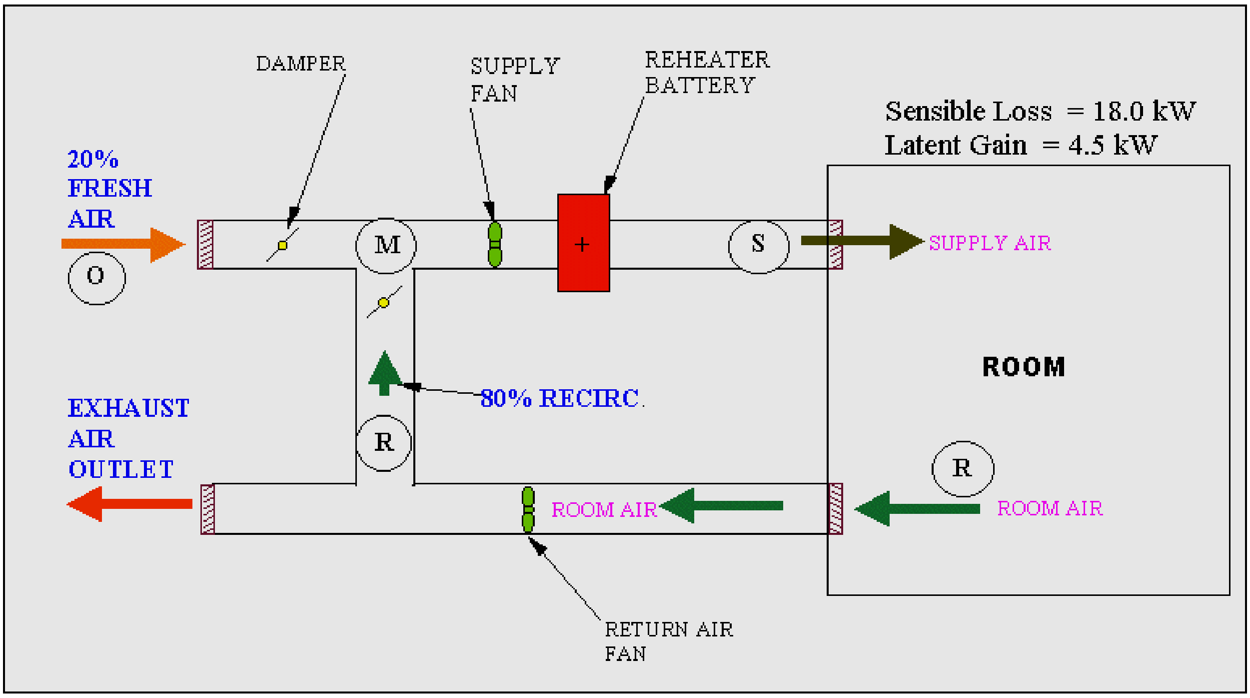


Example 2. Winter Cycle

A room has a 18.0 kW sensible heat loss in winter and a 4.5 kW latent heat gain from the occupants.

Determine the supply air temperature and heater battery load using the following information.

DATA:
Indoor condition: 21°C dry-bulb temperature, 50% saturation.
Outdoor condition: -2°C d.b., 80% saturation.
The outdoor air and recirculated air ratio is 20%/80%.
No preheating or humidification takes place in this simplified example.



Procedure (Winter Cycle)

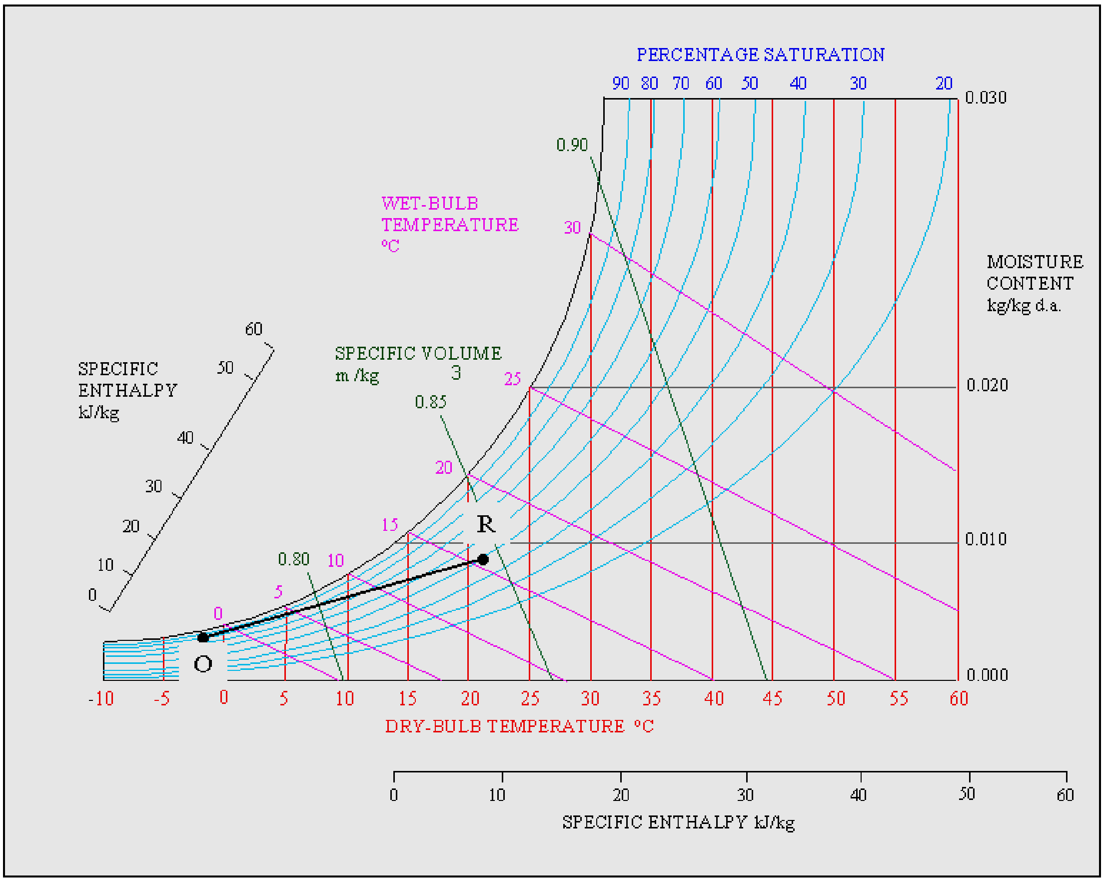
1. Draw schematic diagram of air-conditioning plant (see above)

2. Plot room condition R on psychrometric chart.

3. Plot outside condition O on psychrometric chart.

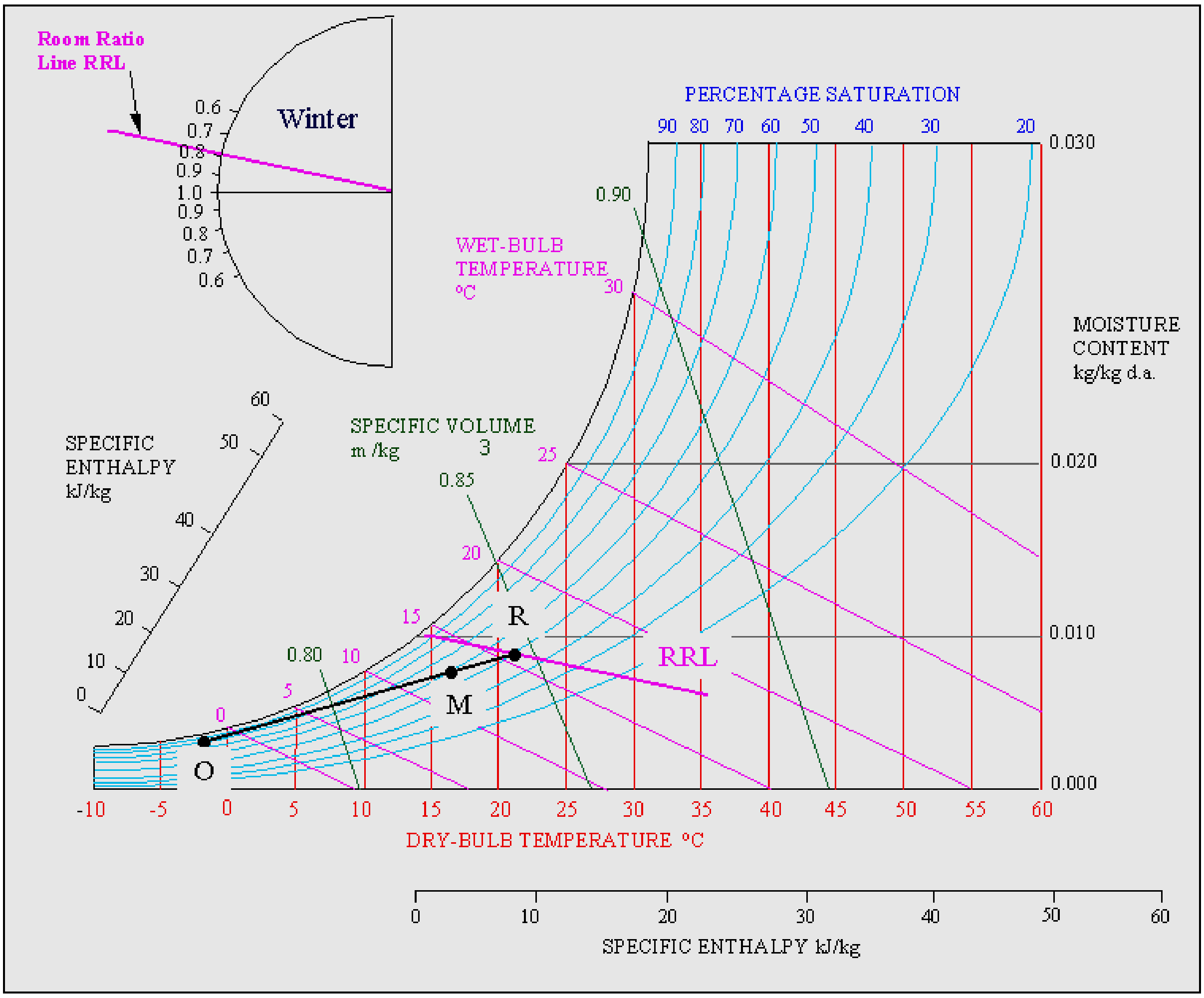
4. No Preheater condition P

5. Join points O and R



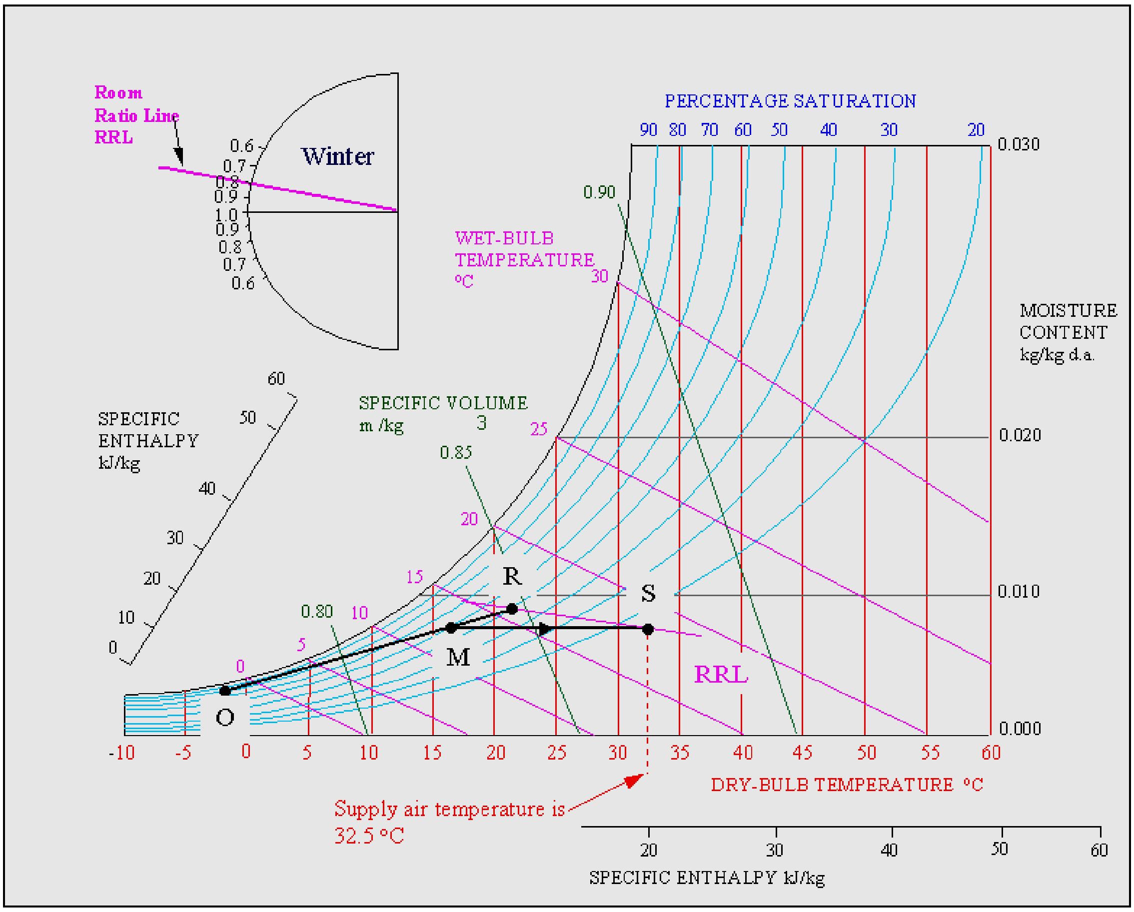
6. Find the mix point M by measuring the length of the line O-R and multiply this by the mixing ratio.
On a full size CIBSE psychrometric chart this measures 110mm.
The ratio of recirculated air is 0. therefore;   110mm x 0.8 = 88mm
Measure up the O-R line from point O by 88mm.
This determines point M .
If there is more recirculated air than outside air at the mix point, then point M will be closer to point R than point O.

7. Find the room ratio.
This is the sensible to total heat ratio.
Neglect signs ie. the total heat for the room will be Sensible loss plus Latent gain.
Total heat = 18 kW sensible + 4.5 kW latent = 22.5 kW total.
Heat ratio = 18 / 22.5 = 0.8
Plot this ratio on the protractor, top segment, on the psychrometric chart and transfer this line onto the chart so that it passes through point R.



8. Find the supply air dry bulb temperature by calculation.

9. Plot the supply air condition S on the room ratio line.
This is on a horizontal line from point M to the right hand side of the chart, and intersects with the RRL.
The supply air Temperature is found to be 32.5°C.



Supply Air Flow Rate

When the sensible heat loss and supply air temperature in winter are known then the mass flow rate of air is calculated from the following formula:

$$Hs = ma \times Cp\ (ts - tr)$$

where:

Hs= Sensible heat loss (kW)

ma=mass flow rate of air (kg/s)

Cp=Specific heat capacity of humid air (approx.1.01 kJ/kg degC)

tr=room temperature (°C)

ts=supply air temperature (°C)

therefore:

ma = Hs / Cp (ts - tr)

ma = 18 / 1.01 (32.5 - 21)

ma = 18 / 11.615

ma = 1.55 kg/s

Heater Battery Output

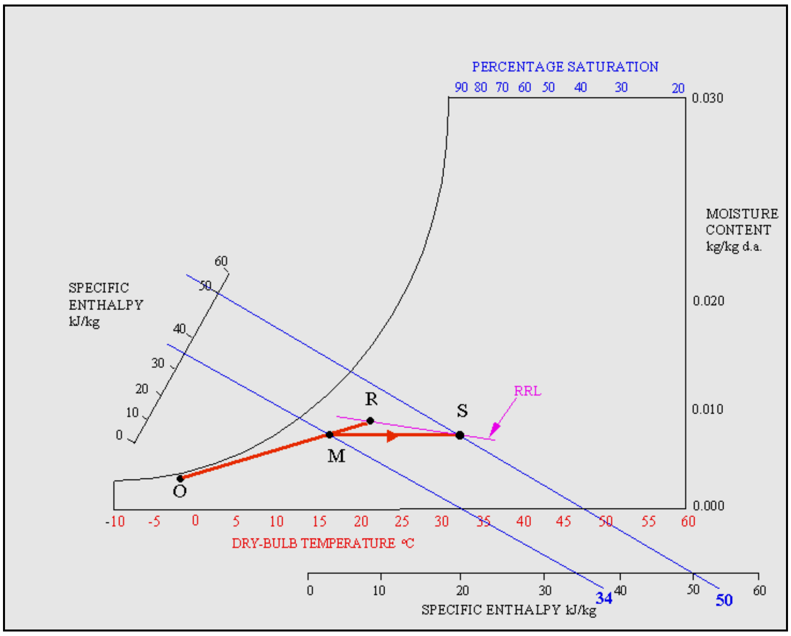
The heater battery output is as follows:

$H_{reheater\ battery} = ma\ (h_S - h_M)$

where:

$H_{reheater\ battery}$ = Reheater battery output (kW)
ma = mass flow rate of air (kg/s)
h_S = specific enthalpy at condition S (kJ/kg)
h_M = specific enthalpy at condition M (kJ/kg)

The specific enthalpies at points S and M are shown on the psychrometric chart below.



$H_{heater\ battery}$ = ma (hS - hM)
$H_{heater\ battery}$ = 1.55 (50 - 34)
$H_{heater\ battery}$ = 24.8 kW
Therefore the heater battery load is **24.8 kW.**

Blank Paper

Northern Technical University
Technical College of Engineering
Department of Power Mechanics Techniques

Fourth Year
Air Conditioning System Design

Chapter Seven
Water Air Conditioning System Design

Prepare By Assist Prof.
Mr. Badran Mohammed Salim
2025 - 2026

Chapter Seven
Water Air Conditioning System Design

1. **Class** : Fourth Year

2. **Subject**: Water Pipe System Design, Fittings, Pumps with Applications.

3. **No. of weeks:** Five weeks

4. Central idea: Study the methods of water pipe design (Open and Closed), Basic of chilled water system and Boiler, how to distribution the water within every zone, size of pump capacity and how to calculate the friction loss and dynamic loss within water pipe.

5. **Test & Problems**:

Introduction:

The advantages of water over steam include the fact that water is safer and more controllable than steam.

Water is safer because the system pressure is not determined by continuously balancing the boiler output with load, and because water does not have the capacity to expand at high velocity in all directions.

Water is more controllable for heating since the water temperature can easily be changed to modify the heat transfer.

