MCE 503

REFRIGERATION AND AIR-CONDITIONING II

PREPARED BY

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1.0 ELEMENTS AND DESIGN OF REFRIGERATION SYSTEMS

Refrigeration is the cooling effect of the process of extracting heat from a lower temperature heat source, a substance or cooling medium, and transferring it to a higher temperature sink, probably atmospheric air and surface water to maintain the temperature of the heat source below that of the surroundings. Therefore, ***Refrigeration is defined as the branch of science that deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surroundings***.

1.1 VAPOUR COMPRESSION REFRIGERATION

The vapour compression cycle is the most widely used refrigeration cycle in practice. In this cycle, a compressor compresses the refrigerant to a higher pressure and temperature from an evaporated vapour at low pressure from an evaporated vapour at low pressure and temperature. The compressed refrigerant is condensed into liquid form by releasing the latent heat of condensation to the condenser water. Liquid refrigerant is then throttled to a low-pressure, low temperature vapour, producing the refrigeration effect during evaporation. Vapour compression is often called mechanical refrigeration, that is, refrigeration by mechanical compression.

1.2 CARNOT REFRIGERATION CYCLE

The Carnot cycle is one whose efficiency cannot be exceeded when operating between two given temperature. The Carnot heat engine receives energy at a high of temperature, converts a portion of the energy into work, and discharges the remainder to a heat sink a low level of temperature.

The Carnot refrigeration cycle performs the reverse effect of the heat engine, because it transfers energy from a low level of temperature to a high level of temperature. The diagram of the equipment and the temperature-entropy diagram of the refrigeration cycle are shown in Figs 1 and 2.

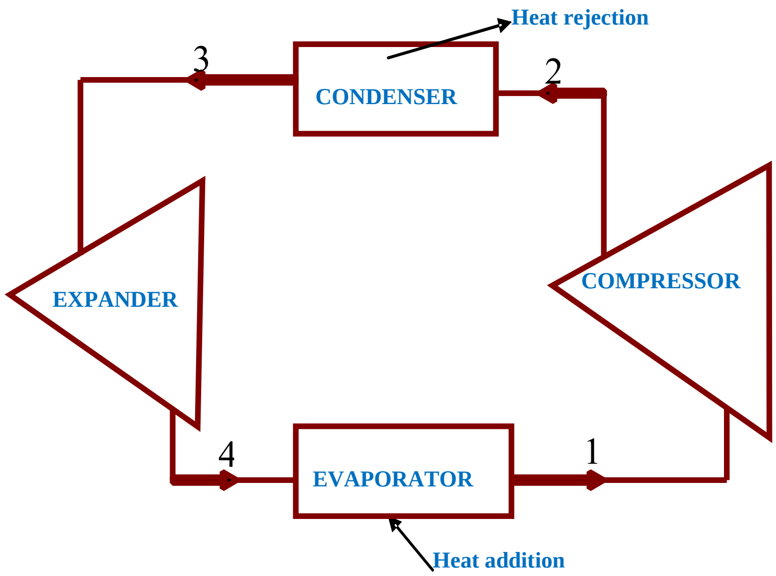


Fig. 1 Carnot Refrigeration cycle

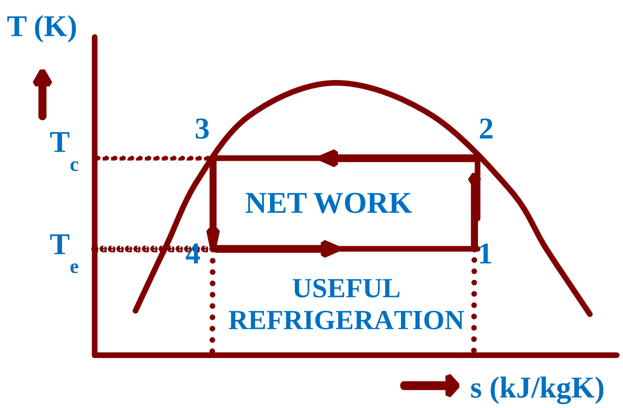


Fig. 2: T-s diagram of the Carnot refrigeration cycle

The processes which constitute the cycle are:

Process 1 - 2: Isontropic compression, $S_1 = S_2$

Process 2 – 3: Isothermal rejection of heat Tc= constant i.e. $T_2 = T_3$

Process 3 – 4: Isentropic expansion $S_3 = S_4$

Process 4 – 1: Isothermal addition of heat (heat absorption from the cold reservoir) at Te – constant i.e. $T_1 = T_4$

All processes in the Carnot cycle are thermo dynamical reversible. Processes 1-2 and 3-4 are consequently reversible adiabatic (isentropic).

The withdrawal of heat from the low temperature source in process 4-1 is the refrigeration step and is the entire purpose of the cycle. All the other processor in the cycle functions so that the low temperature energy can be discharged to some convenient high-temperature sink.

The Carnot cycle, consist of reversible which make its efficiency high than could be achieved in an actual cycle. Although Carnot cycle is an unattainable ideal cycle, it necessary to study the cycle because of the following reason:

(i) It serves a standard of comparison, and

(ii) It provides a convenient guide to the temperatures that should be maintained to achieve maximum effectiveness.

1.3 COEFFICIENT OF PERFORMANCE (COP)

The index of performance is not called efficiency, because that term is usually reserved for the ratio of output to input. The ratio of output to input would be misleading applied to a refrigeration system because the input in process 2-3 (Fig. 2) is usually wasted. The performance term in the refrigeration cycle is called the coefficient of performance, (COP), defined as

$$COP_{ref} = \frac{\text{Useful Refrigeration}}{\text{Net Work}} = \frac{\text{Refrigerating Effect}}{\text{Net work}}$$

The two terms which make up the coefficient of performance (COP_{ref}) must be in the same unit, so that the COP_{ref} is dimensionless. From Fig. 2, refrigerating effect,

$Q_e = Tds = T_1 (S_1 – S_4)$ and

Network $W = Q_c – Q_e = T_2 (S_2 – S_3) - T_1 (S_1 –S_4)$.

The Net Work is the area enclosed in rectangle 1-2-3-4 (Fig. 2). An expression for the coefficient of performance of the Carnot refrigeration cycle is:

$$COP_{ref,Carnot} = \frac{Q_e}{W} = \frac{Q_e}{Q_c - Q_e}$$

$$COP_{ref,Carnot} = \frac{T_1(s_1 - s_4)}{T_2(s_2 - s_3) - T_1(s_1 - s_4)}$$

$s_2 - s_3 = s_1 - s_4$, since $s_2 = S_1$ and $S_3 = S_4$

Therefore, $COP_{ref,Carnot} = \dfrac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)}$

$$COP_{ref,Carnot} = \frac{T_1}{T_2 - T_1} = \frac{T_e}{T_c - T_e} \qquad (1.1)$$

Where, T_e = temperature in the evaporator

T_c = temperature in the condenser

HEAT PUMP

For heat pump $COP_{hp} = \dfrac{\text{Heat Rejected from the Cycle}}{\text{Net Work}}$

$$COP_{hp} = \frac{Q_c}{Q_c - Q_e} = \frac{T_2(s_2 - s_3)}{T_2(s_2 - s_3) - T_1(s_1 - s_4)}$$

$$COP_{hp} = \frac{T_2}{T_2 - T_1} \text{ or } \frac{T_c}{T_c - T_e} \qquad (1.2)$$

The coefficient of performance of the Carnot cycles is entirely a function of the temperature limits and vary from zero to infinity. The thermodynamic analysis per unit mass of reversed Carnot cycle with vapour as a refrigerant is given below:

Refrigerating effect, $Q_e = h_1 - h_4$

Heat rejected, $Q_c = h_2 - h_3$

Compressor Work, $W_{cp} = h_2 - h_1$

Expander Work, $W_{ex} = h_3 - h_4$

Net work, $W = W_{cp} - W_{ex} = (h_2 - h_1) - (h_3 - h_4)$

Or $W = Q_c - Q_e = (h_2 - h_{3=}) - (h_1 - h_4)$

$$COP_{ref,Carnot} = \frac{Q_e}{Q_c - Q_e} = \frac{h_1 - h_4}{(h_2 - h_3) - (h_1 - h_4)} \qquad (1.3)$$

