Refrigeration & Air-Conditioning Laboratory

REPORT

Prepared by:

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10ME10069

Experiment No. 1(a)

Objectives: Study of compressors used in refrigeration and air conditioning applications

Experimental Procedure: Removed all the parts of the two open-type reciprocating compressors and one hermetically sealed reciprocating compressor. Identified all the parts and understood the working principle. Made suitable free hand isometric drawings, and assembled the compressors back.

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 (up to 16 cylinders).

Components:

1. <u>Cylinder Head:</u> 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. <u>Valve Plate:</u> 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. <u>Inlet and Outlet valves:</u> 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. <u>Crank Case:</u> 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. <u>Crank Shaft:</u> 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. <u>Connecting Rods:</u> 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. <u>Piston:</u> 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. <u>Lubrication:</u> 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.

Discussions:

Q 1. How do we define stroke, bore, clearance volume and displacement volume of the compressor?

Ans. The distance traveled by the piston from bottom dead center (BDC) to top dead center (TDC) within the cylinder is known as the stroke of the compressor. The bore is the diameter of the compressor cylinder. The volume between the piston and the cylinder head when it is at its minimum, i.e. when the piston is at the position of Inner Dead Centre is called the clearance volume. The volume of the compressor may be defined as the volume swept by the piston in its stroke between the bottom and the top dead centers respectively.

Q 2. 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. While mounting, the sharp edge should be on the bottom.

Q 3. 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 lubricating oil should be scrapped towards the crank-case by the piston rings. Care should be taken while mounting of multiple piston rings. The piston-ring which is to be farthest from the cylinder-head should be mounted first and so on.

Q 4. What are the types of bearings that can be used in compressors?

Ans. Bearings that are used are deep groove ball bearings and tapered roller bearings.

Q 5. What are the methods used for capacity control of compressors?

Ans. “On and Off Control”, “Holding the Valves Open” ,”Hot Gas Bypass” and “Using Multiple Units” are some of the control methods used.

Q 6. What are the safety devices used with refrigerant compressors?

Ans. Some of the safety devices used are as follows:

1. Low suction pressure cutoff devices
2. Freeze protectors
3. Proofing switches
4. Exhaust fan interlocks
5. Temperature and pressure pop-off valves
6. Low gas pressure cutoff valves

Q 7. What types of motors are normally used with refrigerant compressors?

Ans. 1. Small electric motors suitable which are suitable for domestic electrical supplies using single phase alternating current.

2. Larger electric motors which can be used where an industrial electrical three phase alternating current supply is available.

Q 8. Can you describe various types of seals used in compressors?

Ans. Open type compressors require seals to minimize leakage. These may be stationary or rotary types. The stationary type is basically a metallic bellow and a hardened shaft. The rotary type employs a synthetic seal tightly fitted to the shaft which prevents leakage using a carbon nose.

Q 9. What are the advantages/disadvantages and applications of open type and hermetic compressors?

Ans. Periodic maintenance is required in open type compressors due to the constant slow leakage of refrigerant. These are still preferred in large capacity systems due to higher efficiency compared to hermetic compressors. Open compressors are flexible in terms of speed of the compressor due to use of belt drives and gears. Belt drives also prevent overloading.

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. There is almost nil possibility of refrigerant leakage from the hermetic type compressor.

In Open type compressors, both compressor and motor normally reject heat to the surrounding air. This is not possible in hermitic compressors since both motor and compressor are enclosed in a shell. 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 motor insulation in the windings may react with some refrigerants in hermitic compressors, so special cooling provisions must be made for them.

Also, hermetically sealed compressors give satisfactory and safe performance 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. However, they are ideal for systems which use capillary tubes and are critically charged. Hermetic compressors are normally not serviceable.

Q 10. 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. Centrifugal compressor are preferred over the reciprocating compressors for higher efficiency over a large range of load and a large volume of the suction vapor and hence have a larger capacity to size ratio. However the capacity of Reciprocating machines is not affected much by an increase in the condensing temperature followed by adverse ambient conditions. Rotary compressors have high volumetric efficiencies due to negligible clearance. Screw compressor combines many advantageous features of both centrifugal and reciprocating systems. 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, continuous capacity control, unloaded starting and no balancing problems. It is hence suitable for large capacity installations.

Experiment No. 1(b)

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

Theory: *Air temperature (t_a)* is the most obvious environmental factor, measured by the dry bulb temperature (DBT), and determines the convective heat dissipation, together with any air movement.

Air movement, (v_r) 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 (t_r) determines the radiant heat exchange between an occupant and her/his surroundings.

Relevant Formulae: Relation between t_g and t_r is given by:

$$t_r = [(t_g + 273)^4 + \frac{1.1 \times 10^8 v_r^{0.6}}{\varepsilon D^{0.4}} \times (t_g - t_a)]^{1/4} - 273$$

WBGT indicates the combined effect of air temperature, low temperature radiant heat, solar radiation and air movement.

For Inside: $WBGT = 0.7 \cdot WBT + 0.3 \cdot t_g$

For Outside: $WBGT = 0.7 \cdot WBT + 0.2 \cdot t_g + 0.1 \cdot DBT$

Tropical Summer Index (TSI) is given by:

$$TSI = 0.308 \cdot WBT + 0.745 \cdot t_g - 2.06\sqrt{v_r + 0.841}$$

Heat Stress Index (HIS) is given by:

$$HSI = \frac{M + 22(t_r - 95) + 2 \cdot v_r^{0.5}(DBT - 95)}{10.3 \cdot v^{0.4}(42 - p_a)}$$

Predicted Mean Vote (PMV) is evaluated as:

$$\begin{aligned} PMV =& (0.352e^{-0.042M} + 0.032)[M - \\ & 0.35[43 - 0.061M - p_a] - \\ & 0.42[M - 50] - \\ & 0.0023M(44 - p_a) - 0.0014M(34 - t_a) - \\ & 3.4 \cdot 10^{-8}[(t_{cl} + 273)^4 - (t_r + 273)^4] - f_{cl}h_c(t_{cl} - t_a)] \end{aligned}$$