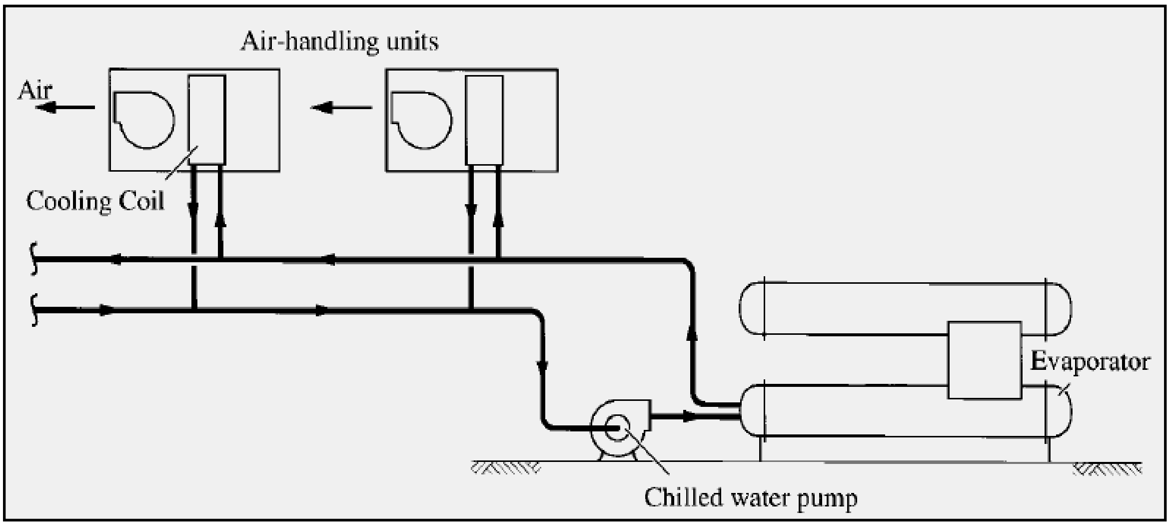
Types of Water System:

Water systems also can be classified according to their operating characteristics into the following categories:

1. **Closed System.**
2. **Open System.**
3. **Once-Through System.**

1. Closed system:

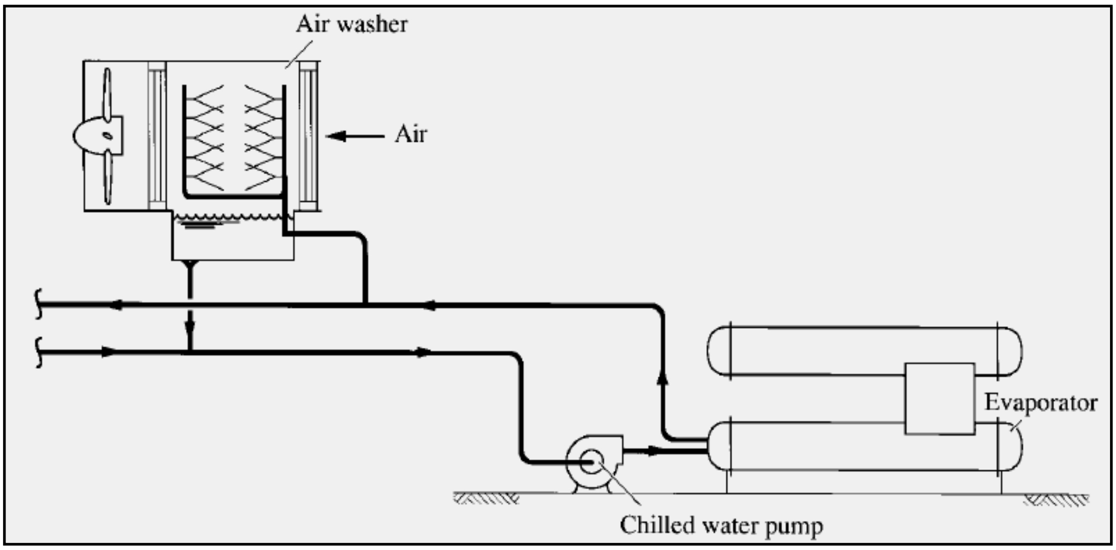
- In a closed system, chilled or hot water flowing through the coils, heaters, chillers, boilers, or other heat exchangers forms a closed recirculating loop.
- In a closed system, water is not exposed to the atmosphere during its flowing process. The purpose of recirculation is to save water and energy.



2. Open System:

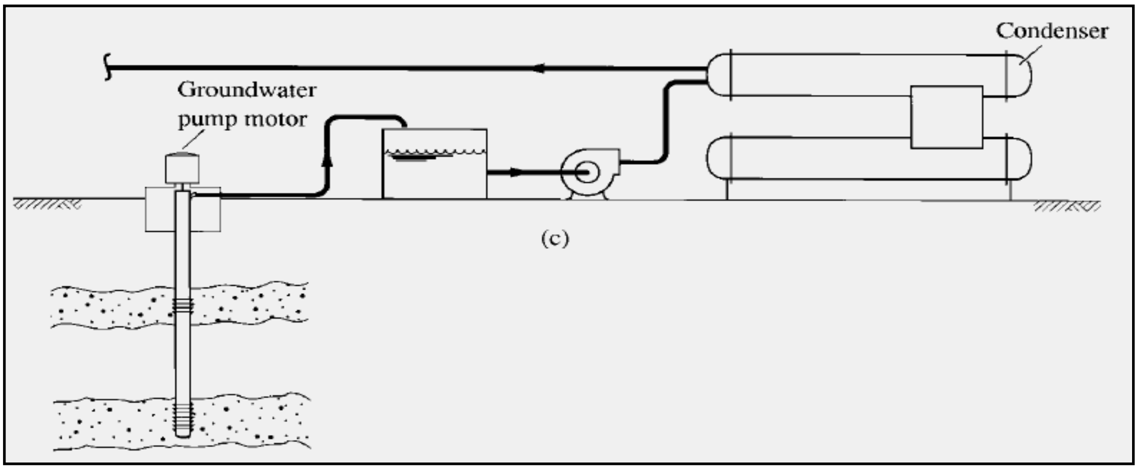
- In an open system, the water is exposed to the atmosphere, For example, chilled water comes directly into contact with the cooled and dehumidified air in the air washer, and condenser water is exposed to atmosphere air in the cooling tower. Recirculation of water is used to save water and energy.

- Open systems need more water treatments than closed systems because dust and impurities in the air may be transmitted to the water in open systems.
- A greater quantity of makeup water is required in open systems to compensate for evaporation, drift carryover, or blow-down operation.



3. Once-through system:

- In a once-through system, water flows through the heat exchanger only once and does not recirculate.
- Lake, river, well, or seawater used as condenser
- cooling water represents a once-through system. Although the water cannot recirculate to the condenser because of its rise in temperature after absorbing the heat of condensation, it can still be used for other purposes, such as flushing water in a plumbing system after the necessary water treatments, to conserve water. In many locations, the law requires that well water be pumped back into the ground.



Heating & Cooling Systems:

Most heating systems for buildings use hot water (Boiler) or cold water (Chiller) which is pumped through pipework from a boiler to heat emitters in the rooms. This has proved to be cheaper than warm air heating because installing pipework is less expensive than ductwork and more equipment is necessary in warm air heating which increases installation costs.

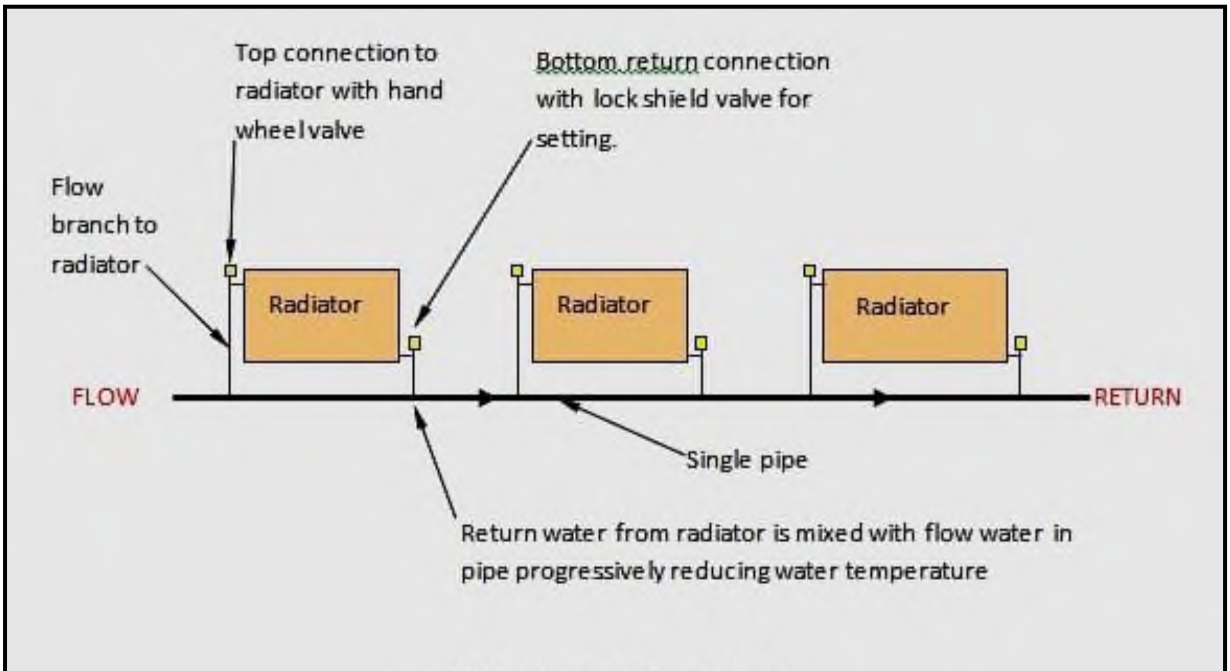
Applications of Water Pipe Design:

- Heating (by Boiler) & Cooling(by Chiller) for A/C system Design.
- Pluming System Design.
- Irrigation System Design.
- Fire Fighting System Design.
- Cooling Tower & Air Washer Design.

Types of Pipe System Supply:

1. Single Pipe (One Pipe System)

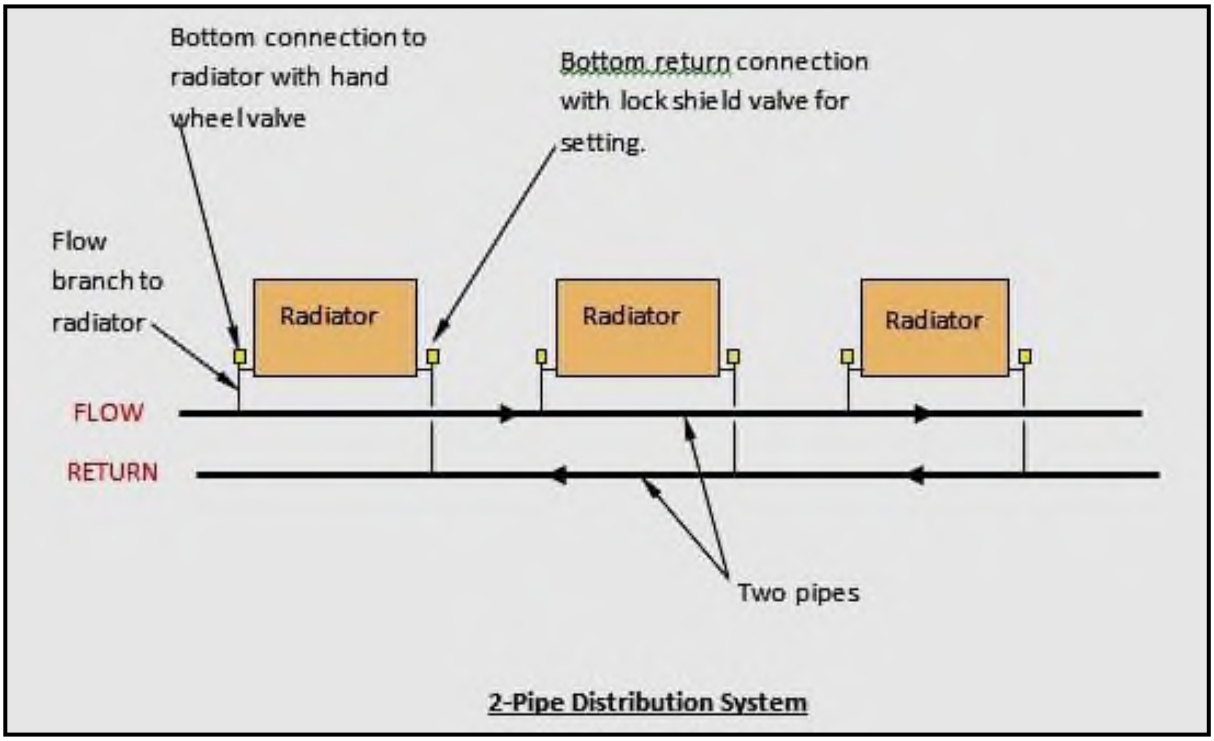
Older heating systems sometimes used a 1-pipe system of distribution.



Single Pipe or 1-Pipe System is

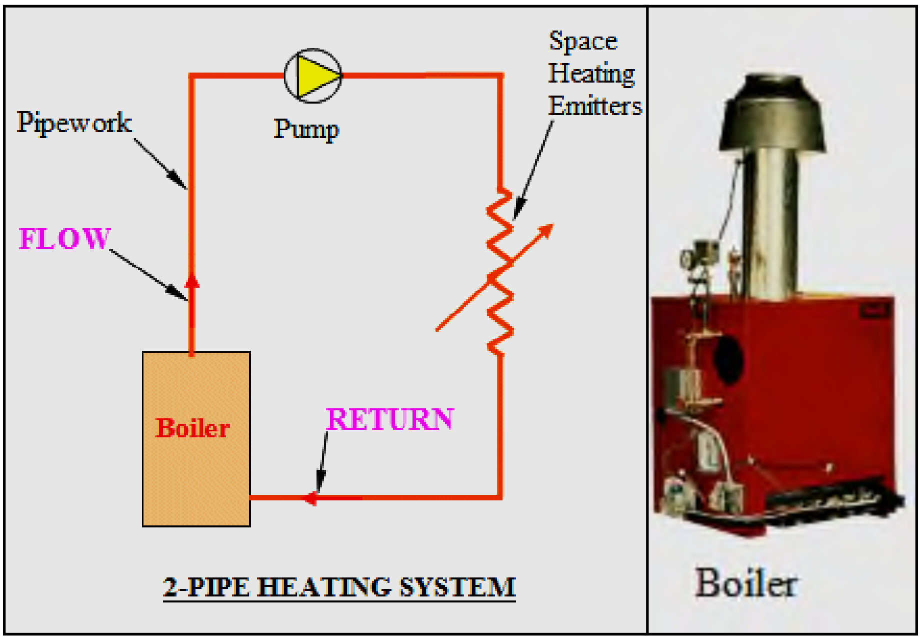
2. Double Pipe (Two Pipe System):

This has been superseded by the more effective 2-pipe system as shown below.



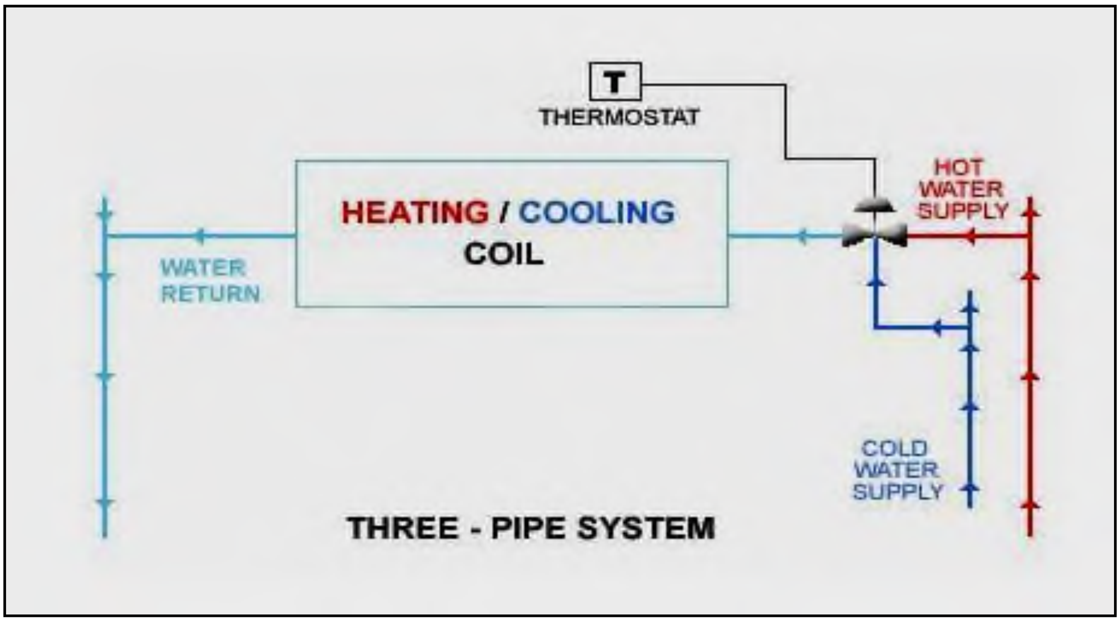
2-Pipe Distribution System

The 2-pipe heating system is most common today and uses a pump to circulate water at about 80°C through heat emitters as shown below:

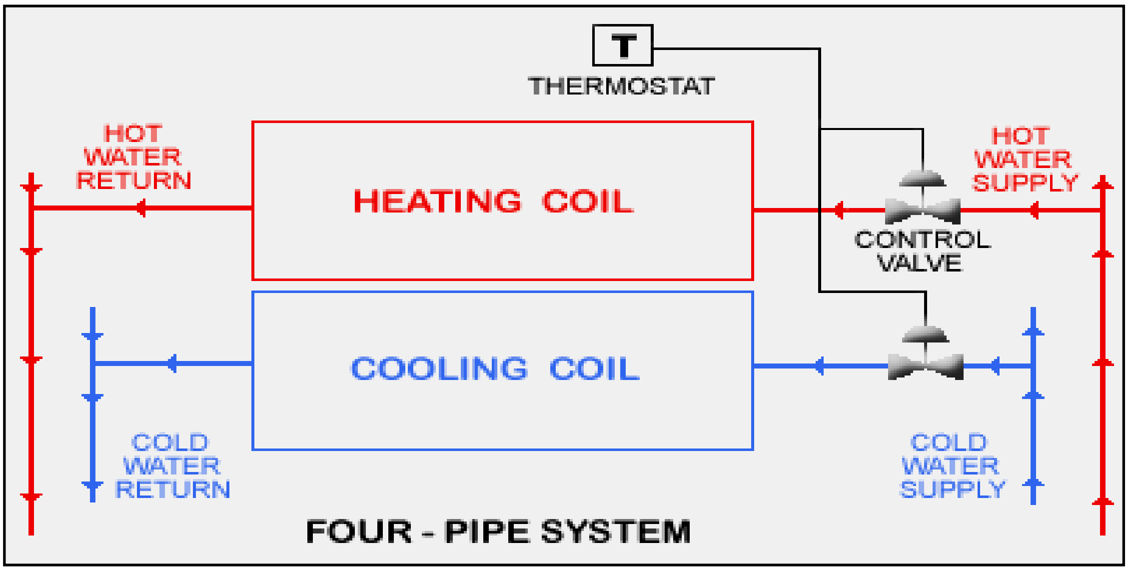


2-PIPE HEATING SYSTEM

3. Three Pipe System:



4. Four Pipe System:

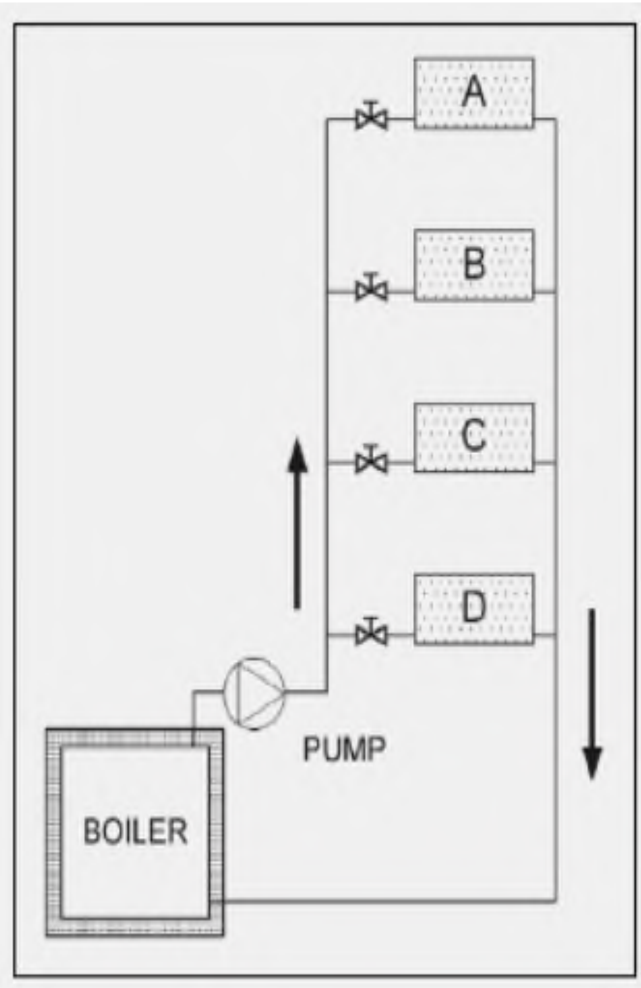


Types of Pipe System Return:

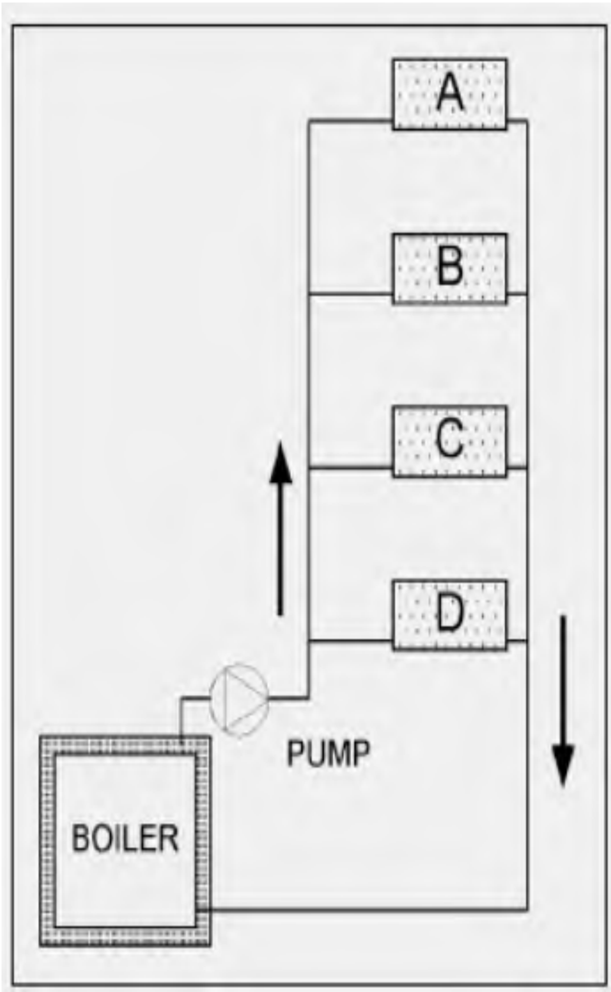
Heating or cooling water can be piped around a building in two ways, either “direct return” or “reverse return.”.