1.4 TONS OF REFRIGERATION

It was common practice to measure amounts of refrigeration in tons of refrigeration. One ton of refrigeration (abbreviation: TR) is the amount of cooling produced by one U.S. ton of ice in melting over a period of 24 hours. Since an American ton is 907.2 kg and the latent of fusion of water amounts to 334.9 kJ/kg, therefore,

$$1\ TR = \frac{907.2 \times 334.9}{24 \times 3600} = 3.5165\ kW$$

If cooling required is x kW of refrigeration, the rate of refrigerant circulation necessary is

as $$\dot{m} = \frac{\text{Useful Refrigeration in kW}}{\text{Refrigerating Effect in kJ/kg}} = \frac{Q_e}{Q_e}$$

$$\dot{m} = \frac{x}{h_1 - h_4} \qquad (1.4)$$

where $\dot{m}$ is the mass floe rate of the refrigerant

ASSIGNMENT 1

A Carnot refrigerator has working temperatures of -30^{0}c and 35^{0}c. If operates with R12 as a working fluid, calculate: (a) the work of isentropic compression and heat that of isentropic expansion, (b) refrigerating effect and heat rejected per kg of the refrigerant and (c) COP of the cycle. If the actual refrigerator operating on the same temperatures has a COP of 0.75 of the maximum, calculate (d) the power consumption and heat rejected to the surroundings per ton of refrigeration (ITR = 3.5165 KW).

1.5 ACTUAL REFRIGERATION SYSTEMS

Although the Carnot cycle is theoretically the most efficient cycle between given temperatures Tc and Te, it sometimes has limitations for practical use. In actual cycle, the COP will be less than its Carnot value. For the purpose of comparison between the actual and Carnot value, the index of performance for cooling or refrigerating efficiency (η_{ref}) is often used.

$$\text{Refrigerating Efficiency, } \eta_{ref} = \frac{COP_{ref(actual)}}{COP_{ref(carnot)}} \tag{1.5}$$

The reverse Carnot cycle with vapour as a refrigerant can be used as a practical cycle with minor modifications. The isothermal processes of heat rejection and heat absorption, accompanying condensation and evaporation respectively, are nearly perfect process and easily achievable in practice. But the isentropic compression and expansion process have limitations which are suitably modified as follows:

(a) The reverse Carnot cycle is simplified by replacing the expansion cylinder with a simple throttle value. Throttling process occur such that the initial enthalpy equals the final enthalpy. The process is highly irreversible so that the whole cycle becomes irreversible. The process is represented by line 3 - 4 on Fig. 3.

(b) The compression process in the Carnot cycle involves the compression of wet-refrigerant vapour. The wet compression is not found suitable in the practical refrigeration; therefore it is desirable to have refrigerant vapour initially dry at the start of compression as shown in Fig. 3. Such compression is known as dry compression by line 1-2 in Fig. 3. The state of the vapour at the end of compression will be at pressure p_c which is the saturation pressure of the refrigerant corresponding to the condensing temperature T_c. The wet compression is not found suitable due to the following reason:

(i) Liquid refrigerant may be trapped in the head of the cylinder and may damage the compression value and the cylinder itself.

(ii) Liquid- refrigerant droplets may wash away the lubricating oil from the walls of the compressor cylinder, thus increasing wear.

SIMPLE SATURATION REFRIGERATION

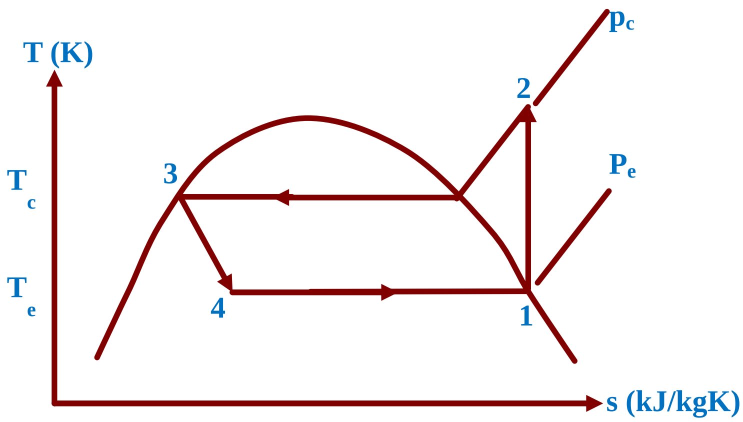


Fig. 3 T-s diagram of a simple saturation refrigeration cycle

Expansion process 3-4:

Liquid refrigerant at condensing pressure pc flows through expansion value where its pressure falls to pc, which is the pressure in the evaporator and at compressor suction. At the expansion value part of the liquid refrigerant flashes into vapour (about 10-20%) vapour produced in this way is known as ‘flash gases’. Point 4 is the state of refrigerant liquid and vapour mixture entering the evaporator. The process at the expansion value is isenthalpic (constant enthalpy; $h_3 = h_4$)

Evaporation process 4—1:

At the evaporator, the vaporization of the remaining liquid occurs at constant temperature T_e.

Refrigerating effect = $h_1 - h_4$ (1.6)

Compression process 1-2:

Compression of refrigerant vapour from Pe to Pc occurs during process 1-2. As vapour is compressed the temperature rises and, finally, leaves the compressor as superheated gas at point 2. Work of compression W_c is given as:

$$W_c = h_2 - h_1 \text{ (kJ/kg)}$$

$$W'_c = m'(h_2 = h_1) \quad (1.7)$$

Where m′ = mass of refrigerant circulated in kg/s.

Condensation process 2-3:

The refrigerant vapour loses its super heat and latent heat of evaporation. At constant pressure pc all vapour will turn to liquid. The total amount of heat rejected to the condenser is given as

$$Q_c = h_2 - h_3 \text{ (kJ/kg)} \quad \text{or} \quad Q'_c = m'(h_2 - h_3) \text{ kW} \quad (1.8)$$

This cycle as described is called the simple saturation cycle. In practice:

(i) The liquid entering the expansion value is usually several degrees cooler than condensing temperature.

(ii) The gas entering the compressor, on the other hand, is several degrees warmer than the temperature of evaporation.

(iii) There are pressure drops in the suction, discharge and liquid pipelines.

(iv) The compression process is not truly isentropic.

(v) The actual power required to drive the compressor is somewhat greater than Wc on account of frictional losses.

All these factors have to be taken into a more exact quantitative treatment of the subject.

1.6. SYSTEM CAPACITY

The capacity of a system is usually expressed as "the rate at which heat is removed from the refrigerated space" usually expressed in kW. For a mechanical refrigerating system the capacity depends on two factors:

(i) the mass of refrigerant circulated per unit of time

(ii) the refrigerating effect per unit mass of refrigerant circulated.

$$Q'_e = m'Q_e \qquad (1.9)$$

where, Q'_e = refrigerating capacity in kW or kJ/s

Q_e = refrigerating effect in kJ/kg

m' = mass flow rate in kg/s

$$V = m'v \qquad (1.10)$$

where V = total volume of vapour generated in the evaporation in m^3/s

v = specific volume of the vapour at the vaporizing temperature in m^3/kg

ASSIGNMENT 2

The evaporating temperature of R 12 mechanical refrigeration system is -10°C whilst the condensing temperature is 40°C. It is assumed that no sub-cooling of the liquid occurs so that temperature of the liquid at the refrigerant control is also 40°C. Calculate:

(a) the refrigerating effect per kilogram

(b) the mass of refrigerant circulated in kg/sec.per kW.

(c) the volume of vapour generated in litre/sec.per kW of refrigerating capacity.

2.0 THE COOLING LOAD CALCULATIONS

2.1 SOURCES OF HEAT

Possible sources of heat that supply the load on the refrigerating equipment are:

(a) heat flow into refrigerated space by conduction through the insulated walls.