Clothing surface temperature is calculated as:

$$\begin{aligned} t_{cl} =& 35.7 - 0.032M - 0.18I_{cl}[M - \\ & 0.35[43 - 0.061M - p_a] - 0.42[M - 50] - \\ & 0.0023M(44 - p_a) - 0.0014M(34 - t_a)] \qquad (^\circ C) \end{aligned}$$

Predicted Percent Dissatisfied (PPD) is calculated as:

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

Observations:

Thermal Comfort (Expt. 1 B)

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

Name of student: AURKO CHATTERJEE **Roll No.:** 10HE10069

Observations:

Barometric Pressure (kPa): 100.375

Outdoors DBT (°C): 22.9
Outdoors WBT (°C): 18.6

Subject Group	t_a (°C)	WBT (°C)	p_a (kPa)	v_r (m/s)	t_g (°C)
1.	25	19.3	1.8667	0	25
2.	25.6	19.2	1.8097	0	25
3.	25.6	19.4	1.8048	0	25

Signature of instructor
Date: 08.02.2013

Calculations:

Subject Group	t_a (^oC)	WBT (^oC)	P_a (kPa)	v_r (m/s)	t_g (^oC)
Group 1	25	19.3	1.8667	0	25
Group 2	25.56	19.2	1.8097	0	25
Group 3	25.6	19.4	1.848	0	25

Group 1:

Mean Radiant Temperature (t_g) : 25 ^oC
WBGT : 21.01 ^oC

Thermal Stress Index : 22.68
Thermal Sensation : Slightly Cool

Value of M : 22 BTU/hr-ft^2
Vapor Pressure (p_a) : 14.0014 mmHg

Heat Stress Index (HSI) : -1.38
Thermal Sensation : Mild Cold Strain

Value of t_{cl} : 29.99 ^oC
Value of f_{cl} : 1.15
Value of h_c : 3.065 kW/m^2-K

Predicted Mean Vote (PMV) : 0.361
Predicted Percent Dissatisfied : 7.61

Group 2:

Mean Radiant Temperature (t_g) : 25 ^oC
WBGT : 20.94 ^oC

Thermal Stress Index : 22.65
Thermal Sensation : Slightly Cool

Value of M : 22 BTU/hr-ft^2
Vapor Pressure (p_a) : 13.5739 mmHg

Heat Stress Index (HSI) : -1.36
Thermal Sensation : Mild Cold Strain

Value of t_{cl} : 30.01 ^oC
Value of f_{cl} : 1.15
Value of h_c : 2.978 kW/m^2-K

Predicted Mean Vote (PMV) : 0.490
Predicted Percent Dissatisfied : 9.68

Group 3:

Mean Radiant Temperature (t_g)	:	25 °C
WBGT :		21.08 °C
Thermal Stress Index	**:**	**22.71**
Thermal Sensation	**:**	**Slightly Cool**
Value of M	:	22 BTU/hr-ft^2
Vapor Pressure (p_a)	:	13.8611 mmHg
Heat Stress Index (HSI)	**:**	**-1.37**
Thermal Sensation	**:**	**Mild Cold Strain**
Value of t_{cl}	:	29.99 °C
Value of f_{cl}	:	1.15
Value of h_c	:	2.969 kW/m^2-K
Predicted Mean Vote (PMV)	**:**	**0.516**
Predicted Percent Dissatisfied	**:**	**10.41**

Discussions:

Q 1. How is thermal comfort defined?

Ans. American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) defines thermal comfort as "that condition of mind which expresses satisfaction with the thermal environment", keeping in mind that comfort is a subjective sensation.

Q 2. What are the parameters affecting thermal comfort?

Ans. Parameters 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.

Q 3. How does the human body regulate its temperature?

Ans. 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.

Q 4. What are the modes of heat transfer from human body?

Ans. Human body transfers hear in three ways, namely convection, radiation and evaporation

Q 5. What scale is used for thermal sensation?

Ans. To determine thermal sensation of occupants, the most popular means is a subjective survey questionnaire using the ASHRAE seven point scale. Anyone at the neutral point is deemed to be comfortable, and the comfort band is generally assumed to be the -1 to +1 range.

Table 2: ASHRAE thermal sensation scale

hot	warm	slightly warm	neutral	slightly cool	cool	cold
3	2	1	0	-1	-2	-3

Q 6. What are the different indices used for measuring thermal comfort and heat stress?

Ans. Various indices that are used to measure thermal comfort and heat stress are Wet Bulb Globe Temperature, Operative Temperature, Tropical Summer Index (TSI), Heat Stress Index (HSI), Predicted Mean Vote (PMV) and Predicted Percent Dissatisfied (PPD).

Q 7. How comfort surveys are conducted to evaluate the comfort standards of a building?

Ans. Normally, some equivalent of the ASHRAE thermal sensation scale is used to collect sensation votes. These subjective data are then statistically analyzed to find a regression fit between comfort vote and indoor temperature, of the form $CV=\alpha T_a - \beta$. Here CV is the comfort vote, T_a is the indoor air temperature, α is the slope of the line (also referred to as sensitivity of the population), and β is the y-axis intercept. Using these relations, the indoor air temperature where CV is zero is determined. This temperature would be the neutral temperature on the thermal sensation scale and neutrality is assumed to represent comfort conditions.