1. Direct Return :

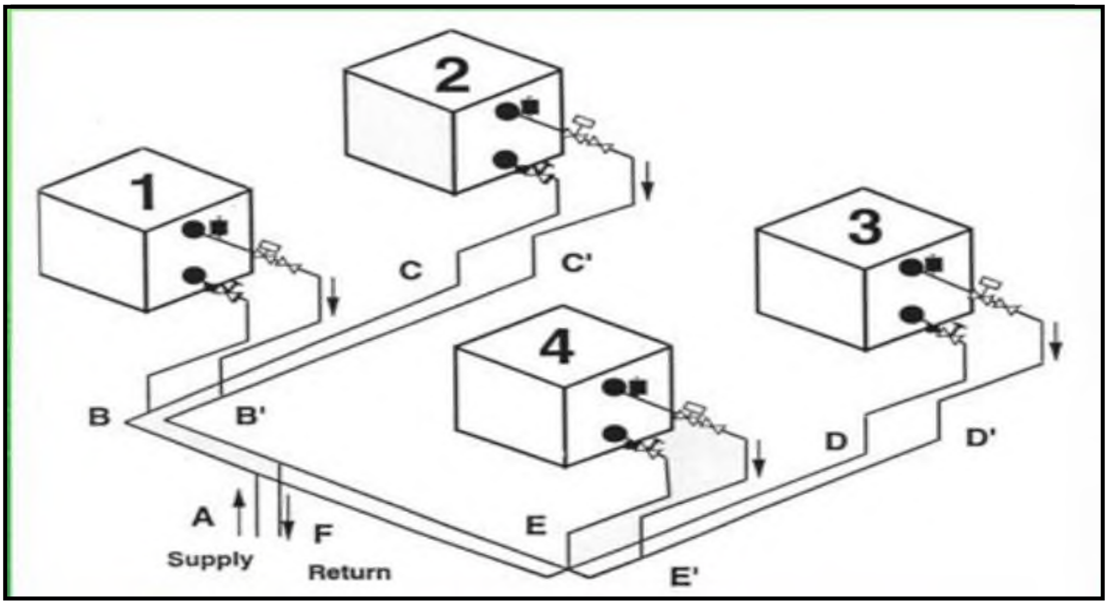
In order to have the same flow through all the heaters, extra resistance has to be added to heaters B, C and D. Adding balancing valves, as shown in Figure below, makes this possible After the system has been installed, a balancing contractor will adjust the balancing valves to create an equal flow through heaters A and B, then an equal low through heaters A and C and finally an equal flow through heaters A and D.



Direct Return with Balance Valve

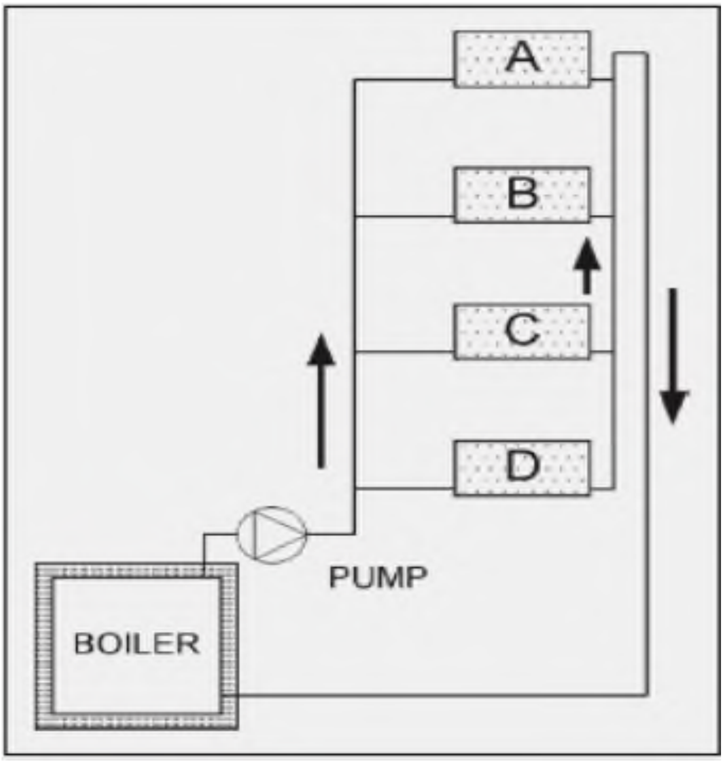


Direct Return without Balance Valve

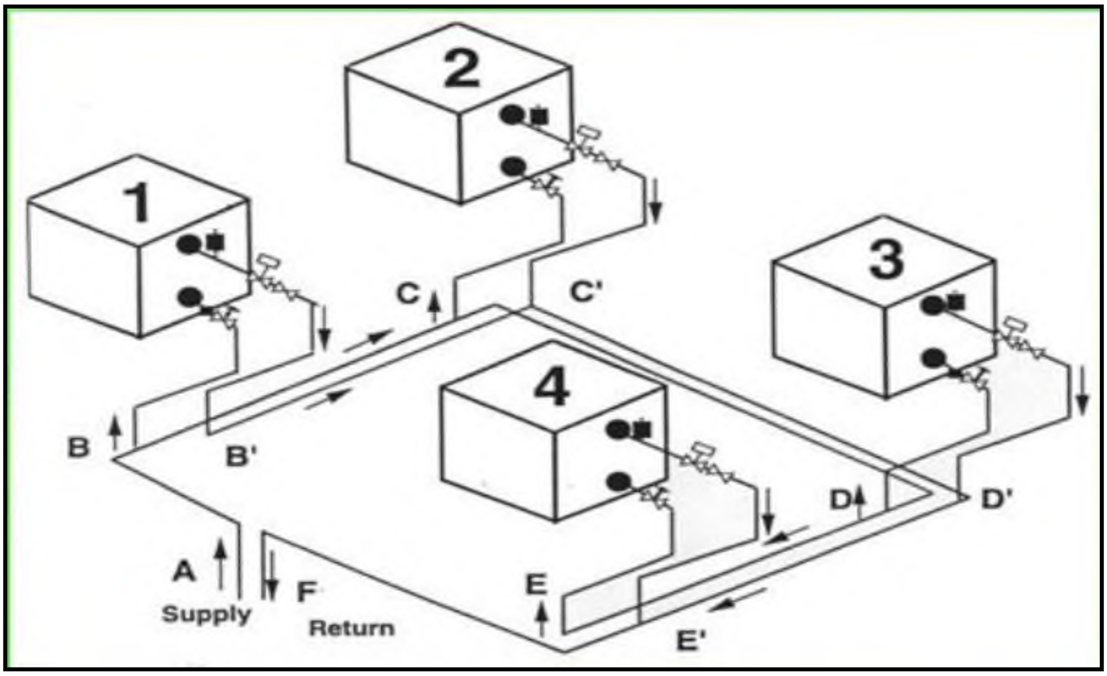


2. Reverse Return :

The reverse return as shown below, Here the pipe length for the flow loop boiler, pump, and heater, boiler is the same for all heaters. As a result, the flow will be the same in each heater; the piping is self-balancing.



Reverse Return



Water Expansion Tank:

The expansion tank is a much more important element of a piping system than generally assumed. It was mentioned earlier that the expansion tank provides for changes in volume. May be part of the elimination system, and establishes a point of constant pressure in the system. This last purpose is very important. A point of constant pressure is necessary to establish the pressure at other points of the closed- loop system; otherwise the system would be like an electrical circuit without a ground.

Expansion tanks are required in a closed loop heating or chilled water HVAC system to absorb the expanding fluid, makeup water due to leakage and limit the pressure within a heating or cooling system. A properly sized tank will accommodate the expansion of the system fluid during the heating or cooling cycle without allowing the system to exceed critical pressure limits. The expansion tank uses compressed air to maintain system pressures by accepting and expelling the changing volume of water as it heats and cools.

The location of the expansion tank becomes an important design consideration. A hot water boiler requires a different approach, because it must be equipped with a safety relief valve, and improper location of the expansion tank and pump may cause unnecessary opening of the relief valve. Therefore, the expansion tank should be located at the boiler outlet or air vent with the pump located just downstream of the boiler. Again the pressures in the system should be analyzed to ensure that positive pressures occur throughout.

Sizing of the expansion tank is important and depends on the total volume of the system, the maximum and minimum system pressures and temperatures, the piping material, the type of tank, and how it is installed.

Types of Expansion Tanks:

1. Open Expansion Tank
2. Close Expansion Tank
3. Diaphragm Expansion Tank

Relations may be derived for sizing of the expansion tanks by assuming that the air behaves as an ideal gas. The type of tank and the way it is employed in the system then influence the results. Consider the free liquid—air interface type where the water in the tank always remains at its initial temperature (uninsulated and connected by a small pipe); the expansion and compression of the air in the tank are isothermal; and the air in the tank is initially at atmospheric pressure. The resulting relation for the tank volume is

$$V_T = \frac{V_w\left[\left(\frac{v_2}{v_1} - 1\right) - 3\alpha\Delta t\right]}{\frac{P_a}{P_1} - \frac{P_a}{P_2}}$$

where:

V_T = expansion tank volume, ft^3 or m^3

V_w = volume of water in the system, ft^3 or m^3

P_a = local barometric pressure, psia or kPa

P_1 = pressure at lower temperature, t_1 (regulated system pressure), psia or kPa

P_2 = pressure at higher temperature, t_2 (some maximum acceptable pressure), psia or kPa

t_1 = lower temperature (initial fill temperature for hot water system or operating temperature for chilled water system), F or C

t_2 = higher temperature (some maximum temperature for both hot and chilled water systems), F or C

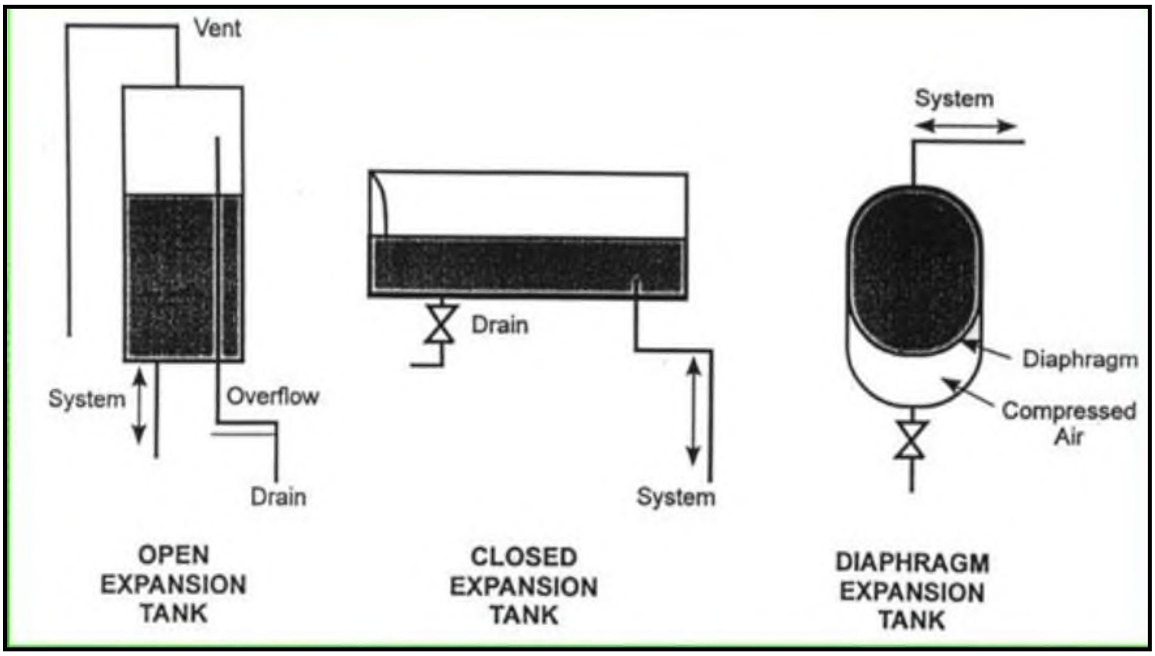
v_1 = specific volume of water at t_1, ft^3/lbm or m^3/kgm

v_2 = specific volume of water at t_2, ft^3/lbm or m^3/kgm

α = linear coefficient of thermal expansion for the piping, F^{-1} or C^{-1}: 6.5×10^{-6} F^{-1} (11.7×10^{-6} C^{-1}) for steel pipe, and 9.3×10^{-5} F^{-1} (16.74×10^{-6} C^{-1}) for copper pipe

Δt = higher temperature minus the lower temperature, F or C

If the initial air charge in the tank is not compressed from atmospheric pressure but rather is forced into the tank at the design operating pressure, as with a bladder-type



Chilled Water system

Introduction:

For large installations the Condenser, Evaporator, Compressor and Expansion device can be purchased as a package unit, known as a **Chiller.** The usual package consists of electrically driven compressor(s) mounted on top of two shell and tube heat exchangers, one for the evaporator and the other for the condenser.

The cooling coil(s) are piped up to the chiller in the conventional manner as shown below. In some countries the Cooling tower is the preferred method of removing heat from the system.

Cooling towers that are open to atmosphere are not often used since the water may become contaminated.

A closed cycle cooling tower or evaporative cooler can be used to reduce contamination risk.

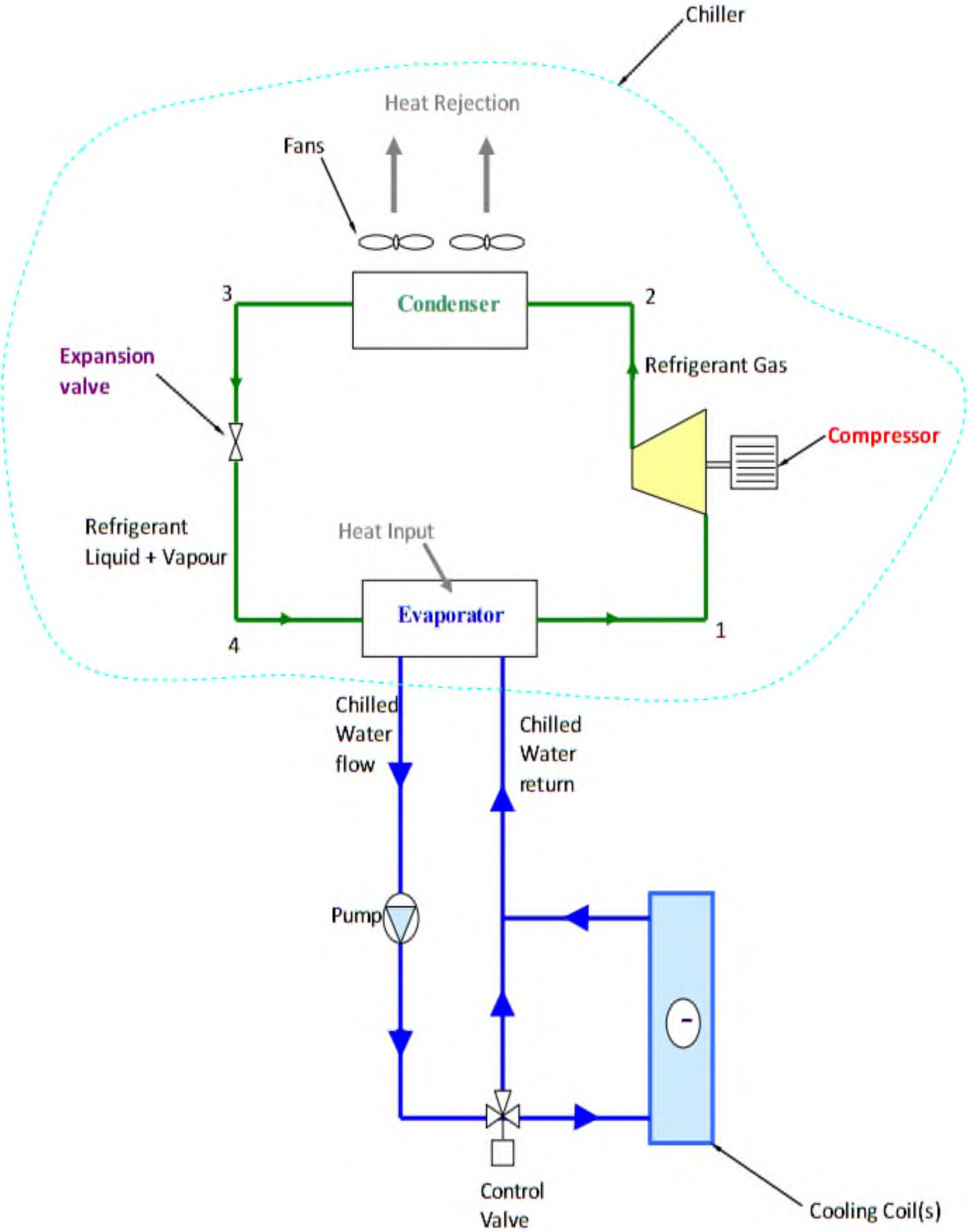


Fig. 7.1: Chiller Schematic Diagram with Air Cooled Condenser

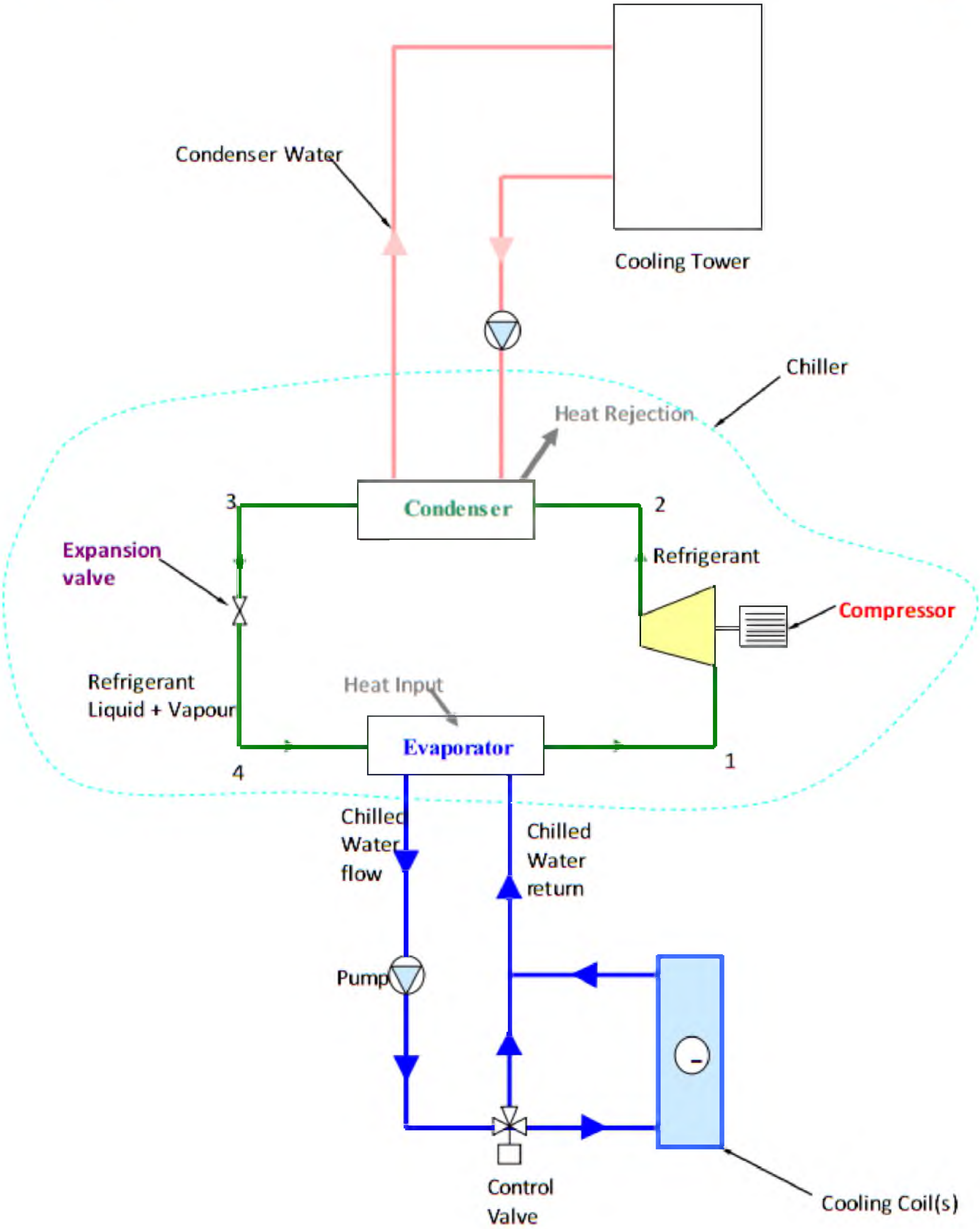


Fig. 7.2: Chiller Schematic Diagram with Cooling Tower

Alternative arrangements are shown for smaller installations where the condenser may be mounted on the roof or external wall of a building and cooled by outside air. The evaporator **may** be installed directly into the ductwork or air handling unit (AHU) for smaller installations .This is known as a Direct Expansion (or DX) coil.

