THERMOFLUIDS LABORATORY-I

(ME39606)

DEPARTMENT OF MECHANICAL ENGINEERING
IIT KHARAGPUR

Spring 2015-16

REFRIGERATION AND AIR-CONDITIONING LABORATORY REPORT

Submitted by :-

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Experiment No.-2

Studies on vapour compression refrigeration test-rig

Objectives:-

To carry-out steady-state measurements on the test-rig of a vapor compression refrigeration system in order to determine:-

(i) Carnot COP, Cycle COP and Actual COP of the refrigeration system

(ii) Overall heat transfer coefficients for the evaporator and the condenser

(iii) Overall volumetric efficiency of the compressor.

Components in our Set-up:-

1. Hermetic compressor	2. an air-cooled condenser
3. a water-cooled evaporator	4. thermostatic expansion valve/ capillary tube
5. rotameter	6. pressure gauges
7. thermometric wells	8. shut-off valve
9. sight glass	10. dryer
11. solenoid valve	12. filter

Theory:-

The test-rig uses R12 (di-chloro di-fluoro methane) as refrigerant. The evaporator used in the test-rig consists of a copper coil immersed in water kept in a stainless steel container. The stainless steel container is housed in a wooden box. The gap between the wooden box and the stainless steel container is filled with fiberglass insulation. An electrical stirrer is used to maintain uniform temperature of water in the SS container. A hermetic compressor with a piston displacement of 2.15 m^3/h is used in the test rig .The compressor is located in such a way that

the ambient air used for cooling the condenser also cools the compressor. The condenser used in the test rig is an air-cooled, fin-and-tube type condenser. A blower is used to blow the ambient air over the condenser and to extract the heat rejected by the condensing refrigerant. A capillary tube is used as an expansion device in the test rig. We can also use a thermostatic expansion valve with solenoid valve instead of the capillary tube.

Observations:-

- Condenser pressure, p_c = 170 psi (gauge)
  = (170 + 14.7) /14.5 bar
  =12.738 bar
- Evaporator pressure, p_e = 37 psi (gauge)
  = (37 + 14.7)/14.5 bar
  = 3.566 bar
- Condenser inlet temperature, t_{ci} = 77 ^{o} C = 350 k
- Condenser outlet temperature, t_{co} = 50 ^{o} C = 323 k
- Evaporator inlet temperature, t_{ei} = 7.0 ^{o} C = 280 k
- Evaporator outlet temperature, t_{eo} = 6.0 ^{o} C = 279 k
- Compressor energy meter reading, W = 450 W
- Heater energy meter reading, Q_e = 550 W
- Number of fins, N_f = 128
- Fin thickness, e = 0.38 mm
- Fin width, b = 0.047 m
- Fin height, h = 0.28 m
- Condenser width, L = 0.33 m
- Number of tubes in condenser, N_{tc} = 24
- Diameter of condenser tubes, d_c = 9.525 mm
- Average air velocity through condenser, V = 1.1 m/s
- Air temperature at condenser inlet, $T_{air,in}$ = 27° C = 300 k
- Air temperature at condenser outlet, $T_{air,out}$ = 38° C = 311 k
- Temperature of water in evaporator, T_{water} = 11.2 ^{o} C = 284.2 k
- Evaporator coil diameter, De = 0.208 m

- Evaporator tube diameter, de = 9.525 mm
- Number of tubes in the coil, Nte = 20
- Room temperature = 27 ^{o}C = 300 k
- Room pressure = 101.325 kPa
- Displacement of compressor,PD = 2.15 m3/h
- Rotameter Reading = 36 Lph

Exp - A

Testing of a vapor compression refrigeration system

Objectives: To carry out steady state measurements on the testing of a vapor compression refrigeration system in order to determine:

(i) Carnot COP, Cycle COP and Actual COP of the refrigeration system

(ii) Overall heat transfer coefficients for the evaporator and the condenser

(iii) Overall volumetric efficiency of the compressor

Name of Student:-

Condenser pressure, p_c [bar]	170 Psi
Evaporator pressure, p_e [bar]	37 Psi
Refrigerant temperature at condenser inlet, t_{ci} [° C]	77
Refrigerant temperature at condenser outlet, t_{co} [° C]	50
Refrigerant temperature at evaporator inlet, t_{ei} [° C]	7.0
Refrigerant temperature at evaporator outlet, t_{eo} [° C]	6.0
Compressor energy meter reading, [W]	450
Heater energy meter reading, Q_e [W]	550
Number of fins, N_f	128
Fin thickness, e [m]	0.38×10^{-3}
Fin width, b [m]	0.047
Fin height, h [m]	0.28
Condenser width, L [m]	0.33
Number of tubes in condenser, N_{tc}	24
Diameter of condenser tubes, d_c [m]	9.525×10^{-3}
Average air velocity through condenser, V [m/s]	1.1 m/s
Air temperature at condenser inlet, $T_{air,in}$ [° C]	27°C
Air temperature at condenser outlet, $T_{air,out}$ [° C]	38°C
Temperature of water in evaporator, T_{water} [° C]	11.2°C
Evaporator coil diameter, D_e [m]	0.208
Evaporator tube diameter, d_e [m]	9.525×10^{-3}
Number of tubes in the coil, N_{te}	20
Room temperature [° C]	27
Room pressure [Pa]	101325
Displacement of compressor, PD [m^3/h]	2.15

Sign of supervisor

Date:

Calculations:-

1. Carnot COP:-

$$COP_{carnot} = T_{evp} / (T_{cond} - T_{evp})$$

T_{evp} = Saturation temperature corresponding to evaporator pressure = 277.7 k

T_{cond} = Saturation temperature corresponding to condenser pressure = 325.1 k

So, COP_{carnot} = 277.7 / (325.1-277.7) = 5.859

COP_{carnot} = 5.859

2. Cycle COP :-

$$COP_{cycle} = (h_1 - h_3)/(h_2 - h_1)$$

h_1 = enthalpy of refrigerant vapour at evaporator outlet = 190.4 kJ/kg

h_2 = enthalpy of refrigerant vapour at compressor outlet = 227.3 kJ/kg

h_3 = enthalpy of refrigerant liquid at condenser outlet = 84.96 kJ/kg

So, COP_{cycle} = (190.4 – 84.96)/(227.3 - 190.4)

COP_{cycle} = 2.857

COP_{cycle} = 2.857

3. COP_{acual} = Actual refrigeration effect / Actual energy input to compressor

= Energy meter reading of heater / Energy meter reading of compressor

= 550 / 450

= 1.222

COP_{acual} = 1.222

Overall Heat Transfer Coefficients :-

1. Evaporator :

U_{evp} = Actual refrigeration effect / [A_{coil} x (T_{water} – T_e)]

Area of Evaporator coil = A_{coil} = π x De x π x de x Nte

= π x 0.208 x π x 9.525 x 20 x 10^{-3}

$= 0.391\ m^2$

So, $U_{evp} = 550 / [.391 \times (284.2 - 277.7)]$

$= 216.408\ W/\ m^2\text{-}k$

$$\boxed{\mathbf{U_{evp} = 216.408\ W/\ m^2\text{-}k}}$$

2. Condenser:

U_{cond} = Condenser Heat Rejection Rate / (A_{cond} x ($LMTD_{cond}$)

$= (m.c_p)_{air} \times (T_{air,out} - T_{air,in}) / (A_{cond} \times (LMTD_{cond})$

Where,

$LMTD_{cond} = (T_{air,out} - T_{air,in}) / \ln [(T_{cond} - T_{air,in}) / (T_{cond} - T_{air,out})]$

$= (311 - 300) / \ln [(325.1\text{-}300)/(325.1\text{-}311)]$

$= 19.07\ k$

T_{cond} = Saturation temperature corresponding to condenser pressure = 325.1 k

A_{cond} = Area of Bare tubes + Area of Fins

Area of Bare tubes = $\pi \times d_c \times N_{tc} \times (L - N_f \times e)$

$= \pi \times 9.525 \times 24 \times (.33\text{-}128 \times .38 \times 10^{-3}) \times 10^{-3}$

$= 0.202\ m^2$

Area of Fins = $2 \times b \times h \times N_f - 2 \times \pi \times d_c^2 \times N_{tc} \times N_f \times 0.25$

$= 2 \times 0.047 \times 0.28 \times 128 - 2 \times \pi \times 9.525 \times 9.525 \times 24 \times 128 \times 0.25 \times 10^{-6}$

$= 2.931\ m^2$

So, $A_{cond} = 2.931 + 0.202 = 3.133\ m^2$

m = Velocity of air x density x Area of condenser inlet

$= 1.1 \times 1.176 \times 0.33 \times .280$

$= 0.120$ kg/s

$U_{cond} = 0.120 \times 1005 \times (311\text{-}300) / (3.133 \times 19.07)$

$= 22.204\ W/\ m^2\text{-}k$

$$\boxed{\mathbf{U_{cond} = 22.204\ W/\ m^2\text{-}k}}$$

Overall Volumetric efficiency of compressor, η_{ov}:

η_{ov} = Actual refrigerant mass flow rate / Maximum possible mass flow rate.

$\eta_{ov} = (Q_{liq} / v_{liq}) / (PD / v_{suction})$

where ,

Q_{liq} = volumetric flow rate of liquid refrigerant as indicated by the rotameter

= $0.036 m^3/h$

v_{liq} = specific volume of liquid refrigerant at condenser outlet

= $0.0008315\ m^3/kg$

$v_{suction}$ = specific volume of refrigerant vapor at compressor inlet

= $0.04832\ m^3/kg$

PD = piston displacement of the compressor

= $2.15 m^3/h$

So, η_{ov} = (0.036/0.0008315) / (2.15/0.04832)

= 0.973

$$\boxed{\eta_{ov} = 97.3\%}$$

Discussion Questions

Q1. What are the errors in measurement of refrigerant temperature? Suggest a better method of measuring the temperature.

Ans: The errors in measurement of refrigerant temperature can be due to:

i. Faulty calibration of thermometer
ii. Improper insulation in the thermometer contacts leading to heat losses in the thermometer contacts
iii. Ineffective wetting of thermometer bulb by the connecting liquid.

Solid state thermoelectric sensors can be used to deal with these errors.

Q2. What are the errors in energy meter reading?

Ans: The errors in the meter reading can be due to following reasons:-

i. Human errors in reading (e.g. parallax error)
ii. Overheating in the equipment, leading to a change in internal resistances and hence a change in the reading
iii. Inaccurate calibration
iv. Unsteady or unknown power factor for AC source.

Q3. What are the assumptions made in calculating the Carnot, cycle and actual COPs?
Ans:

i. Carnot COP: a saturated refrigerant cycle is assumed between the reference temperatures
ii. Actual COP: we are directly using the readings from the energy meters, but there may be errors in the readings, along with electrical losses. Hence the actual heat input and compressor works can differ.
iii. Cycle COP: the processes in the evaporator and the compressors are considered ideal flow processes, with no other losses, and the expansion during throttling is assumed isenthalpic and ideal.

Q4. How does one find cooling capacity and condenser heat rejection rate from the refrigerant side? What differences do you observe when you compare cooling capacity and heat rejection rate calculated from refrigerant side with those calculated from water/air side ?
Ans: The properties of the working fluid required to calculate cooling capacity and heat rejection rate are:

i. Flow rate: rotameter
ii. Property tables or EES is used to calculate thermodynamics properties like enthalpy from knowing the temperatures and pressures at evaporator inlet and outlet. Same used for the condenser.

These differ from the conventional methods since these can be more error free. These do not involve many external measuring devices and gives a more direct approach to calculate the system properties.

Q5. Why is thermostat used in the water tank?

Ans: A thermostat is used to keep the temperature more or less constant in the water tank. When the temperature falls below a permissible limit, the power is switched off and cooling is momentarily halted. This enables to maintain steady state fairly easily for the measurements, to find the cooling capacity from the power consumption of the heater.

Q6. What are the safety devices used in the setup?

Ans: The safety devices used in the system are:

i. Strainer or filter: to filter out any dust particles mixed in the refrigerant.
ii. Drier: to silicate gel is used to absorb any moisture that might be present in the refrigerant while charging or improper filling
iii. H.P.L.P. Cutoff for the compressor: if the fluid pressure drops below a certain a level or shoots up beyond a certain limit, this cut off switch cuts power form the compressor, or else the compressor load may be transferred to the motor, which would draw high currents and burn out the motor.

Q7. The experiment gives COP at a fixed temperature. Suggest a method to find COP at different condenser and evaporator temperatures.

Ans: COP of evaporator and condenser at different temperatures can be found by conducting the experiment at various ambient temperatures through room temperature control and loads, by varying the heat input.

Q8. What are the assumptions made in calculating LMTD for the condenser?

Ans: The assumptions are:

i. The cycle is operated at saturation
ii. The heat transfer is assumed to take place at steady state conditions
iii. It is assumed that condensation is taking place throughout the condenser.

Q9. What is the significance of volumetric efficiency of compressor?

Ans: Volumetric efficiency is the ratio of the actual volume of refrigerant (Acf/min) drawn into the cylinder to the piston displacement (cf/min). It determines the critical pressure ratio for a

compressor, and is used to find the limit for the condenser pressure for a given evaporator pressure.

Q10. Carry out an error analysis of your experimental results.