Experiment No. 2

Objective: To carry-out steady-state measurements on the test-rig of a Vapor Compression Refrigeration System in order to determine:

- Carnot COP, Cycle COP and Actual COP of the refrigeration system
- Overall heat transfer coefficients for the evaporator and the condenser
- Overall volumetric efficiency of the compressor

Components in our Setup:

I. Hermetic compressor
II. Air cooled condenser
III. Capillary tube / Thermostatic expansion valve
IV. Water cooled evaporator
V. Shut-off valves
VI. Sight glass
VII. Drier
VIII. Filter
IX. Solenoid valve
X. Pressure gauges
XI. Thermometric wells
XII. Rotameter

Refrigerant used: R-12

Theory: 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. An electrical heater controlled by a thermostat provides the required cooling load to the evaporator.

A hermetic compressor with a piston displacement of 2.15 m3/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 compressor. 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. Provision is also there to use a thermostatic expansion valve with solenoid valve instead of the capillary tube.

Procedure: Once steady-state is reached, we measured the pressure gage readings, temperature readings of all the thermometers, heater and compressor energy meter (or voltmeter and ammeter) readings, and temperature of water around the evaporator, velocity of air over the condenser at 5 locations, inlet and outlet temperature of air at condenser, barometric pressure. Then we opened the valve in the rotameter by-pass line and took the rotameter reading. Then the system was switched-off and measurements of condenser and evaporator dimensions were taken. For the condenser, we measured the width and height of condenser, number of condenser tubes, number of fins, thickness, width and height of fins and number of tubes passing through the fins. For the evaporator, we measured the copper tube diameter, coil diameter and number of coils immersed in water.

Observations:

Test rig of a vapor compression refrigeration system

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

Name of Student:- AURKO CHATTERJEE

Condenser pressure, p_c [bar]	15.15
Evaporator pressure, p_e [bar]	4.15
Refrigerant temperature at condenser inlet, t_{ci} [° C]	65.50
Refrigerant temperature at condenser outlet, t_{co} [° C]	55.00
Refrigerant temperature at evaporator inlet, t_{ei} [° C]	11.00
Refrigerant temperature at evaporator outlet, t_{eo} [° C]	11.00
Compressor energy meter reading, [W]	491.80
Heater energy meter reading, $Q_{e.}$ [W]	1016.95
Number of fins, N_f	127
Fin thickness, e [m]	4×10^{-4}
Fin width, b [m]	0.05
Fin height, h [m]	0.30
Condenser width, L [m]	0.2772
Number of tubes in condenser, N_{tc}	24
Diameter of condenser tubes, d_c [m]	0.0096
Average air velocity through condenser, V [m/s]	1.04
Air temperature at condenser inlet, $T_{air,in}$ [° C]	28.00
Air temperature at condenser outlet, $T_{air,out}$ [° C]	45.00
Temperature of water in evaporator, T_{water} [° C]	19.50
Evaporator coil diameter, D_e [m]	0.208
Evaporator tube diameter, d_e [m]	0.0096
Number of tubes in the coil, N_{te}	19
Room temperature [° C]	28.00
Room pressure [Pa]	100104.16
Displacement of compressor, PD [m^3/h]	2.15

Rotameter Reading (LPH) 40.00

Sign of supervisor

Date: 01.03.2013

$COP_{carnot} = 5.61$

$COP_{cycle} = 4.71$

$COP_{actual} = 2.07$

01/03/13

Calculations:

1. Carnot COP

$$COP_{Carnot} = \frac{T_{evap}}{(T_{cond} - T_{evap})}$$

$T_{evap} = 11°C$

$T_{cond} = 59.78°C$

$\therefore COP_{Carnot} = 5.91$

2. Cycle COP

$$COP_{Cycle} = \frac{(h_1 - h_3)}{(h_2 - h_1)}$$

Enthalpy at state 1 (exit of evaporator), $h_1 = 357.97 \text{ kJ/kg}$
Enthalpy at state 2 (exit of compressor), $h_2 = 379.88 \text{ kJ/kg}$
Enthalpy at state 3 (exit of condenser), $h_3 = 255.05 \text{ kJ/kg}$
Enthalpy at state 4 (inlet of evaporator), $h_4 = 255.05 \text{ kJ/kg}$

$\therefore COP_{Cycle} = 4.69$

3. Actual COP

$$COP_{Actual} = \frac{\text{Actual refrigeration effect}}{\text{Actual energy input to compressor}}$$

Compressor energy input = 491.8 W
Heater energy meter reading = 1016.45 W

$\therefore COP_{Actual} = 2.07$

4. Overall heat transfer coefficient for evaporator

Diameter of coil (D_e) = 0.208
Diameter of tube (d_e) = 0.0096
Number of tubes in the coil (*N)* = 1
$A_{coil} = \pi^2 D_e d_e N$ => $A_{coil} = 0.374\ m^2$

$T_{water} = 19.5°C$
$T_{evap} = 11°C$

So, $U_{evap} = 319.7\ W/m^2K$

5. Overall heat transfer coefficient for condenser

$$U_{cond} = \frac{mC_{P,air}(T_{air,out} - T_{air,in})}{A_{cond} LMTD_{cond}}$$

$T_{air,in} = 28°C$
$T_{air,out} = 45°C$
$T_{cond} = 59.78°C$

Number of fins (N_f) = 127
Fin thickness (*e*) = 0.004
Fin height (*h*) = 0.3
Fin width (*b*) = 0.05
Condenser tube length (*L*) = 0.2772
Number of tubes in condenser (N_{tc}) = 24
Diameter of condenser tubes (d_c) = 0.0096

$A_{cond} = \pi d_c L N_{tc} - \pi d_c e N_f N_{tc} + A_{fin}$
$A_{fin} = 2N_f(hb - N_{tc}\pi d_c^2/4)$
$A_{fin} = 3.36\ m^2$
$A_{cond} = 3.52\ m^2$

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

So, $LMTD_{cond} = 22.2°C$

$m_{air} = \rho A_{air} v$
$A_{air} = Lh - d_c L N_{tc}/2 - N_f eh$
$A_{air} = 0.036$
$\rho_{air} = 1.14\ kg/m^3$
$v = 1.04\ \mathrm{m/s}$