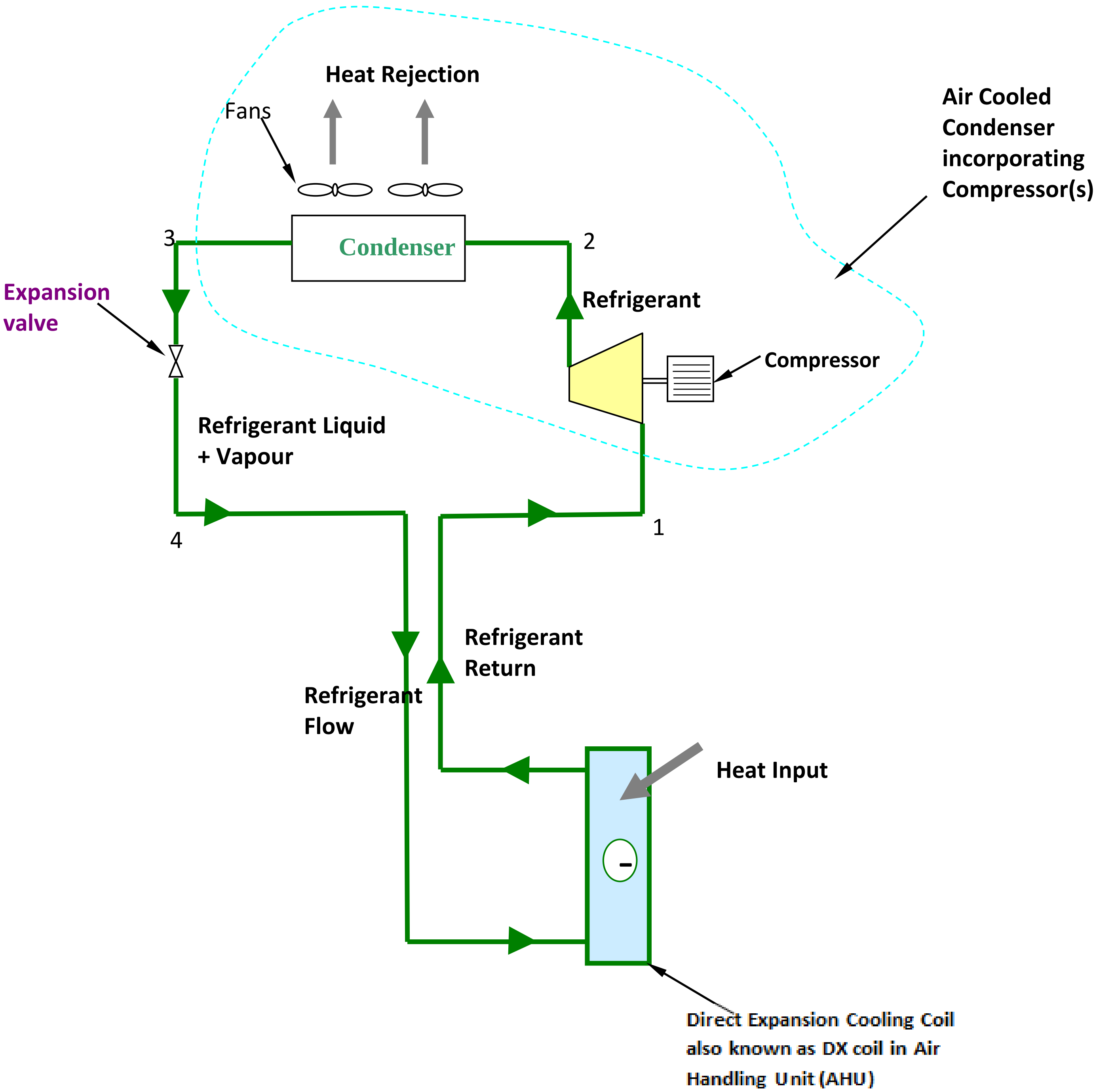


Fig. 7.3: Direct Expansion (DX) System

Chilled-water System:

In larger buildings and particularly in multi-story buildings, the split-system approach begins to run into problems. Either running the pipe between the condenser and the air handler exceeds distance limitations (runs that are too long start to cause lubrication difficulties in the compressor), or the amount of duct work and the length of ducts becomes unmanageable.

At this point, it is time to think about a **chilled-water system**. In a chilled-water system, the entire air conditioner is situated on the roof or behind the building. It cools water to between 4.0°C and 8.0°C.

This chilled water is then piped throughout the building and connected to the cooling coils in air handlers as needed. There is no practical limit to the length of a chilled-water pipe if it is well-insulated.

Outdoor Chiller

Carrier Air cooled liquid chiller

Carrier – Air cooled Water Chiller. Indoor unit

Daikin – Air cooled liquid chiller

Daikin – Air cooled Chiller

Fig. 7.4: Types of Chiller

Chilled Water Temperatures:

In a chilled water system, water is first cooled in the water chiller—the evaporator of a reciprocating, screw, or centrifugal refrigeration system located in a centralized plant to a temperature of 40 to 50°F (4.4 to 10.0°C). It is then pumped to the water cooling coils in AHUs and terminals in which air is cooled and dehumidified. After flowing through the coils, the chilled water increases in temperature up to 60 to 65°F (15.6 to 18.3°C) and then returns to the chiller.

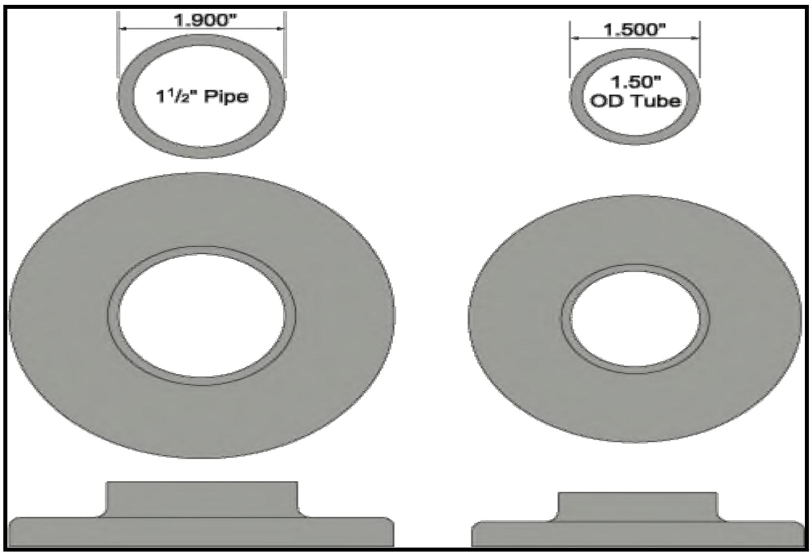
Chilled water is widely used as a cooling medium in central hydronic air conditioning systems. When the operating temperature is below 38°F (3.3°C), inhibited glycols, such as ethylene glycol or propylene glycol, may be added to water to create an aqueous solution with a lower freezing point.

Ethylene/glycol solutions may be used in cooling coils in order to allow lower air temperatures to be obtained.
The temperatures of the fluid circulating may be -7°C from the evaporator and -3°C returning to it, or lower as required.

% ethylene glycol by volume	5	10	15	20	25	30	35	40	45	50
Freezing point °C	–1.1	–2.2	–3.9	–6.7	–8.9	–12.8	–16.1	–20.6	–26.7	–33.3
Specific gravity $d^{15.6°}$	1.004	1.006	1.012	1.017	1.020	1.024	1.028	1.032	1.037	1.040

Pipe VS Tube:

- A pipe is a vessel - a tube is structural.
- A pipe is measured ID - a tube is measured OD.
- For example: 1-1/2" pipe size flanges have an opening to fit over 1.90" OD – the actual outside diameter of 1-1/2" pipe – while 1.50" OD tubing has a true 1.50" outside diameter and the flange is sized accordingly



Piping Material:

- For water systems, the piping materials most widely used are steel, both black (plain) and galvanized (zinc-coated), in the form of either welded-seam steel pipe or seamless steel pipe; ductile iron and cast iron; hard copper; and polyvinyl chloride (PVC).
- The piping materials for various services are shown in the table:
- Copper, galvanized steel, galvanized ductile iron and PVC pipes have better corrosion resistance than black steel pipes. Technical requirements, as well as local customs, determine the selection of piping materials.

Steel Pipes:

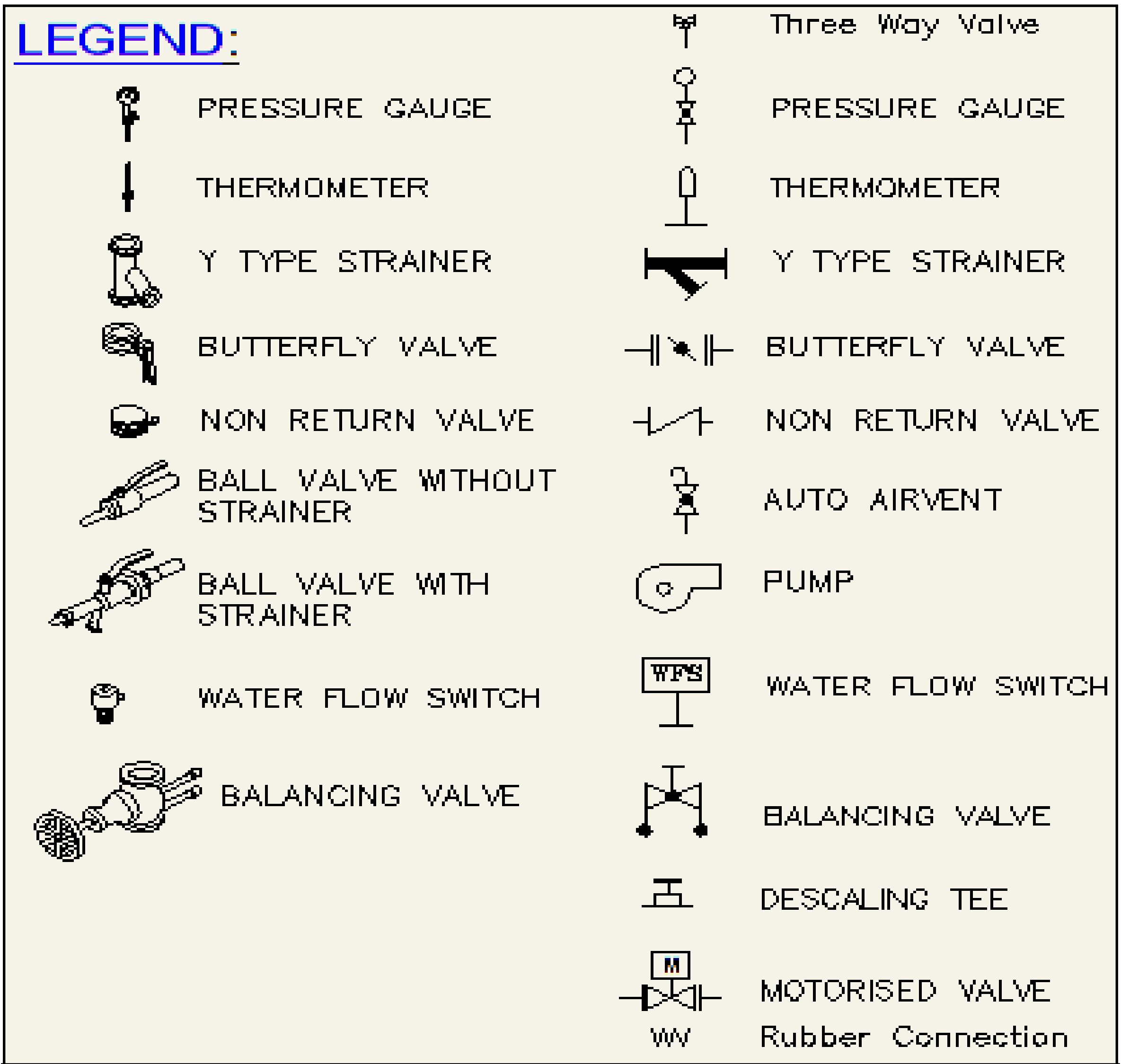
- Steel Pipe is manufactured by several processes:
  - Seamless Pipe, made by piercing or extruding, has no longitudinal seam.
  - CW- Continuous weld furnace-butt-welding process forces and joints the edge together at high temperature.
  - ERW- Electric Resistance Welded pipe.
- NPS – Nominal Pipe Size.
- Up to 12in diam. NPS do not match the internal and External diameters.
- For 14in and larger pipe, the size corresponds to the outside diameter.
- Schedule 40 (standard wall)
- Schedule 80 (extra heavy wall)

Wall Thickness:

Nominal Pipe Size	Outside Diameter	Schedule 5	Schedule 10	Schedule 40	Schedule 80
1/2"	.840"	.065"	.083"	.109"	.147"
3/4"	1.050"	.065"	.083"	.113"	.154"
1"	1.315"	.065"	.109"	.133"	.179"
1-1/4"	1.660"	.065"	.109"	.140"	.191"
1-1/2"	1.900"	.065"	.109"	.145"	.200"
2"	2.375"	.065"	.109"	.154"	.218"
2-1/2"	2.875"	.083"	.120"	.203"	.276"
3"	3.500"	.083"	.120"	.216"	.300"
3-1/2"	4.000"	.083"	.120"	.226"	.318"
4"	4.500"	.083"	.120"	.237"	.337"
5"	5.560"	.109"	.134"	.258"	.375"
6"	6.630"	.109"	.134"	.280"	.432"

Pipe Joints:

- Steel pipes of small diameter (2 in. or 50 mm less) threaded through cast-iron fittings are the most widely used type of pipe joint.
- For steel pipes of diameter 2 in. (50 mm) and more, welded joints, bolted flanges, and grooved ductile iron joined fittings are often used.
- Galvanized steel pipes are threaded together by galvanized cast iron or ductile iron fittings.
- Copper pipes are usually joined by soldering and brazing socket end fittings.
- Plastic pipes are often joined by solvent welding, fusion welding, screw joints, or bolted flanges.
- Vibrations from pumps, chillers, or cooling towers can be isolated or dampened by means of flexible pipe couplings. Arch connectors are usually constructed of nylon, dacron, or polyester and neoprene.
- They can accommodate deflections or dampen vibrations in all directions. Restraining rods and plates are required to prevent excessive stretching. A flexible metal hose connector includes a corrugated inner core with a braided cover. It is available with flanged or grooved end joints.



Gate Valve .... Usually used for isolation (on/off) valve

Globe Valve .... Usually used for throttling and regulating flow

Butterfly, Plug, Ball, Needle, Diaphragm .... are usually used for both regulation and stop or start but they differ according to media or temperature and pressure

for example: diaphragm valve is very good with chemicals*

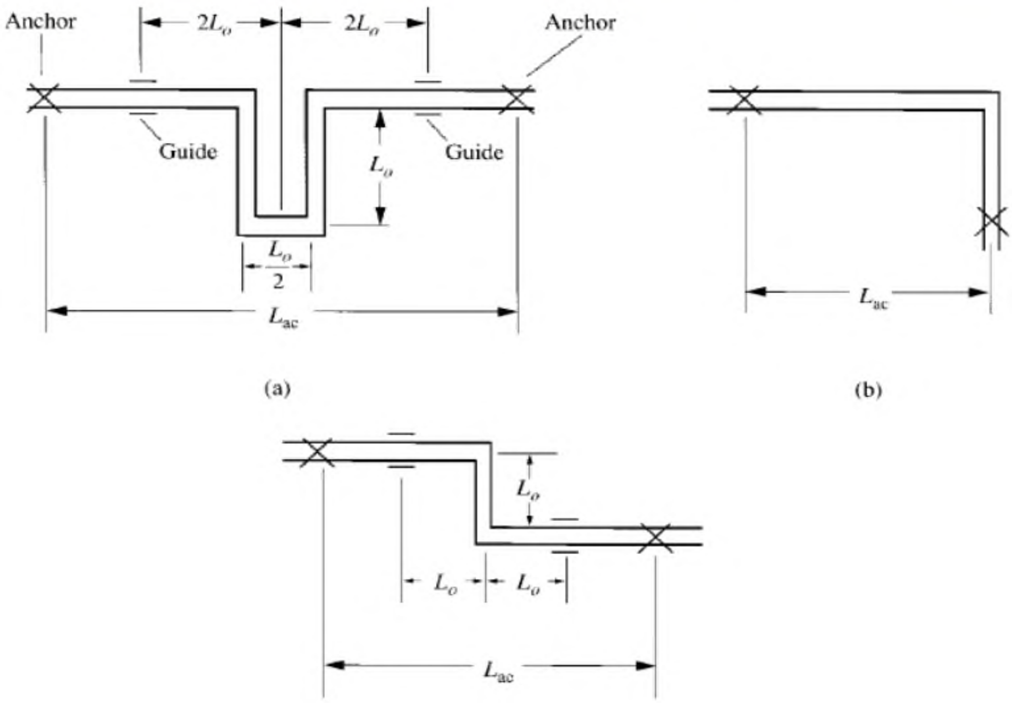
Check valves .... are usually used to prevent reverse flow

Safety and Relief valves .... are usually used to relief excess pressure from the system and to protect system from being damaged

Expansion and Contraction:

- During temperature changes, all pipes expand and contract. The design of water pipes must take into consideration this expansion and contraction.
- Both the temperature change during the operating period and the possible temperature change between the operating and shutdown periods should also be considered.
- For chilled and condenser water, which has a possible temperature change of 40 to 100°F (4.4 to 37.8°C), expansion and contraction cause considerable movement in a long run of piping.
- Unexpected expansion and contraction cause excess stress and possible failure of the pipe, pipe support, pipe joints, and fittings.
- Expansion and contraction of hot and chilled water pipes can be better accommodated by using loops and bends. The commonly used bends are U bends, Z bends, and L bends
- Anchors are the points where the pipe is fixed so that it will expand or contract between them. Reaction forces at these anchors should be considered when the support is being designed.
- *ASHRAE Handbook 1992*, *HVAC Systems and Equipment*, gives the required calculations and data for determining these stresses. Guides are used so that the pipes expand laterally.

Expansion loops.
(*a*) U bends; (*b*) L bends; (*c*) Z bends.



