Refrigeration & Air Conditioning Laboratory

Laboratory Manual

Spring 2016

Department of Mechanical Engineering
Indian Institute of Technology Kharagpur
Kharagpur – 721 302, INDIA
Ph. (091) 03222-281577

Refrigeration Laboaratory (Thermo-Fluids Lab.-I: ME39606)

The syllabi for this laboratory consists of 4 experiments:

1a) Studies on refrigerant compressors

1b) Studies on thermal comfort

2) Performance evaluation of a vapour compression refrigeration system

3) Pull-down and cycling performance of a cold storage

4) Performance evaluation of a summer air conditioning system

Experiment 1(a): Studies on refrigerant compressors

1.1. Objective:

Study of compressors used in refrigeration and air conditioning applications

1.2. Experimental Procedure:

Remove all the parts of the two open reciprocating compressors and one hermetically sealed reciprocating compressor. Identify all the parts and understand the working principle. Make suitable free hand isometric drawings, and assemble the compressors.

1.3. Introduction:

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 ($\approx$ 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.

1.4.1. Main parts of a reciprocating compressor

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.

1.4.2. Open vs Hermetic Compressors:

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 150^{o}C 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 NH_3 compressors. Cylinder cooling is a must for NH_3 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 70^{o}C for adequate sealing and lubrication. NH_3 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.

At the end of your study, you must try to find answers to the following questions:

1. 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?
2. How do we define stroke, bore, clearance volume and displacement volume of the compressor?
3. Do the piston rings have a taper? While mounting the ring the sharp edge should be on the top or the bottom?

4. In which direction should the piston rings scrap the lubricating oil? If more than one piston ring is used what precautions should be used?
5. What are the types of bearings that can be used in compressors?
6. What are the methods used for capacity control of compressors?
7. What are the safety devices used with refrigerant compressors?
8. What type of motors are normally used with refrigerant compressors?
9. Can you outline the manufacturing methods for the various parts of the compressor.
10. Can you describe various types of seals used in compressors.
11. What are the advantages/disadvantages and applications of open type and hermetic compressors?
12. 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.

Department of Mechanical Engineering

Refrigeration and Air Conditioning Laboratory

Manual

of

Studies on indoor thermal comfort

1 Objective

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

2 Introduction

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". According to this definition, comfort is a subjective sensation. Since the industrial revolution, human beings have been spending progressively larger proportions of their time indoors. So, here, we are primarily concerned about the sensation of thermal comfort indoors.

Our bodies continuously generate heat within due to a miscellany of ongoing metabolic activities. An essential requirement for continued normal body function is that the deep body temperature be maintained within a very narrow limit of ±1℃ around the acceptable resting body core temperature of 37℃. Achieving this body temperature equilibrium requires a constant exchange of heat between the body and the environment. The rate and amount of the heat exchanged is governed by the fundamental laws of thermodynamics. The amount of heat that must be exchanged is a function of: the total metabolic heat produced and the heat gained from the environment. If the heat generated within can not be dissipated to the environment or if more heat is dissipated than is being generated, then it can lead to changes in body temperature causing discomfort and in extreme cases, permanent damage to our physiology too. Poor thermal comfort conditions not only have consequences for occupant health, they also affect work efficiency of the occupants. There is a continuous transport of heat from inside the body to skin surface, where from it is dissipated by radiation, convection, and evaporation (minimally by conduction too). Some heat is also lost during expiration. Most descriptions of thermal comfort assume that for a feeling of comfort, the body needs to be in thermal equilibrium and both sweating rate and average skin temperature are within comfort limits. Thermal equilibrium is achieved when the metabolic rate equals rate of heat loss, less any mechanical work done. So to find if a certain environment would be suitable for human occupation or not, we need to evaluate it in terms of the variables that would affect heat transfer from our bodies to the surroundings.

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 (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. In this manual, DBT and t_a will be used interchangeably to denote indoor air temperature.