Ans: Absolute error in T_{evp} =1 K

Absolute error in T_{cond} =1 K

Relative error in T_{evp} =1/(277.7) =0.0036

Relative error in T_{cond} - T_{evp} = 0.0422

Relative error in COP_{carnot} = 0.0422 + 0.0036 = 0.0458

Experiment No. 3

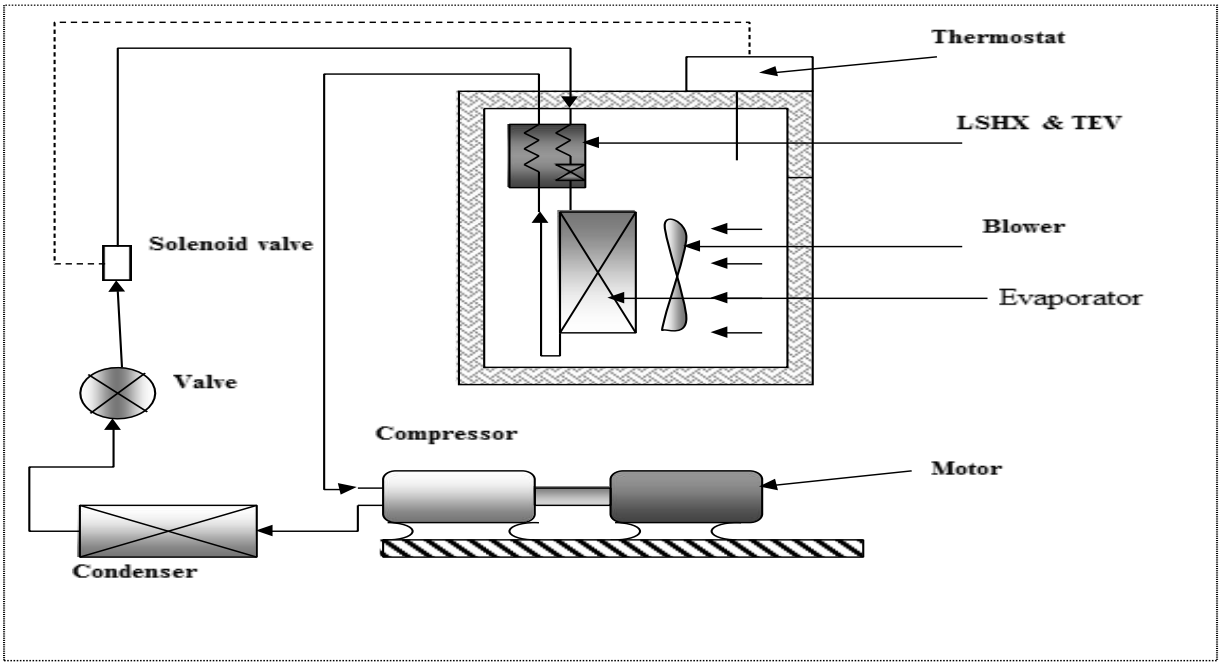
Studies on a small cold storage

Objectives:-

To study the pull down and cycling characteristics of a cold storage

Components in our Set-up:-

1. Condensing unit	2. Air cooled condenser
3. Open compressor 3TR	4. Dryer
5. Solenoid valve	6. Electric bulb
7. Sub-cooling heat exchanger	8. Thermostatic expansion valve
9. Direct expansion coil	10. Thermostat
11. HP/LP cut-out	12. Hand valve
13. Sight glass	14. Fan



Schematic of the cold storage with refrigeration plant

Introduction:-

A cold storage maintains temperature, relative humidity and air velocity in a given space as required for storage of perishable goods such as vegetables, fruits etc. A cold storage mainly consists of an insulated chamber, which is maintained at the required conditions with the help of a refrigeration system. The storage life and quality of goods stored in a cold storage depend on the type of insulation provided, capacity of the refrigeration system, type of controls provided etc. The pull-down and cycling are the most important characteristics of any cold storage system. The word pull-down implies the process of bringing down the temperature of the cold storage from ambient to the set, desired conditions. It is study of the events, which occur as the temperature decreases with time. To obtain the pull-down characteristics, all parts of the cold storage are first brought up to the ambient conditions by switching-off the system and leaving the door of the cold storage open to ambient for about 24 hours. As per the test standards, the ambient conditions must be maintained within certain standard limits. This process is known as soaking of the cold storage. Once the system is thoroughly soaked, then the door is closed and the refrigeration system is switched on. The temperature of the cold storage decreases continuously with time till it reaches the set cut-out point, at which the refrigeration system is switched off by the control system. The time taken for the cold storage to reach the desired set point from the initial ambient temperature is known as the pull-down time.

The term cycling refers to the on-and-off operation of the system under normal working conditions. The cooling load on the refrigeration system of a cold storage varies depending upon the ambient conditions, quantity and type of goods stored, frequency of product loading etc. To take care of the varying load conditions, the cold storages are normally designed for the worst possible conditions (i.e. for maximum cooling load). A control system regulates the capacity of the refrigeration system to suit the load under off-design conditions. One of the simplest types of capacity control is to switch off the refrigeration system once the desired condition (known as cutout point) is attained. Since the load on the system is continuous, the temperature of the cold storage starts increasing when the refrigeration system is switched-off. Once the temperature rises to a set point (known as cut-in point) the refrigeration system is switched-off. This process of

switching on and off of the refrigeration system is known as cycling. The cycle time is equal to the sum of on-time and off-time. The ratio of the on-time to the cycle time (percentage) is known percent run time and is an indication of the energy consumption of the system. The cycling characteristics are obtained after the initial pull-down and once the system reaches stable cycling conditions. The cycling characteristics here refer to the recording information on cut-out and cut-in point values (temperatures, pressures etc.), on-time, off-time, cycle time, percent run time etc.

Description of the cold storage under study:

The cold storage under study consists of an insulated chamber with cork as insulating material. The refrigeration plant uses open compressor of 3 TR capacity mounted on a condensing unit with R-22 as refrigerant. The condenser is air-cooled type with six rows of tubes and two fans driven by compressor motor. The set-up also consists of a hand valve, dryer, solenoid valve, a electrical bulb connected to solenoid valve to indicate its on or off position, sight glass, sub-cooling heat exchanger, thermostatic expansion valve and direct expansion coil (fin-tube evaporator). There is a fan in front of the evaporator to circulate the air inside the cold storage and to maintain uniform temperature in the cold storage. The feeler elements of the thermostat and the dial temperature gauge are mounted in front of the fan. The control system consists of a thermostat, which controls the opening/closing of the solenoid valve and a HP/LP cut-out which controls the operation of the compressor.

The control system in the present set-up works as follows:

When the desired temperature is achieved in the cold storage (cut-out point) as sensed by the feeler bulb of the thermostat, the thermostat closes the solenoid valve that stops the flow of liquid refrigerant to the evaporator. The compressor however continues to run and remove whatever refrigerant is left in the evaporator. This reduces the pressure in the evaporator. When evaporator pressure becomes slightly less than the LP setting (LP cut-out) of the HP/LP switch, the power supply to the compressor motor is cut-off.

Due to leakage of heat from the surroundings to the cold storage, the temperature of the cold storage starts rising up. When the cold storage temperature becomes equal to the cut-in point of thermostat, the thermostat opens the solenoid valve, the refrigerant starts to flow to the

evaporator and evaporator pressure starts to rise. When evaporator pressure reaches the cut-in point pressure setting of the HP/LP Cut-out switch, the power supply to the compressor motor is resumed. This process of on-off cycling of the compressor maintains the temperature in the cold storage within the limits of the sensitivity of the thermostat (cut-in – cut-out).

The **HP/LP** switch has three settings:

HP cut-out: a safety measure to avoid the build-up of excessive pressure in the condenser. This may occur if the condenser water pump or fan does not work satisfactory or there is oil logging of the condenser, or there is excess charge in the system. Once the pressure exceeds the safe limit, the HP Cut-out switches off the compressor. The system has to be started manually after troubleshooting.

LP cut-out: The LP cut-out switches off the compressor, when the suction pressure falls below the set point. The pressure falls below the set point when the solenoid valve is closed.

LP cut-in: This is used for restarting the compressor after it has been switched off by LP cut-out. The cut-in switch restarts the compressor when the suction pressure exceeds the set cut-in point. This happens when refrigerant starts flowing into the evaporator after the solenoid valve is opened.

The cut-in setting is decided by the starting torque characteristics of the compressor motor. In small capillary tube based systems, pressure balancing takes place when refrigerant flows from condenser to the evaporator during off cycle. Hence, the compressor starts against zero pressure difference. On the other hand in large systems, starting the compressor at the LP Cut-out pressure will require a large starting torque.

The Solenoid Valve: This has a primary winding and a soft iron core, which is connected to a plunger. The plunger is lifted up and the refrigerant continues to flow as long as current flows through the winding. The thermostat acts as a switch, which remains closed as long as the cold storage temperature is greater than the set value. The thermostat puts off the flow of current to the winding when the desired temperature is reached. The winding gets de-energized and the plunger drops down due to gravity in the refrigerant line to stop the flow of liquid refrigerant.

Obviously the solenoid should be mounted on the horizontal run of the liquid refrigerant tube to take advantage of gravity to drop the plunger. There is condenser pressure on topside of the plunger and evaporator pressure on the lower side of the plunger. Hence to open the valve electromagnetic force is required to take care of the weight of the plunger and the pressure difference across it. A pilot valve is provided inside the plunger. This is has a smaller area so that when winding gets energized it opens up and balances the pressure on the two sides of the plunger which can then lifted up be a smaller magnetic force.

The solenoid valve is self-closing type valve, that is higher pressure acts from the top and keeps it closed (the common water taps in water line are self-opening type, that is, water pressure acts from down below and helps it in opening). The solenoid valve has an arrow marked outside, which indicates the flow direction to be followed while mounting it. *A Cut-out Solenoid valve will be shown and its working principle will be demonstrated during the class.*

The Expansion valve and sub-cooling heat exchanger: Thermostatic expansion valve is used in the system. The sub-cooling heat exchanger and the expansion valve are enclosed in a casing.

Observations:-

Exp - D

Cold Storage

Objectives: - To study the pull down and cycling characteristics of cold storage

Name of Student:- Naveen Bhati, Naman Joshi, Nilesh, Nirav, Yashwant

Clock Time (hh:mm:sec)	Temperature	Evaporator Pressure	Condenser Pressure	Solenoid Condition	Compressor condition
0	81	16	9.4	Open Bulb glows	On
35.47	46	10	145	Closed Bulb goes off	On
36.00	45	-4in	142	Closed	Off
42.14	46	2	131	Open	Off
42.28	46	18	132	Open	On
46.17	46	13	145	Closed	On
46.32	46	-2in	144	Closed	Off
52.19	47	2	132	Open	Off
52.23	47	18	134	Open	On
56.12	46	13	147	Closed	On
56.27	46	-3in	144	Closed	Off
1.02.45	47	2	132	Open	Off
1.02.48	47	20	129	Open	On
1.06.06	47	10	146	Closed	On
1.06.21	47	-3in	144	Closed	Off
				Open	Off
				Open	On
				Closed	On
				Closed	Off

Signature of Supervisor: RC

Date : 24/02/2016

Calculations:-

Pull Down Time, On Time and Off Time are calculated as follows:

Pull Down Time = 35 Minutes 47 Seconds

CYCLE 1:

Off Time = 6 Minutes 28 Seconds

On Time = 4 Minutes 4 Seconds

% cycle run time = 244/(244 + 388) *100 = 38.6

CYCLE 2:

Off Time = 5 Minutes 51 Seconds

On Time = 4 Minutes 4 Seconds

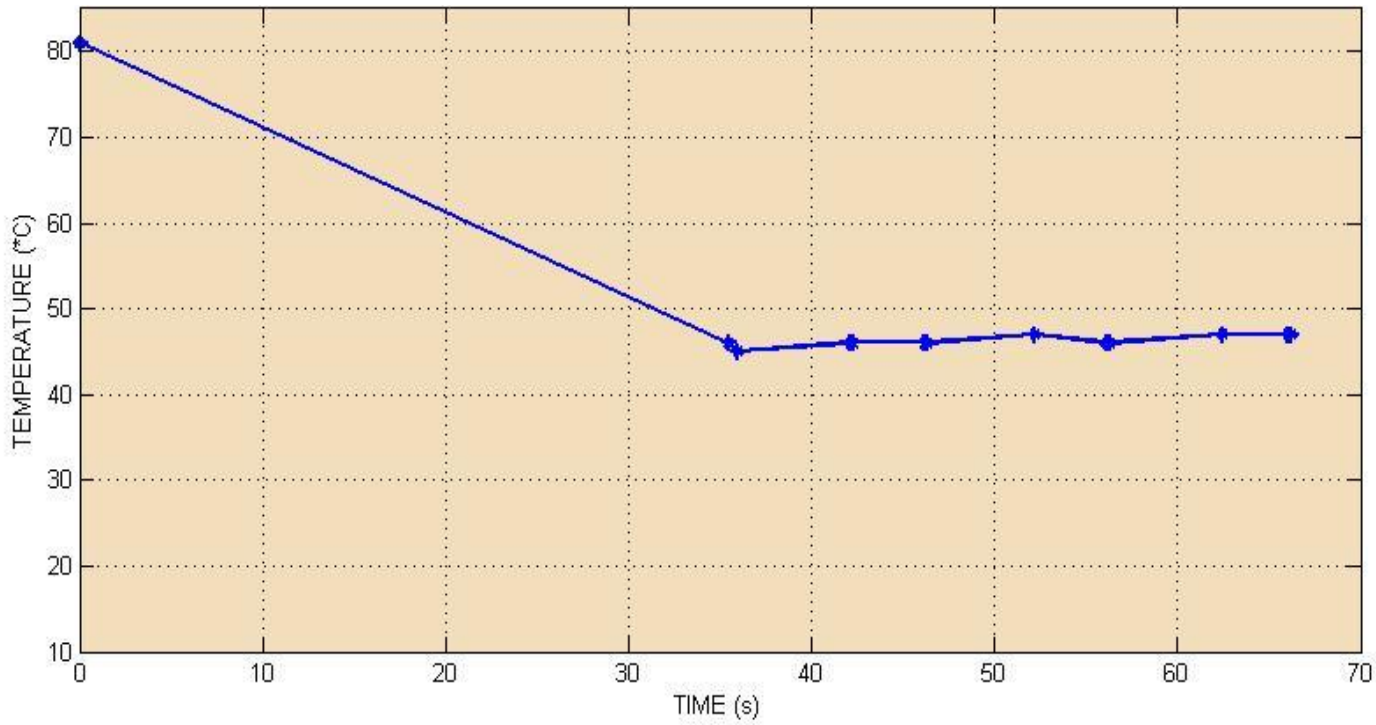
% cycle run time = 244/(244 + 351) *100 = 41.0

CYCLE 3:

Off Time = 6 Minutes 21 Seconds

On Time = 3 Minutes 33 Seconds

% cycle run time = 213/(213 + 381) *100 = 35.86



Discussion Questions

Q1. Describe various methods of capacity control of refrigeration systems.