Empirical formulas are often used instead of detailed stress analyses to determine the dimension of the offset leg Lo [ft,(m)]. Waller (1990) recommended the following formulas:

U bends: $L_o = 0.041 D^{0.48} L_{ac}^{0.46} \Delta T$

Z bends: $L_o = (0.13 D L_{ac} \Delta T)^{0.5}$

L bends: $L_o = (0.314 D L_{ac} \Delta T)^{0.5}$

where D = diameter of pipe, in. (mm)

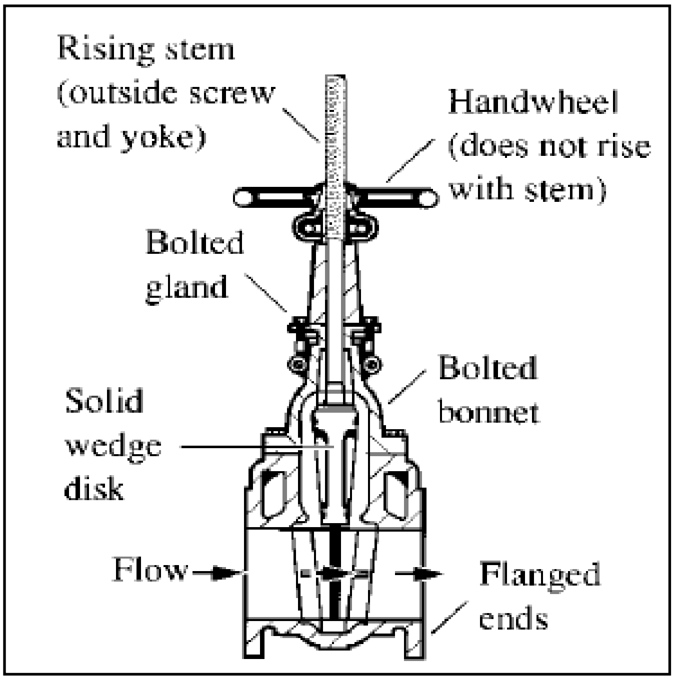
L_{ac} = distance between anchors, hundreds of ft (m)

ΔT = temperature difference, °F (°C)

Types of Valve:

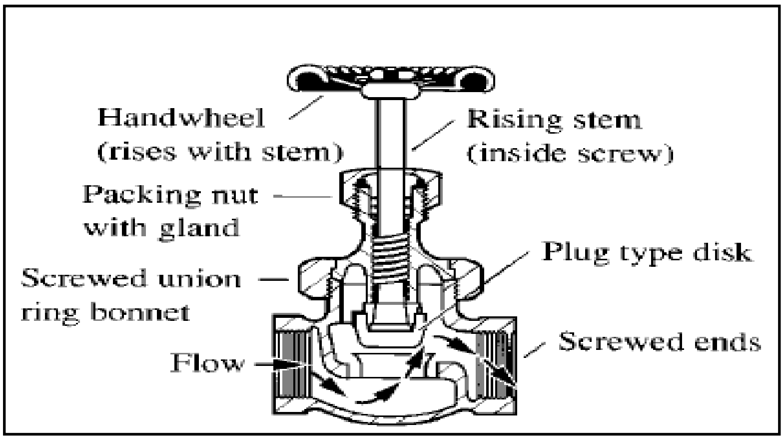
1. Gate Valves.

- The disk of a gate valve is in the shape of a "gate" or wedge.
- When the wedge is raised at the open position, a gate valve does not add much flow resistance.
- The wedge can be either a solid wedge, which is most commonly used, or a split wedge, in which two disk halves being forced outward fit tightly against the body seat.
- Gate valves are used either fully opened or closed, an on/off arrangement. They are often used as isolating valves for pieces of equipment or key components, such as control valves, for service during maintenance and repair.



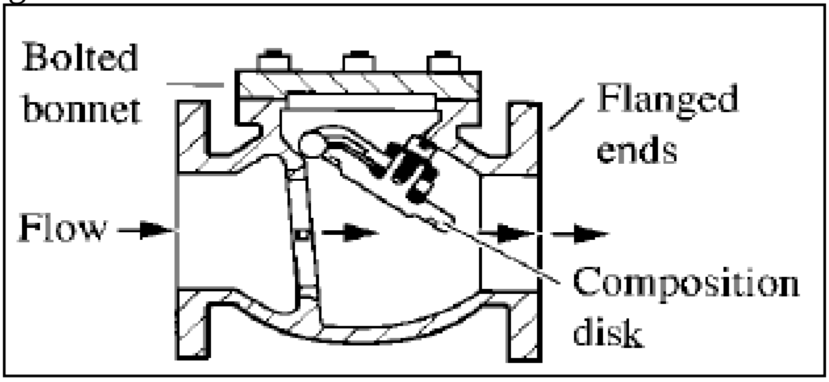
2. Globe Valves:

- They are so named because of the globular shape of the valve body
- Globe valves have a round disk or plug-type disk seated against a round port.
- Water flow enters under the disk. Globe valves have high flow resistances.
- substantially faster than gate valves. Angle valves are similar to globe valves in their seats and operation. The basic difference is that the valve body of an angle valve can also be used as a 90° elbow at that location.
- Globe valves are used to throttle and to regulate the flow.
- They are sometimes called balancing valves. They are deliberately designed to restrict fluid flow, so they should not be used in applications for which full and unobstructed flow is often required.



3. Check Valves:

- As their name suggests, are valves used to prevent, or check, reverse flow.
- There are basically two types of check valves: swing check and lift check. A swing check valve has a hinged disk. When the water flow reverses, water pressure pushes the disk and closes the valve.
- In a lift check valve, upward regular flow raises the disk and opens the valve, and reverse flow pushes the disk down to its seat and stops the backflow.
- A swing check valve has a lower flow resistance than a lift check valve.



4. **Balance Valves:**

These valves are used to balance the water flow in a water system. There are two kinds of balancing valves: manual balance valves and automatic balance valves. A globe valve can be used as a manual balance valve. A manual balance valve can also be a valve with integral pressure taps for flow measurement and a calibrated port to adjust the flow. An automatic balancing valve is also called an automatic flow-limiting valve. There is a moving element that adjusts the flow passage area according to the water pressure differential across the valve.

5. **Pressure Relief Valves:**

These valves are safety valves used to prevent a system that is over pressurized from exceeding a predetermined limit. A pressure relief valve is held closed by a spring or rupture member and is automatically opened to relieve the water pressure when it rises above the system design working pressure.

6. **Plug Valves:**

These valves use a tapered, cylindrical plug disk to fit the seat. They vary from fully open to fully closed positions within a quarter-turn. Plug valves may be used for throttling control during the balancing of a water system.

7. **Ball Valves:**

These valves use a ball as the valve disk to open or close the valve. As with plug valves, they vary from fully open to fully closed positions within a quarter-turn. As with gate valves, ball valves are usually used for open/ close service. They are less expensive than gate valves.

8. **Butterfly Valves:**

A butterfly valve has a thin rotating disk. Like a ball or plug valve, it varies within a quarter-turn from fully open to fully closed. As described in Sec. 5.6, a butterfly valve exhibits low flow resistance when it is fully opened. The difference between a butterfly valve used for control purposes and a hand-operated butterfly valve is that the former has an actuator and can be operated automatically. Butterfly valves are lightweight, easy to operate and install, and lower in cost than gate valves. They are primarily used as fully open or fully closed, but they may be used for throttling purposes. Butterfly valves are gaining in popularity, especially in large pipes.

Valve Connections and Ratings:

The commonly used types of valve connection are as follows:

- **Threaded ends**. for small pipes with diameters from to 2in.

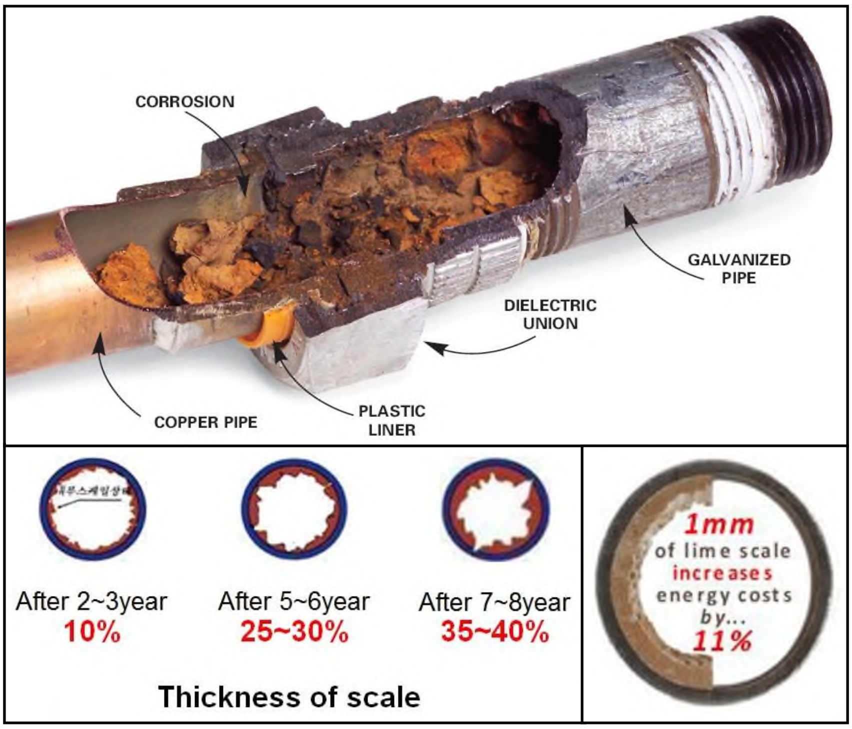
- **Flanged ends**. for larger pipes (2.5in. or 63 mm and above)
- **Welded ends**. higher pressure and temperature
- **Grooved ends**. These connections use circumferential grooves in which a rubber gasket fits and are enclosed by iron couplings.
- **Soldered ends**. Bronze valves in copper piping systems

Pipe Fittings and Water System Accessories:

- Water pipe fittings include elbows, tees, and valves.
- Water pipe elbows and tees are often made of cast iron, ductile iron, or steel.
- Pressure losses due to the water pipe fittings are usually expressed in terms of an equivalent length of straight pipe, for the sake of convenience.

Water Impurities:

In hot and chilled water systems, the problems associated with water mainly concern water's dissolved impurities, which cause corrosion and scale, and the control of algae, bacteria, and fungi.



PRESSURE DROP EQUATIONS

Darcy-Weisbach Equation

Pressure drop caused by fluid friction in fully developed flows of all "well-behaved" (Newtonian) fluids is described by the Darcy-Weisbach equation:

$$\Delta p = f\left(\frac{L}{D}\right)\left(\frac{\rho V^2}{2}\right)$$

where

Δp = pressure drop, Pa
f = friction factor, dimensionless (from Moody chart

L = length of pipe, m
D = internal diameter of pipe, m
ρ = fluid density at mean temperature, kg/m^3
V = average velocity, m/s

This equation is often presented in specific energy form as

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

where

Δh = energy loss, m
g = acceleration of gravity, m/s^2

In this form, the density of the fluid does not appear explicitly (although it is in the Reynolds number, which influences f).

The friction factor f is a function of pipe roughness ε, inside diameter D, and parameter Re, the Reynolds number:

$$\text{Re} = DV\rho/\mu$$

where

Re = Reynolds number, dimensionless
ε = absolute roughness of pipe wall, m
μ = dynamic viscosity of fluid, Pa·s

A useful fit of smooth and rough pipe data for the usual turbulent flow regime is the **Colebrook equation**:

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log\left(\frac{2\varepsilon}{D} + \frac{18.7}{\mathrm{Re}\sqrt{f}}\right)$$

Hazen-Williams Equation

A less widely used alternative to the Darcy-Weisbach formulation for calculating pressure drop is the Hazen-Williams equation, which is expressed as

$$\Delta p = 6.819L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167}(\rho g)$$

or

$$\Delta h = 6.819L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167}$$

where C = roughness factor.

Typical values of C are 150 for plastic pipe and copper tubing, 140 for new steel pipe, down to 100 and below for badly corroded or very rough pipe.

Valve and Fitting Losses

Valves and fittings cause pressure losses greater than those caused by the pipe alone. One formulation expresses losses as

$$\Delta p = K\rho\left(\frac{V^2}{2}\right) \quad \text{or} \quad \Delta h = K\left(\frac{V^2}{2g}\right)$$

where K = geometry- and size-dependent loss coefficient (Tables 1, 2, and 3).

Table 1 K Factors—Screwed Pipe Fittings

Nominal Pipe Dia., mm	90° Ell Reg.	90° Ell Long	45° Ell	Return Bend	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve	Bell Mouth Inlet	Square Inlet	Projected Inlet
10	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
15	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
20	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
25	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
32	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
40	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
50	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
65	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
80	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
100	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: Engineering Data Book (HI 1979).

Table 2 K Factors—Flanged Welded Pipe Fittings

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

Source: Engineering Data Book (HI 1979).

Table 3 Approximate Range of Variation for *K* Factors

90° Elbow	Regular screwed	±20% above 50 mm ±40% below 50 mm	Tee	Screwed, line or branch Flanged, line or branch	±25% ±35%
	Long-radius screwed	±25%	Globe valve	Screwed	±25%
	Regular flanged	±35%		Flanged	±25%
	Long-radius flanged	±30%	Gate valve	Screwed	±25%
45° Elbow	Regular screwed	±10%		Flanged	±50%
	Long-radius flanged	±10%	Angle valve	Screwed	±20%
Return bend (180°)	Regular screwed	±25%		Flanged	±50%
	Regular flanged	±35%	Check valve	Screwed	±50%
	Long-radius flanged	±30%		Flanged	+200% −80%

Source: Engineering Data Book (HI 1979).

Example 1. Determine the pressure drop for 15°C water flowing at 1 m/s through a nominal 25 mm, 90° screwed ell.

Solution: From Table 1, the *K* for a 25 mm, 90° screwed ell is 1.5.

$$\Delta p = 1.5 \times 1000 \times 1^2/2 = 750 \text{ Pa}$$

Calculating Pressure Losses

The most common engineering design flow loss calculation selects a pipe size for the desired total flow rate and available or allowable pressure drop.

Because either formulation of fitting losses requires a known diameter, pipe size must be selected before calculating the detailed influence of fittings. A frequently used rule of thumb assumes that the design length of pipe is 50 to 100% longer than actual to account for fitting losses. After a pipe diameter has been selected on this basis, the influence of each fitting can be evaluated.

Table 4 Water Velocities Based on Type of Service

Type of Service	Velocity, m/s	Reference
General service	1.2 to 3.0	a, b, c
City water	0.9 to 2.1	a, b
	0.6 to 1.5	c
Boiler feed	1.8 to 4.6	a, c
Pump suction and drain lines	1.2 to 2.1	a, b

[a]Crane Co. (1976). [b]Carrier (1960). [c]Grinnell Company (1951).

Table 5 Maximum Water Velocity to Minimize Erosion

Normal Operation, h/yr	Water Velocity, m/s
1500	4.6
2000	4.4
3000	4.0
4000	3.7
6000	3.0

Source: Carrier (1960).

Erosion

Erosion in piping systems is caused by water bubbles, sand, or other solid matter impinging on the inner surface of the pipe. Generally, at velocities lower than 30 m/s, erosion is not significant as long as there is no cavitation. When solid matter is entrained in the fluid at high velocities, erosion occurs rapidly, especially in bends. Thus, high velocities should not be used in systems where sand or other solids are present or where slurries are transported.

Noise Generation

Velocity-dependent noise in piping and piping systems results from any or all of four sources: turbulence, cavitation, release of entrained air, and water hammer. In investigations of flow-related noise, Marseille (1965), Ball and Webster (1976), and Rogers (1953, 1954, 1956) reported that velocities on the order of 3 to 5 m/s lie within the range of allowable noise levels for residential and commercial buildings. The experiments showed considerable variation in the noise levels obtained for a specified velocity. Generally, systems with longer pipe and with more numerous fittings and valves were noisier. In addition, sound measurements were taken under widely differing conditions; for example, some tests used plastic-covered pipe, while others did not. Thus, no detailed correlations relating sound level to flow velocity in generalized systems are available.

Water Hammer

When any moving fluid (not just water) is abruptly stopped, as when a valve closes suddenly, large pressures can develop. While detailed analysis requires knowledge of the elastic properties of the pipe and the flow-time history, the limiting case of rigid pipe and instantaneous closure is simple to calculate. Under these conditions,

$$\Delta p_h = \rho c_s V$$

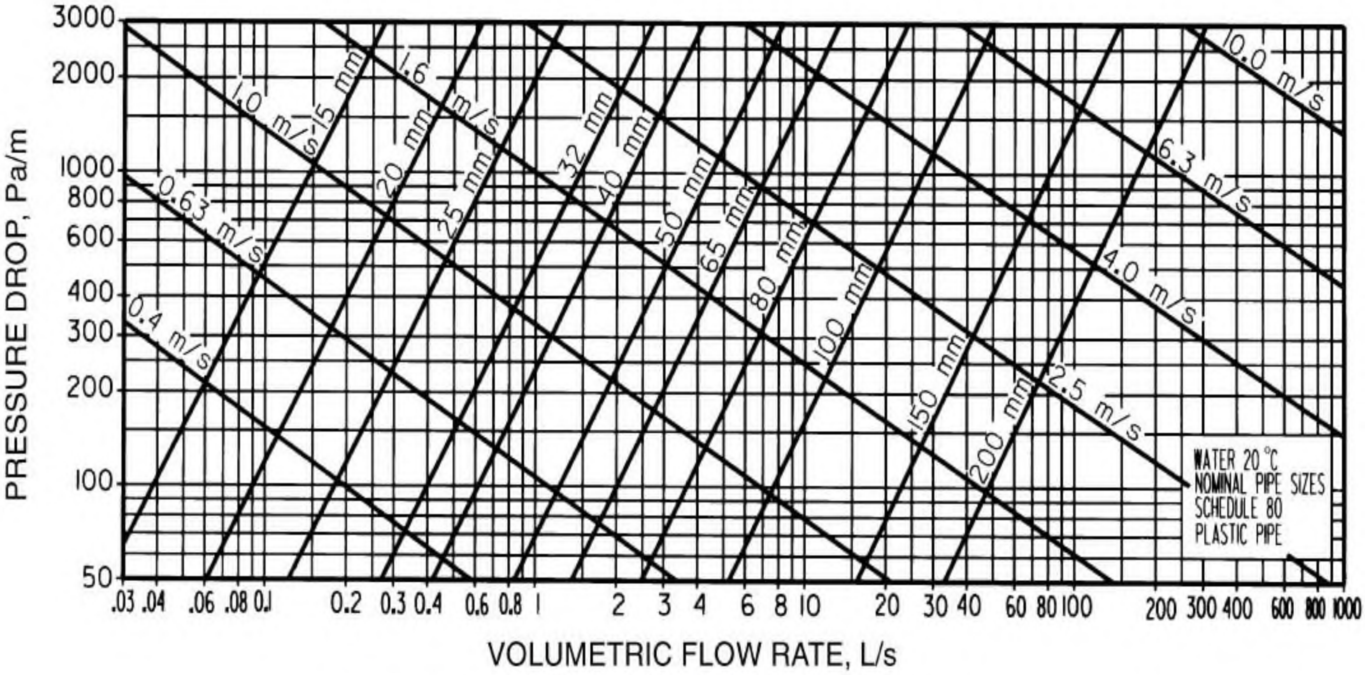
where

Δp_h = pressure rise caused by water hammer, Pa
ρ = fluid density, kg/m^3
c_s = velocity of sound in fluid, m/s
V = fluid flow velocity, m/s

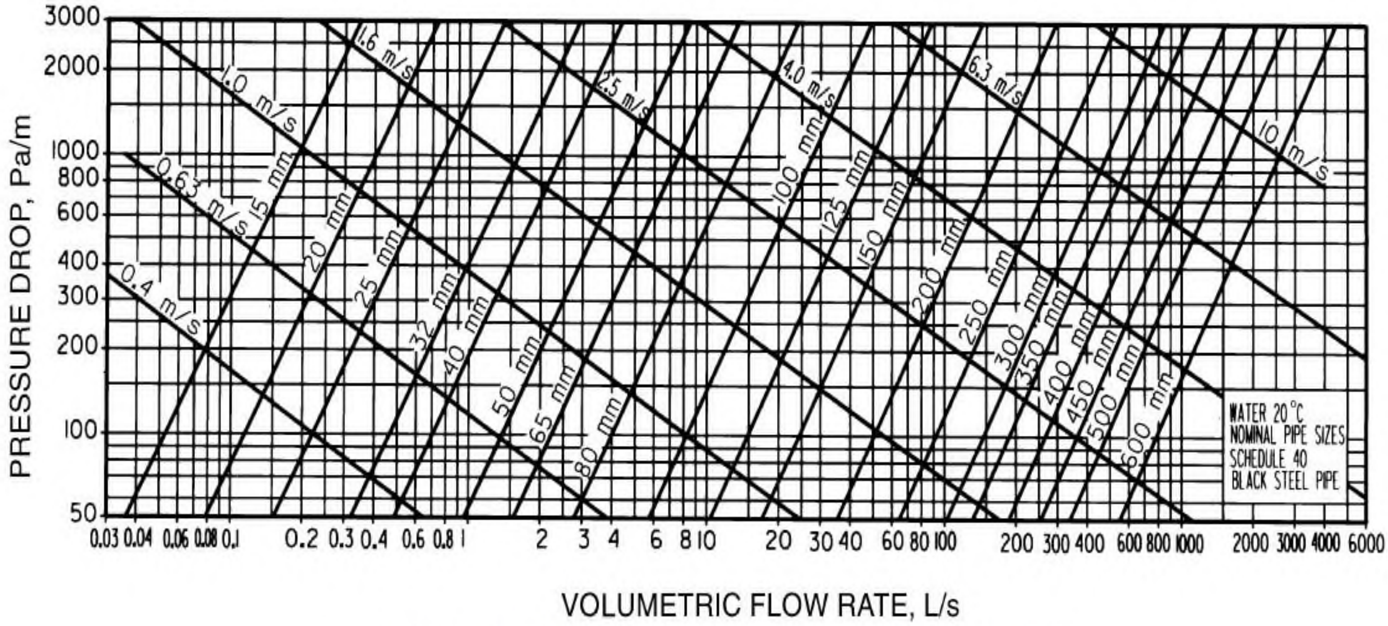
The c_s for water is 1439 m/s, although the elasticity of the pipe reduces the effective value.