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. 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. Mean radiant temperature cannot be measured directly, but it can be approximated by globe temperature (t_g) measurements. The globe thermometer is a mat black sphere with a thermometer located at its centre. Positioned in a room, after equilibrium is reached (in 10-15 minutes), the globe will respond to the net radiation to or from the surrounding surfaces. Relation between t_g and t_r is given by:

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

where ε is the emissivity of the globe surface and D is the globe diameter. For the given instrument, the globe diameter is 67 mm and emissivity of the surface can be taken as 0.95

The human body continuously produces heat. This metabolic heat production can be of two kinds: basal metabolism, due to biological processes and muscular metabolism, whilst carrying out work. *Metabolic activity* (M) is expressed per unit surface area of the human body, in terms of the unit 'met'. One met is approximately equivalent to 58.2 W/m^2. An average human being has a total body surface area of around 1.8 m^2. The metabolic rates of certain common activities are given in Table 1.

Table 1: Metabolic rates for some typical tasks

Activity	Metabolic rate		
	met	W/m^2	BTU/h ft^2
Seated, quiet	1.0	60	18
Standing, relaxed	1.2	70	22
Seated, reading or writing	1.0	60	18
Typing	1.1	65	20
Cooking	1.6–2.0	95–115	29–37

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** (I_{cl}). This corresponds to an insulating cover over the whole body of a resistance of 0.155 m^{2}K/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.

2.1 Thermal sensation scale

To determine thermal sensation of occupants, the most popular means is a subjective survey questionnaire using the ASHRAE seven point scale. The scale is represented in Table 2. On this scale, the mid point i.e. 0 is called the neutral point and anyone recording their thermal sensation at the neutral point are deemed to be comfortable. Since individual differences would make it impossible for a particular thermal environment to be neutral to everyone, a wider comfort band is considered between ± 1. Any occupant voting within the comfort band is deemed to be reasonably comfortable and not stressed by her/his thermal environment.

Table 2: ASHRAE thermal sensation scale

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

2.2 Thermal comfort indices

The main purpose of defining a thermal comfort index is for evaluating thermal comfort performance of a building. Thermal indices try to combine two or more variables that affect thermal comfort into a single expression. Indices may either be empirical (i.e. derived from the results of experimental studies done with people) or rational (i.e. rigorously derived using laws of heat transfer).

Some commonly used indices in field of human thermal comfort are given here and their values can be obtained through this experiment.

Wet bulb globe temperature (WBGT)

WBGT indicates the combined effect of air temperature, low temperature radiant heat, solar radiation and air movement. It is the weighted average of DBT, naturally ventilated WBT and globe temperature, for outdoor use (including the presence of solar radiation). For indoor use the DBT term is dropped out.

$$\begin{aligned} for \quad indoors: &\qquad WBGT = 0.7 \cdot WBT + 0.3 \cdot t_g \\ for \quad outdoors: &\qquad WBGT = 0.7 \cdot WBT + 0.2 \cdot t_g + 0.1 \cdot DBT \end{aligned} \tag{2}$$

Operative temperature (t_o)

Operative temperature is defined as the temperature of a uniform, isothermal, "black" enclosure in which an occupant would exchange heat by radiation and convection at the same rate as in the given non-uniform environment. For calculations, t_o is also given by the average of t_r and DBT weighted by their respective heat transfer coefficients.

$$t_o = \frac{h_r \cdot t_r + h_c \cdot t_a}{h_r + h_c} \tag{3}$$

where h_r is the radiative heat transfer coefficient and h_c is the convective heat transfer coefficient of the environment. Fortunately, h_r is nearly constant for typical indoor temperatures, and a value of 4.7 W/m^{2}K will suffice for our calculations.

The convective heat transfer coefficient for a seating person can be evaluated from the following empirical expression:

$$\begin{aligned} h_c &= 8.3 v_r^{0.6} \qquad 0.2 < v_r < 4.0 \\ h_c &= 3.1 \qquad 0 < v_r < 0.2 \end{aligned} \tag{4}$$

Tropical summer index (TSI)

The TSI has been developed in the mid-1980s at the Central Building Research Institute, Roorkee, for the climatic conditions prevalent in India and to suit the living habits of our population. It is defined as the temperature of still air, at 50% relative humidity, which causes the same thermal sensation as the given environmental condition. TSI is given by:

$$TSI = 0.308 \cdot WBT + 0.745 \cdot t_g - 2.06\sqrt{v_r + 0.841} \tag{5}$$

Ranges and optimum values of TSI, for different thermal sensations, are given in Table 3.