Ans: The various methods used to control the capacity of refrigeration systems are:

Bypass control method

i. On-off control
ii. Sensor controllers
iii. Suction internal unloading
iv. Valve open hold control
v. Using a thermostat to automatically switch on and off the cooling by shutting off the compressor

Q2. Describe a method for precise control of temperature.

Ans: A solenoid valve guided and coupled to a thermostat and set at a desired pre-set temperature, can act as a temperature control system.

When the pre-determined temperature is attained in the cold storage, cut-out point is said to be reached. The thermostat senses this, and causes the electric circuitry to close the solenoid valve, hence arresting the flow of refrigerant in liquid form to the evaporator. The compressor continues to operate and removes the remaining refrigerant in the evaporator, and reducing the pressure in the evaporator. When the pressure in the evaporator falls below the LP setting of the compressor, the compressor itself is shut down. As the temperature now rises as a result of no cooling work done, it reaches a pre-set upper limit value, which is sensed by the thermostat. This in turn causes the compressor to work and supply evaporator with refrigerant, thus cooling the system down again. This is repeated.

Q3. Describe methods for humidity control.

Ans: Dry and wet bulb temperatures can be measured by the use of two temperature sensitive elements. Humidity measurement can be done by using hygroscopic compounds, especially salts, viz. calcium chloride, lithium chloride and magnesium chloride. The water or moisture absorbed

can be calculated and from there the amount of humidity can be estimated. Passage over heating and cooling coils that are exposed to the ambient can be used to control humidity.

Q4. Some old types of cold storage have the cooling coil mounted near the ceiling. These are called bunker coils. These do not have any fan for air circulation. How does the air circulate in such a cold storage?

Ans: Bunker coiled cold storages use natural convection to circulate air around the storage. These are more energy efficient than forced systems, but can be only used for systems requiring moderate amount of cooling.

Q5. In some cold storages the chilled air is injected at high velocity into the cold storage. What are the disadvantages and advantages of this method?

Ans: This system has the advantage of higher cooling or higher heat transfer rates, since it leads to a higher heat transfer coefficient. This results in faster cooling. But it also requires additional power input to pump or force the air at high velocities, in addition to the energy requirement to chill the air down to a low temperature to transfer heat fast.

Q6. What is the advantage of buying a condensing unit rather buying separate components and assembling them?

Ans: A single compressor unit is properly sealed and hence can work efficiently and without significant amounts of leakage. On assembly of separate components, faulty assembly, misfit and slight errors and mismatches can lead to leakage. It can also lead to more noise and vibration, in addition to the reduction in efficiency.

Q7. Where and why is TEV with external pressure equalizer used?

Ans: A Thermostatic Expansion Valve is used to maintain a constant superheating of the refrigerant. External pressure equalizer is used when the pressure drop in the evaporator is substantial resulting in superheat of vapour to be higher than desired. This desired value is adjusted with follow up springs, built into the TEV. The equalizer transmits the pressure at the evaporator outlet to the inside of the bellows in the TEV. So the pressure in the inside of the bellows is

‘equalized’ with that at the evaporator outlet, and adjusts and filters out any fluctuation in the pressure through the evaporator tubes, irrespective of the drop in pressure.

Q8. Why are cross charge, liquid charge and vapor charge used in TEV.

Ans: The thermostatic charge is the substance in the TEV's sensing bulb which responds to suction line temperature to create the bulb pressure, and it is designed to allow the TEV to operate at a satisfactory level of superheat over a specific range of evaporating temperatures.

The categories of thermostatic charges are the following:

1. Liquid Charge
2. Gas Charge (or Vapour Charge)
3. Liquid-Cross Charge
4. Gas-Cross Charge

The liquid charge consists of the same refrigerant in the thermostatic element that is used in the refrigeration system, while the liquid-cross charge consists of a refrigerant mixture. The term cross charge arises from the fact that the pressure-temperature characteristic of the refrigerant mixture used within the sensing bulb will cross the saturation curve of the system refrigerant at some point. Both the liquid and liquid-cross charges possess sufficient liquid such that the bulb, capillary tubing, and diaphragm chamber will contain some liquid under all temperature conditions. This characteristic prevents charge migration of the thermostatic charge away from the sensing bulb if the sensing bulb temperature becomes warmer than other parts of the thermostatic element. Charge migration will result in loss of valve control. An additional characteristic of these charges is their lack of a maximum operating pressure (MOP) feature. A thermostatic charge with an MOP feature causes the TEV to close above a predetermined evaporator pressure, thereby restricting flow to the evaporator and limiting the maximum evaporator pressure at which the system can operate.

Similarly, the gas charge consists of the same refrigerant in the thermostatic element that is used in the refrigeration system, while the gas-cross charge consists of a refrigerant mixture. Unlike the liquid type charges, both gas charges are distinguished by having a vapour charge in the thermostatic element which condenses to a minute quantity of liquid when the TEV is in its normal operating range. This characteristic provides an MOP for the valve at the bulb temperature

of which the liquid component of the charge becomes vapour. Above this bulb temperature, a temperature increase does not significantly increase thermostatic charge pressure, limiting the maximum evaporator pressure at which the system can operate. A disadvantage of this type of thermostatic charge is the possibility of charge migration.

Cross charge uses "power fluids" which have a more flat pressure-temperature curve than the refrigerant hence helping maintain a constant degree of superheat for different evaporator temperatures

Q9. Describe pull down characteristics of a compressor and outline the procedure to bypass the power peak while starting a compressor.

Ans: There is an evaporator temperature at which the power reaches a maximum value. If the design evaporator temperature of the refrigeration system is less than the evaporator temperature at which the power is maximum, then the design power requirement is lower than the peak power input. However, during the initial pull-down period, the initial evaporator temperature may lie to the left of the power peak. Then as the system runs steadily the evaporator temperature reduces and the power requirement passes through the peak point. If the motor is designed to suit the design power input then the motor gets overloaded during every pull-down period as the peak power is greater than the design power input.

The pressure difference between the condenser and evaporator should be as minimum as possible as possible, even none, in order to bypass the power peak during start-up.

Q10. What decides the cut-in pressure?

Ans: The evaporator pressure at which the compressor starts working, after passing through the HPLP cut-off system is called the cut-in pressure. The prescribed evaporator pressure designed into its system and the starting torque capacity of the condenser motor decide the cut-in pressure.

Q11. What happens if instead of using the LP cut-out/cut-in, the thermostat itself simultaneously closes the solenoid valve and switches off compressor?

Ans: Such a system will work just fine, as the solenoid valve is capable of preventing slogging in the compressor. But in the absence of the LP cut-off, on restarting the compressor will have to work against a high pressure difference so a high starting torque will be required.

Q12. What differences did you notice between the components of the refrigeration system of the cold storage and the vapor compression refrigeration test rig (Experiment 1)?

Ans: Here, a thermostatic expansion valve was used, whereas in VCRC, a capillary tube was used for expansion. Here, the compressor used is an open type, whereas there it was a hermetically sealed one. In VCRC, R12 was used, whereas here, R22 is used. Solenoid valve is used here, but not in VCRC.

Experiment No. 4

Studies on a summer air conditioning system

Objectives:-

To determine:

i) The COP of the air-conditioning system

ii) Condensate removal rate (moisture removal rate)

iii) Apparatus dew-point and the by-pass factor of the cooling coil

iv) To plot the processes on psychrometric chart for readings 2, 4 and 6.

Components in our Set-up:-

1. Hermetic compressor 1 TR capacity	2. finned tube direct expansion coil type evaporator
3. Air-cooled condenser	4. Thermostatic expansion valve Dryer
5. Fan	6. a blower

There is a provision to re-circulate a part of the conditioned air through a tube, which has an orifice meter to determine the flow rate. An orifice meter can also measure the flow rate of inlet air. The test chamber duct has provisions to preheat, reheat and humidify the air.

Principle and procedure:

The COP of the air conditioning system is given by:

COP = Cooling capacity / Power input to compressor

= Qe / Wc

From energy balance across the evaporator coil, the actual cooling capacity in kW is given by:

$$Qe = m_a (h_{a,I} - h_{a,o}) - m_w . h_w$$

where m_a is the mass flow rate of dry air (kg/s); $h_{a,i}$ and $h_{a,o}$ are the enthalpies of air at inlet and outlet of the cooling coil (kJ/kg of dry air), m_w is the moisture removal rate (condensate rate, kg/s) and h_w is the specific enthalpy of the condensate (kJ/kg of water).

From mass balance of water vapor across the cooling coil, the condensate rate m_w is given by:

$$m_w = m_a (W_i - W_o)$$

where W_i and W_o are the humidity ratios of inlet and outlet air (kg of water/kg of dry air).

For a given barometric pressure, the condition of the moist air at the inlet and outlet of the cooling (i.e. properties such as enthalpy, humidity ratio) are obtained by measuring the two independent thermodynamic properties of moist air, namely dry bulb temperature (DBT) and wet bulb temperature WBT. The DBT and WBT are measured by using a sling type of psychrometer. Then from the DBT & WBT values other properties of moist air can be obtained either directly from the psychrometric chart or by using empirical equations developed.

The mass flow rate through the evaporator duct is obtained by direct measurement of average velocity of air at the outlet with a vane type manometer and also by measuring the pressure drop across the orifice meter (as shown by the two manometers). The outlet area is determined by measuring the outlet diameter of the test chamber. Perfect gas relation at outlet temperature determines the density of dry air. The mass flow rate of the dry air from measured average velocity is given by:

$$m_a = \rho_a . A_{outlet} . V_{avg}$$

where ρ_a is the density of dry air at outlet temperature, A_{outlet} is the cross sectional area of the duct at the outlet and V_{avg} is the average air velocity at the duct outlet.

The condensate rate is measured by collecting the condensate in a measuring cylinder for a measured time. This can also be obtained from the mass flow rates of dry air and the inlet outlet humidity ratios as given above. The condensate enthalpy is obtained by measuring the temperature on the surface of the cooling coil (assumed to be same as condensate temperature).

The power consumption of the compressor is obtained from the measured refrigerant mass flow rate ($\mathbf{m_r}$) and computed refrigerant enthalpy change across the compressor ($\Delta\mathbf{h_c}$). The enthalpy change across the compressor is obtained from the measured values of pressure and temperature at the inlet and exit of the compressor and using refrigerant (R134a) property data. It is to be noted that this procedure gives only an approximate value of COP as it is based on the assumption of adiabatic compression. Power consumption is given by:

$$Wc = m_r . \Delta h_c$$

iii) Apparatus dew point (t_{ADP}) and by-pass factor (X)

The condition line of the cooling and dehumidification process on psychrometric chart follows a straight line. This is known as *straight-line law.* The inlet and outlet states of the moist air are located on the psychrometric chart; joined by a straight line and extended to intersect the saturation curve. Its point of intersection with the saturation curve is known as Apparatus Dew Point of the cooling coil. Apparatus Dew Point temperature may be considered as an effective surface temperature of the coil. The bypass factor X is defined as:-

$$X = (T_{a,o} - T_{ADP}) / (T_{a,i} - T_{ADP})$$

The bypass factor is an indication of the inefficiency of the cooling coil. Refrigerant used is R134a.