$\therefore m_{air} = 0.0426\ \mathrm{kg/s}$
$C_{P,air} = 1.005\ \mathrm{kJ/kg\ K}$

$\therefore U_{cond} = 9.31\ W/m^2K$

6. Overall volumetric efficiency of compressor

$$\eta_{ov} = \frac{\text{Actual refrigerant mass flow rate}}{\text{Maximum possible mass flow rate}}$$

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

$Q_{liq} = 40\ \mathrm{l/h}$
$v_{liq} = 0.000838$
$v_{suction} = 0.0424$
$\eta_{ov} = 94.1\%$

Error Analysis:

1. Carnot COP

Absolute error in T_{evap} = 1 K
Absolute error in T_{cond} = 1K
Relative error in $T_{evap} = 0.0035$
Relative error in $T_{cond} - T_{evap} = 0.029$
Relative error in $COP_{Carnot} = 0.0292$

2. Cycle COP

Relative error in $(h_1 - h_3) = 0.0123$
Relative error in $(h_2 - h_1) = 0.048$
Relative error in $COP_{Cycle} = 0.0499$

Discussions:

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

Ans. The following errors may creep into refrigerant temperature measurement:

- Faulty calibration of thermometer.
- Due to heat losses in the thermometer contacts
- Ineffective wetting of thermometer bulb by the connecting liquid.

These may be remedied by using solid state thermoelectric sensors.

Q2. What are the errors in energy meter reading?

Ans. The following are some sources of error in an energy meter reading:

- Human error in measurements
- Inaccurate Calibration
- Unsteady or unknown power factor for AC source.

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

Ans. The assumptions were:

- Carnot COP: A saturation cycle has been assumed between the measured temperatures.
- Actual COP: We directly calculated the values from the energy meter readings. These may have errors and also do not account for electrical losses.

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 flow rate is determined from the rotameter and the thermodynamic state functions and variables of the refrigerant at inlet and outlet of the condenser and evaporator are calculated from the property tables knowing the pressure and temperature etc. The same is used to calculate the cooling capacity and condenser heat rejection rate.

Q5. Why is thermostat used in the water tank?

Ans. A thermostat is used to keep the water temperature constant in the tank. This is useful to maintain a steady rate of heat rejection by the water to the refrigerant, and find the cooling capacity from the power consumption of the water heater.

Q6. What are the safety devices used in the set-up?

Ans. The following are the safety devices used in the system:

- Filter, drier absorb moisture and dust.
- High and Low pressure cut off systems for the compressor.

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

Ans. COP at different temperatures of Evaporator and Condenser can be found out by carrying out the experiment at various ambient temperatures (Room Temp control) and loads (varying heater output).

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

Ans. It has been assumed that the rate of change for the temperature of both fluids is proportional to the temperature difference; this assumption is valid for fluids with a constant specific heat. However, if the specific heat changes, the LMTD approach will no longer be accurate. It has also been assumed that the heat transfer coefficient (U) is constant, and not a function of temperature. Also, The LMTD is a steady-state concept, and cannot be used in dynamic analyses.

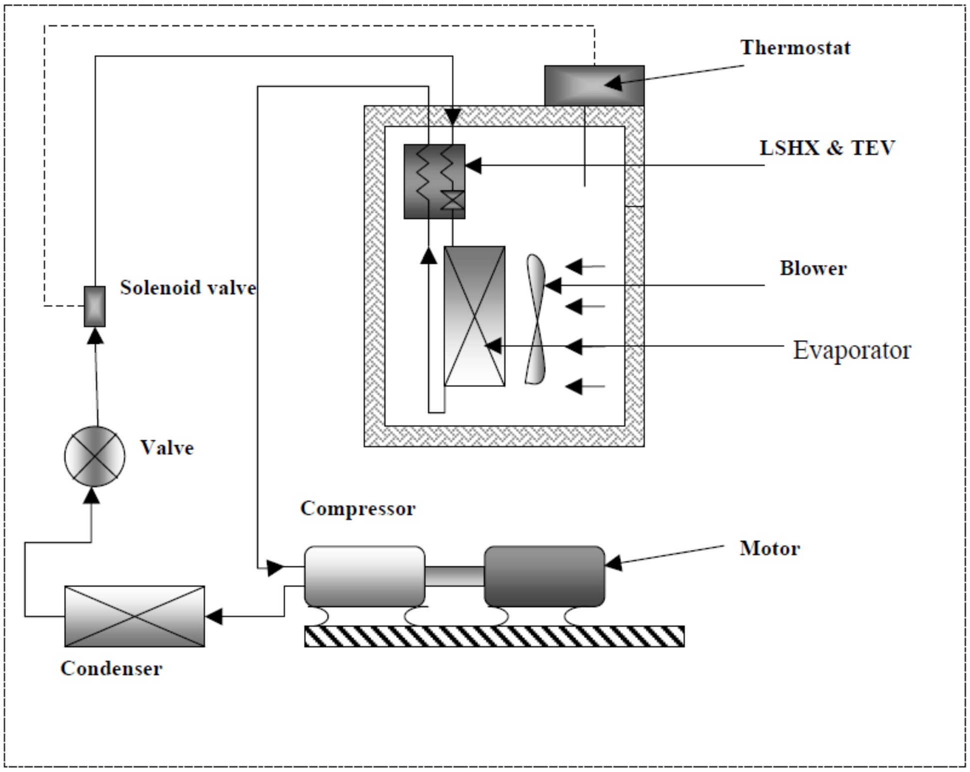
Q9. What is the significance of volumetric efficiency of compressor?

Ans. Volumetric efficiency determines the critical pressure ratio for a particular compressor. It gives us a limit for the condenser pressure, for a given evaporator pressure.