(b) heat flow into space by indirect radiation through glass or other transparent materials

(c) heat flow into refrigerated space through opening of doors or through cracks around windows and doors.

(d) product load,i.e heat given off by a warm product as its temperature is lowered to the designed level.

(e) heat given off by any heat producing equipment located inside the space such as lights, electric motors, electronic equipment e.t.c.

(f) heat given off by occupants of the refrigerated space.

2.2 EQUIPMENT RUNNING TIME

In refrigerated system it is necessary to defrost the evaporator at frequent intervals.

(a) Off- cycle Defrosting Method

Defrosting by switching off the equipment thereby allowing the temperature in the refrigerated space to rise above the freezing point of water and maintained at this level until the frost has melted off the cooling coil and drained off is known as "off-cycle" defrosting method.

(b) Defrosting using Artificial Methods

Some artificial means of defrosting can be employed whereby electric heaters or hot gas can be used to achieve defrosting.

In any of these methods, defrosting takes some time but the off-cycle method takes a longer period. For off-cycle defrosting method 16hours running time is allowed out of every 24 hours period. With artificial methods of defrosting 18 to 20 hours running time is usually allowed out of every 25 hours period depending upon how often defrosting is necessary for any particular application. In any of these cases, the refrigerating machine must have sufficient capacity to accomplish the equivalent of 24 hours of cooling during the actual running time.

Required Equipment Capacity

In general $Q = (24/RT)(q_u)$ (2.1)

where, Q = required equipment capacity in kW

RT = running time in hours (h)

q_u = total cooling load in kW (the sum of heat loads)

Cooling Load Calculations

For commercial refrigeration total cooling load is divided into four as follows:

i. the wall gain load

ii. the air change

iii. the product load

iv. the miscellaneous or supplementary load.

2.3 THE WALL GAIN LOAD

This is a measure of the heat flow by conduction through the walls of the refrigerated space. Heat will usually flow from outdoors to inside since the indoor temperature is lower than the outside temperature.

$$Q = AU\Delta T \qquad (2.2)$$

where Q = rate of heat transferred (W)

A = outside surface area of the wall (m^2)

U = the overall coefficient of heat transmission ($W/m^2.K$)

ΔT = temperature differential across the wall (K)

2.4 AIR CHANGE LOAD

$$Q = m'(h_o - h_i) \qquad (2.3)$$

Where Q = air change load (kW)

m' = mass of air entering space (kg/s)

h_o = enthalpy of outside air (kJ/kg)

h_i = enthalpy of inside air (kJ/kg)

To determine air change load in kW multiply the infiltration rate in litre/s by the enthalpy change factors.

2.5 PRODUCT LOAD

$$Q = m'C_p\Delta T \qquad (2.4)$$

Where Q = quantity of heat from product (kJ/kg)

m' = mass of the product (kg)

C_p = specific heat above freezing (kJ/kg.K)

ΔT = the change in temperature (K)

Equation 2.4 merely gives the quantity of heat the product will give off in cooling to the space temperature, therefore, the produce load or product cooling is determined by the following equation:

$$Q = \frac{mC\Delta T}{Desire\ cooling\ time\ in\ second} \text{ (W or kW)} \qquad (2.5)$$

2.5.1 Chilling Rate Factor

Because of high temperature difference between the product and the space air at the start of chilling operation the chilling rate is higher during the period and the product load tends to concentrate in the early part of the chilling period. It is therefore good practice to select equipment with sufficient capacity to carry the load during the initial stage of chilling. In order to make reasonable estimates a "***chilling rate factor***" is sometimes introduced into chilling load calculations. This factor is to compensate for the uneven distribution of the chilling loads. It also has the effect of increasing the product load calculation by an amount sufficient to make the average hourly cooling rate approximately equal to the hourly of the peak condition.

Chilling rate factors for various products are listed in tables. When a chilling rate factor is used, equation 2.5 is written as:

$$Q = \frac{\dot{m}C\Delta T}{(Chilling\ time\ in\ second)(Chilling\ rate\ factor)} \qquad (2.6)$$

2.5.2 Product Freezing and Storage

For a product to be frozen and stored at some temperature below its freezing temperature, the heat involved is calculated in three parts;

i. heat given off by product in cooling from entering to freezing temperature.

ii. heat given off in solidifying or freezing

iii. heat given off by product in cooling from freezing to final storage temperature.

The quantity of heat resulting from freezing is calculated using equation:

$$Q = mh_{fg} \qquad (2.7)$$

Where m = mass of the product (kg)

h_{fg} = product latent heat (kJ/kg)

To determine the equivalent product load in kW divide the summation of the three parts with the processing time in seconds.

2.5.3 Respiration Heat

Where considerable quantities of fruits and of vegetables are held in storage at temperatures above the freezing temperature, respiration heat load must be considered. This is because fruits and vegetables are still and alive after harvesting and continue to undergo changes during storage.

The product load accruing from respiration heat is given by equation:

Q(W) = mass of product (kg) x respiration rate (W/kg) (2.8)

2.5.4 Containers and Packaging Materials

When products are chilled in containers, the heat given off by the containers and packaging materials in cooling from entering to space temperature must be considered as part of the product load.

2.6 MISCELLANEOUS LOADS

Miscellaneous loads are the heat given off by lights and electric motor operating in the refrigerated space, and the people working in the space.

(i) For lights

$$Q = \frac{\text{Watts x hours (W)}}{\text{24 hours}} \tag{2.9}$$

$$Q = \frac{\text{Wattage of light x No of hours/day}}{\text{24 hours}} \text{ (W)}$$

(ii) Motors Working in a Refrigerated Space

$$Q(kW) = \frac{\text{Motor output (kW) x factor (from Table) x hour in use}}{\text{24 hours}} \tag{2.10}$$

(iii) People

$$Q(kW) = \frac{\text{No of people x heat equivalent, kW (from Table) x hour occupied}}{\text{24 hours}} \tag{2.11}$$

Tale 2.1: Heat Equivalent of Electric Motors

Motor Rating kW output	Motor efficiency %	Multiplying factor		
		Connected load in refrigerated space (a)	Motor losses outside refrigerated space (b)	Connected load outside refrigerated space (c)
0.1 – 0.5	33.3	1.67	1.0	0.67
0.5 – 2.0	55.0	1.45	1.0	0.45
2.0 – 150.0	85.0	1.15	1.0	0.15

Foot note for table 2.1:

a. For use when both useful output and motor losses are dissipated within refrigerate space; motors, chilling fans for forced circulation and unit coolers.

b. For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; pump on a circulating brine or chilled water system, fan-motor outside refrigerated space driving for circulating air within refrigerated space.

c. For use when motor losses are dissipated within refrigerated space and useful work expended outside of refrigerated space; motor in refrigerated space driving pump or fan located outside the space.

2.7 FACTOR OF SAFETY

A factor of 5% to 10% of calculated total cooling load is normally added. The value added depends on the reliability of data use in the computation but 10% is used as a general rule.

After applying the factor of safety the cooling load is multiplied by 24 hours and divided by the desired running time in hours to determine the average load. This average cooling load is used as a basis for equipment selection.