Observations:-

Static duct pressure, z2=3.6 (mm of H_2O)

Readings	One	Two	Four	Six
First pre-heater, 1kW	off	off	on	off
First re-heater, 1kW	off	off	off	on
Fan supply voltage(V_f) [V]	130	150	150	150

Air at fan inlet DBT(T1) [°C]	32.8	32.9	32.9	33.0
Air at fan inlet WBT(T2) [°C]	25.6	26.0	26.2	26.2
Air after pre-heater DBT(T3) [°C]	33.0	33.3	44.2	33.6
Air after pre-heater WBT(T4) [°C]	25.1	25.7	27.8	25.8
Air after cooling/dehumidification DBT(T5) [°C]	21.1	22.8	26.6	23.5
Air after cooling/dehumidification WBT(T6) [°C]	19.5	20.6	23.1	21.4
Air after re-heater DBT(T7) [°C]	21.5	22.9	26.1	34.5
Air after re-heater WBT(T8) [°C]	21.4	22.4	24.4	24.6
Temperature at evaporator outlet (T13) [°C]	16.2	16.9	18.7	17.4
Temperature at condenser inlet (T14) [°C]	84.0	85.5	86.6	86.3
Temperature at condenser outlet (T15) [°C]	51.2	51.1	53.8	51.9
Evaporator outlet pressure (P1) [kPa]	275	285	325	275
Condenser inlet pressure (P2) [kPa]	1300	1350	1425	1350
Condenser outlet pressure (P3) [kPa]	1275	1325	1400	1325
Duct pressure z1 [mm of H_2O]	6.9	8.8	9	9
Duct differential pressure $\Delta z=z1-z2$ [mm of H_2O]	3.3	5.2	5.4	5.4
Condensate collected (m_c) [mL]	72	74	40	76
Time interval for condensate collected (x) [s]	300	300	300	360
R134a mass flow rate (m_{ref}) [g/s]	13.5	14.0	14.25	13.75
Ambient temperature [°C]	32.8	32.9	32.9	33.0
Ambient WBT [°C]	25.6	26.0	26.2	26.2

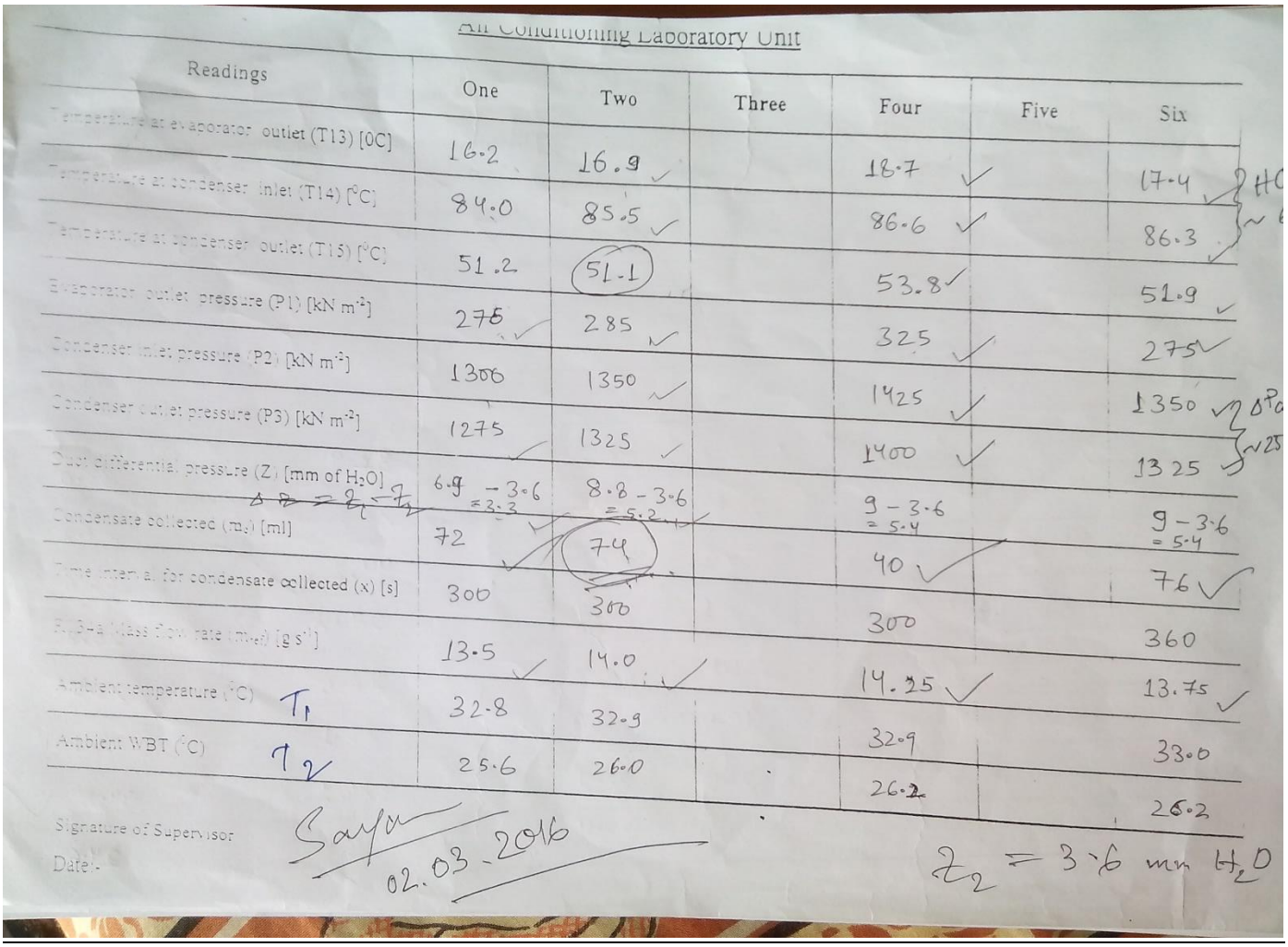
a) To find COP, condensate removal rate and by-pass factor for different operating conditions.

b) To plot the processes on psychrometric chart for readings 2, 4 & 6.

Exp – B

Student's Name :- Noman Joshi, Naveen Bhati, Nilesh Kumar, Nirav V. Doshi, Yashwant

Readings	One	Two	Three	Four	Five	Six
First pre-heater, 1kW	Off	Off	On	On	Off	Off
First re-heater, 1kW	Off	Off	Off	Off	On	On
Fan supply voltage (Vf) [V]	130	150	130	150	130	150
Air at fan inlet DBT (T1) [°C]	32.8	32.9		32.9		33.0
Air at fan inlet WBT (T2) [°C]	25.6	26.0		26.2		26.2
Air after pre-heater DBT (T3) [°C]	33.0	33.3		44.2		33.6
Air after pre-heater WBT (T4) [°C]	25.1	25.7		27.8		25.8
Air after cooling / dehumidification DBT (T5) [°C]	21.1	22.8		26.6		23.5
Air after cooling / dehumidification WBT (T6) [°C]	19.5	20.6		23.1		21.4
Air after re-heater DBT (T7) [°C]	21.5	22.9		26.1		34.5
Air after re-heater WBT (T8) [°C]	21.4	22.4		24.4		24.6



Air Conditioning Laboratory Unit

Readings	One	Two	Three	Four	Five	Six
Temperature at evaporator outlet (T13) [0C]	16.2	16.9		18.7		17.4
Temperature at condenser inlet (T14) [°C]	84.0	85.5		86.6		86.3
Temperature at condenser outlet (T15) [°C]	51.2	51.1		53.8		51.9
Evaporator outlet pressure (P1) [kN m⁻²]	275	285		325		275
Condenser inlet pressure (P2) [kN m⁻²]	1300	1350		1425		1350
Condenser outlet pressure (P3) [kN m⁻²]	1275	1325		1400		1325
Duct differential pressure (Z) [mm of H_2O]	6.9 − 3.6 = 3.3	8.8 − 3.6 = 5.2		9 − 3.6 = 5.4		9 − 3.6 = 5.4
Condensate collected (m_w) [ml]	72	74		40		76
Time interval for condensate collected (x) [s]	300	300		300		360
Evaporator Mass flow rate ($m_{w(e)}$) [g s⁻¹]	13.5	14.0		14.25		13.75
Ambient temperature (°C) T_1	32.8	32.9		32.9		33.0
Ambient WBT (°C) T_2	25.6	26.0		26.2		26.2

Signature of Supervisor: Sayan

Date: 02.03.2016

$Z_2 = 3.6$ mm H_2O

System Properties and Results:

Property	Case 2	Case 4	Case 6
Condensate m_w(exp)	0.2467 g/s	0.1333 g/s	0.2111 g/s
Humidity Ratio w_{in}	0.01773	0.01684	0.01777
Humidity Ratio w_{out}	0.01434	0.01637	0.01517
Air Velocity	9.375 m/s	9.618 m/s	9.727 m/s
Volume flow rate-air	0.1769 m^3/s	0.1815 m^3/s	0.1835 m^3/s
Specific vol. of air	0.8614 m^3/kg	0.8731 m^3/kg	0.8931 m^3/kg
m_a	0.2054 kg/s	0.2079 kg/s	0.2055 kg/s

Enthalpy of air, $h_{a,in}$	78.95 kJ/kg	87.99 kJ/kg	79.37 kJ/kg
Enthalpy of air, $h_{a,out}$	59.41 kJ/kg	68.53 kJ/kg	62.25 kJ/kg
Qair	3.995 kW	4.034 kW	3.501kW
Theoretical m_w	0.6963 g/s	0.0977 g/s	0.5343 g/s
% error in w	-64.57%	36.44%	-60.51%
mref	0.0140 kg/s	0.01425 kg/s	0.01375 kg/s
Enthalpy, R134a, h1	263.3 kJ/kg	264 kJ/kg	264 kJ/kg
Enthalpy, R134a, h2	313.7 kJ/kg	313.8 kJ/kg	314.6 kJ/kg
Enthalpy, R134a, h4	125.2 kJ/kg	129.4 kJ/kg	126.4 kJ/kg
Qref	1.933 kW	1.918 kW	1.89 kW
Wcompressor	0.7056 kW	0.7096 kW	0.6958 kW
COP|air	5.662	5.685	5.031
COP|refrigerant	2.740	2.703	2.716
TADP	18.2°C	21.6°C	19.5°C
Bypass factor, X	0.3046	0.2212	0.2695

Calculations:-

Case 2: Pre-Heater OFF, Re-Heater OFF, Fan Supply Voltage: 150V

Mass flow rate of air, $m_a = \rho_a . A_{outlet} . V_{avg} = 1.1609 \times \pi \times 0.155^2 \times .25 \times 9.375 = 0.2054$ kg/s

Specific enthalpy of air before cooling, $h_{a,in}$ = 78.95 kJ/kg of dry air

Specific enthalpy of air after cooling, $h_{a,out}$ = 59.41 kJ/kg of dry air

Condensate removal rate, m_w = 0.2467 g/s (volume collected x density/time)

Specific Enthalpy of condensate = 76.31 kJ/kg

w_{in} = humidity ratio for DBT=T_3=33.3°C, WBT=T_4=25.7°C

= 0.01773

w_{out} = humidity ratio for DBT=T_5=22.8 °C, WBT=T_6=20.6 °C

= 0.01434

Theoretical condensate collected $m_w = m_a(w_{in} - w_{out})$ = 0.6963 g/s

Cooling Capacity = $Q_{air} = m_a*(h_{a,in} - h_{a,out}) - m_w*h_w = 3.995$ kW

Mass flow rate of refrigerant, $m_r = 0.0140$ kg/s

Specific Enthalpy at evaporator outlet, h1 = 263.3 kJ/kg

Specific Enthalpy at condenser inlet, h2 = 313.7 kJ/kg

Specific Enthalpy at condenser outlet, h4 = 125.2 kJ/kg

Work done by compressor, $W_c = m_r*(h2 - h1) = 0.7056$ kW

$COP_{air} = Q_{air} / W_c = 5.662$

$Qref = m_r (h1 - h4)$

$= 0.0140(263.3 - 125.2)$

$= 1.933$ kW

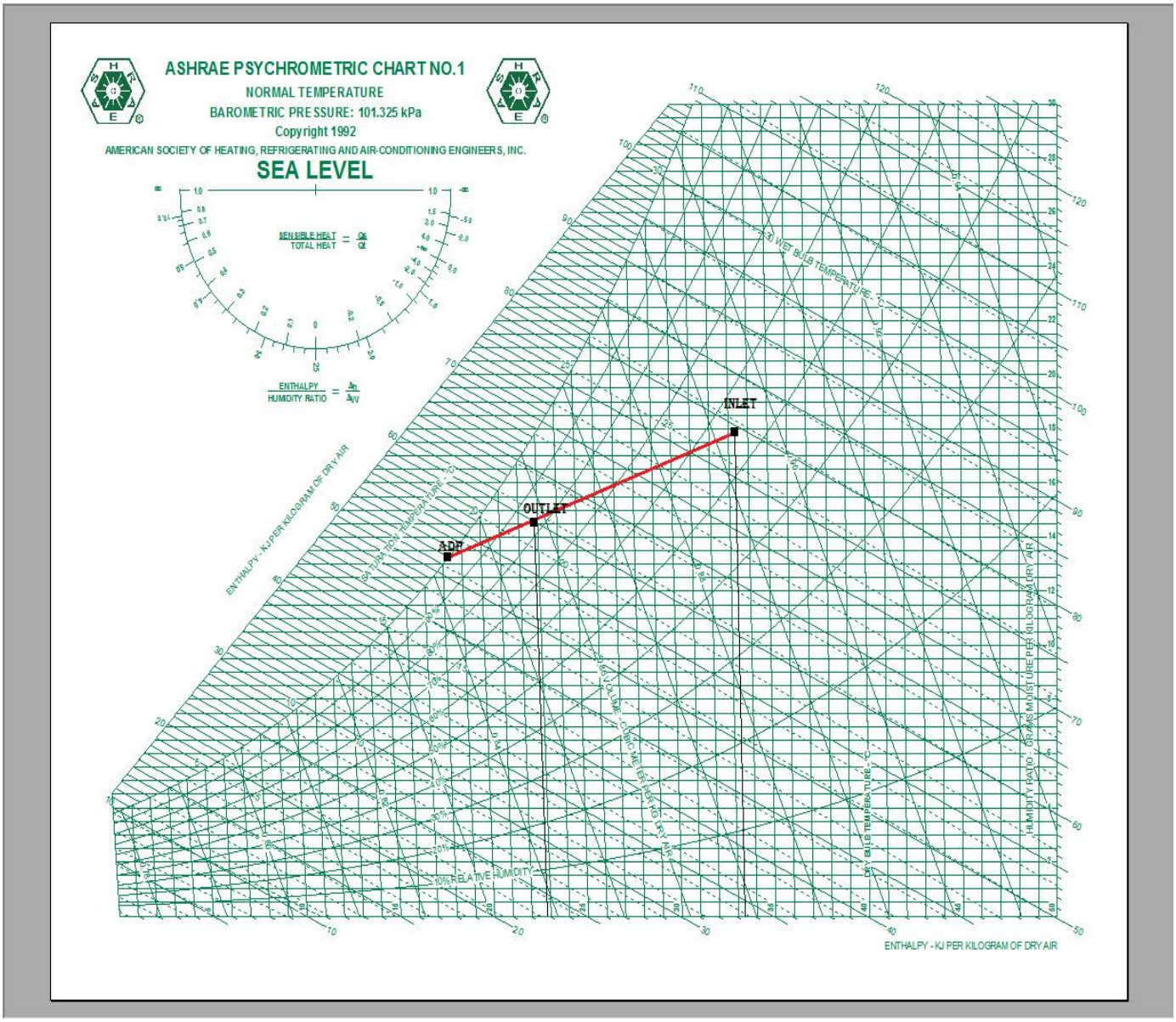
$COP|refrigerant = Qref/ W_c$

$= 2.74$

From psychrometric chart,

Apparatus Dew point Temperature, $T_{ADP} = 18.2\ ^oC$

By-pass factor of cooling coil, $X = (T_{a,o} - T_{ADP})/ (T_{a,i} - T_{ADP}) = (22.8-18.2)/(33.3-18.2) = 0.3046$