Experiment No. 3

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

Components of a cold storage:



I. 3 TR Open-type Compressor
II. Condensing unit
III. Air cooled condenser
IV. Hand valve
V. Dryer
VI. Solenoid valve
VII. Electric bulb
VIII. Sight Glass
IX. Sub-cooling Heat Exchanger
X. Thermostatic Expansion Valve
XI. Direct Expansion Coil
XII. Thermostat
XIII. HP/LP Cutout

Theory: 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 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. 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 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.

Procedure:

1. Noted down the room temperature and temperature set on the thermostat.
2. Started the system and by closing and opening the hand valve, noted down the cut-out and cut-in pressures.
3. Recorded the time, temperature, pressure, and solenoid valve condition and compressor condition at solenoid closing, compressor closing, solenoid opening and compressor re-starting. Tabulated the observations as shown below:

Observations:

$P_{atm} = 29.54$ inches Hg

$T_{room} = 29^{\circ}C$

Cold Storage

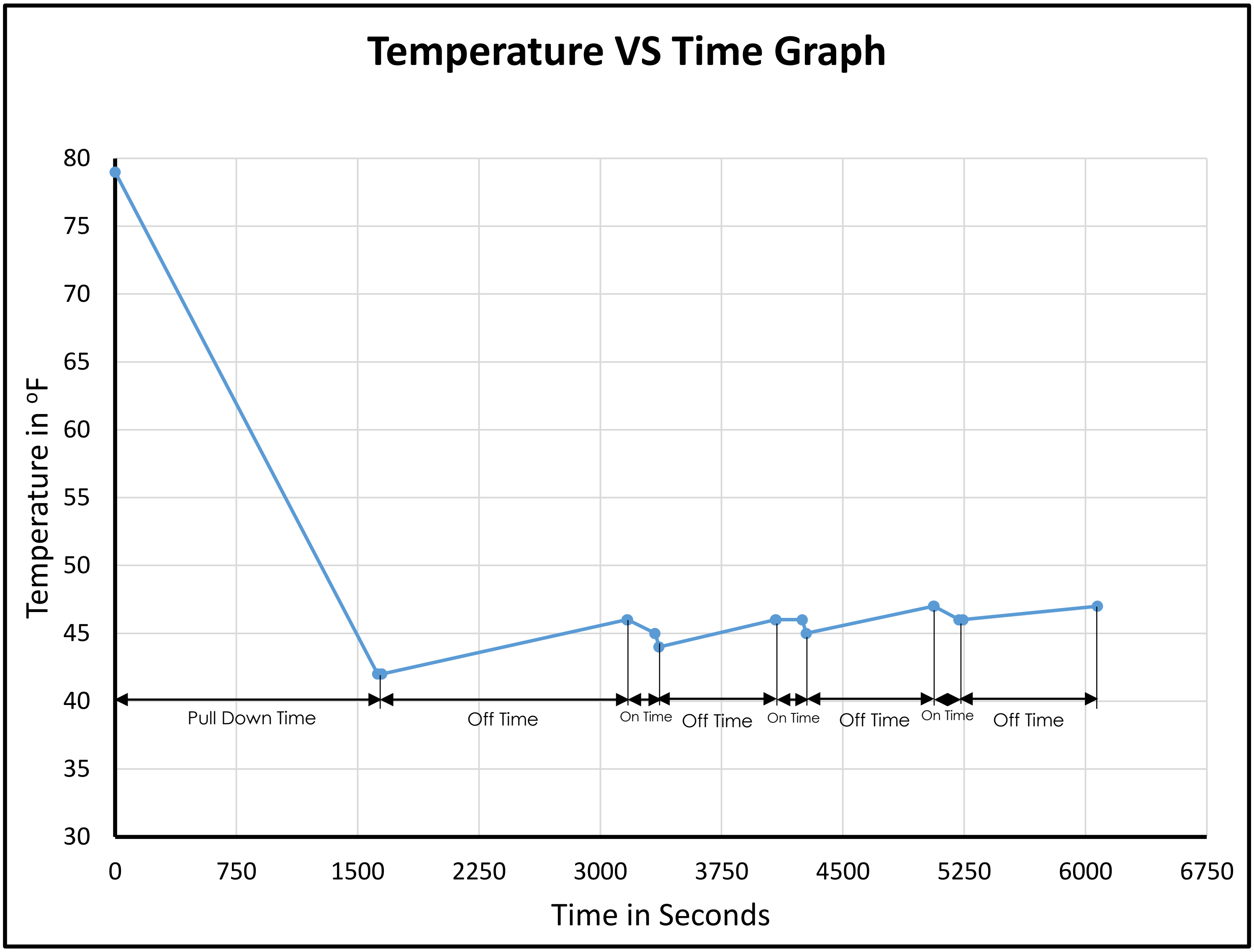
Objectives: - To study the pull down and cycling characteristics of cold storage

Name of Student:- AURKO CHATTERJEE

Clock Time (hh:mm:sec)	Temperature (°F)	Evaporator Pressure	Condenser Pressure	Solenoid Condition	Compressor condition
0	79	10	140	Open	On
27:04	42	8	152	Closed	On
27:29	42	-10 inches Hg	140	Closed	Off
52:46	46	20	128	Open	Off
52:49	46	14	135	Open	On
55:39	45	10	152	Closed	On
56:04	44	-10 inches Hg	150	Closed	Off
1:08:04	46	20	128	Open	Off
1:08:07	46	14	140	Open	On
1:10:49	46	10	150	Closed	On
1:11:14	45	-4 inches Hg	148	Closed	Off
1:24:21	47	20	128	Open	Off
1:24:24	47	13	134	Open	On
1:26:59	46	10	150	Closed	On
1:27:23	46	-10 inches Hg	148	Closed	Off
1:41:14	47	20	126	Open	Off
				Open	On
				Closed	On
				Closed	Off

Signature of Supervisor

Date: 8/3/2013



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

CYCLE 1:
Pull Down Time = **27 Minutes 29 Seconds**
Off Time = **25 Minutes 20 Seconds**

CYCLE 2:
On Time = **3 Minutes 15 Seconds**
Off Time = **12 Minutes 3 Seconds**

CYCLE 3:
On Time = **3 Minutes 7 Seconds**
Off Time = **13 Minutes 10 Seconds**

CYCLE 4:
On Time = **2 Minutes 59 Seconds**
Off Time = **13 Minutes 51 Seconds**

Discussions:

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

Ans. On and Off Control, Holding the Valves Open, Hot Gas Bypass and Using Multiple Units are some of the control methods used.