Table 3: Range and optimum values of TSI for thermal sensation

Thermal sensation	Range (°C)	Optimum value (°C)
Slightly cool	19.0–25.0	22.0
Comfortable	25.0–30.0	27.5
Slightly warm	30.0–34.0	32.0

Heat stress index (HSI)

HSI is defined as the ratio of evaporative cooling required for maintaining heat balance, to the maximum evaporative cooling possible under the given conditions. So, $HSI = (E_{reqd}/E_{max}) \cdot 100$. E_{max} has a physiological limit of 700 W which is equivalent to a little over one litre of sweat evaporating in an hour. HSI can also be expressed as an empirical function of met rate, air movement, humidity, DBT, and t_r.

$$HSI = \frac{M + 22(t_r - 95) + 2 \cdot v_r^{0.5}(DBT - 95)}{10.3 \cdot v^{0.4}(42 - p_a)} \tag{6}$$

where M is in Btu/h, t_r is in °F, DBT is in °F, v_r is in ft/min, and p_a is the vapour pressure in mm Hg.

HSI is mostly intended for places where occupants are likely to be under considerable thermal duress. It is not suitable for the comfort zone or conditions below comfort level. Over an eight hour exposure, HSI values may be interpreted as follows:

-10 to -20	mild cold strain
0	no strain
10 to 30	mild to moderate strain
40 to 60	severe strain, health threat, decreased work performance
70 to 90	very severe, tolerated only by fit and acclimatised people
100	the maximum tolerated only by the most fit and acclimatised young men; the upper limit of thermal equilibrium, with a sweat rate of 1 L/h, above which body heating would occur which can be tolerated for short periods and only up to 1.8 K

PMV

The PMV index is based on Fanger's work in the field of human thermal comfort. Fanger took all the six variables affecting comfort and combined them into a single index. The thermal situation of the body, as determined from heat transfer, is related to the feeling of thermal comfort through PMV (predicted mean vote) and PPD (predicted percent dissatisfied) relations. These relations, that were developed from extensive climate chamber surveys, still act as the most accepted modelling for human thermal comfort. The PMV scale is similar in form, purpose, and number of divisions to the seven point ASHRAE thermal sensation scale (Table 2). The equation for PMV needs to be solved iteratively. Close to neutral conditions though, PMV can be evaluated with the help of a set of simplified formulas:

$$\begin{aligned} PMV =&(0.352e^{-0.042M} + 0.032)[M- \\ &0.35[43 - 0.061M - p_a]- \\ &0.42[M - 50]- \\ &0.0023M(44 - p_a) - 0.0014M(34 - t_a)- \\ &3.4 \cdot 10^{-8}[(t_{cl} + 273)^4 - (t_r + 273)^4] - f_{cl}h_c(t_{cl} - t_a)] \end{aligned} \tag{7}$$

where p_a is in mm Hg, M is in kcal/hr m^2 (to convert from met to kcal/hr m^2, multiply by 50), temperatures are in ℃,f_{cl} is the ratio of the surface area of the clothed body to the surface area of the nude body and is approximated as $f_{cl} = 1 + 0.3 \cdot I_{cl}$, and t_{cl} is the clothing surface temperature. Clothing surface temperature is calculated as:

$$\begin{aligned} t_{cl} =&35.7 - 0.032M - 0.18I_{cl}[M- \\ &0.35[43 - 0.061M - p_a] - 0.42[M - 50]- \\ &0.0023M(44 - p_a) - 0.0014M(34 - t_a)] \qquad (℃) \end{aligned} \tag{8}$$

where p_a is in mm Hg, I_{cl} is in clo, M is in kcal/hr m^2, and temperatures are in ℃. Convective heat transfer coefficient h_c is taken to be the larger of the following two values: $2.05(t_{cl} - t_a)^{0.25}$ and $10.4\sqrt{v_r}$

Equation 9 gives the relation between PMV and PPD.

$$PPD = 100 - 95exp[-(-0.03353 \times PMV^4 + 0.2179 \times PMV^2)] \tag{9}$$

As is apparent from equation 9, whatever be the conditions, we can not expect to have a 100% satisfaction amongst occupants. This is due to the difference in physiology amongst persons. A single environmental condition can not be suitable to everyone. The best that can be aimed for is to have a majority of occupants satisfied. For most buildings, satisfied percentage between 80 to 90 is considered adequate.