ASHRAE PSYCHROMETRIC CHART NO.1
NORMAL TEMPERATURE
BAROMETRIC PRESSURE: 101.325 kPa
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SEA LEVEL
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
INLET
OUTLET
ADP
ENTHALPY - KJ PER KILOGRAM OF DRY AIR
WET BULB TEMPERATURE - °C
DRY BULB TEMPERATURE - °C
HUMIDITY RATIO - GRAMS MOISTURE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY

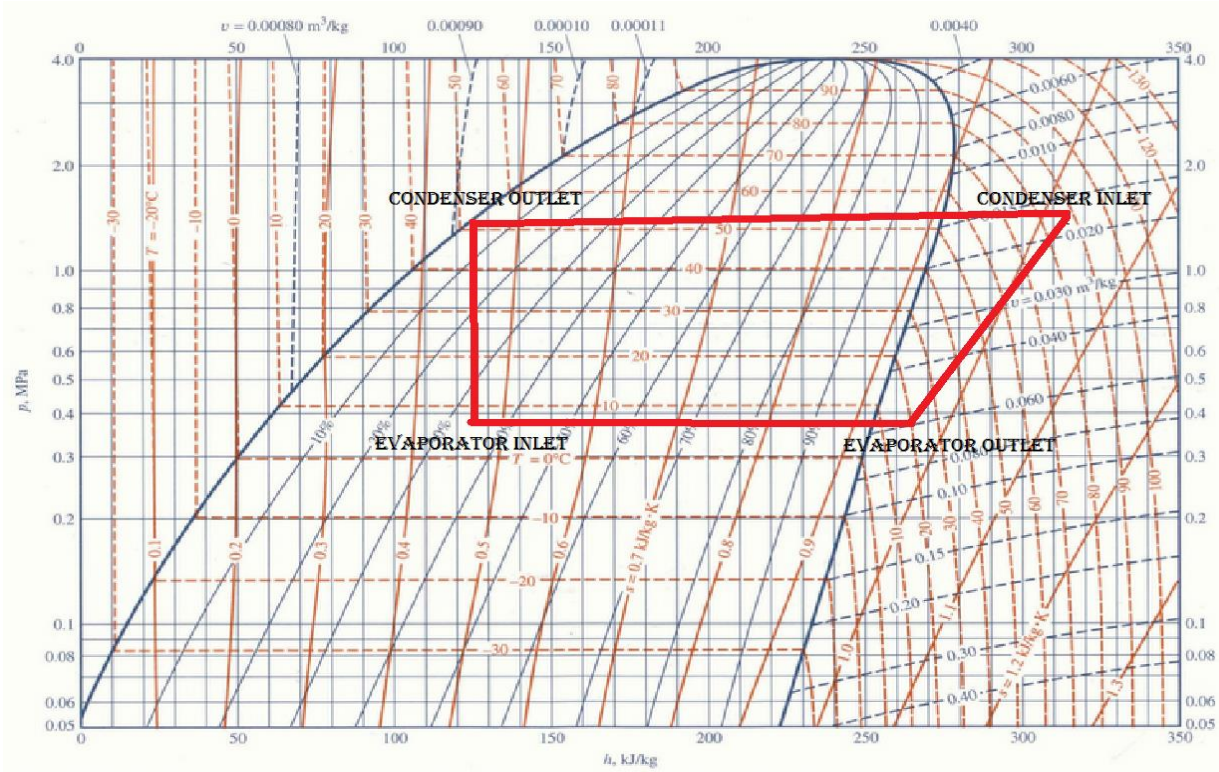


Chart A-11 R134a *ph* diagram. (*Source:* Based on *Thermodynamic Properties of HFC-134a (1,1,1, 2-tetrafluoroethane)*, DuPont Company, Wilmington, Delaware, 1993, with permission.)

<u>Case 4: Pre-Heater ON, Re-Heater OFF, Fan Supply Voltage: 150V</u>

Mass flow rate of air, $m_a = \rho_a . A_{outlet} . V_{avg}$ = 1.1453 x π x 0.155^2 x .25 x 9.618 = 0.2079 kg/s

Specific enthalpy of air before cooling, $h_{a,in}$ = 87.99 kJ/kg of dry air

Specific enthalpy of air after cooling, $h_{a,out}$ = 68.53 kJ/kg of dry air

Condensate removal rate, m_w = 0.1333 g/s (volume collected x density/time)

Specific Enthalpy of condensate = 90.53 kJ/kg

w_{in} = humidity ratio for DBT=T_3=44.2°C, WBT=T_4=27.8°C

= 0.01684

w_{out} = humidity ratio for DBT=T_5=26.6 °C, WBT=T_6=23.1 °C

= 0.01637

Theoretical condensate collected $m_w = m_a(w_{in} - w_{out})$ = 0.0977 g/s

Cooling Capacity = $Q_{air} = m_a*(h_{a,in} - h_{a,out}) - m_w*h_w$ = 4.034 kW

Mass flow rate of refrigerant, m_r = 0.01425 kg/s

Specific Enthalpy at evaporator outlet, h1 = 264 kJ/kg

Specific Enthalpy at condenser inlet, h2 = 313.8 kJ/kg

Specific Enthalpy at condenser outlet, h4 = 129.4 kJ/kg

Work done by compressor, $W_c = m_r*(h2 - h1) = 0.7096$ kW

$COP_{air} = Q_{air} / W_c = 5.685$

$Q_{ref} = m_r (h1 - h4)$

$= 0.0140(264 - 129.4)$

$= 1.918$ kW

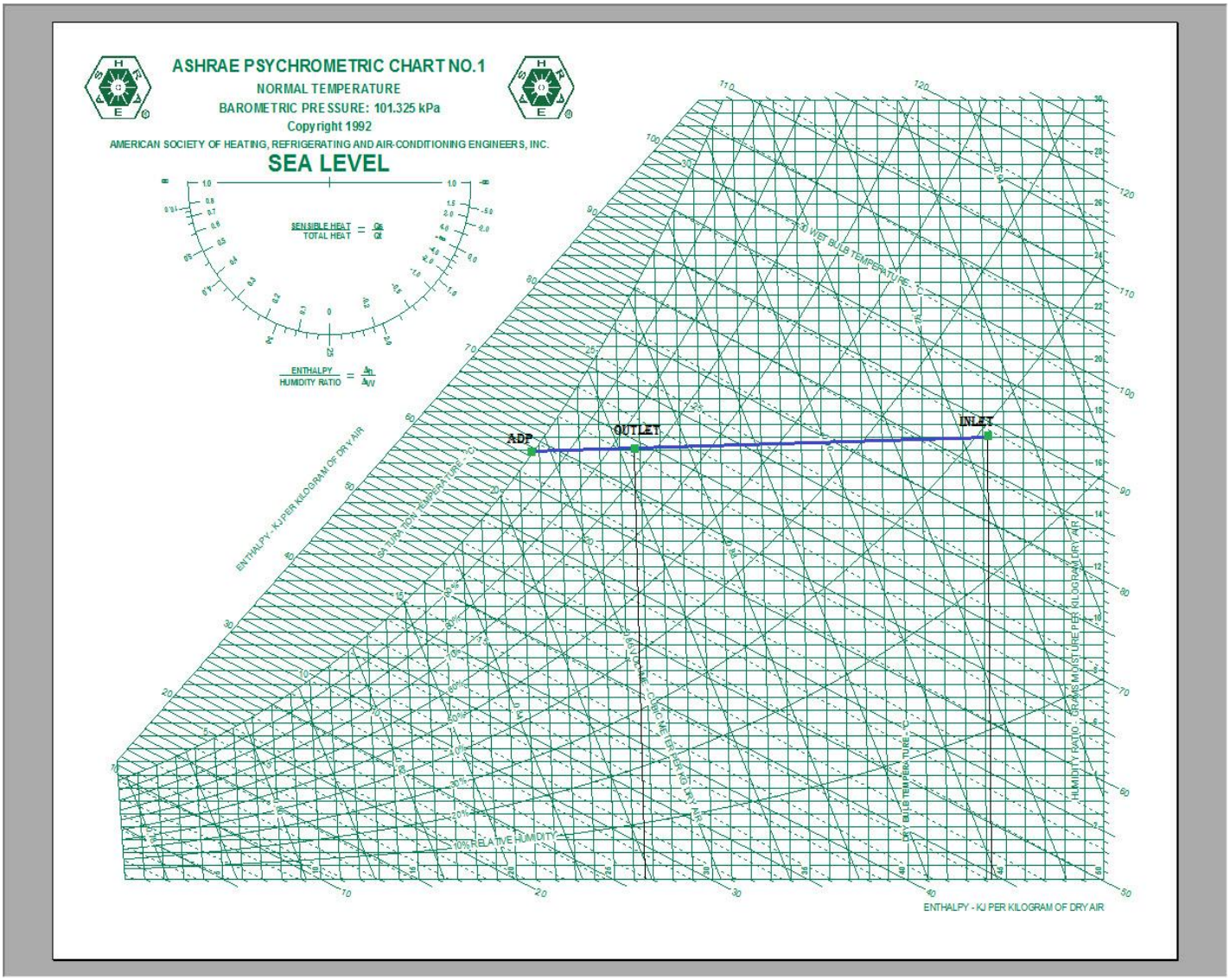
$COP|_{refrigerant} = Q_{ref} / W_c$

$= 2.703$

From psychrometric chart,

Apparatus Dew point Temperature, $T_{ADP} = 21.6\ ^{o}C$

By-pass factor of cooling coil, $X = (T_{a,o} - T_{ADP}) / (T_{a,i} - T_{ADP}) = (26.6-21.6)/(44.2-21.6) = 0.2212$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ENTHALPY / HUMIDITY RATIO = Δh/ΔW
ADP
OUTLET
INLET
WET BULB TEMPERATURE - °C
ENTHALPY - KJ PER KILOGRAM OF DRY AIR
SATURATION TEMPERATURE - °C
VOLUME - CUBIC METERS PER KG DRY AIR
RELATIVE HUMIDITY
DRY BULB TEMPERATURE - °C
HUMIDITY RATIO - GRAMS MOISTURE PER KILOGRAM DRY AIR
ENTHALPY - KJ PER KILOGRAM OF DRY AIR

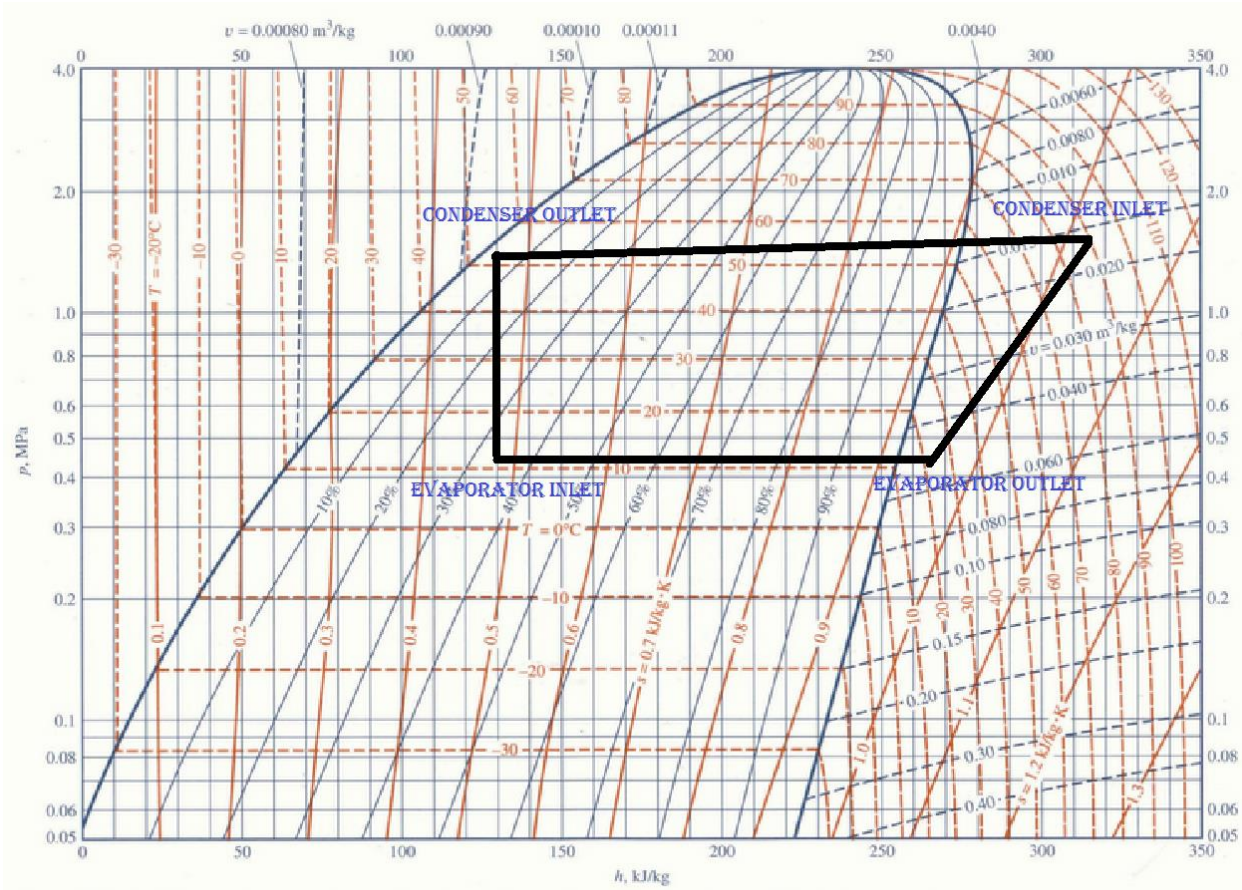


Chart A-11 R134a *ph* diagram. (*Source:* Based on *Thermodynamic Properties of HFC-134a (1,1,1, 2-tetrafluoroethane)*, DuPont Company, Wilmington, Delaware, 1993, with permission.)