Q2. Describe a method for precise control of temperature.

Ans. A thermostat used in combination with a solenoid valve is a good method for temperature control. When a pre-determined desired temperature is achieved in the cold storage (cut-out point), a thermostat placed there effects the electrical circuit to close the solenoid valve, hence arresting the flow of liquid refrigerant to the evaporator. The compressor continues to operate and removes the leftover refrigerant in the evaporator, reducing the pressure in the evaporator. When evaporator pressure becomes slightly less than the Low Pressure setting (LP cut-out) of the HP/LP switch, the power supply to the compressor motor itself is cut-off.

Q3. Describe methods for humidity control.

Ans. Two temperature sensitive elements may be used to measure the dry and wet bulb temperatures and hence the humidity. Hygroscopic nature of some salts such as Calcium chloride, lithium chloride etc. has been used in humidity measurement. Passage over heating and cooling coils exposed to the atmosphere may 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. In such systems, the air circulates by natural convection.

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. The high velocity results in high heat transfer coefficient and hence faster cooling.

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

Ans. In any vapor based assembly, slight mismatches in the components may lead to leakage, besides loss of 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. An external pressure equalizer is used when the pressure drop in the evaporator is substantial resulting in superheat of the vapor to be higher than the superheated adjusted with the follow up springs. The equalizer transmits the pressure at the evaporator outlet to the inside of the bellows. Thus the pressure at the inside of the bellows is always equal to the pressure at the exit of the evaporator irrespective extent of drop in evaporator tubing.

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

Ans. 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. In order to bypass the power peak while starting a compressor the condenser and evaporator should be at the same pressure at the time of starting or the pressure difference should be minimum.

Q10. What decides the cut-in pressure?

Ans. The pressure of the evaporator at which the compressor; through the HP LP setup starts working is called the cut in temperature. The design evaporator pressure and the starting torque capacity of the motor determine it's magnitude.

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. The solenoid valve will prevent compressor slogging. However on turning on again 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 2)?

Ans. Here, a thermostatic expansion valve was used, while in the case of Vapor Compression Refrigeration test rig, we used a float valve for expansion purpose. We used an open type compressor in this case while we used a closed hermitic compressor for Vapor Compression Refrigeration test rig. IN the VCRS, the system was critically charged, and there was no HP LP cut off used. The refrigerant normally used in cold storage is ammonia, while R22 was used in the VCRS.

Experiment No. 4

Objective: 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

Components:

I. Hermetic compressor of 1 TR capacity
II. Air-cooled condenser
III. Finned tube direct expansion coil as an evaporator
IV. Capillary tube/thermostatic expansion valve
V. Orifice meter
VI. Pre-heat coil
VII. Re-heat coil
VIII. Humidification components

Refrigerant used: R134a

Principle and Formulae: The COP of the air conditioning system is given by: W_c

$$COP = \frac{Cooling\ capacity}{Power\ input\ to\ compresor} = \frac{Qe}{Wc}$$

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 mw is given by:

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

The mass flow rate of the dry air from measured average velocity is given by:

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

Power consumption is given by:

$$Wc = m_r \Delta h_c$$

The bypass factor X is defined as:

$$X = \frac{t_{a,o} - t_{ADP}}{t_{a,i} - t_{ADP}}$$

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

Observations:

Air-conditioning laboratory unit

Objectives:- 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 2,5 & 8.

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Readings	One	Two	Three	Four	Five	Six	Seven	Eight	Nine
First Pre-Heater, 1kW	Off	Off	Off	On	On	On	Off	Off	Off
First Re-Heater, 1kW	Off	Off	Off	Off	Off	Off	On	On	On
Fan supply Voltage (V_f) [V]	130	150	170	130	150	170	130	150	170
Air at fan inlet DBT (T1) [^oC]	34.5	34.4	34.7	34.7	34.9	34.5	34.2	34.1	34.5
T12 Air at fan inlet WBT (T2) [^oC]	26.7	26.8	26.6	27.3	27.0	27.2	27.4	27.6	27.5
Air after pre-heater DBT (T3) [^oC]	35.3	34.7	35.1	47.7	45.0	44.0	34.7	34.5	34.5
8 Air after pre-heater WBT (T4) [^oC]	26.4	26.1	26.4	29.8	29.0	28.9	26.5	26.8	26.9
Air after cooling / dehumidification DBT (T5) [^oC]	22.7	23.3	24.4	27.2	27.7	28.2	23.3	24.4	25.2
c Air after cooling / dehumidification WBT (T6) [^oC]	20.7	21.5	22.1	23.4	23.6	24.2	21.9	22.6	23.2
Air after reheating DBT (T7) [^oC]	23.5	24.4	25.0	27.9	28.1	28.6	37.3	35.7	35.7
T10 Air after reheating WBT (T8) [^oC]	22.5	23.3	23.7	25.3	25.5	25.8	25.9	26.2	26.5