Table 2.2: Heat equivalent for occupancy

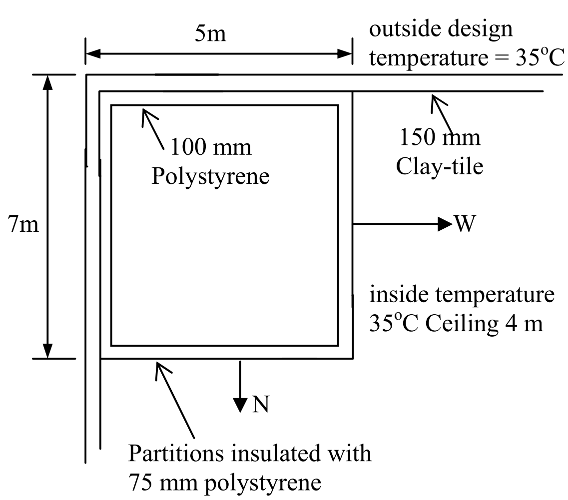
Cooler Temperature	10	5	0	-5	-10	-15	-20
Head Equivalent/person (kW)	0.211	0.242	0.275	0.305	0.347	0.378	0.407

ASSIGNMENT 3

A walk in cooler 6m x 7m x 3m high is located in the southeast corner of a store building in an area where the outdoor design dry-bulb temperatures in summer and winter are 35°C and -6°C respectively. The south and east walls of the cooler are adjacent to and a part of the south and east walls of the store building. The store has a 4m ceiling so that there is a 1m clearance between the top of the cooler and the ceiling of the store. The store is air conditioned and temperature inside the store is maintained at approximately 26°C. The inside design temperature for the cooler is 2°C. The north and west (inside) walls, floor and ceiling are insulated with 75 mm of closed cell (smooth surface) polystyrene and the south east walls are insulated with 100 mm of close-cell polystyrene. Determine the wall load in KW using he following data:

- R for polystyrene (smooth) = 0.029 W/mK
- U-value for north and west walls, floor and ceiling

  (75 mm thickness = 0.346 W/m^2K
- U-value for south and east walls (100 mm thickness) = 0.267 W/m^2K
- The design ground temperature based on outside writer design term of 6°C dry bulb = 25°C
- Allowance for solar radiation on south and east walls is 2°C and 3°C respectively to be added to the outdoor – indoor temperature difference.



ASSIGNMENT 4

300 kg of poultry enters a chiller of 5^{o}C and are frozen and chilled to a final temperature of -15^{o}C for storage in 12 hour. Compute the product load in KW. The specific heat of poultry above and below frozen are 3.18 and 1.55kJ/kgK respectively. The latent heat is 246kJ/kg whilst the freezing point is -3^{o}C.

ASSIGNMENT 5

3,000 lug boxes of apples are stored at 2^{o}C in a storage cooler, 16m x 12m x 3.4m high. The apple enters the cooler at a temperature of 30^{o}C and at the rate of 200 lugs per day each day for the 15 day harvesting period. The walls including floor and ceiling are constructed using 100 mm boards and are insulated with 100mm of mineral wool. All of the walls are shaded and the ambient temperature is 30^{o}C. The average weight of apples per lug box is 27 kg. The boxes have an average weight of 2 kg and a specific heat value of 2.5 kJ/kgK. The lighting load is 500 W for 3 hours per day. Two people and one battery operated lift truck (4.17 kW) are in the space 3 hours per day. Determine the average load on a 16 hour per day equipment operating time using the following data:

- Insulation K factors = 0.045 W/mK.
- U-value based on 100 mm insulation = 0.383 W/m^{2}K
- Air change rate assuming heavy usage = 33.5 litre/sec.
- The enthalpy change factor (Assuming 50% R.H.) = 0.0598 kJ/litre
- The specific heat of apples = 3.72 kJ/kg and chilling rate factor = 0.67
- Respiration heat of apple at 2^{o}C = 0.0145 W/kg.
- Heat gain per person = 261.8 W

3.0 VENTILATION AND AIR DISTRIBUTION SYSTEMS CALCULATIONS

3.1 VENTILATION

Ventilation is defined as supplying air by natural or mechanical means to a space. Normally, ventilation air is made up of outdoor air and recirculated air. The outdoor air is provided for dilution.

Air quality must also be maintained to provide a healthy, comfortable indoor environment. Sources of pollution exist in both the internal and external environment. Indoor air quality is controlled by removal of the contaminant or by dilution. Ventilation plays important role in both processes. Table 3.1 presents outdoor air requirements for ventilation for three occupancy types listed in the standard. As noted in the table, much larger quantities of air are required for dilution in areas where smoking is permitted.

Table 3.1: Outdoor – air requirements for ventilation

Function	Estimated occupancy per 100 m^2 floor area	Outdoor air requirements per person (litre/sec)	
		Smoking	Non smoking
Offices	7	10	2.5
Meeting and waiting spaces	60	17.5	3.5
Lobbies	30	7.5	2.5

Ventilation imposes a significant load on heating and cooling equipment and therefore, is a major contribution to energy use. Space occupancies and the choice of ventilation rates should be considered carefully. For example, if smoking is permitted in part of a building but restricted in another part of the building,

Ventilation rates for smoking should not be assumed uniformly. The use of re-circulated air will conserve energy whenever the outdoor-air temperature is extremely high or low. The following is the procedure for determining the allowable rate for recirulation.

$$V = V_r + V_m \tag{3.1}$$

Where, V = rate of supply air for ventilation purposes, (litre/sec)

V_r = recirculation air rate, (litre/sec).

V_m = minimum outdoor-air rate for specified occupancy, for example the nonsmoking value from Table 3.1, but never less than 2.5 litre/s per person

Also, $$V_r = \frac{V_o - V_m}{E} \tag{3.2}$$

Where V_o = outdoor-air rate from table 3.1 for specified occupancy (smoking or non-smoking, as appropriate (litre/s).

E = efficiency of contaminant by air cleaning device. The efficiency must be determined relative to the contaminants to be removed.

ASSIGNMENTS 6

Determine the ventilation rate, outdoor-air rate, and recirculated-air rates for an office-building meeting room if smoking is permitted. An air cleaning device with E = 60% for removal of tobacco smoke is available. The outdoor-air requirements for ventilation are shown in Table 3.1.

4.0 THE DESIGN AND ANALYSIS OF AIR CONDITIONING

In the design and analysis of air conditioning plants, the fundamental requirement is to identify the various processes being performed on air. Once identified, the processes can be analyzed by applying the laws of conservation of mass and energy. All these processes can be plotted easily on a psychrometric chart. This is very useful for quick visualization and also for identifying the changes taking place in important properties such as temperature, humidity ratio, enthalpy etc.

4.1 AIR MIXTURES

Figure 4.1 shows what happens when two air streams meet and mix adiabatically on the psychrometric chart. Moist air at state 1 mixes with moist air at state 2, forming a mixture at state 3. The principle of the conversation of mass allows two mass balance equations to be written.

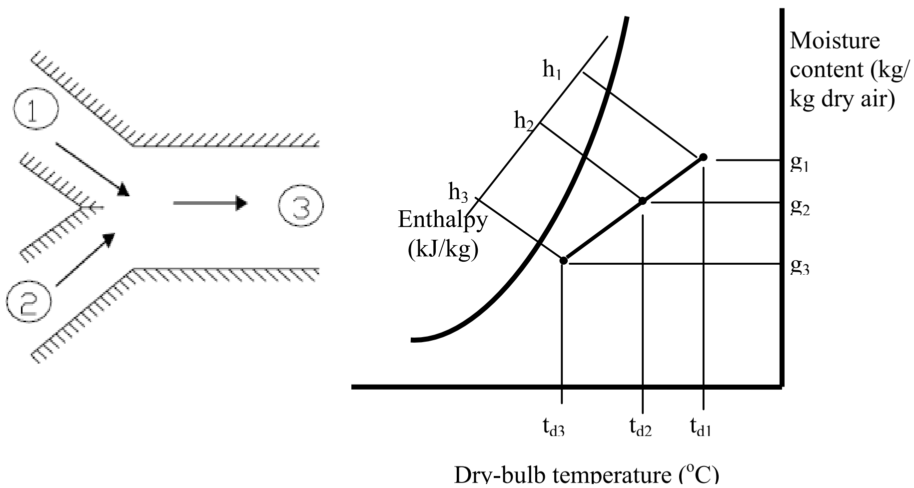


Fig. 4.1: Adiabatic mixing of two airstreams on psychometric chart.