<u>Case 6: Pre-Heater OFF, Re-Heater ON, Fan Supply Voltage: 150V</u>

Mass flow rate of air, $m_a = \rho_a . A_{outlet} . V_{avg} = 1.119 \times \pi \times 0.155^2 \times .25 \times 9.727 = 0.2055$ kg/s

Specific enthalpy of air before cooling, $h_{a,in} = 79.37$ kJ/kg of dry air

Specific enthalpy of air after cooling, $h_{a,out} = 62.25$ kJ/kg of dry air

Condensate removal rate, $m_w = 0.2111$ g/s (volume collected x density/time)

Specific Enthalpy of condensate = 81.74 kJ/kg

w_{in} = humidity ratio for DBT=T_3=33.6°C, WBT=T_4=25.8°C

= 0.01777

w_{out} = humidity ratio for DBT=T_5=23.5 °C, WBT=T_6=21.4 °C

= 0.01517

Theoretical condensate collected $m_w = m_a(w_{in} - w_{out}) = 0.5343$ g/s

Cooling Capacity = $Q_{air} = m_a*(h_{a,in} - h_{a,out}) - m_w*h_w = 3.501$ kW

Mass flow rate of refrigerant, $m_r = 0.01375$ kg/s

Specific Enthalpy at evaporator outlet, h1 = 264 kJ/kg

Specific Enthalpy at condenser inlet, h2 = 314.6 kJ/kg

Specific Enthalpy at condenser outlet, h4 = 126.4 kJ/kg

Work done by compressor, $W_c = m_r*(h2 – h1) = 0.6958$ kW

$COP_{air} = Q_{air} / W_c = 5.031$

$Qref = m_r (h1 – h4)$

$= 0.01375(264 – 126.4)$

$= 1.89$ kW

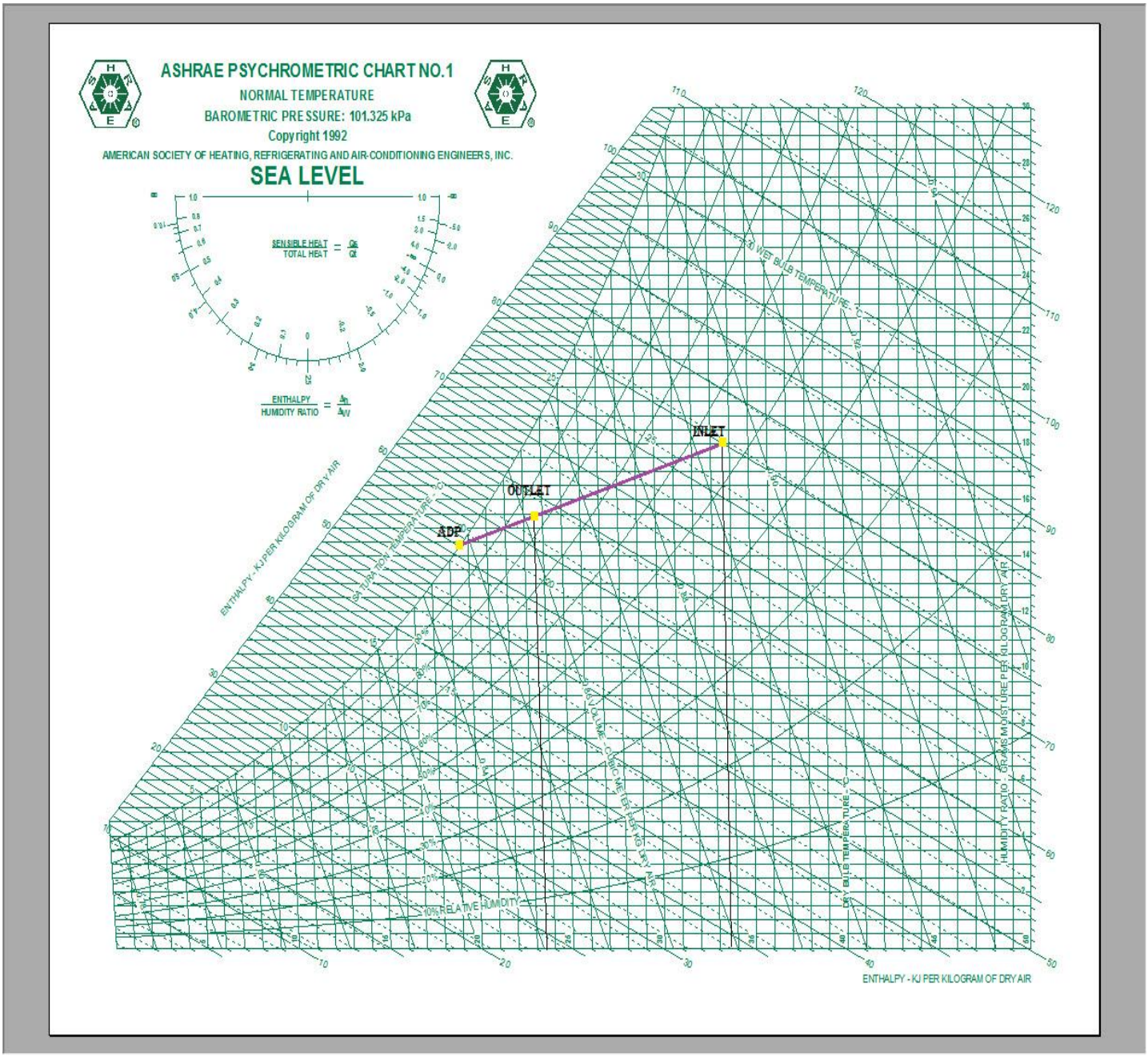
$COP|refrigerant = Qref/ W_c$

$= 2.716$

From psychrometric chart,

Apparatus Dew point Temperature, $T_{ADP} = 19.5\ ^oC$

By-pass factor of cooling coil, $X = (T_{a,o} - T_{ADP})/ (T_{a,i} - T_{ADP}) = (23.5-19.5)/(33.6-19.5) = 0.2695$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RELATIVE HUMIDITY
DRY BULB TEMPERATURE - °C
HUMIDITY RATIO - GRAMS MOISTURE PER KILOGRAM DRY AIR
ENTHALPY - KJ PER KILOGRAM OF DRY AIR

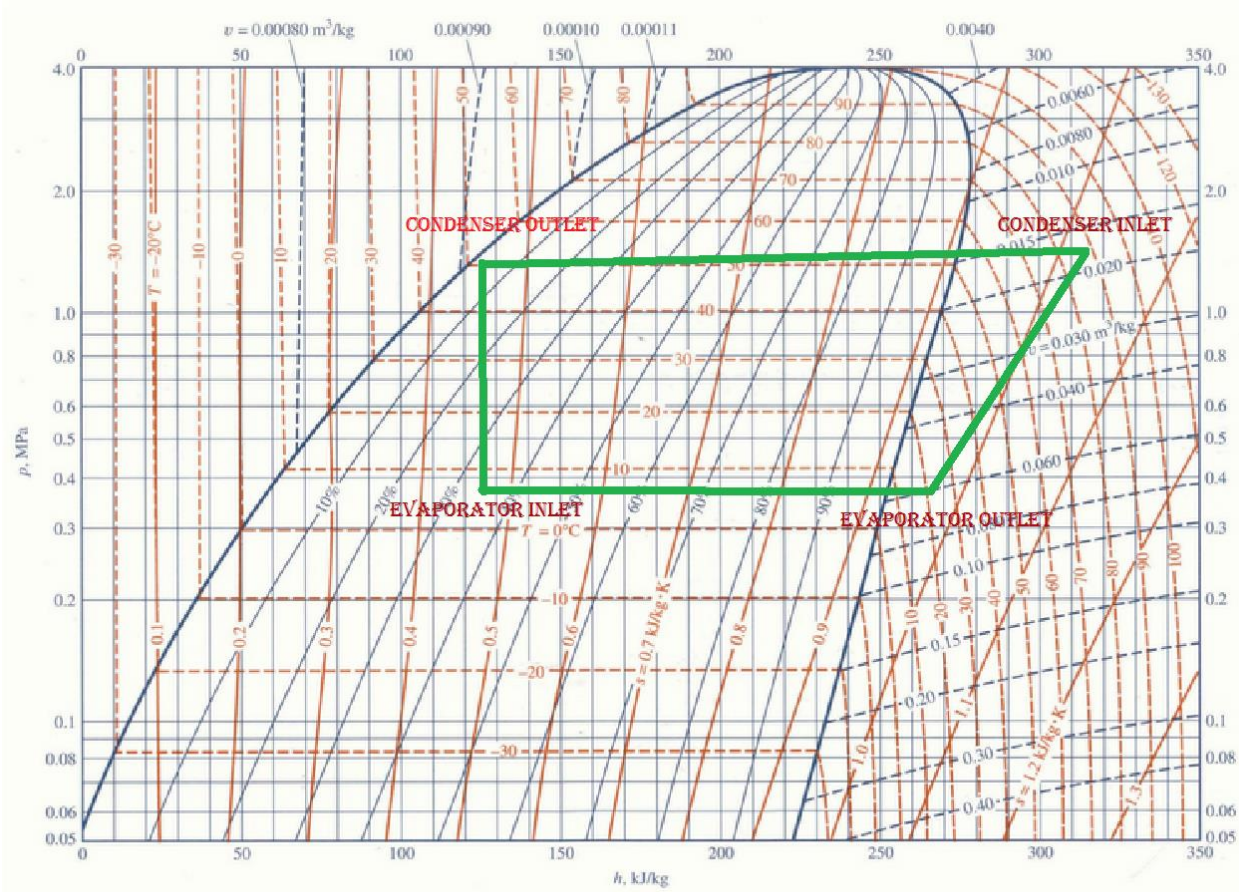


Chart A-11 R134a *ph* diagram. (*Source:* Based on *Thermodynamic Properties of HFC-134a (1,1,1, 2-tetrafluoroethane)*, DuPont Company, Wilmington, Delaware, 1993, with permission.)

Discussion Questions

Q1. How does the bypass factor vary with air velocity, answer with reason?

Ans: With increase in velocity, the bypass factor increases, and it does so because with increase in velocity the air gets less time to be in contact with the cooling oil and hence lesser cooling and dehumidification happens.

Q2. What are the effects of fin spacing and number of rows on bypass factor?

Ans: The contact area of the fins with the air increases with increase in the number of rows of fins. This decreases the bypass factor. Reduction in fin spacing leads to a decrease in bypass factor as contact area increases. But as fins are brought further closer, bypass factor increases since the flow gets restricted.

Q3. What is the relationship between by-pass factor and the performance of the system?
Ans: Bypass factor is a measure of the efficiency of the system. It indicates how well the cooling and dehumidification system is working. A higher bypass factor means a less efficient system.

Q4. How is the relative humidity controlled in A/C systems?
Ans: Refrigeration air conditioning equipment usually reduces the humidity of the air processed by the system. The relatively cold (below the dew point) evaporator coil condenses water vapour from the processed air, sending the water to a drain and removing water vapour from the cooled space and lowering the relative humidity.

Q5. Describe methods for independent control of DBT and RH.
Ans: For controlling DBT and RH independently components such as air washers (RH) and pre-heater and re-heater (DBT) can be utilized.

Q6. What are the errors in the measurements of condensate rate?
Ans: Condensate formation may not take place only at the cooling coil surface. Not all the condensate can be collected by the collecting mechanism. There may be spillage of condensate during measurements. One can also make errors while reading the condensate reading.

Q7. What precautions should be taken in measurement of WBT?
Ans: The bulb of the thermometer must be always kept moist, and air must be kept circulating around. The bulb must hence be properly isolated from the surroundings through the use of a proper wick.