Example 3. What is the maximum pressure rise if water flowing at 3 m/s is stopped instantaneously?

Solution: $\Delta p_h = 1000 \times 1439 \times 3 = 4.32$ MPa



Friction Loss for Water in Plastic Pipe (Schedule 80)



Friction Loss for Water in Commercial Steel Pipe (Schedule 40)

Table 6 Equivalent Length in Metres of Pipe for 90° Elbows

Velocity,	Pipe Size, mm													
m/s	15	20	25	32	40	50	65	90	100	125	150	200	250	300
0.33	0.4	0.5	0.7	0.9	1.1	1.4	1.6	2.0	2.6	3.2	3.7	4.7	5.7	6.8
0.67	0.4	0.6	0.8	1.0	1.2	1.5	1.8	2.3	2.9	3.6	4.2	5.3	6.3	7.6
1.00	0.5	0.6	0.8	1.1	1.3	1.6	1.9	2.5	3.1	3.8	4.5	5.6	6.8	8.0
1.33	0.5	0.6	0.8	1.1	1.3	1.7	2.0	2.5	3.2	4.0	4.6	5.8	7.1	8.4
1.67	0.5	0.7	0.9	1.2	1.4	1.8	2.1	2.6	3.4	4.1	4.8	6.0	7.4	8.8
2.00	0.5	0.7	0.9	1.2	1.4	1.8	2.2	2.7	3.5	4.3	5.0	6.2	7.6	9.0
2.35	0.5	0.7	0.9	1.2	1.5	1.9	2.2	2.8	3.6	4.4	5.1	6.4	7.8	9.2
2.67	0.5	0.7	0.9	1.3	1.5	1.9	2.3	2.8	3.6	4.5	5.2	6.5	8.0	9.4
3.00	0.5	0.7	0.9	1.3	1.5	1.9	2.3	2.9	3.7	4.5	5.3	6.7	8.1	9.6
3.33	0.5	0.8	0.9	1.3	1.5	1.9	2.4	3.0	3.8	4.6	5.4	6.8	8.2	9.8

Chilled Water

Chilled water typically has a supply temperature of between 42°F and 48°F. Historically, the return temperature was often chosen to be 10°F above the flow temperature. With the higher cost of fuel and the concern over energy usage, it is usually cost effective to design for a higher difference of 15°F or even 20°F. The higher return temperatures require larger coils, and create challenges when high dehumidification is required.

On the other hand, doubling the temperature difference halves the volume flow, and, consequently, reduces the purchase cost of piping and pumps, as well as substantially reducing ongoing pumping power costs.

With a flow temperature in the range 42°F to 48°F, the piping must be insulated to reduce heat gain and avoid condensation. The insulation requires a moisture barrier on the outside to prevent condensation on the pipe.

Chillers, the refrigeration equipment used to produce chilled water, mostly use a direct expansion evaporator. Therefore, the flow must be maintained fairly constant to prevent the possibility of freezing the water. The chiller

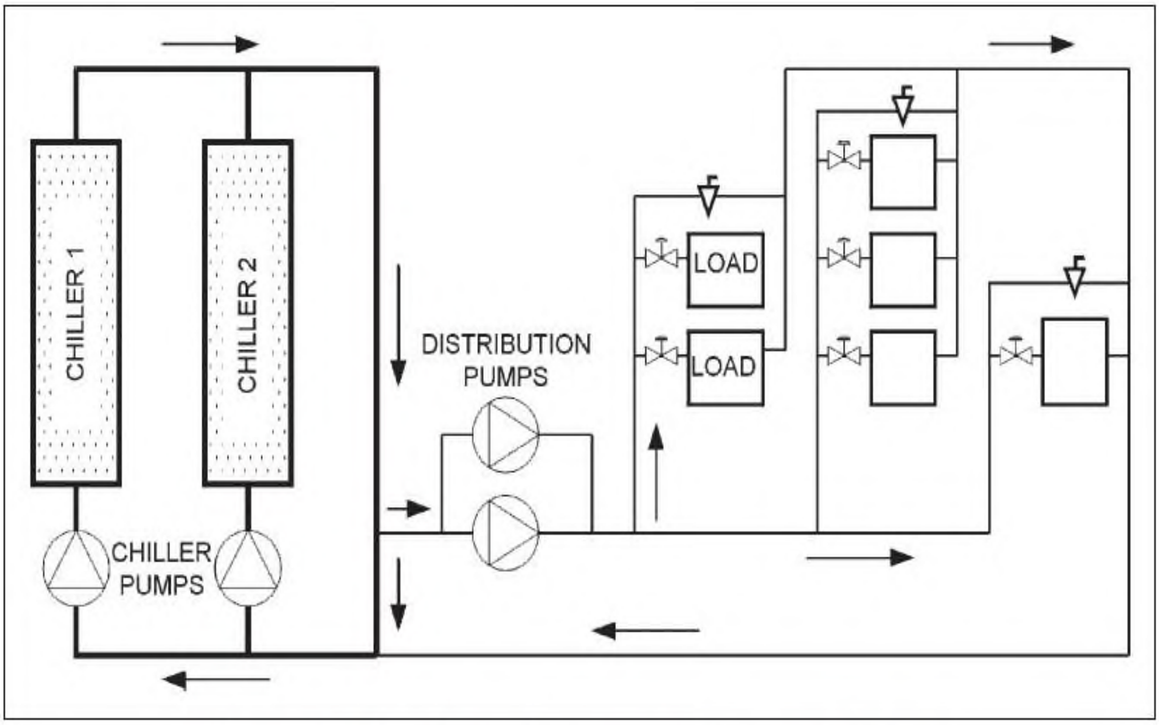


Figure (1) : Chiller System with Decoupled Flows

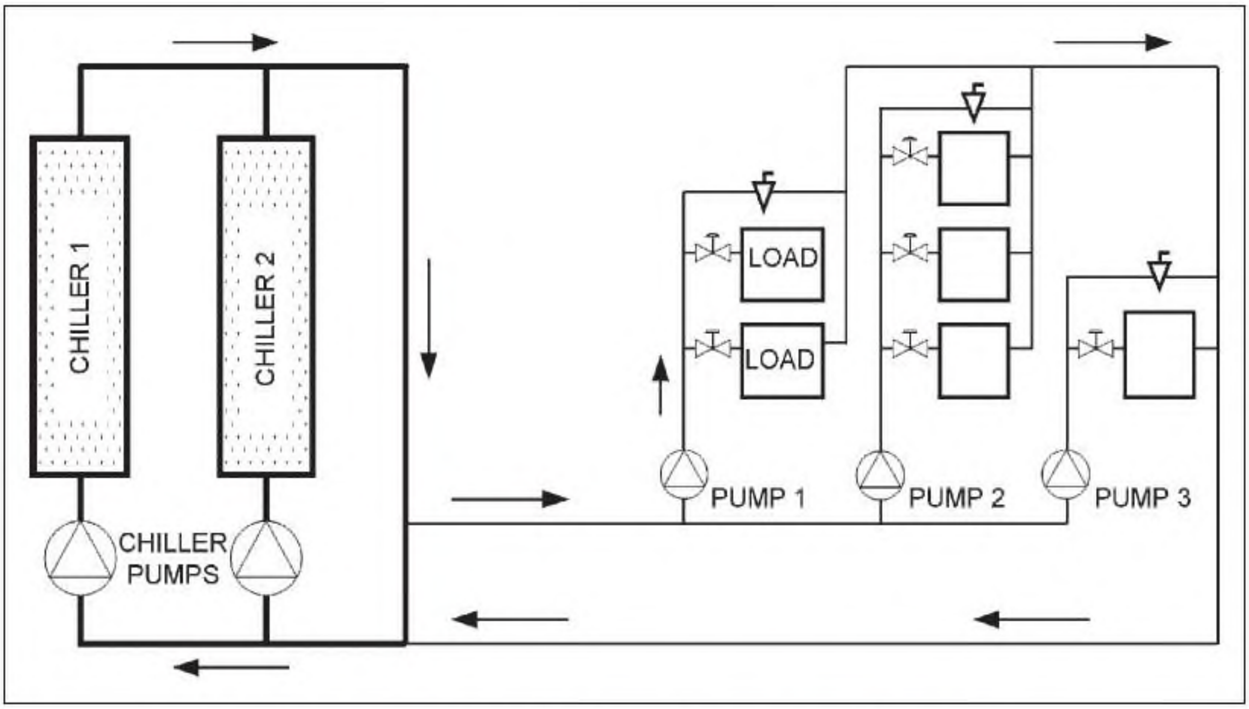


Figure (2) : Distributed Secondary Pumping

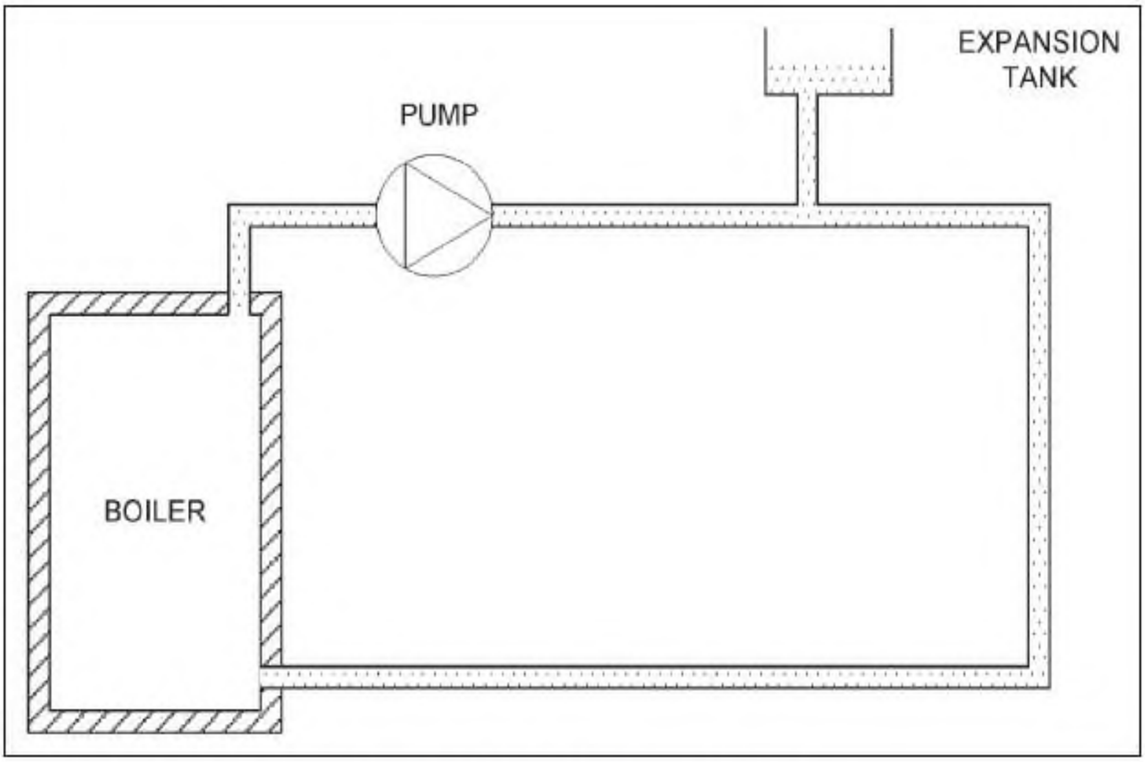


Figure (3) : Closed Water Circuit

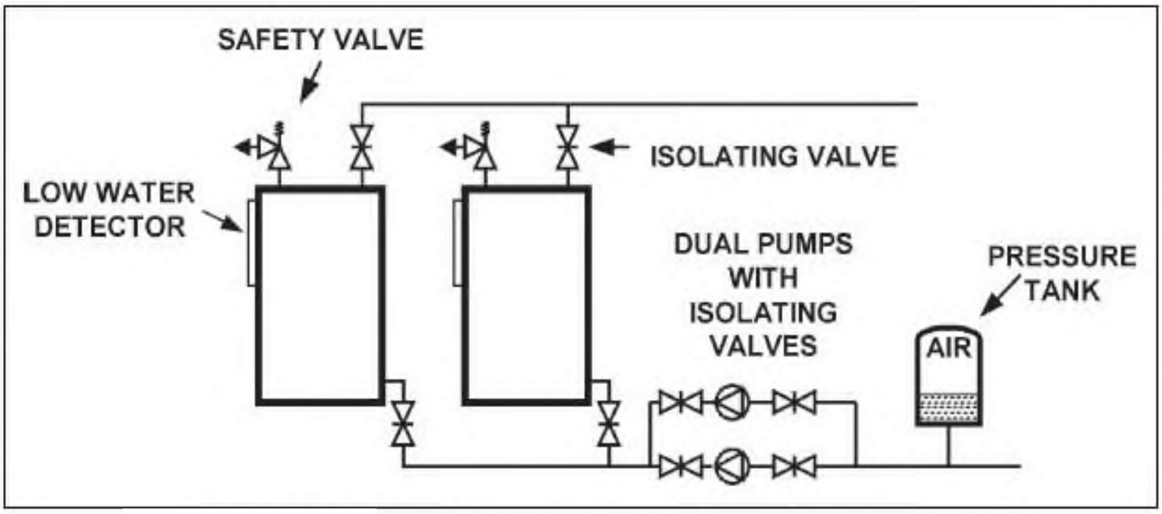


Figure (4) : Hot Water Heating System with Two Boilers

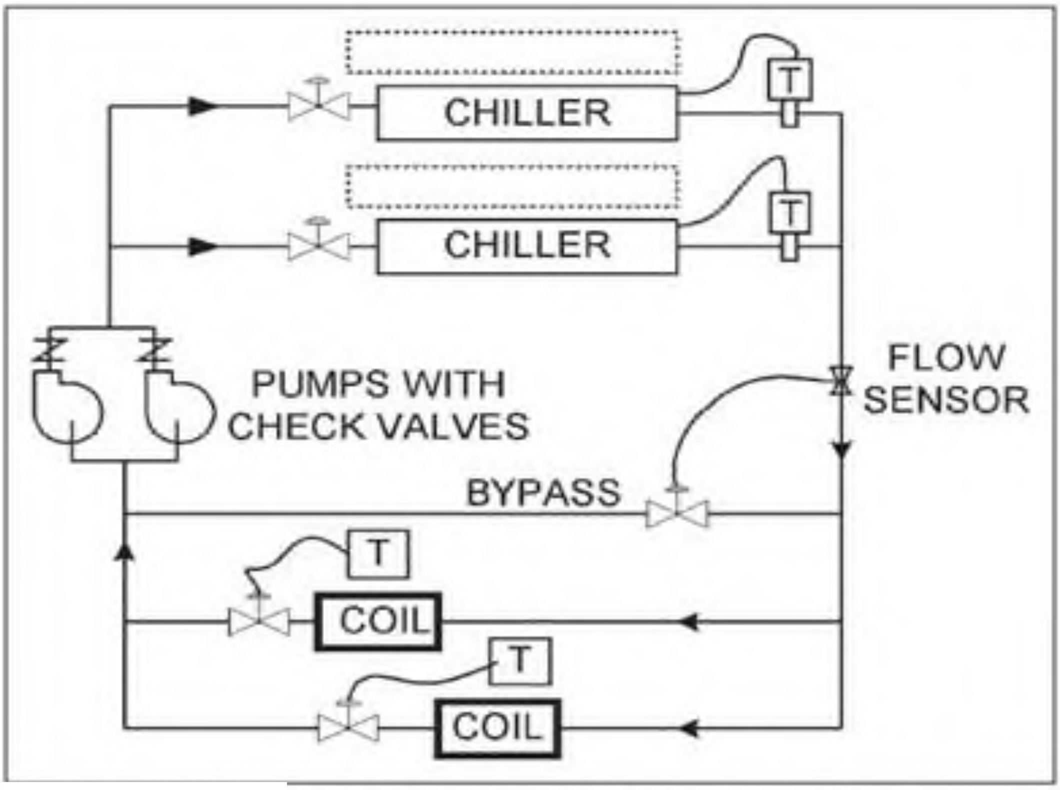
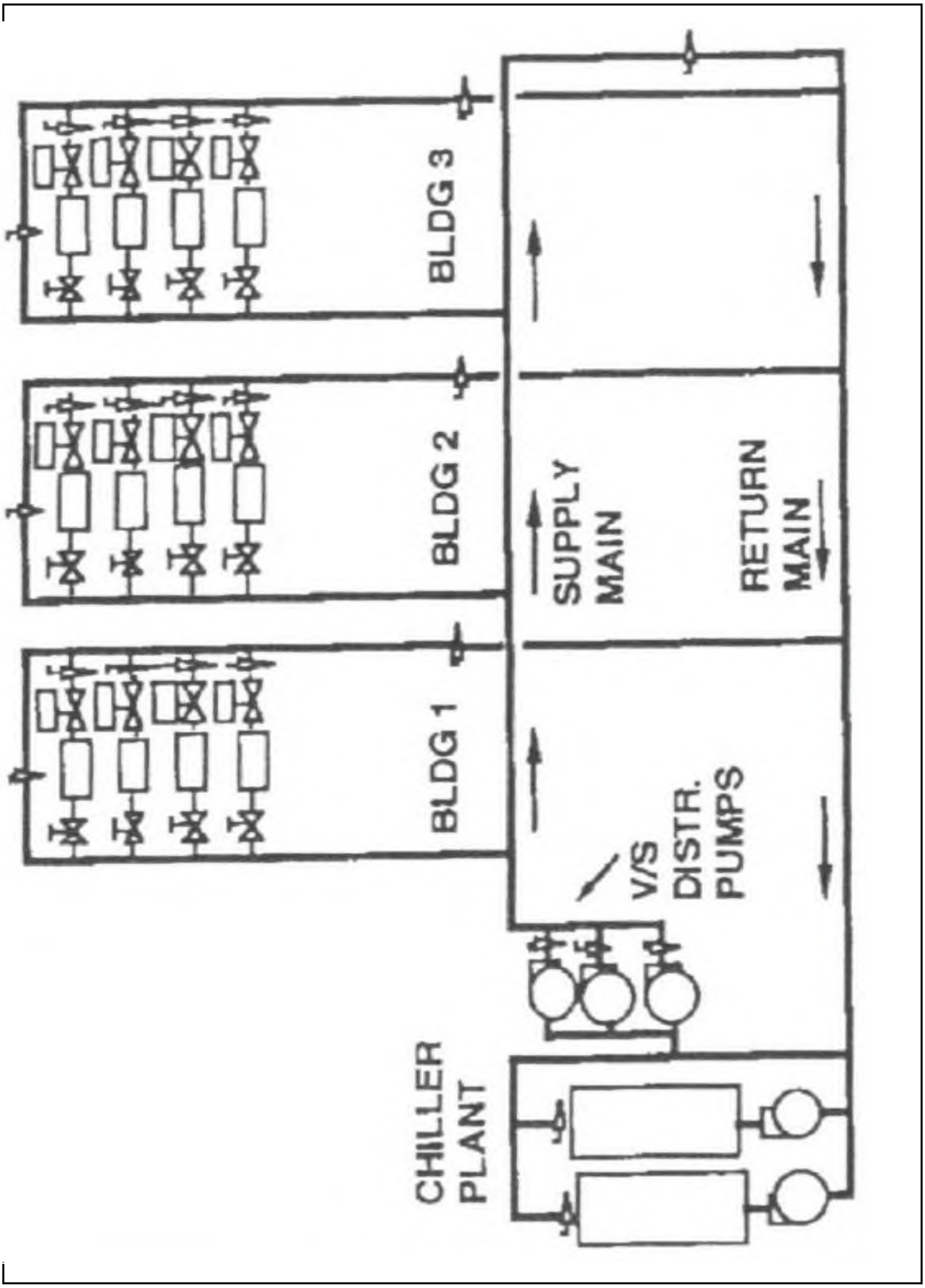
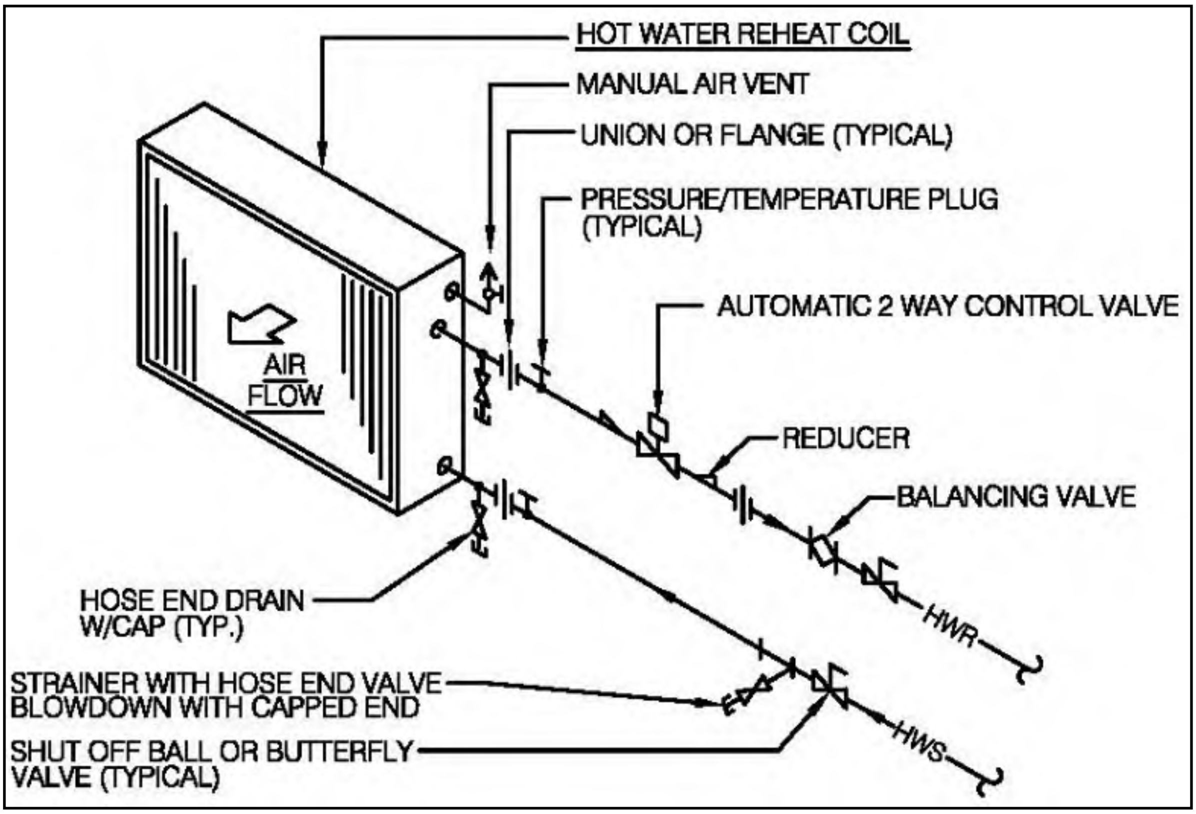