Conservation of mass:

$m_1 + m_2 = m_3$ for the dry air (4.1)

$g_1m_1 + g_2m_2 = g_3m_3$ for the associated water vapour (4.2)

Substituting Eq. (4.1) in (4.2):

$g_1m_1 + g_2m_2 = g_3(m_1 + m_2)$

Therefore, $(g_1 - g_3)m_1 = (g_3 - g_2)m_2$

Hence $$\frac{g_1 - g_3}{g_3 - g_2} = \frac{m_2}{m_1} \qquad (4.3)$$

Conservation of energy:

$$m_1h_1 + m_2h_2 = m_3h_3$$

Therefore, $$\frac{h_1 - h_3}{h_3 - h_2} = \frac{m_2}{m_1} \qquad (4.4)$$

To determine the dry-bulb temperature of the mixture, the following practical equation can be used:

$$h_3 = (1.007t_3 - 0.026) + g_3\,(2501 - 1.94t_3) \qquad (4.5)$$

The equation above shows that the states point 1, 2 and 3 must line on a straight line on a mass energy coordinate system. Hence the principle of this mass-energy coordinate system can be stated as:

"*when two air streams mix adiabatically the mixture state his on the straight line which joins the state points of the constituents and the position of the mixture state points is such that the line is divided inversely as the ratio of the masses of the dry air in the constituent air streams*"

Approximately temperature of the mixture is obtained as:

$$td_3 = \frac{(td_1 \text{ x } m_1) + (td_2 \text{ x } m_2)}{m_1 + m_2} \qquad (4.6)$$

This equation employed direct proportionally between temperature and mass.

ASSIGNMENT 7

A stream of moist air at a state of 21^{o}C dry-bulb and 14.5^{o}C wet-bulb (sling) mixes with another stream of moist air at a state of 28^{o}C dry-bulb and 20.2^{o}C wet-bulb (sling), the respective masses of the associated dry air being 3kg and 1kg. With the aid of psychrometric chart, calculate the moisture content, enthalpy and the dry-bulb temperature of the mixture.

ASSIGNMENT 8

Moist air at a state of 39^{o}C dry-bulb, 25^{o}C wet-bulb (sling) and pressure of 1 atm mixes adiabatically with moist air at 25^{o}C dry-bulb, 18^{o}C wet-bulb and pressure of 1 atm. If the masses of dry air are 2 kg and 5 kg respectively, with the aid of psychrometric chart calculate the moisture content, enthalpy and the dry-bulb temperature of the mixture.

4.2 SENSIBLE HEATING AND COOLING

4.2.1 Sensible Heating

Sensible heating occurs when moist air flow across a heater. Requirements are that the heating medium shall be at a higher temperature than the air. During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The change of state of the air can be sketched on a psychometric chart. Fig. 4.2 shows the sensible heating process O-A on a psychrometric chart. The heat transfer rate during this process is given by:

$$Q_h = m_a(h_A - h_O) = m_a C_{pm}(T_A - T_O) \qquad (4.7a)$$

Where C_{pm} is the humid specific heat (≈1.0216 kJ/kg dry air) and m_a is the mass flow rate of dry air (kg/s).

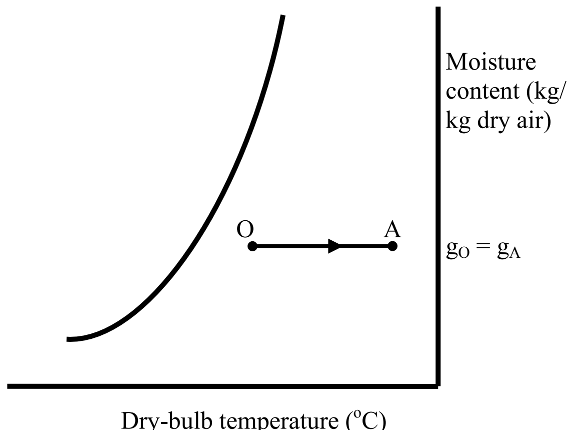


Fig. 4.2: Sensible heating on psychometric chart

4.2.2 Sensible Cooling

Sensible cooling occurs when it over the coils of a sensible cooler. Requirements are that the temperature of the chilled water should not be so low as to cause condensation to occur on the cooler ceils as dehumidification will take place. During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil.

For moisture content to remain constant, the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. The change of state of the air can be sketched on a psychometric chart. Fig. 4.3 shows the sensible cooling process O-B on a psychrometric chart. The heat transfer rate during this process is given by:

$$Q_h = m_a(h_B - h_O) = m_aC_{pm}(T_B - T_O) \qquad (4.7b)$$

Where C_{pm} is the humid specific heat (≈1.0216 kJ/kg dry air) and m_a is the mass flow rate of dry air (kg/s).

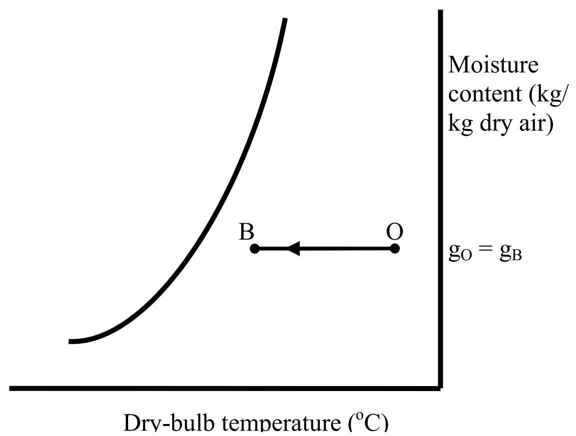


Fig. 4.3 Sensible cooling on psychometric chart

ASSIGNMENT 8

Calculate the load on a battery which heats 1.5m^3/s of moist air, initially at a state of 21^{o}C dry-bulb, 15^{o}C wet bulb (sling) and 1013.25 mbar barometric pressure, by 20^{o}C. If low pressure hot water at 85^{o}C flow and 75^{o}C return, is used in kilograms of water per second. Take specific heat of water to be 4.2 kJ/kg.

4.3 DEHUMIDIFICATION

There are four principal methods whereby moist air can be to dehumidify.

(a) Cooling to a temperature below the dew point

(b) Adsorption

(c) Absorption

(d) Compression followed by cooling

The first method will be discussed here.

Cooling to a temperature less than dew point is done by either passing the moist air over a cooler coil or through an air washer provided with chilled water.

Some of the spray water in the air washer or some part of the cooler coil, must be at a temperature less than the dew point of the air entering the equipment. In Fig. 4.4, T_s

is the dew point temperature of the moist air ‘on’ the coil or washer (point O). The temperature T_s, corresponding to the point ‘s’ on the saturation curve, is termed the ***apparatus dew point***. This term is in use for both coils and washers but, in the case of cooler coils alone, T_s is also sometimes termed the ***mean coil surface temperature***. It is obvious that the following has happened to the moist air between points O and C:

(i) enthalpy of the moist air is reduced

(ii) the moisture content is reduced

(iii) the wet-bulb temperature (WBT) is reduced

(iv) the dry-bulb temperature (DBT) is reduced

(v) percentage of saturation increases (but not up to 100%).

The contact factor (β) or by-pass factor (1 - β) is used to represent the efficiency of a cooler coil and they are defined as follow:

Contact factor, $\beta = \dfrac{g_a - g_b}{g_a - g_c} = \dfrac{h_a - h_b}{h_a - h_c}$

and By-pass factor, $1 - \beta = 1 - \dfrac{g_a - g_b}{g_a - g_c} = \dfrac{g_b - g_c}{g_a - g_c}$

or $1 - \beta = 1 - \dfrac{h_a - h_b}{h_a - h_c} = \dfrac{h_b - h_c}{h_a - h_c}$

4.3.1 Cooling and Dehumidification

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in Fig. 4.4, some of the water vapor in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. This is the process air undergoes in a typical air conditioning system. Although the actual process path will vary depending upon the type of cold surface, the surface temperature, and flow conditions, for simplicity the process line is assumed to be a straight line. The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations.