Q8. How do you design the thermometer wells used for measurement of temperature?
Ans: The thermometer wells must isolate the bulb from the surroundings, and the thermocouples connected to them must be connected properly. The wells must be sealed properly, and care must be taken to mount them in an upright position.

Q9. Suggest alternate methods (other than measuring WBT) for finding state points of moist air?

Ans: Other means of finding state points are:

i. moisture-absorbing polymer films
ii. chilled gold mirrors with optical sensors etc.

Q10. Suggest methods for estimating the system performance more accurately.

Ans:

i. Collect condensates from every place where it may potentially form.
ii. Using moisture absorbing polymers gives better results than WBT.
iii. Use fans within a passage to allow more uniform temperature at a point throughout the area.

Q11. Carry out error analysis of your results.

Ans: Absolute error in T = 0.1 K

Error in TADP = 0.5 K

Error in enthalpy of input = 0.33 kJ/kg

Error in enthalpy of air after cooling = 0.26 kJ/kg

Error in enthalpy of water at evaporator temperature= 0.42 kJ/kg

Error in $Q_e = m_a(\Delta h_{a,in} - \Delta h_{a,out}) - m_w \Delta h_w$=0.101 kJ/kg

Relative error in Q_e=4.2%

Error in enthalpy of refrigerant at compressor outlet=0.11 kJ/kg

Error in enthalpy of refrigerant at compressor inlet=0.05 kJ/kg

Error in compressor work=1.6×10^{-3} kJ/kg

Relative error in compressor work=0.3%

Relative Error in COP=4.5%

Relative Error in bypass factor = (0.1+0.5)(1/(To – TADP)+1/(Ti – TADP))=1.55

Experiment No.-1

Part A *: Studies on Refrigerant compressors*

Objectives:-

Study of compressors used in refrigeration and air conditioning applications.

Theory:-

The compressor is the most critical component of a mechanical vapor compression refrigeration system. The four types of compressors commonly used in refrigeration and air conditioning applications are: Reciprocating, Vane, Centrifugal and Screw type. Recently scroll type compressors have been introduced into the market.

Based upon the operating principle, compressors may be classified as positive displacement type or dynamic type.

In positive displacement type of compressor, certain amount of vapor is trapped in an enclosure usually with the help of valves. The rise of pressure is obtained by mechanically decreasing the volume of the enclosure. This may be reciprocating type or rotary type or rotary screw type

In dynamic type (also called roto-dynamic type) of compressor, the rotating blades impart kinetic energy to the vapor. Some pressure rise may take place during the flow of vapor through the diverging passage between the rotating blades, but bulk of the pressure rise occurs in the volute casing where the vapor velocity reduces resulting in a rise in static pressure, essentially a part of kinetic energy is converted into pressure head. Centrifugal and axial flow type compressors are examples of dynamic compressors.

The reciprocating, vane, screw and scroll type of compressors are positive displacement type of compressors, while centrifugal compressor is of dynamic type.

1.4. Reciprocating Compressors:

Reciprocating compressor is the workhorse of the refrigeration and air conditioning industry. It is the most widely used compressor with cooling capacities ranging from a few Watts to

hundreds of kilowatts. Modern day reciprocating compressors are high speed (≈ 3000 to 3600 rpm), single acting, single or multi-cylinder (upto 16 cylinders).

Reciprocating compressors consist of a piston moving back and forth in a cylinder, with suction and discharge valves to achieve suction and compression of the refrigerant vapor. Its construction and working are somewhat similar to a two-stroke engine, as suction and compression of the refrigerant vapor are completed in one revolution of the crank. The suction side of the compressor is connected to the exit of the evaporator, while the discharge side of the compressor is connected to the condenser inlet. The suction (inlet) and the discharge (outlet) valves open and close due to pressure differences between the cylinder and inlet or outlet manifolds respectively. The pressure in the inlet manifold is equal to or slightly less than the evaporator pressure. Similarly the pressure in the outlet manifold is equal to or slightly greater than the condenser pressure. The purpose of the manifolds is to provide stable inlet and outlet pressures for the smooth operation of the valves and also provide a space for mounting the valves.
The valves are of reed or plate type, which are either floating or clamped. Usually, backstops are provided to limit the valve displacement and springs may be provided for smooth return after opening or closing. The piston speed is decided by valve type. Too high a speed will give excessive vapor velocities that will decrease the volumetric efficiency and the throttling loss will decrease the compression efficiency.

<u>Main parts of a reciprocating compressor :-</u>

1. Cylinder Head: This may have a built-in three-way service valve (usually for small compressors). The inlet and outlet manifolds are also built-in. Material: cast iron
2. Valve Plate: This may house both the inlet and outlet valves. Gaskets are required on both the sides of the valve plate to prevent leakage, especially in open type compressors. Material: Mild steel or cast iron
3. Inlet and Outlet valves: The valves generally made of high speed, spring steel, as they have to open and close properly for years together. Inlet and outlet valves are designed differently, as the pressures against which they operate are different.

4. Crank Case: This is the bottom part of the housing of the compressor, which contains the lubricating oil and the crankshaft. The top part of the compressor body houses the cylinders and oil return path. Material: Cast iron

5. Crank Shaft: This is generally made of forged steel with hardened bearing surfaces. Current practice is to use cast shafts of either alloy or nodular iron or high tensile gray iron. It is dynamically balanced to reduce noise and vibrations.

6. Connecting Rods: It is either eccentric strap (non-split) or scotch yoke type or blade and cap (split) type. The first two types are made of bronze or aluminum, whereas the last one may be made of forged steel or cast iron.

7. Piston: This is usually made of cast iron or aluminum. Cast iron gives a tighter fit. The clearance varies from 0.003 mm per cm diameter for small cast iron pistons to 0.03 mm per cm for large aluminum pistons. Piston rings are usually provided to decrease the leakage. Oil scrapping rings are provided for lubrication. The piston is recessed under the scrapper ring for smooth return of oil.

8. Lubrication: The lubrication may be the simple splash lubrication system or the elaborate forced feed system with filters, vents and equalizers. In splash lubrication, a dipper/scoop attached to the connecting rod throws the oil upwards. This is collected by oil scrapper ring for lubrication. This system works satisfactorily for small compressors. Large compressors require gear pumps to lift the oil and throw it at the level of scrapper ring. The oil lubricates the gudgeon pin also apart from providing some cooling. Crank case heaters are provided to prevent the condensation of refrigerant on the surface of lubricating oil.

9. Oil return Path: A separating chamber is built into the compressor on the suction manifold side with a narrow opening to the crankcase so that lubricating oil returns to the crankcase.

10. Seals: Seals are required in open type of compressors to minimize refrigerant leakage. The seals are either stationary or rotary types. The stationary type employs a metallic bellow and a hardened shaft. The rotary type is a synthetic seal tightly fitted to the shaft and prevents leakage against a carbon nose, which has the stationary cover plate.

<u>Open vs Hermetic Compressors:-</u>

a) Open Type Compressors:

A compressor whose crankshaft extends through its housing and is externally connected to a motor is called Open type compressor. A seal is required at the point where the rotating shaft comes out of the compressor housing to prevent the leakage of refrigerant. The cylinder head is another point, which is prone to leakage of refrigerant hence; gaskets are used to minimize leakage from the cylinder head. Normally, the discharge temperature of the refrigerant vapor must be below 150oC to prevent oil and refrigerant breakdown and damage to discharge valves. In case of open type compressors, fins are provided on the crankcase and cylinder head for air-cooling. Flow passages are provided in the cylinder head for water-cooling of NH3 compressors. Cylinder cooling is a must for NH3 and R-22 compressors due to the high discharge temperatures. In small compressors the pulley has braces, which act as fan, otherwise a fan is connected to the compressor motor in condensing unit.

The lubricating oil temperature should be kept below 70oC for adequate sealing and lubrication. NH3 compressors always have oil-cooling arrangement and in some cases R-22 compressors also require oil cooling.

Though open type compressors require periodic maintenance due to the continuous refrigerant leakage, they are preferred in large capacity systems due to their higher efficiency compared to hermetic type compressors. They also offer the flexibility of changing the speed of the compressor using the belt or gear drives. A belt drive also prevents the overloading of the compressor. Open type compressors are not suitable for small and critically charged systems such as domestic refrigerators and air conditioners, as it is not practical to provide continuous and periodic maintenance for these systems.

b) Hermetically Sealed Compressors:

In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor. All motors reject a part of the power supplied to it due to eddy currents and friction, that is, inefficiencies. Similarly the compressor also gets heated-up due to friction and also due to temperature rise of the vapor during compression. In Open type, both the compressor and the motor normally reject heat to the surrounding air for efficient operation. In hermetic compressors heat cannot be rejected to the surrounding air since both are enclosed in

a shell. Hence, the cold suction gas is made to flow over the motor and the compressor before entering the compressor. This keeps the motor cool. The motor winding is in direct contact with the refrigerant hence only those refrigerants, which have high dielectric strength, can be used in hermetic compressors. The cooling rate depends upon the flow rate of the refrigerant, its temperature and the thermal properties of the refrigerant. If flow rate is not sufficient and/or if the temperature is not low enough the enamel on the winding of the motor can burn out and short-circuiting may occur. Hence, hermetically sealed compressors give satisfactory and safe performance over a very narrow range of design temperature and should not be used for off-design conditions.

There is complete mixing of refrigerant and the lubricating oil in the shell hence there is a possibility of significant amount of lubricating oil being pumped along with the refrigerant depending upon the solubility of the oil in refrigerant. The refrigerant under favorable conditions gets adsorbed on the surface of lubricating oil. When the amount of adsorbed refrigerant becomes large it evaporates giving rise to froth of refrigerant and the lube oil. This frothing of the compressor reduces the volumetric efficiency of the compressor drastically.

The COP of the hermetic compressor based systems is lower than that of the open compressor based systems since a part of the refrigeration effect is lost in cooling the motor and the compressor. However, hermetic compressors are almost universally used in small systems such as domestic refrigerators, water coolers, air conditioners etc, where efficiency is not as important as customer convenience (due to absence of continuous maintenance). In addition to this, the use of hermetic compressors is ideal in systems, which use capillary tubes and are critically charged systems. Hermetic compressors are normally not serviceable. They are not very flexible as it is difficult to vary their speed to control the cooling capacity.

In some (usually larger) hermetic units, the cylinder head is usually removable so that the valves and the piston can be serviced. This type of unit is called a Semi-hermetic compressor.

Discussion Questions

Q1. What are the various performance, material and manufacturing issues involved in the design of refrigerant compressors. In what way the refrigerant compressors are different from air or other industrial gas compressors?

Ans: The various issues concerning the performance, operation, material and manufacturing of refrigerant compressors are:

i. Leakage: this problem arises in open type compressors, wherein the lubricant leaks through the crankshaft from the cylinder. This also requires regular refilling of the lubricant into the chamber. Hermetic and semi-hermetic type compressors do not face this problem.

ii. Cooling: the fast movement of the pistons generate a lot of heat, and this heat has to be effectively extracted or else the equipment could be damaged. Hence fins and lubricants, coolants are to be designed into the unit, which in itself poses a problem during manufacturing.

iii. Lubrication: this is to ensure smooth working of the compressor. This reduces friction and hence arrests overheating due to frictional heat. Lubrication could be provided through a lubricant pump or could be done through scoops which spray lubricant from an oil storage tank or a sump.

iv. Noise and vibrations: since compressors that are used in domestic purposes need to be noiseless, they need to be made as silent as possible, and reduce vibrations. Hermetic compressors are less efficient but more compact and smaller in size, hence can be used for domestic purposes. Larger compressors in industrial uses need not be overly monitored for noise and vibration.

v. Slugging: intake systems for compressors operating at high speeds need to be constructed such that they do not take in liquid, or else these can be damaged, since they are designed to handle only gases.

Q2. How do we define stroke, bore, clearance volume and displacement volume of the compressor?

Ans:

i. The distance travelled by the piston between the top dead centre and the bottom dead centre constitutes one stroke of the compressor.

ii. The inner diameter of the piston cylinder is called its bore.

iii. The volume enclosed by the piston when it us at the topmost point i.e. top dead centre is called clearance volume.

iv. The volume swept by the piston when travelling once between the top dead centre and the bottom dead centre is called as displacement volume.

Q3. Do the piston rings have a taper? While mounting the ring the sharp edge should be on the top or the bottom?
Ans: Yes, the piston rings have a taper, for the ease of mounting and maintenance. While mounting, the sharp edge should be at the bottom.

Q4. In which direction should the piston rings scrap the lubricating oil? If more than one piston ring is used what precautions should be used?
Ans: The piston rings should scrap the lubricant towards the crank-case. While mounting multiple piston rings, the one farthest from the cylinder-head should be mounted first and then the remaining in serial order.

Q5. What are the types of bearings that can be used in compressors?
Ans: For reciprocating compressors, the crankshaft is usually supported on journals, but in rotor type compressors e.g. screw and centrifugal, roller bearings can be used.

Q6. What are the methods used for capacity control of compressors?
Ans: The various methods used are :

i. Bypass control method
ii. On-off control
iii. Sensor controllers
iv. Suction internal unloading
v. Valve open hold control

Q7. What are the safety devices used with refrigerant compressors?
Ans: The various safety devices are:

i. Low pressure-high pressure power cut-off switches
ii. Low suction pressure cut-off device

iii. Low gas pressure cut-off valve
iv. Freeze protectors
v. Exhaust fan interlocks
vi. Temperature and pressure pop-off valves

Q8. What type of motors are normally used with refrigerant compressors?
Ans: Small electric motors using single phase AC can be used for small compressors used in domestic applications. Larger motors working on 3 phase AC are used in industrial applications.