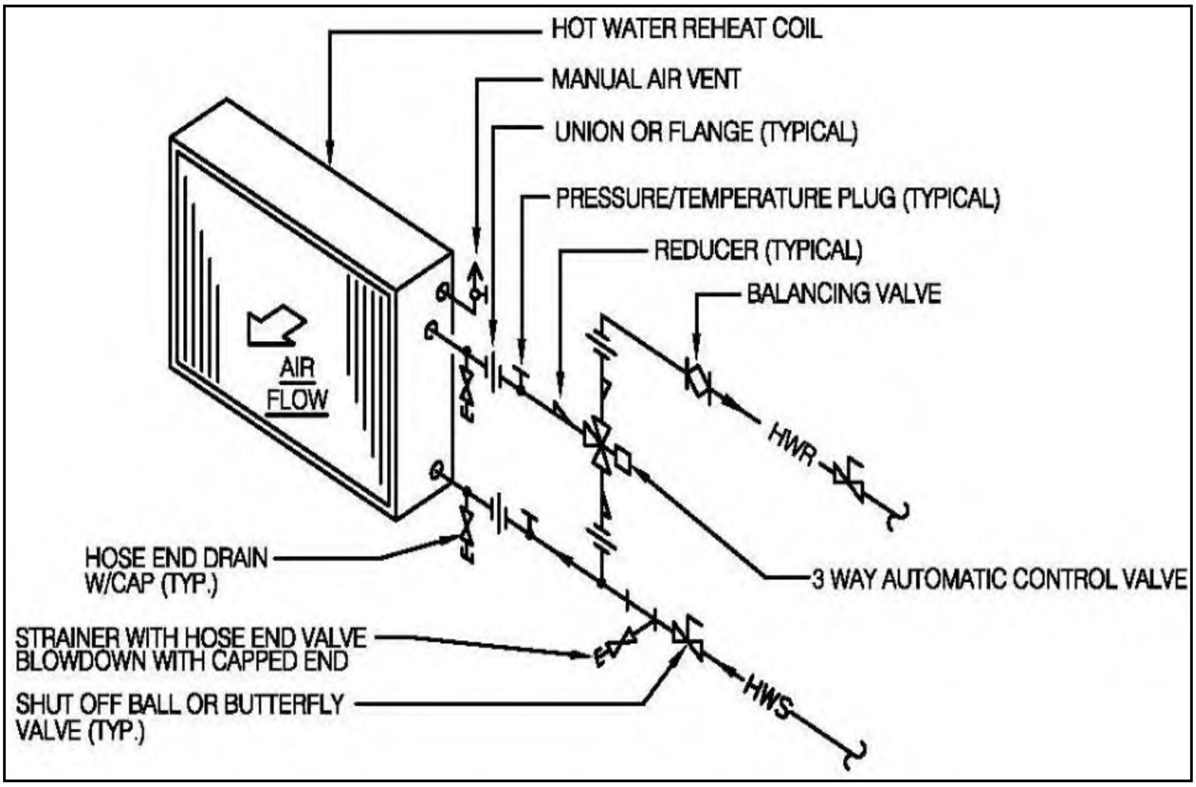
Figure (5) : Two Chiller Piping with Constant Chiller Flow



Water System Layout



Two Way Valve Fan Coil Unit Details



Three Way Valve Fan Coil Unit Details

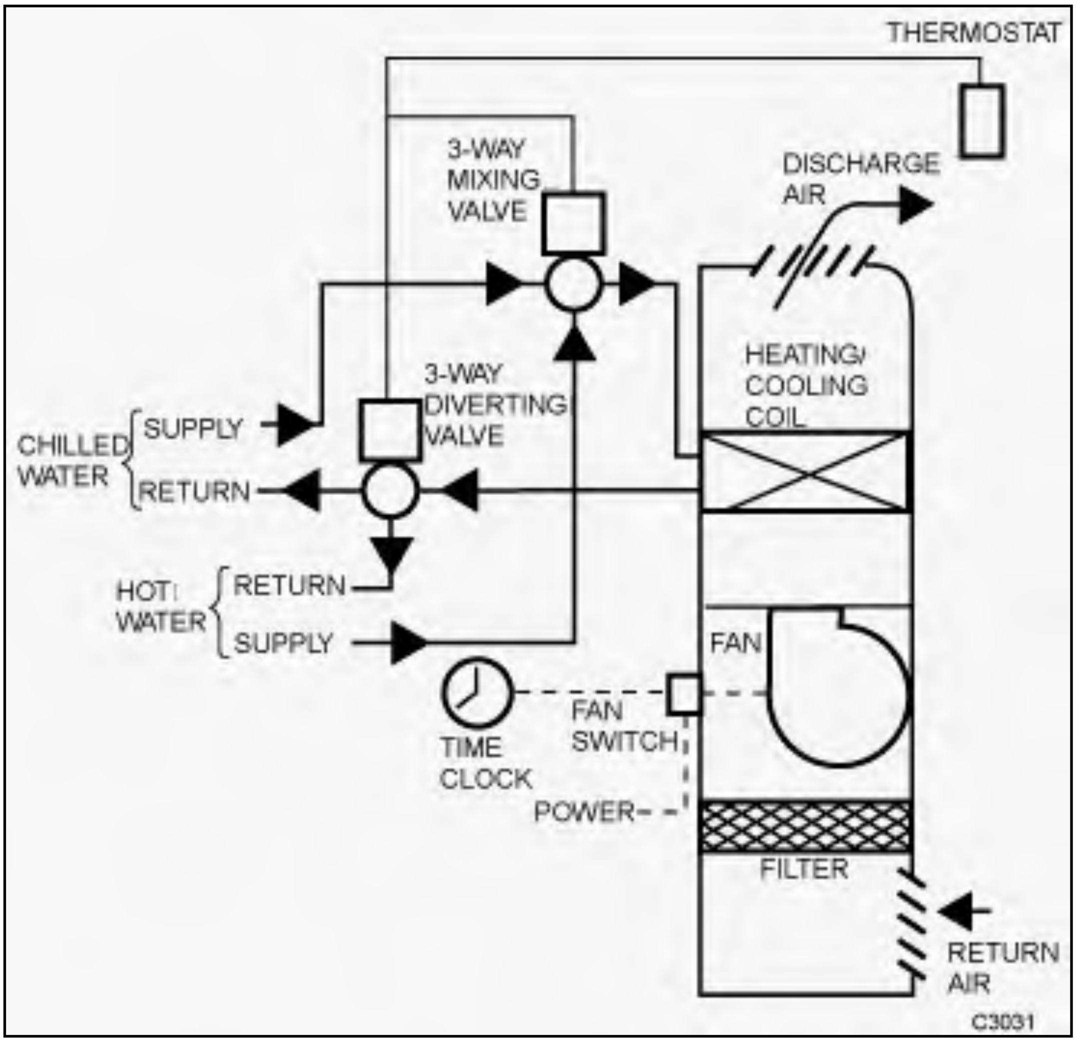
Difference between AHU and FCU:

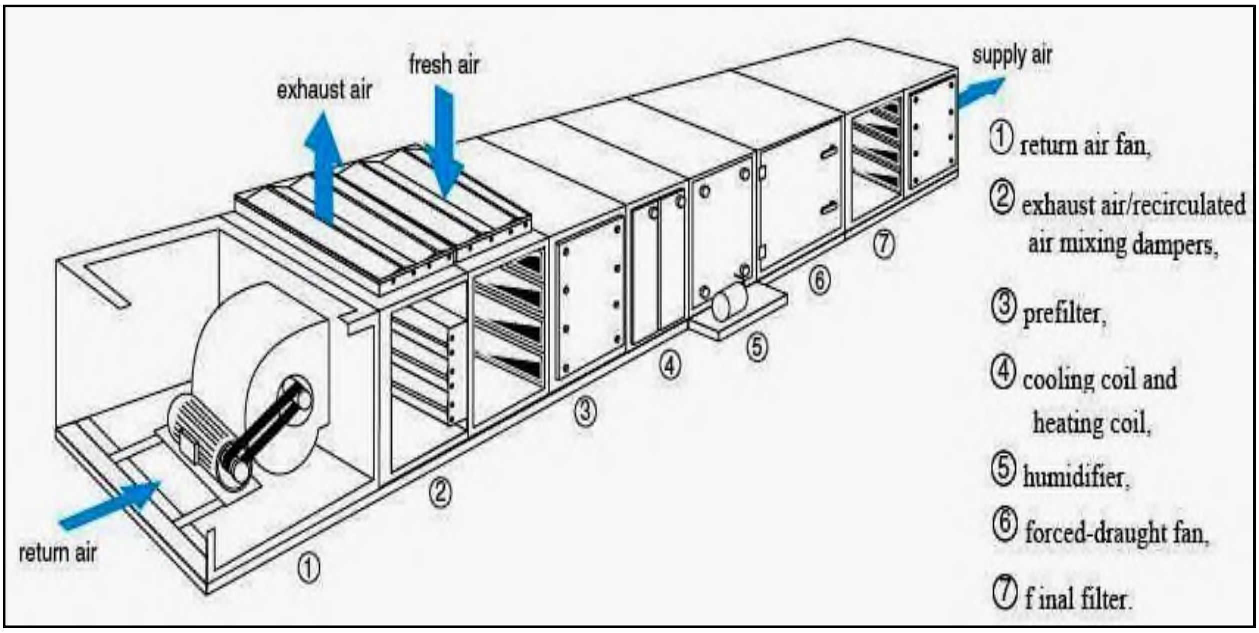
AHU and FCU are both included in the HVAC system. The latter is an acronym that describes multiple systems of heating, ventilating and A/C. AHU, completely known as air handling unit is different from FCU or the fan coil unit. AHUs are usually connected to a central HVAC system whereas an FCU can function or be installed itself. Because of this, it is often the AHU that is used to ventilate an entire building whereas FCUs are used in smaller and often local spaces only. It is no surprise that AHU is the bigger HVAC system not to mention, the FCU is also regarded as the smaller version of the AHU. In this regard, the smaller FCUs can be called terminal units. Because of the magnitude of the AHU system, it usually makes use of the air from the outside environment. Thus, it treats outside air and brings it inside the vicinity with the use of special ducts while an FCU system on the contrary just circulates the inside air. The latter usually does not have any duct system that's why it is obviously smaller in size as previously mentioned. It is only composed of a simple coil and fan. Ordinarily, AHUs have certain features that are not present in the FCU type.

A.H.U is designed and manufactured to meet the international standard for air ventilation (fresh air/wet system) indoor air quality, comfort air conditioning and specialized application like clean room. A.H.U. for clean room consists of mixing chamber, pre filter, fine filter, coil section, fan section, optional pan humidifier and dehumidifier (strip heater). These are double screen section panels with aluminum hollow extruded profiles and thermoplastic, corner joints. AHUs have several sections for reheating and even humidifying. FCUs don't have sections such as these. Perhaps the only advantage that the FCU has over the AHU is that generally, AHUs can have several fans or blowers installed in its system. Traditionally, the AHU's blowers are placed at the point where the ducts originate or at the terminal end of the air handler unit. FCUs have a different approach to blowers.

Summary:

1. AHU is generally a bigger system than FCU.
2. AHU is more complex than the FCU and that AHU are often used in bigger establishments or spaces.
3. The AHU system usually channels air through ducts whereas the FCU don't have any ductworks.
4. AHU system treats outside air while FCUs basically recycle or re-circulates air.
5. AHU have sections for reheating and humidifying whereas the FCU does not have any.
6. FCU are often observed to be noisier than the AHU.





Air Handling Unit Details

Net Positive Suction Head:

- The lowest absolute water pressure at the suction inlet of the centrifugal pump must exceed the saturated vapour pressure at the corresponding water temperature. If the absolute pressure is lower than the saturated vapour pressure, the water evaporates and a vapour pocket forms between the vanes in the impeller. As the pressure increases along the water flow, the vapour pocket collapses and may damage the pump.
- This phenomenon is called *cavitation*.
- The sum of the velocity head at the suction inlet and the head loss (due to friction and turbulence) between the suction inlet and the point of lowest pressure inside the impeller is called the *net positive suction head required* (NPSHR), in feet (meters). This factor is determined by the pump manufacturer for a given centrifugal pump.

$$\text{NPSHA} = H_{at} + H_{suc} - H_f - 2.31 p_{vap}$$

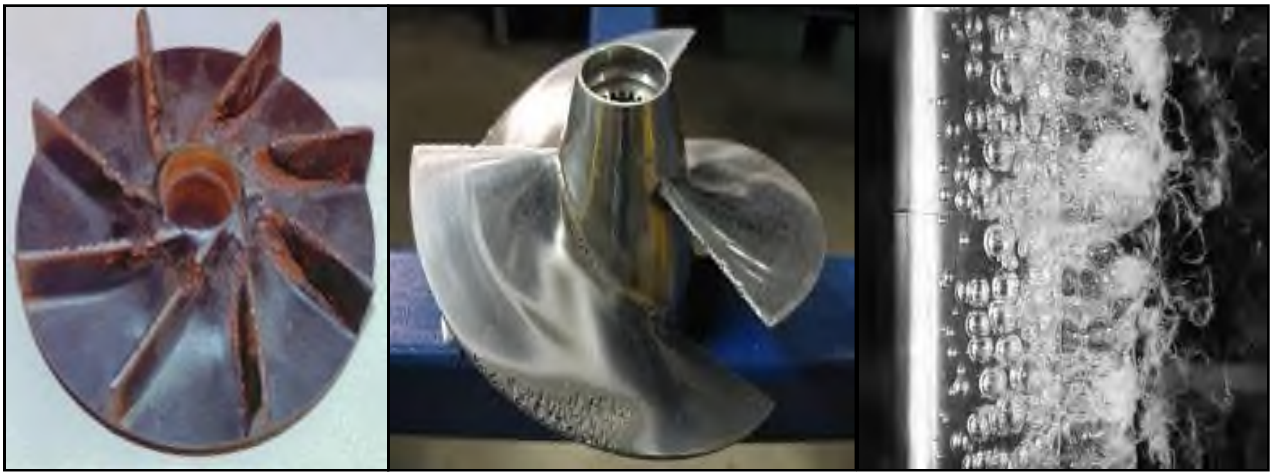
where H_{at} = atmopheric pressure, usually expressed as 34 ft (10.3 m or 101 kPa) of water column

H_{suc} = static suction head, ft (m)

ΔH_f = head loss due to friction and dynamic losses of suction pipework and fittings, ft (m)

p_{vap} = saturated water vapor pressure corresponding to water temperature at suction inlet, psia (kPa abs.)

NPSHA must be greater than NPSHR to prevent cavitation.



Effect of Cavitation

Pump Sizing

There are two items required to size a pump;

- Fluid flow rate
- Pressure to be developed.

The pressure that should be developed by the pump should equal the Pressure Drop in the system.

This is usually found from pipe sizing tables or from other methods.

See Science section of these notes - Fluids section; Head Loss due to friction in a pipe.

The flow rate of fluid is also found from pipe sizing tables or given in other data.

Add 20% margin to pump pressure to allow for future extensions and the system getting less efficient.

The designer must be careful when adding a margin to pump pressure since too much pressure can lead to ‘pumping over’ in open systems and other problems.

Some pump catalogues have units of head instead of pressure.

For conversion;

Pressure (Pa) = density of water x acceleration due to gravity x head (m)

Or **$P = \rho \times g \times H$**

Where, P=Pump pressure (Pa)

ρ =Density of water ρ approx.1000 kg/m^3.

g=Acceleration due to gravity 9.81 m/s^2.