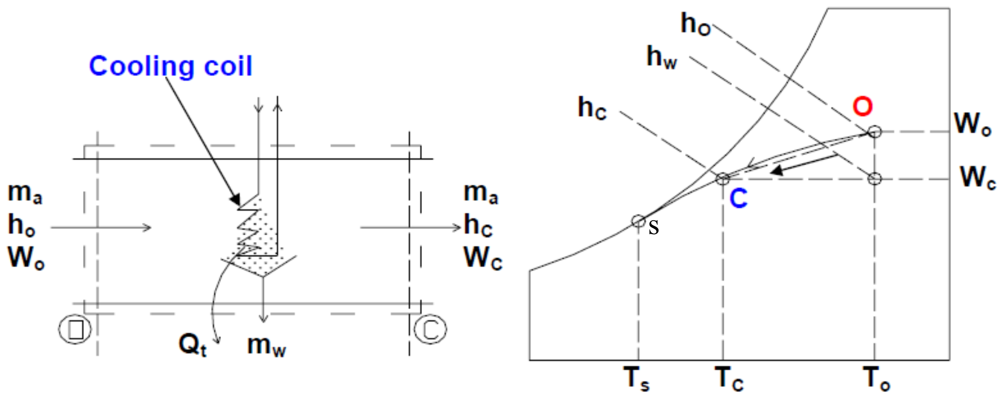


Fig. 4.4: Cooling dehumidification using cooler coil

4.3.2 Heating and Dehumidification

This process can be achieved by using a hygroscopic material, which absorbs or adsorbs the water vapor from the moisture. If this process is thermally isolated, then the enthalpy of air remains constant, as a result the temperature of air increases as its moisture content decreases as shown in Fig. 4.5. This hygroscopic material can be a solid or a liquid. In general, the absorption of water by the hygroscopic material is an exothermic reaction, as a result heat is released during this process, which is transferred to air and the enthalpy of air increases.

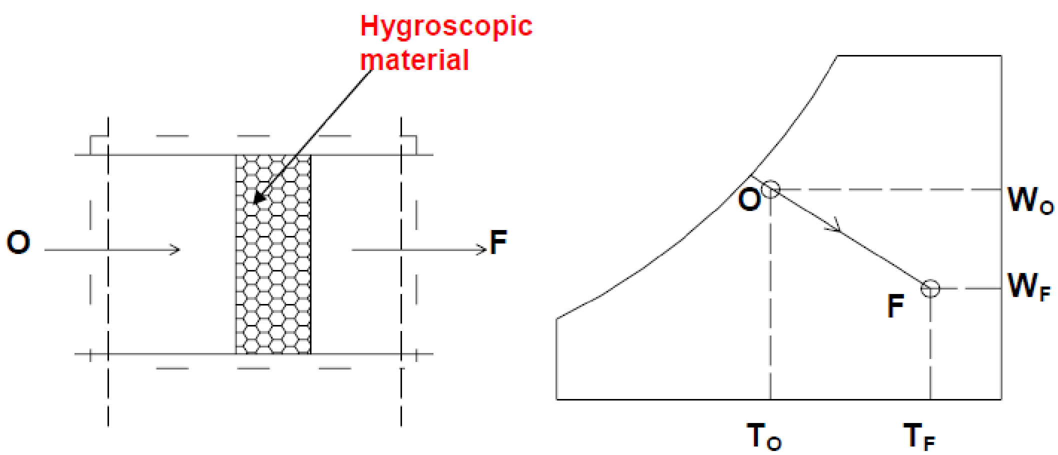


Fig. 4.5: Chemical dehumidification process

4.4 HUMIDIFICATION

4.4.1 Heating and Humidification

During winter it is essential to heat and humidify the room air for comfort. As shown in Fig. 4.6, this is normally done by first sensibly heating the air and then adding water vapour to the air stream through steam nozzles as shown in the figure.

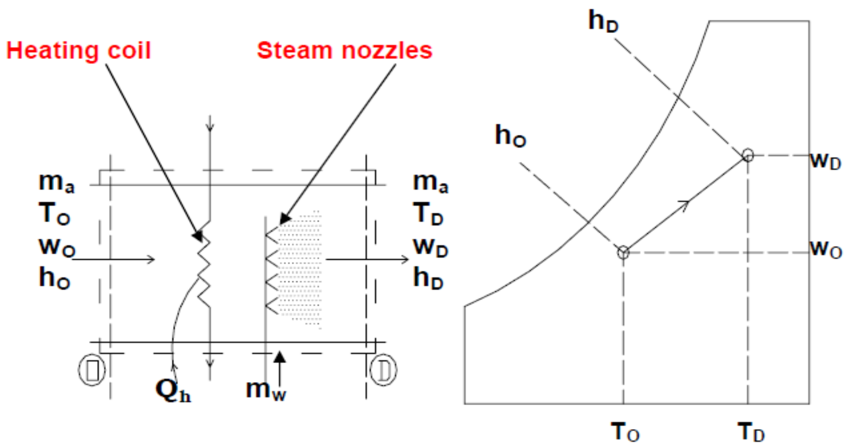


Fig. 4.5: Heating humidification process

4.4.2 Cooling and Humidification

As the name implies, during this process, the air temperature drops and its humidity increases. This process is shown in Fig. 4.6. As shown in the figure, this can be achieved by spraying cool water in the air stream. The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew-point temperature to avoid condensation ($T_{DPT} < T_w < T_O$).

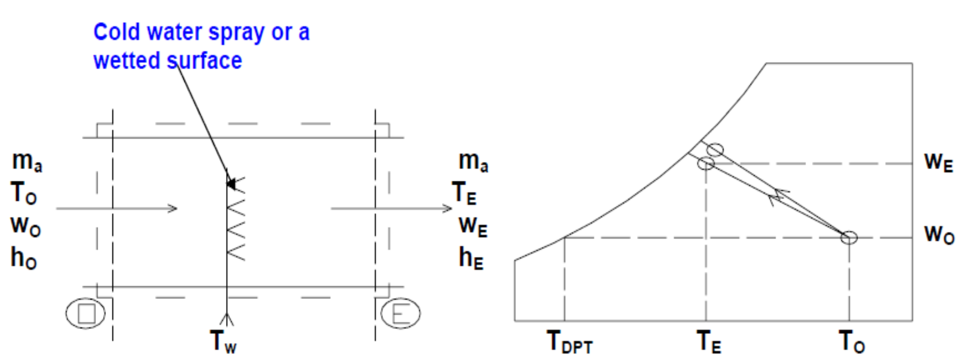


Fig. 4.6: Cooling and humidification process

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature. If the temperature of the water sprayed is equal to the wet-bulb temperature of air, then the net transfer rate will be zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water. Under a special case when the spray water is entirely recirculated and is neither heated nor cooled, the system is perfectly insulated and the make-up water is supplied at WBT, then at steady-state, the air undergoes an adiabatic saturation process, during which its WBT remains constant. The process of cooling and humidification is encountered in a wide variety of devices such as evaporative coolers, cooling towers etc.

4.5 AIR WASHERS

An air washer is a device for conditioning air. As shown in Fig. 4.7, in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes. In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer.

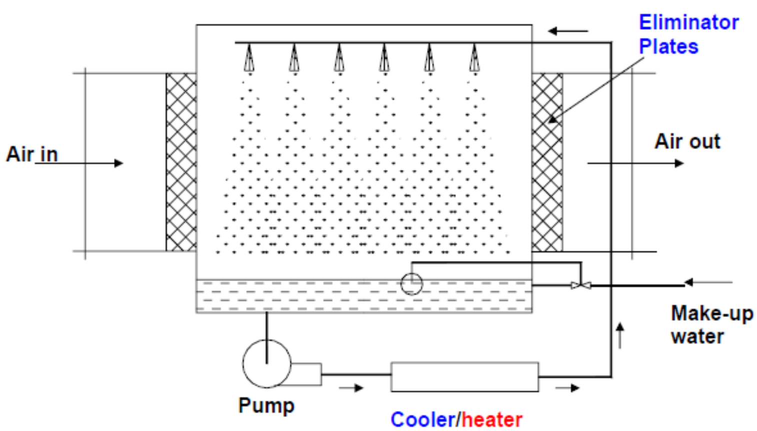


Fig. 4.7: Air washer

5.0 ELECTRO-MECHANICAL CONTROLS IN REFRIGERATION AND AIR-CONDITIONING

5.1 AUTOMATIC CONTROL IN RFA SYSTEMS

In general, the design capacity of any component in refrigeration and air-conditioning systems will not match the load. If the load on the refrigerant plant remains constant, there would be no need of having controls in the system, but the loads are continually changing. This means that the state maintained within the conditioned space will not stay constant and if the plant is left to run wild, the capacity will exceed the load for most of the time. Therefore, the automatic control of air-conditioning aims to satisfy the temperature and humidity design specifications throughout the year. This can be for the benefit of the thermal comfort of the occupants or a plant or process environments.