Air conditioning laboratory unit

Temperature at evaporator outlet (T13) [^oC]	14.8	15.8	16.1	18.0	18.8	20.1	14.5	16.0	17.8	
Temperature at condenser inlet (T14) [^oC]	89.0	89.9	89.3	90.5	90.7	90.1	89.2	89.5	89.8	
Temperature at condenser outlet (T15) [^oC]	55.7	56.5	57.0	57.4	57.0	55.3	55.3	56.2	56.5	
Evaporator outlet pressure (P1) [kN m^{-2}]	300	300	325	350	350	350	300	325	325	
Condenser inlet pressure (P2) [kN m^{-2}]	1475	1500	1500	1550	1575	1575	1475	1500	1500	
Condenser outlet pressure (P3) [kN m^{-2}]	1450	1475	1450	1500	1550	1550	1450	1450	1475	
Duct differential pressure (Z) [mm of H_2O]	8.7	10.3	12.4	8.4	10.4	12.6	8.0	10.5	12.7	
Condensate collected (m_c) [ml]	98	76	74	38		36	91	75	105	
Time interval for condensate collected (x) [s]	501	503	525	756	428	507	520	343	500	
R134a Mass flow rate (m_{ref}) [g s^{-1}]	15	15	15	17	17	17	15	16	16	
Ambient Temperature [^oC]	34.5	34.4	34.7	34.7	34.9	34.5	34.2	34.1	34.5	
Ambient WBT [^oC]	26.7	26.8	26.6	27.3	27.0	27.2	27.4	27.6	27.5	

15/3/13

Sign of supervisor

Date:- 15/3/13

End z = 4.1

Calculations:

Note: Psychrometric and refrigerant properties are calculated using Computer Aided Thermodynamic Tables

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

Mass flow rate of air, $\mathbf{m_a = 0.274\ kg/s}$
Specific enthalpy of air before cooling, $\mathbf{h_{a,i} = 80.63}$ **kJ/kg of dry air**
Specific enthalpy of air after cooling, $\mathbf{h_{a,o} = 62.61}$ **kJ/kg of dry air**
Condensate removal rate, $\mathbf{m_w = 0.195\ g/s}$
Specific Enthalpy of condensate **= 97.73 kJ/s**
Cooling Capacity = $\mathbf{Q_e = m_a*(h_{a,i} - h_{a,o}) - m_w*h_w = 4.91\ kJ/s}$

Mass flow rate of refrigerant, $\mathbf{m_r = 15\ g/s}$
Specific Enthalpy at evaporator outlet, $\mathbf{h_{e,o} = 415.35\ kJ}$
Specific Enthalpy at condenser inlet, $\mathbf{h_{c,i} = 465.98\ kJ}$
Work done by compressor, $\mathbf{W_c = m_r*(h_{c,i} - h_{e,o}) = 759.45\ J/s}$

$\mathbf{COP = Q_e / W_c = 6.47}$

From psychrometric chart,
Apparatus Dew point Temperature, $\mathbf{T_{ADP} = 19\ ^oC}$

By-pass factor of cooling coil = $\mathbf{(T_{a,o} - T_{ADP}) / (T_{a,i} - T_{ADP}) = (23.3-19)/(34.7-19) = 0.27}$

Case 5: Pre-Heater ON, Re-Heater OFF, Fan Supply Voltage: 150V

Mass flow rate of air, $\mathbf{m_a = 0.273\ kg/s}$
Specific enthalpy of air before cooling, $\mathbf{h_{a,i} = 93.80}$ **kJ/kg of dry air**
Specific enthalpy of air after cooling, $\mathbf{h_{a,o} = 70.44}$ **kJ/kg of dry air**
Condensate removal rate, $\mathbf{m_w = 0.086\ g/s}$
Specific Enthalpy of condensate **= 116.13 kJ/s**
Cooling Capacity = $\mathbf{Q_e = m_a*(h_{a,i} - h_{a,o}) - m_w*h_w = 6.37\ kJ/s}$

Mass flow rate of refrigerant, $\mathbf{m_r = 17\ g/s}$
Specific Enthalpy at evaporator outlet, $\mathbf{h_{e,o} = 414.89\ kJ}$
Specific Enthalpy at condenser inlet, $\mathbf{h_{c,i} = 465.73\ kJ}$
Work done by compressor, $\mathbf{W_c = m_r*(h_{c,i} - h_{e,o}) = 864.28\ J/s}$

$\mathbf{COP = Q_e / W_c = 7.37}$

From psychrometric chart,
Apparatus Dew point Temperature, $\mathbf{T_{ADP} = 20\ ^oC}$

By-pass factor of cooling coil = $\mathbf{(T_{a,o} - T_{ADP})/(T_{a,i} - T_{ADP}) = (27.7-20)/(45-20) = 0.31}$