H=Head (m)

$H = P / \rho \times g$

$H = P/1000 \times 9.81$

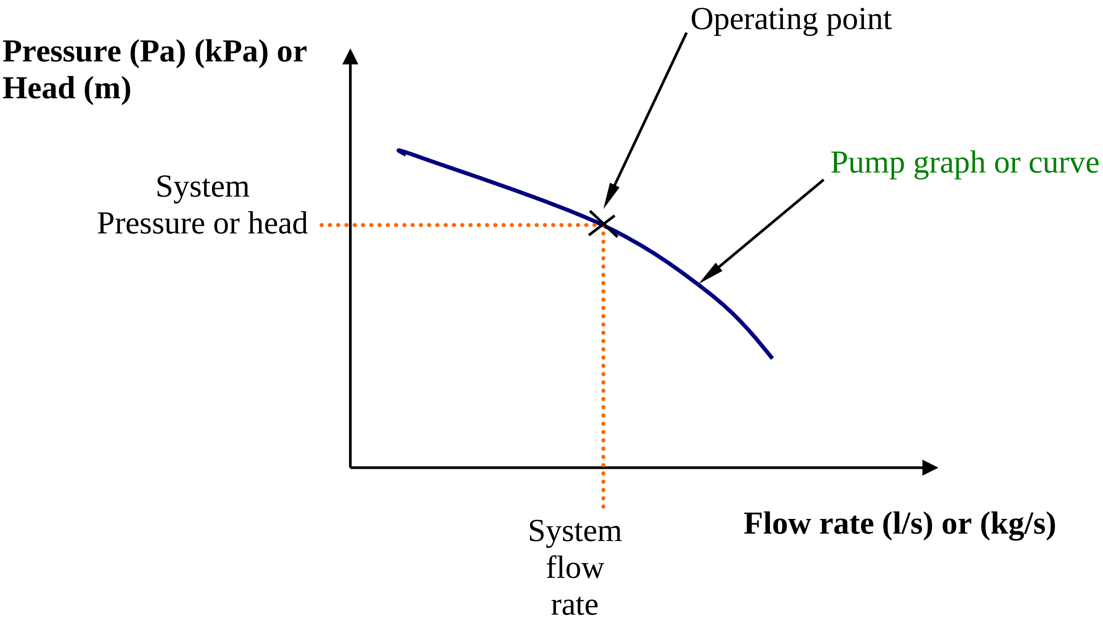
$H = P/9810$

The flow rate of water that the pump delivers will be the flow rate in the section in which the pump is installed.

A 20% margin may be added to this flow rate to allow for future extensions to the system.

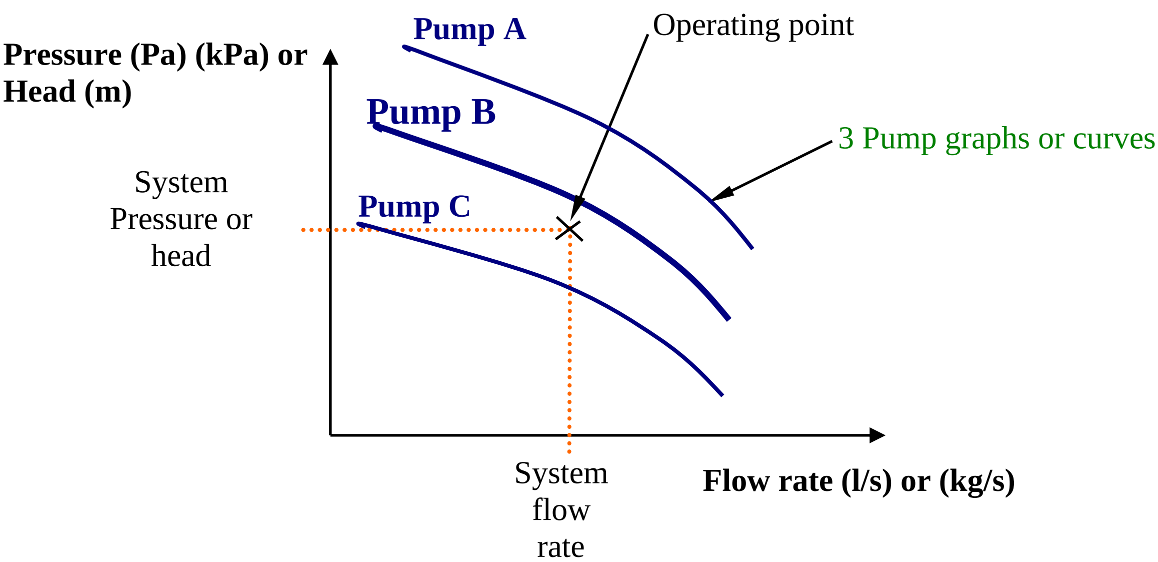
A pump catalogue may be consulted to choose a suitable unit.The operating point can be super-imposed on the pump graph for pressure (head) against flow rate in kg/s or l/s. It is best to choose a pump with the operating point near the lower speeds or the bottom end of the performance curve so that the pump will not be operating at its maximum capacity, thus allowing little room for error or margin.

A typical pump sizing curve is shown below with a system operating point superimposed on the curve.



Typical Pump Sizing Curve

Not all system operating points are directly on top of a pump graph or curve as shown below.It would be best to choose a pump on the curve above the operating point, i.e. Pump B since the output of both pressure and flow rate will be slightly above that required and not below.



Typical Pump Sizing Curve - 3 Pumps

Similarly if a pump has three speeds then three curves will be shown above .It would be best to operate a pump at a lower speed if possible to prolong the life of the pump and bearings. The diagram below shows a 3-speed pump with the operating point between speed No.1 and No.2. The pump would then be installed to run at speed No.2; this means that if the system is extended at a later date the pump speed may be increased to accommodate this increase in flow rate and pressure.

The pressure **that should be developed by the pump should equal the** Pressure Drop **in the Index Circuit.**

The Index Circuit is the part of the system with the highest pressure drop.
Therefore:

Pump pressure =pressure drop in Index Circuit.

Add 20% margin to **pump pressure** to allow for future extensions and the system getting less efficient. The designer must be careful when adding a margin to **pump pressure** since too much pressure can lead to 'pumping over' in open systems and other problems.Some pump catalogues have units of head instead of pressure.

For conversion;

Pressure (Pa) = density of water x acceleration due to gravity x head (m)

Or $$\mathbf{P = \rho \times g \times H}$$

Where, Density of water (ρ) =1000 kg/m^3.
Acceleration due to gravity (g) =9.81 m/s^2.

Therefore;

$$\mathbf{H = P / \rho \times g = P / (1000 \times 9.81)}$$

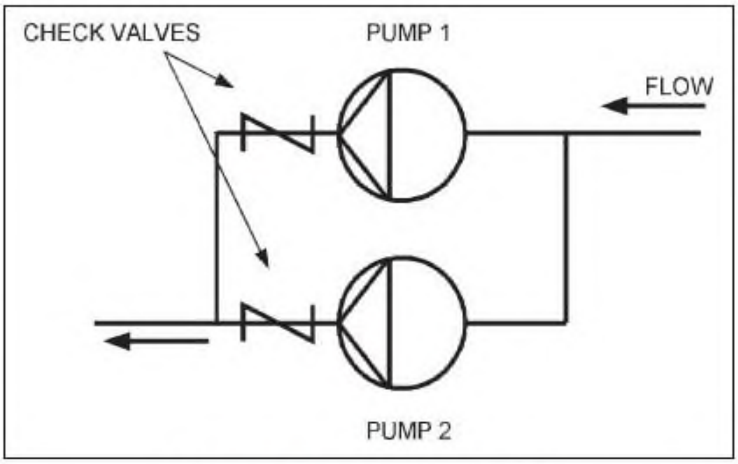
$$\mathbf{H = P / 9810}$$

The flow rate of water that the pump delivers will be the flow rate in the section in which the pump is installed.

A 20% margin may be added to this flow rate to allow for future extensions to the system. A pump catalogue may be consulted to choose a suitable unit. Smaller pumps can be in-line, that is, installed in the pipeline. Larger pumps may be seated on a concrete base, these tend to be end suction pumps where the water is sucked into the pump end and comes out at 90 degrees at the outlet.

The operating point can be super-imposed on the pump graph for pressure (head) against flow rate in kg/s or l/s.

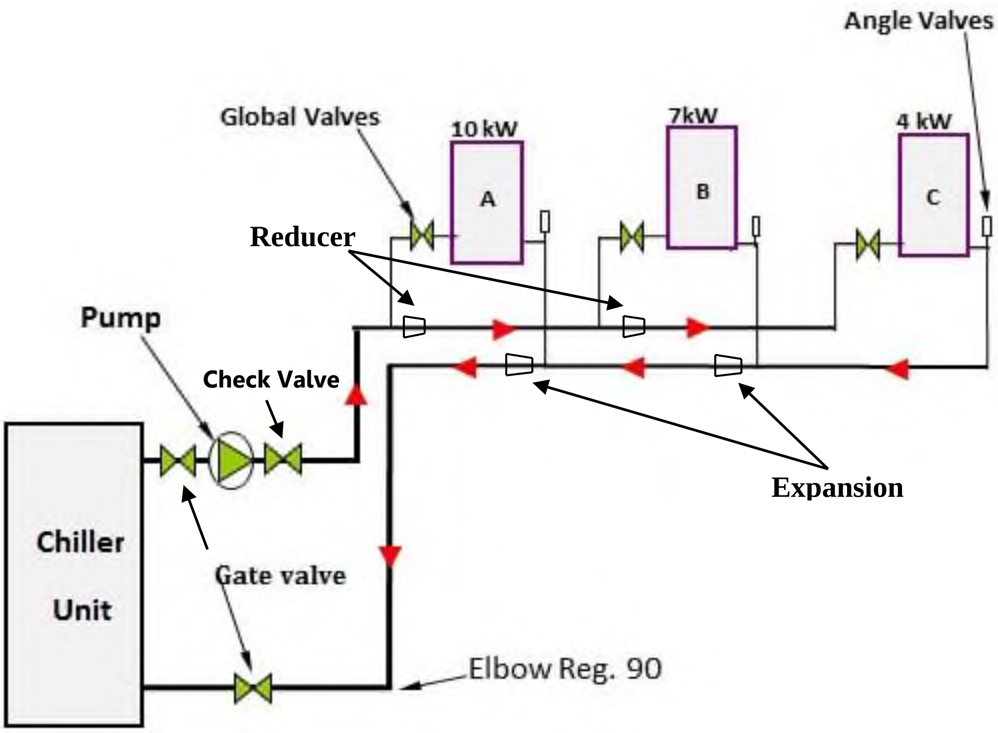
It is best to choose a pump with the operating point near the lower speeds or the bottom end of the performance curve so that the pump will not be operating at its maximum capacity, thus allowing little room for error or margin.



Pump in Parallel

Problem: Size the pipework for the water cooling system shown below, also determine the pressure drops. The total lengths of the section are:

Section	Length of Discharge Flow (m)	Length of Return Flow(m)
1	16	6
2	6	6
3	6	16



Solution:

Mt =(Q1+Q2+Q3) / (5.5*4.2) =0.91 kg/s

Now, we divided the pipe of water system into following sections:

Discharge Line:

Section (1)

Section (1) ; Mt=0.91kg/s			
D (mm)	25	32	40
V (m/s)	1.7	1	0.73
ΔP (pa/m)	1600	380	180
Le (m)	0.9	1.3	1.5
Statue	V. large ΔP	large ΔP	Acceptable

We will select pipe diameter =40mm for section (1)

Friction Loss for section **(1)** =180×16=2880 Pa

Dynamic Loss for section **(1)** =0.5(1000) $(0.73)^2$ [2×1.2 Gate Valve +1.6 Tee Branch +0.19 Gate Valve+2.5 Swing Check Valve +2.9 Reducer] =2555 Pa

Total Loss for section (1) =5435 Pa

Section (2)

Section (2) ; Mt=0.47kg/s			
D (mm)	20	25	32
V (m/s)			0.52
ΔP (pa/m)	1500	480	120
Le (m)	0.7	0.9	1.1
Statue	V. large ΔP	large ΔP	Acceptable

We will select pipe diameter =32mm for section (2)

Friction Loss for section (2) =120*6=720 Pa

Dynamic Loss for section (2) =0.5(1000) $(0.52)^2$ × [1.7 T.Branch+3.6 reducer] =716.5 Pa

Total Loss for section (2) =1436.5 Pa

Section (3)

Section (3) ; Mt=0.17kg/s		
D (mm)	20	25
V (m/s)	0.54	0.33
ΔP (pa/m)	240	77.5
Le (m)	0.6	0.7
Statue	Acceptable	Low velocity

We will select pipe diameter =20mm for section (3)
Friction Loss for section (3) =240*6=1440 Pa
Dynamic Loss for section (3) =0.5× (1000) × $(0.54)^2$ × [2×1.7 Elbow +10 Global Valve] =1953.72 Pa
Total Loss for section (3) =3393.72 Pa

Total Pressure Loss for Discharge Line =10265.22 Pa

Return Line:
Section (1)
D=20mm
Friction Loss for section (1) =240×6=1440 Pa
Dynamic Loss for section (1) =0.5× (1000) $(0.54)^2$ × [2×1.7 Elbow +6.1 Angle Valve +0.9 T Line + 6.1 Expansion]= 2405.7 Pa
Total Loss for section (1) = 3845.7 Pa

Section (2)
D=32mm
Friction Loss for section (2) =120*6=720pa
Dynamic Loss for section (2) =0.5(1000) $(0.52)^2$ (3.6 Expansion +0.9T Line) = 608.4 Pa
Total Loss for section (2) = 1328.4 Pa

Section (3)
D=40mm
Friction Loss for section (3) =180×16= 2880 Pa
Dynamic Loss for section (2) =0.5× (1000) ×$(0.73)^2$ ×[2×1.2 Elbow + 0.19 Gate Valve]= 690.1055 Pa
Total Loss for section (3) = 3570.1 Pa

Total Pressure Loss for Return Line =8744.2 Pa

Total Pressure Loss for Discharge & Suction Lines (Pump Capacity) = 8744.2 + 10265.2 = 19009.4 Pa
Pump Head (m) =19009.4 / (9.8*1000) =1.94 m

Table 3.23 Heat emission or absorption from insulated pipes

Nominal pipe size (mm)	Heat emission or absorption from insulated pipework / $W.m^{-1}.°C^{-1}$ temperature difference for given values of insulation thermal conductivity / $W.m^{-1}.K^{-1}$																			
	Thickness of insulation (mm)																			
	0.025					0.040					0.055					0.070				
	12.5	19	25	38	50	12.5	19	25	38	50	12.5	19	25	38	50	12.5	19	25	38	50
15	0.18	0.14	0.12	0.10	0.09	0.27	0.22	0.19	0.16	0.14	0.34	0.29	0.25	0.21	0.19	0.41	0.35	0.31	0.27	0.24
20	0.21	0.16	0.14	0.11	0.10	0.31	0.25	0.22	0.18	0.16	0.40	0.33	0.29	0.24	0.21	0.47	0.40	0.36	0.30	0.26
25	0.25	0.19	0.16	0.13	0.11	0.36	0.29	0.25	0.20	0.18	0.47	0.38	0.33	0.27	0.24	0.56	0.46	0.41	0.34	0.30
32	0.29	0.22	0.19	0.15	0.13	0.43	0.34	0.29	0.23	0.20	0.55	0.45	0.39	0.31	0.27	0.66	0.54	0.47	0.38	0.34
40	0.32	0.25	0.21	0.16	0.14	0.48	0.37	0.32	0.25	0.21	0.61	0.49	0.42	0.33	0.29	0.72	0.59	0.52	0.42	0.36
50	0.39	0.29	0.24	0.18	0.16	0.57	0.44	0.37	0.29	0.24	0.73	0.58	0.49	0.39	0.33	0.86	0.70	0.60	0.48	0.41
65	0.47	0.35	0.29	0.22	0.18	0.69	0.55	0.44	0.34	0.28	0.88	0.69	0.58	0.45	0.38	1.04	0.83	0.71	0.56	0.48
80	0.54	0.40	0.33	0.24	0.20	0.79	0.60	0.50	0.38	0.32	1.0	0.78	0.66	0.50	0.43	1.19	0.94	0.80	0.63	0.53
100	0.67	0.49	0.40	0.29	0.24	0.98	0.74	0.61	0.45	0.38	1.25	0.96	0.80	0.61	0.51	1.47	1.16	0.98	0.75	0.63
125	0.81	0.58	0.47	0.34	0.28	1.18	0.88	0.72	0.53	0.44	1.49	1.14	0.95	0.71	0.59	1.76	1.38	1.16	0.88	0.73
150	0.96	0.69	0.55	0.40	0.32	1.37	1.02	0.83	0.61	0.50	1.74	1.32	1.09	0.81	0.67	2.05	1.59	1.33	1.01	0.84
200	1.22	0.88	0.70	0.50	0.40	1.78	1.32	1.07	0.77	0.63	2.26	1.70	1.40	1.03	0.84	2.66	2.05	1.71	1.27	1.05
250	1.50	1.07	0.86	0.60	0.48	2.19	1.61	1.30	0.94	0.75	2.77	2.09	1.71	1.25	1.01	3.27	2.51	2.08	1.54	1.26
300	1.77	1.26	1.00	0.70	0.56	2.58	1.89	1.52	1.09	0.87	3.26	2.44	2.00	1.45	1.17	3.84	2.94	2.48	1.79	1.46

Note: The pipes sizes are to BS1387 and BS3600[37, 38]. It is assumed that the outside surface of the insulation has been painted, is in still air at 20°C and h_{so} = 10 $W.m^{-2}.K^{-1}$

Table 3.17 Heat emission from single horizontal steel pipes (ε = 0.95) freely exposed in surroundings at 20°C

Temp. difference between surface and surroundings (°C)	Heat emission/W.m^{-1} for stated pipe nominal size* and outside diameter† d_{op} (mm)															
	* 15	20	25	32	40	50	65	80	100	125	150	200	250	300	350	400
	† 21.3	26.9	33.7	42.4	48.3	60.3	76.1	88.9	114.3	139.7	168.3	219.1	273.0	323.9	355.6	406.4
−15	−12	−15	−18	−22	−24	−29	−36	−41	−52	−62	−71	−92	−112	−130	−142	−160
−10	−7.7	−9.4	−11	−14	−16	−19	−23	−27	−33	−40	−46	−60	−73	−85	−92	−104
−5	−3.5	−4.3	−5.3	−6.5	−7.3	−8.9	−11	−13	−16	−19	−22	−28	−35	−40	−44	−50
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
5	3.6	4.5	5.5	6.7	7.5	9.1	11	13	16	19	23	29	36	42	46	52
10	8.1	9.8	12	15	16	20	25	28	35	42	49	63	77	90	98	111
15	13	16	19	23	26	32	39	45	56	67	78	100	123	143	156	176
20	18	22	27	33	37	45	55	63	78	94	109	140	171	199	217	245
25	24	29	35	43	48	58	71	81	102	122	141	182	221	258	281	317
30	29	36	44	53	60	72	88	101	126	151	175	226	275	320	349	393
35	35	43	53	64	72	87	106	122	152	182	211	272	331	385	419	473
40	42	51	62	75	84	102	125	143	179	214	248	319	389	453	493	556
45	48	59	71	87	98	118	145	166	207	247	287	369	449	524	570	642
50	55	67	81	99	111	135	165	189	236	281	327	421	512	597	649	732
55	62	75	92	112	125	152	186	213	266	317	368	474	577	673	732	825
60	69	84	102	125	140	169	207	238	297	354	411	529	644	751	817	921
65	77	93	114	138	155	188	230	263	329	392	456	586	714	832	905	1020
70	84	103	125	152	170	206	253	290	362	432	502	646	786	916	997	1120
75	92	112	137	167	186	226	277	317	396	473	549	706	860	1000	1090	1230
80	100	122	149	181	203	246	301	345	431	515	598	769	937	1090	1190	1340
100	135	164	200	244	273	331	406	466	582	695	808	1040	1270	1480	1610	1820
120	173	211	257	314	352	427	523	600	750	897	1040	1340	1640	1910	2080	2350
140	215	262	320	391	438	532	653	750	937	1120	1310	1680	2050	2400	2610	2950
160	261	318	389	476	534	648	796	915	1140	1370	1600	2060	2520	2940	3200	3620
180	311	380	465	569	638	776	954	1100	1370	1650	1920	2480	3030	3540	3860	4360
200	366	447	547	670	753	916	1130	1300	1620	1950	2270	2940	3600	4200	4580	5180
220	425	520	637	781	878	1070	1320	1510	1900	2280	2660	3450	4220	4930	5380	6090
240	490	600	735	902	1010	1240	1520	1750	2200	2650	3090	4000	4900	5740	6260	7080
260	560	686	842	1030	1160	1420	1750	2010	2530	3040	3550	4610	5650	6620	7220	8180
280	635	780	957	1180	1320	1620	2000	2300	2890	3480	4060	5280	6470	7590	8280	9380
300	717	881	1080	1330	1500	1830	2260	2610	3280	3950	4620	6010	7370	8650	9440	10 700
320	806	990	1220	1500	1690	2060	2550	2940	3710	4470	5230	6800	8350	9800	10 700	12 100

*These pipe sizes are to BS1387: 1967 and BS 3600: 1973[37,38] †These outside diameters are to BS3600: 1997[39]

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