5.1 BASIC FUNCTIONS OF CONTROL SYSTEMS

For the operation of air conditioning systems a wide variety of automatic a wide variety of automatic controls and many types of control systems are available. All modern air conditioning installations, even the smallest, have some automatic controls. There are many different kinds of automatic controls for air conditioning application. They can be classified or grouped in two ways:

(i) according to their major functions

(ii) in terms of their specific action

5.1.1 Classification of Automatic control in R & A According to their major function

(a) Controls which govern the condition of the air within the space: these controls are sensing and activating devices such as thermostsats, humidistats, dampers, valves and switches connected electrically or pneumatically into control systems for the purpose of keeping temperature and relative humidity in the conditioned space within the design limits. They control such variables as the temperature, pressure and flow rate of air, water, refrigerant and steam.

(b) Controls which function as protective devices: these controls act as safety devices to protect machinery from damage and also to guard against high temperatures, low temperatures and fire hazards. Examples are pressure controls, antifeeeze devices, safety valves, limit switches, oil failure switches, motor-overload protection devices, fusible link dampers, smoke sensors and the like.

(c) controls whose primary purpose is to produce economy of operation: these devices automatically reduce the amount of power, water or fuel being consumed when the load on the air-conditioning system drops below the design maximum. Examples are water-control valves or refrigeration compressors, autouloaders on compressors etc.

5.1.2 Classification of control in R & A in terms of their specific action

(a) Starting controls: they singly or sequentially start electric motors which drive compressors, fans, and pumps.

(b) Operating controls: they may function as safety devices to protect machinery from damage by guarding against excessively high temperatures or pressures of the hazards, and they may provide capacity control for operation or starting of equipment. Examples of such controls are condensing unit high-low pressure controls, time-delay relays, freeze protection devices, temperature limit controls. Such controls provide protection and economy of operation, and match the capacities of the various components of a system.

(c) Space controls: they govern the condition of air in the space. Although the controls may sense temperature, pressure, or humidity, they also function to maintain the conditioned space within the design limits. These controls are sensing and actuating devices.

5.2 CONTROL SYSTEMS

Control systems can be of two types

(a) Open loop or feed forward control

(b) Close loop or feedback control

5.2.1 Open Loop Control

An open-loop system does not make use of negative feedback from the controlled variable, but regulates the manipulated variable in some pre-arranged manner. The system has no precise control on inside conditions, because it does not have any element to detect the variations in inside conditions and to take any correction action accordingly.

5.2.2 Closed Loop Control

A closed-loop system is one which measures a departure from the desired value of the controlled variable and feeds this information back to the device which regulates the capacity of the manipulated variable, so that corrective action may be taken (illustrated in Fig. 5.1).

The controller is that which compares the value of the controlled variable with a reference point (fixed temperature setting). In this particular circuit the controller is a thermostat. Other types of controller are humidistat and pressure controller. The controlled device reacts to the signal received from a controller and varies the flow of the control agent. Control agent is the medium manipulated by the controlled device.

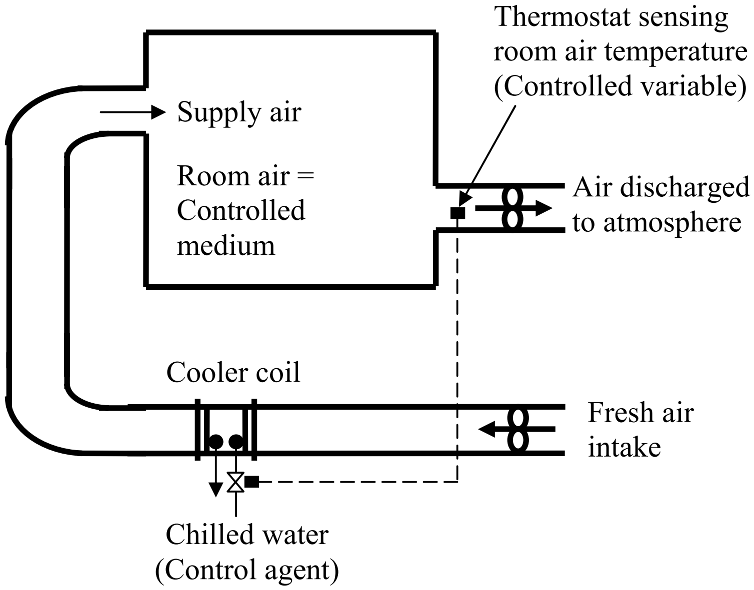


Fig. 5.1. Closed loop system

5.3 DEFINITIONS

The terms in common use in automatic controls are defined below:

i. **Controlled medium:** the substance which has a physical property that is under control e.g. the air in a room.

ii. **Controlled variable:** the quantity or physical property measured and control e.g. room air temperature.

iii. **Manipulated variable:** the physical property or quantity regulated by the control system in order to achieve a change of capacity which will match the change of load e.g. the flow rate of chilled water through a cooler coil.

iv. **Control agent:** the substance whose physical property or quantity is regulated by the control system e.g. the chilled water fed to a cooler coil.

v. **Desired value:** the value of the controlled variable which it is desired that the control system will maintain.

vi. **Set point or set value:** the value on the scale of the controller at which it is set. For example a thermostat may have its pointer set against the figure 21^{o}C on the scale.

vii. Control point: the value of the controlled condition which the controller is trying to maintain. This is a function of the mode of control. For example, with proportional control and a set point of 21^{o}C atfull load and only 21^{o}C at 50 percent load.

viii. **Deviation:** the difference between the set point and the measured value of the controlled condition, at any instant. For example, although the set point on the thermostat is 21^{o}C, the measured value of the room air temperature may be 19^{o}C. The deviation is then -2^{o}C.

ix. **Offset:** a sustained deviation caused by some inherent characteristic of the control system. For example, if the set point is 21^{o}C and the measured room air temperature is at a steady value of 19^{o}C for some period of time, then the offset is -2^{o}C, for this period.

x. **Primary Element or Measuring unit:** the part of the controller which responds to the value of the controlled condition (Detecting element) and gives a measured value of the condition (measuring element). E.g. a bimetallic strip type of thermostat.

xi. **Final control element or correcting unit:** this is the mechanism which alters the value of the manipulated variable in response to a signal initiated at the primary element, for example, a motorized valve.

xii. **Automatic controller:** a device which compares a signal from the detecting element with the set point and, hence, initiates a corrective action to reduce the deviation e.g., a room thermostat.

xiii. **Differential gap:** this is also termed differential. It is the smallest range of values through which the controlled variable must pass for the final control element to move from one to the other of its two possible positions. For examples, if a two-position controller has a set point of 21^{o}C + 1^{o}C, then the differential gap is 2^{o}C.

xiv. **Proportional band:** also known as the throttling range, this is the range of values of the controlled variable which corresponds to the movement of the final control element between its extreme positions. For example, if a proportional controller has a set point of 21^{o}C, then the proportional band is 4^{o}C (illustrated in fig. 1.2)

xv. **Cycling or hunting:** this is a persistent periodic change in the controlled condition which is self-induced

5.4 CONTROL ELEMENTS

The Control elements for HVAC System are subdivided into four groups

i Sensing elements (detectors)

ii Controllers

iii Controlled devices

iv Auxiliary devices

5.4.1 SENSING ELEMENTS (DETECTORS)

A sensing element is a device, which measures changes in the controlled variable and produces a proportional effect for use by controller. It performs the function of detection as well as measurement. Examples are:

(a) Temperature sensing elements.