Q9. Can you outline the manufacturing methods for the various parts of the compressor.
Ans:

i. Piston cylinders: can be cast and then the inner walls must be machined for a proper surface finish
ii. Pistons: can be cast and then machined on the outer surface for a smooth operation
iii. Connecting rods, crankshaft: cast or rolled, later machined for slots and grooves for mounting
iv. Casing, oil housing: forged or cast
v. Valves and valve plate: CNC machined from metal slabs for precision

Q10. Can you describe various types of seals used in compressors.
Ans: Seals are essential to prevent leakage in open-type compressors. Stationary type sealing uses a bellow and a hardened shaft. Rotary type seals use a synthetic seal tightly fitted to the shaft.

Q11. What are the advantages/disadvantages and applications of open type and hermetic compressors?
Ans:

i. Periodic maintenance is required in open type compressors due to the constant slow leakage of refrigerant, but are preferred in large capacity systems due to higher efficiency compared to hermetic compressors.

ii. Open compressors are flexible in terms of speed of the compressor due to use of belt drives and gears. Belt drives also prevent overloading.
iii. Open type compressors are unsuitable for small and critically charged systems, particularly domestic systems such as refrigerators and air conditioners, as it is impractical to provide for periodic maintenance for these systems. So hermetic-type compressors are used in these systems.
iv. There is almost no possibility of refrigerant leakage from the hermetic type compressor.
v. In Open type compressors, both compressor and motor reject heat to the surrounding air which is not possible in hermitic compressors since both motor and compressor are enclosed in a casing. The motor winding is in direct contact with the refrigerant. Hence only the refrigerants with high dielectric strength can be used in hermetic compressors.
vi. The motor insulation in the windings may react with some refrigerants in hermitic compressors, so special cooling provisions must be made for them.
vii. Hermetically sealed compressors perform well only over a very narrow range of design temperature. The COP of the hermetic compressor based systems is lower than that of the open compressor based systems since a part of the refrigeration effect is lost in cooling the motor and the compressor.
viii. They are ideal for systems which use capillary tubes and are critically charged. Hermetic compressors are normally not conveniently serviceable.

Q12. What are the advantages, disadvantages and applications of other types of positive displacement (screw, rolling piston, rotary vane and scroll types) and dynamic (centrifugal) compressors.

Ans:

i. Centrifugal compressors are preferred over the reciprocating compressors for higher efficiency over a large range of load and a large volume of the suction vapour and hence have a larger capacity to size ratio.
ii. The capacity of reciprocating machines is not affected much by an increase in the condensing temperature followed by adverse ambient conditions.
iii. Rotary compressors have high volumetric efficiencies due to negligible clearance.

iv. Screw compressor has traits of both the above. As it is a positive displacement machine, high pressure refrigerants may be used. And due to high speeds large volumes may be handled. It has no surging problems. It has small pipe dimensions and positive pressures due to the use of high pressure refrigerants. It has high compression efficiency, unloaded starting, and continuous capacity control among other things. It is hence suitable for large capacity installations.

v. Scroll type compressors are introduced only recently, and also share the essential features of screw type compressors. These are more efficient than reciprocating ones as they do not have a dynamic discharge valve that introduces additional throttling losses. These also have fewer moving parts and hence is more reliable.

Part B *: Studies on indoor thermal comfort*

Objectives:-

To measure different environmental parameters of human thermal comfort and using them, calculate indices of heat stress and thermal discomfort.

Theory:-

Variables that affect heat dissipation from the body (thus also thermal comfort) can be grouped under environmental and personal factors. Under environmental factors we have air temperature, air movement, humidity, and mean radiant temperature. Under personal factors we have metabolic activity of the person and insulation of the clothing worn.

Air temperature (ta) is the most obvious environmental factor, measured by the dry bulb temperature (DBT), and determines the convective heat dissipation, together with any air movement. In this manual, DBT and ta will be used interchangeably to denote indoor air temperature.

Air movement, (vr) measured in m/s with an anemometer, affects both evaporative and convective heat transfer from the skin.

Humidity (RH) of the air also affects evaporation rate as well as non-thermal factors of comfort like feelings of dryness or stuffiness. Humidity can be found from a psychrometric chart, after one measures the DBT, wet bulb temperature (WBT), and barometric pressure.

Mean radiant temperature (tr) determines the radiant heat exchange between an occupant and her/his surroundings. The mean radiant temperature is defined as the uniform temperature of an imaginary enclosure in which the radiant heat transfer from the human body is equal to the radiant heat transfer in the actual non-uniform enclosure.

Clothing is one of the dominant factors affecting heat dissipation from our bodies. The variety of clothing that we wear is a major contributing factor behind how humans have flourished in diverse climatic conditions. For the purposes of thermal comfort studies a unit has been devised, named the clo (Icl). This corresponds to an insulating cover over the whole body of a resistance of 0.155 m2K/W. Typical clothing patterns during summer have a clo value of 0.5 while the

value is 1.0 for winter clothing ensembles. For seated occupants, the thermal insulation of chairs also needs to be considered and added to the clothing insulation.

The body has several thermal adjustment mechanisms to survive in drastic temperature environments. To warm conditions(or increased metabolic rates) the body responds by vasodilation: subcutaneous blood vessels expand and increase the skin blood supply, thus the skin temperature, which in turn increases heat dissipation. If this cannot restore thermal equilibrium, the sweat glands are activated to enhance evaporative heat transfer. To cold conditions the response is firstly vasoconstriction: reduced circulation to the skin, lowering of skin temperature, thus reduction of heat dissipation rate. If this is insufficient, shivering takes place, involuntarily forcing the muscles to work and increasing the metabolic heat production. In extreme conditions, shivering can increase the metabolic rate to ten times that of basal values.

Observations:-

Outdoors WBT: 26.67°C

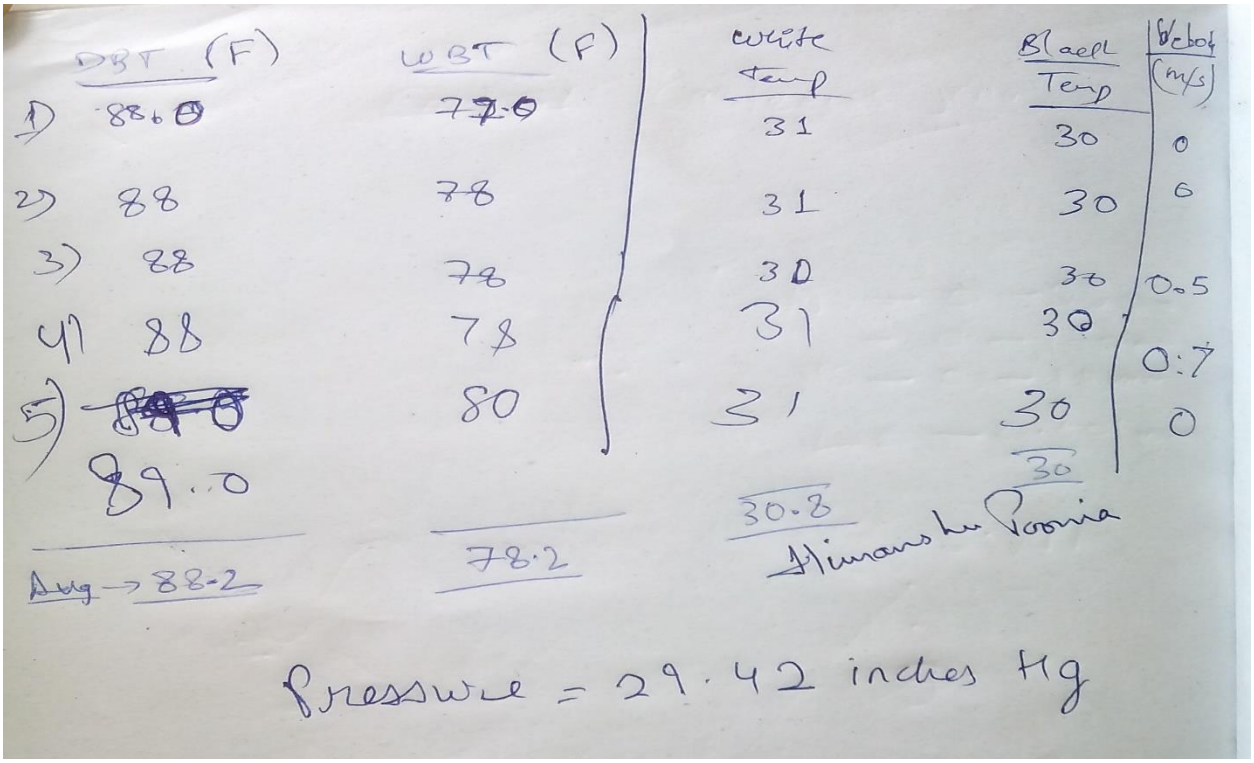
Outdoors DBT: 31.67°C

Tg (black and white): 31 °C

Barometric Pressure: 99.628 kPa

Wind Velocity: 0 m/s

location	DBT (°C)	WBT (°C)	Globe Black (°C)	Globe White (°C)	Wind Velocity(m/s)
1	31.11	25	31	30	0
2	31.11	25.56	31	30	0
3	31.11	25.56	30	30	0.5
4	31.11	25.56	31	30	0.7



	DBT (F)	WBT (F)	White temp	Black Temp	Velocity (m/s)
1)	88.0	77.0	31	30	0
2)	88	78	31	30	0
3)	88	78	30	30	0.5
4)	88	78	31	30	0.7
5)	89.0	80	31	30	0
			30.8	30	
Avg →	88.2	78.2			

Himanshu Poonia

Pressure = 29.42 inches Hg

Calculations:-

Average indoor DBT = 31.11 °C

Average indoor WBT = 25.416°C

Average indoor wind velocity = 0.3 m/s

For indoors:

WBGT = 0.7xWBT + 0.3xTg = 27.09°C

Ta = 31.11°C

Tg = 31°C

Emissivity ε = 0.95

V = 0.3 m/s

WBT = 25.416°C

D = 67 mm

Mean temperature gradient:

$Tr = [(Tg + 273)^4 + 1.1 \times 10^8 \times (Vr)^{0.6} / (0.95 \times D^{0.4}) \times (Tg - Ta)]^{0.25} - 273 = 30.838°C$

For outdoors:

WBT = 26.67°C, DBT = 31.67°C

WBGT = 0.7xWBT + 0.2xTg + 0.1xDBT = 28.036°C

Operating temperature (To) = (hr x Tr + hc x Ta)/(hr+hc)

= 30.96°C

Where hr = radiative heat transfer coefficient = 4.7 W/m^2K

And hc = convective heat transfer coefficient = 4.03 W/m^2K

Tropical summer index (TSI) = $0.308xWBT + 0.745xTg – 2.06(Vr + 0.841)^{0.5}$

= 28.72 °C

So it is comfortable.

PMV = (0.352exp(-0.042M) + 0.032)[M - 0.35[43 – 0.061M - pa]- 0.42[M - 50] - 0.0023M(44 - pa) – 0.0014M(34 - ta) - $3.4 \times 10^{-8}[(tcl + 273)^4 - (tr + 273)^4]$ - fclhc(tcl - ta)]

= 2.614

where tcl =35.7 – 0.032M – 0.18Icl[M - 0.35[43 – 0.061M - pa] – 0.42[M - 50] - 0.0023M(44 - pa) – 0.0014M(34 - ta)]

= 30.79°C

hc = $10.4*(0.3)^{0.5}$

= 5.696

M = 60 kcal/hr-m^2, pa=12.9 mm Hg, lcl = 1 clo

PPD = 100 - 95exp[-(-0.03353 x PMV^4 + 0.2179 x PMV^2)]

= 31.19%

HSI = M+22(tr-95)+2x$Vr^{0.5}$(DBT-95)/10.3x$Vr^{0.4}$(42-pa)

= -0.166

So almost no strain.

Discussion Questions

Q1. How is thermal comfort defined?

Ans: It is the condition of mind which expresses satisfaction with thermal environment.

Q2. What are the parameters affecting thermal comfort?

Ans: Air temperature, air humidity, mean radiant temperature, humidity, metabolic activities, clothing etc.

Q3. How does the human body regulate its temperature?
Ans: The body responds to warm conditions by vasodilation, subcutaneous blood vessels expand and increase skin blood supply. To cold conditions, response is through vasocontraction, lowering of skin temperature.

Q4. What are the modes of heat transfer from human body?
Ans: Different modes of heat transfer from human body are:-

i. Convection
ii. Radiation
iii. Conduction
iv. Evaporation
v. Expiration.

Q5. What scale is used for thermal sensation?
Ans: ASHRAE seven point scale used.

Hot	Warm	Slightly Warm	Neutral	Slightly Cool	Cool	Cold
3	2	1	0	-1	-2	-3

Q6. What are the different indices used for measuring thermal comfort and heat stress?
Ans: Wet bulb globe temperature, tropical summer index, heat stress index, operative temperature, PMV, PPD.

Q7. How comfort surveys are conducted to evaluate the comfort standards of a building?
Ans: A typical comfort survey for a building is centred around its occupied zone. The survey would involve measurement of the four environmental parameters within one meter distance from the occupants at head height. The occupants are given the subjective survey questionnaires to fill up.

The role of occupants is taken by the students in other three experiments. Readings of WBT, DBT, Vr, Tg are to be measured in the near vicinity of the occupants. Enough time should be given for readings to stabilize. After measurements, the indices need to be calculated.