Case 8: Pre-Heater OFF, Re-Heater ON, Fan Supply Voltage: 150V

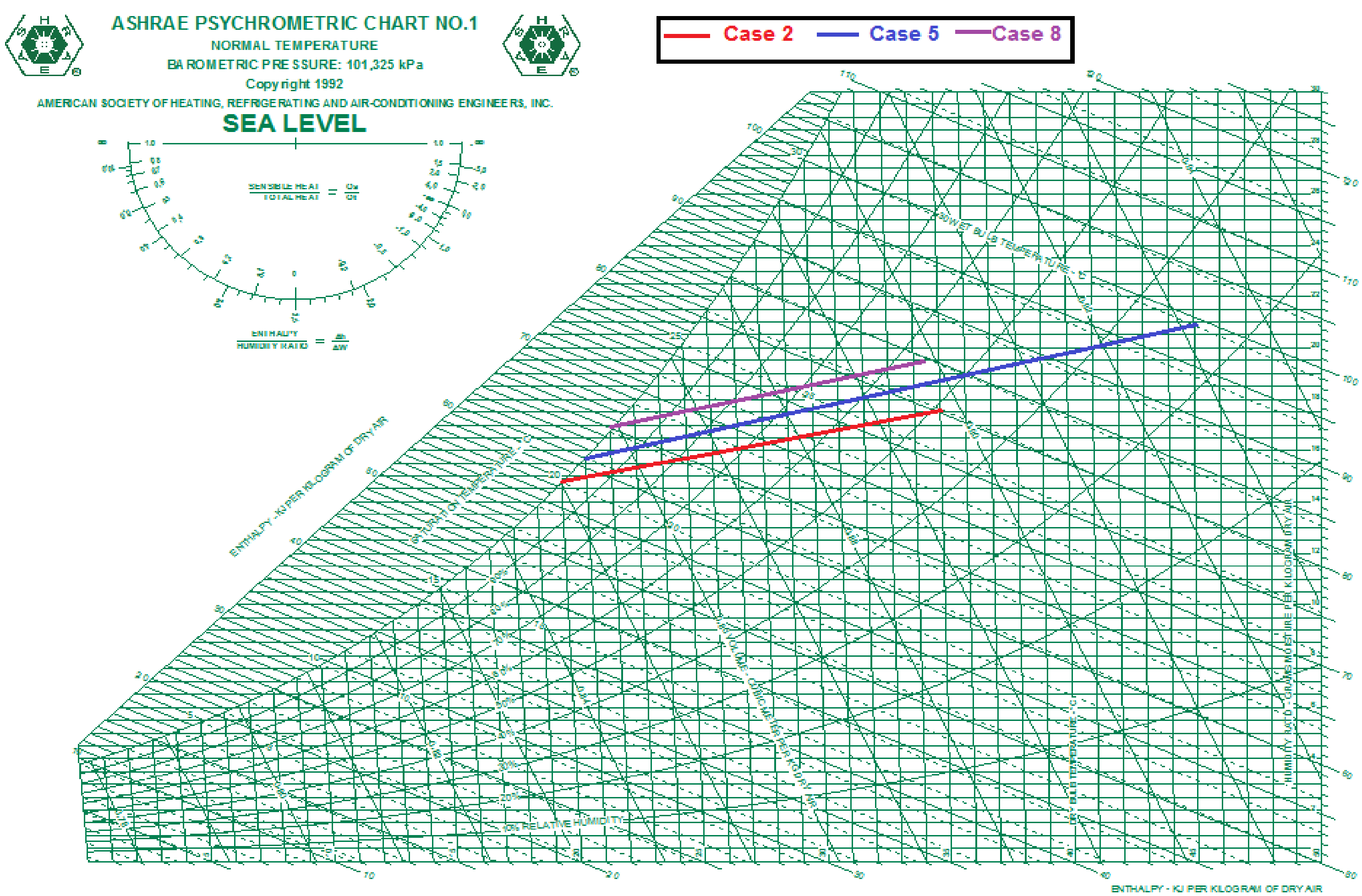
Mass flow rate of air, $\mathbf{m_a = 0.272}$ **kg/s**
Specific enthalpy of air before cooling, $\mathbf{h_{a,i} = 83.79}$ **kJ/kg of dry air**
Specific enthalpy of air after cooling, $\mathbf{h_{a,o} = 66.68}$ **kJ/kg of dry air**
Condensate removal rate, $\mathbf{m_w = 0.218}$ **g/s**
Specific Enthalpy of condensate **= 102.33 kJ/s**
Cooling Capacity = $\mathbf{Q_e = m_a*(h_{a,i} - h_{a,o}) - m_w*h_w = 4.63}$ **kJ/s**

Mass flow rate of refrigerant, $\mathbf{m_r = 16}$ **g/s**
Specific Enthalpy at evaporator outlet, $\mathbf{h_{e,o} = 415.19}$ **kJ**
Specific Enthalpy at condenser inlet, $\mathbf{h_{c,i} = 465.54}$ **kJ**
Work done by compressor, $\mathbf{W_c = m_r*(h_{c,i} - h_{e,o}) = 805.6}$ **J/s**

$\mathbf{COP = Q_e / W_c = 5.75}$

From psychrometric chart,
Apparatus Dew point Temperature, $\mathbf{T_{ADP} = 21\ ^oC}$

By-pass factor of cooling coil = $\mathbf{(T_{a,o} - T_{ADP}) / (T_{a,i} - T_{ADP}) = (24.4-21) / (34.5-21) = 0.25}$



Discussions:

Q 1. How does the bypass factor vary with air velocity, answer with reason?

Ans. The Bypass factor of the system increases with increase in air velocity. This is because as the air velocity increases, the air gets less time to be in contact with the cooling coil and therefore less cooling and -dehumidification takes place.

Q 2. What are the effects of fin spacing and number of rows on bypass factor?

Ans. As the number of rows of fins is increased the contact area with the air increases which leads to decrease in bypass factor. Reduction in fin spacing first leads to a decrease in bypass factor as contact area increases. However, as fin space is decreased further, the bypass factor increases as flow is restricted.

Q 3. 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 is taking place. A higher bypass factor means a less efficient system.

Q 4. 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 can be utilized (for DBT).

Q 5. What are the errors in the measurements of condensate rate?

Ans. Condensate formation may not take place only at the cooling coil surface. Also, not all the condensate can be collected by the collecting mechanism.

Q 6. What precautions should be taken in measurement of WBT?

Ans. We must take care that the wet bulb should not be provided excess water. We must also see to it that the water we use must be at the air DBT. We must let the reading stabilize before we start reading off, once the bulb is set properly. The readings may also be erroneous if there is too much oil around the bulb, or the water used is too rich in salts and other foreign material, so distilled water should only be used, and not reused again.

Q 7. Suggest alternate methods (other than measuring WBT) for finding state points of moist air?

Ans. Other means of finding state points include moisture-absorbing polymer films, chilled gold mirrors with optical sensors etc.

Q 8. How do you design the thermometer wells used for measurement of temperature?

Ans. Thermometer wells are used in measuring the temperature of a moving fluid in a conduit, where the stream exerts an appreciable force. For velocities of 300 ft./s or less, tapered thermometer wells are used. For velocities in excess of 300 ft./s, a fixed beam type thermometer well is recommended.