(b) Humidity sensing elements.

a. Water flow rate sensing elements: It can be a orifice plate, pitot tube, ventury, flow nozzles e.t.c.

5.4.2 TEMPERATURE SENSING ELEMENTS

(a) Contacts Thermometer: These are mercury-in-glass thermometers with electrical contacts within the stem. A rise or fall in the height of the column of mercury makes or breaks an electrical circuit, thus providing a signal. A proper choice of thermometer with the appropriate degree of accuracy, gives a reliable, direct and precise, form of measurement, within the range -40^{o}C (the freezing point of mercury) to 538 ^{o}C (the softening point of glass). They are suitable for two-position control.

(b) Bi-metallic strip thermometers: As the implies, pair of dis- similar metal strips is joined firmly together and their differing coefficients of thermal expansion cause the strips to bend together, making or breaking an electrical circuit, on change in temperature. Their range of suitability is from - 180 ^{o}C to 420 ^{o}C.

(c) Fluid Expansion Phials: A phials filled with a fluid having a suitable coefficient of thermal expansion is connected by a capillary tube to a bellows or to a Bourdon tube. The latter, which is a spirally – wound tube, increase its length and unwinds as the liquid expands. This unwinding is arranged to produce a rotational movement and so operate a proportional or other controller. If a bellows is used, a linear movement is produced as the bellows fill upon liquid expansion. These devices are suitable for use in the range - 45 ^{o}C to 650 ^{o}C.

(d) Thermocouples: When a pair of dissimilar metallic wires are joined together to form a loop, a current flows around the loop if the two junctions are at different temperature. The magnitude and direction of the current depend on the temperature difference and the pairing of the metals. The electrical current is used as the feedback variable. Depending on the metals used, the range of application varies from -260 ^{o}C to 2600 ^{o}C.

(e) **Resistance Thermometers:** The electrical resistance of a conductor varies with its temperature, and this is used to provide an accurate measurement of feedback. The range of use, depending on the material used, is from -265 ^{o}C to 650 ^{o}C.

(f) **Thermistors:** These are made of oxides of metals and give an inverse, exponential relationship between temperature and electrical resistance. Their response is non- linear but very sensitive up to 100 ^{o}C.

5.4.3 CONTROLLERS

Controller takes the sense or effect for detectors, compares it with the desired control condition and regulates an output signal to cause various types of control actions. This is the actualizing element to perform corrective action by minimizing the error. Controllers are essentially transducers, which convert one type of energy input into another type of energy output. There are numerous controllers, which are classified as:

(a) Electrical\Electronic controllers

(b) Pneumatic controllers

(c) Transducers

(d) Thermostats

5.4.4 CONTROLLED DEVICES

Controlled devices may be (a) Automatic valves or (b) Dampers

(a) Automatic Value

An automatic value is designed to control the flow of steam water, chilled water, and other fluids and is having a variable orifice which is positioned by an electric or pneumatic operator in response to impulses from the controller. The functions of automatic values are:

(i) **Single seated value:** It is designed for tight shut off.

(ii) **Pilot piston value:** It uses the pressure of the control agent as an aid in operating the value. It is usually a single seated value and used where large forces are required for value operation.

(iii) **Double seated or Balanced value**: It is designed so that the media pressure acting against the value disc is essentially balanced thereby reducing the force required by operator. It is widely used where fluid pressure is too high to permit single seated value to close but cannot be used where tight shut if is required.

(iv) **Three way mixing value:** It has got two in let connections and one outlet connection and a double faced disc operating between two seats. It is used to mix two fluids.

(v) **Butterfly value:** It consults of heavy ring en closing a disc, which rotates on an axis at or near its center and in principle similar to round single blade damper. The disc seats against a ring machined within the body or resilient liner in the body.

(b) Automatic Dampers

These are designed to control the flow at air or gases and function like values in this respects. They can be further of single or multi blade design and having parallel or opposed operations.

1.5 THE MAIN CONTROL SYSTEMS

There are five types of main control systems:

(i) Self- acting

(ii) Pneumatic

(iii) Hydraulic

(iv) Electrical

(v) Electronic

5.5.1. SELF- ACTING

With this form of system the pressure, force or displacement, produced as a signal by the measuring element, is used directly as the source of power at the final control element. A temperature-sensitive, liquid-filled phial, produces a force which may action a attached The force exerted by the expanding or contracting liquid moves the Value spindle and so its position is related to the temperature measured, without any external power supply or amplification. This form of control is simple proportional in nature and usually has a fairly wide throttling range because the temperature sensed must change sufficiently to produce the force necessary for value movement

5.5.2 Pneumatic

Compressed air is piped to each controller and the controller reduces the pressure in a manner related to the value of the controlled variable that it senses. This is achieved by bleeding some air to waste. The reduced output, or the control pressure, is then transmitted to the relevant final control element, which is caused to move as the control pressure changes.

5.5.3 HYDRAULIC:

This form of system is similar to the pneumatic, but oil, water, or other fluid is used to transmit the signals. Hydraulic control systems are use for higher force transmission than are pneumatics.

5.5.4 Electrical

Variations in voltage are used to transmit signals and to provide the current necessary for moving the final control elements. Normally, these symptoms are used with 24 volts.

5.5.5 ELECTRONIC

This system is similar to electrical type, but much smaller signal strengths are transmitted by the sensing elements (usually resistance thermometers or thermocouples for temperature control). Electronic amplifiers change the magnitude of the signals to values suitable for activating the final control elements.

5.6 METHODS OF CONTROL

There are three methods of control in common use in air-conditioning: (i) two position (ii) proportional and to a lesser extent (iii) floating. All are capable of sophistication, aimed at improvement, whereby offset is reduced or stability enhanced. The form of control chosen must-suit the application.

5.6.1 SIMPLE TWO-POSITION CONTROL

There are only two valves of the manipulated variable: maximum and zero. The sensing element switches on full capacity when the temperature falls to say, the lower value of the differential and switches the capacity to zero when the upper value of the differential is reached.

5.6.2 SIMPLE PROPORTIONAL CONTROL

If the output signal from the controller is directly proportional to the deviation, then the control action is termed simple proportional. If this output signal is used to vary the position of a modulating value, then there is one and only one position of the valve for each value of the controlled variable. Offset is thus an inherent feature of simple proportional control. Only when the valve is half open will the valve of the controlled variable equal the set point. At all other times there will be a deviation, when the load is a maximum the deviation will be greatest in one direction, and vice-versa.

5.6.3 FLOATING ACTION

Floating control is a so-called because the final control element floats in a fixed position as long as the valve of the controlled variable lies between two chosen limits. When the value of the controlled variable reaches the upper of these limits, the final control element is activated to open, say, at a constant rate. Suppose that the value of the controlled variable then starts to fall in response to this movement of the final control element. When it falls back to the value of the upper limit, movement of the final control element is stopped and it stays in its new position partly open. It remains in this position until the controlled condition again reaches a value equal to one of the limits.

5.7 FUNCTION OF CONTROL ELEMENTS

The various automatic controls in the air conditioning and refrigerator system are incorporated to achieve the following functions among others:

(i) The automatic control elements are required to star and stop air conditioning and refrigeration equipment without manual operation.

(ii) They are to maintain stable or constants with a controllable device

(iii) It makes the plant to be independent of skilled supervision

(iv) It achieves a higher degree of accuracy in Maintaining the required temperatures, pressures and humidification

(v) It offers flexibility in the operation of complicated plants on various impulses from different parts of the system ensuring over-all control and protection.

(vi) It regulates the equipment to operates within the designs boundaries

(vii) It operates the equipment to efficiently

(viii) It regulates the system such that comfort conditions are maintained in the occupied space.