2.3 Adaptive thermal comfort and comfort surveys

The underlying philosophy of adaptive comfort is: if a change occurs in the surroundings that causes discomfort, then people will react so as to restore their feeling of comfort. In comfort field studies, simultaneously objective data on the thermal environment (DBT, humidity, air velocity, t_r) and subjective thermal sensation votes of people are gathered. Normally, some equivalent of the ASHRAE thermal sensation scale is used to collect sensation votes. These subjective data are then statistically analysed 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.

A typical thermal comfort survey questionnaire is presented here. It contains questions regarding thermal sensation, thermal perception, acceptability feeling of the environment etc.

1. How do you feel at this moment?

   a) Cold b) Cool c) Slightly cool d) Neutral e) Slightly warm f) Warm g) Hot

2. Do you feel comfortable now?

   a) Much too cool b) Too cool c) OK(cool) d) OK(just right) e) OK(warm) f) Too warm g) Much too warm

3. Would you like to be?

   a) Cooler b) No change c) Warm

4. How would you rate the overall acceptability of the temperature at this moment?

   a) Acceptable b) Not acceptable

5. How do you feel about the air flow at this moment?

   a) still b) just right c) breezy

6. How do you feel in terms of humidity?

   a) too dry b) slightly dry c) just right d) slightly humid e) too humid

Occupants answer these multiple choice questions by tick marking their choices according to their perception of the environment.

3 Instrument Description

DBT and WBT are measured using a sling psychrometer. Air velocity is measured with a hot wire anemometer. The globe thermometer is used for measuring values of t_g.

4 Experimental Procedure

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 and at head height. Simultaneously, the occupants are given the subjective survey questionnaires to fill up. Care is to be taken that the filling of survey questionnaire by the occupants and measurement of environmental parameters are to be done simultaneously, without any appreciable time gap between them.

For this experiment, the role of occupants is taken by the student groups engaged in the other three experiments. The students are to be handed one questionnaire sheet each. As they fill up the questionnaire, readings of DBT, WBT, v_r, and t_g are to be measured in the near vicinity of the "occupants". Enough time should be given for the instrument readings to stabilise. The longest time for obtaining a stable reading is for the globe thermometer — approximately 15 minutes.

Clothing resistance value can be approximated to be 0.5 clo for observations being taken in summer and 1.0 clo for observations taken in winter. For laboratory work, metabolic activity can be approximated as 1.6 met.

After recording the observations, the following comfort and heat stress indices have to be calculated using the formulas from Section 2.2: WBGT, TSI, HSI, and PMV.

Calculations

Using the measured environmental parameters and estimated values of met and I_{cl}, WBGT, TSI, HSI, PMV, and PPD values have to calculated. The calculated values of PMV may later be verified using the thermal comfort tool at the following website: `http://www.cbe.berkeley.edu/comforttool/`. It is expected that the results will be different considering for use in laboratory conditions, an explicit form of the PMV calculations is given in this manual.

5 Learning objectives

At the end of this experiment, you should be able to answer questions like:

1. How is thermal comfort defined?
2. What are the parameters affecting thermal comfort?
3. How does the human body regulate its temperature?
4. What are the modes of heat transfer from human body?
5. What scale is used for thermal sensation?
6. What are the different indices used for measuring thermal comfort and heat stress?
7. How comfort surveys are conducted to evaluate the comfort standards of a building?

Useful references:

[1] *ANSI/ASHRAE Standard 55-2004. Thermal comfort conditions for human occupancy*, ASHRAE, Atlanta, 2004.

[2] *ASHRAE Handbook, Fundamentals*, SI ed., ASHRAE, Atlanta, GA, 2009.

[3] P. O. Fanger, *Thermal comfort. Analysis and applications in environmental engineering.*, Copenhagen: Danish Technical Press., 1970.

[4] F. Nicol, *Adaptive thermal comfort standards in the hot–humid tropics*, Energy and Buildings **36** (2004), no. 7, 628–637.

[5] M. R. Sharma and S. Ali, *Tropical summer index — a study of thermal comfort of Indian subjects*, Building and Environment **21** (1986), no. 1, 11–24.

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: **Roll No.:**

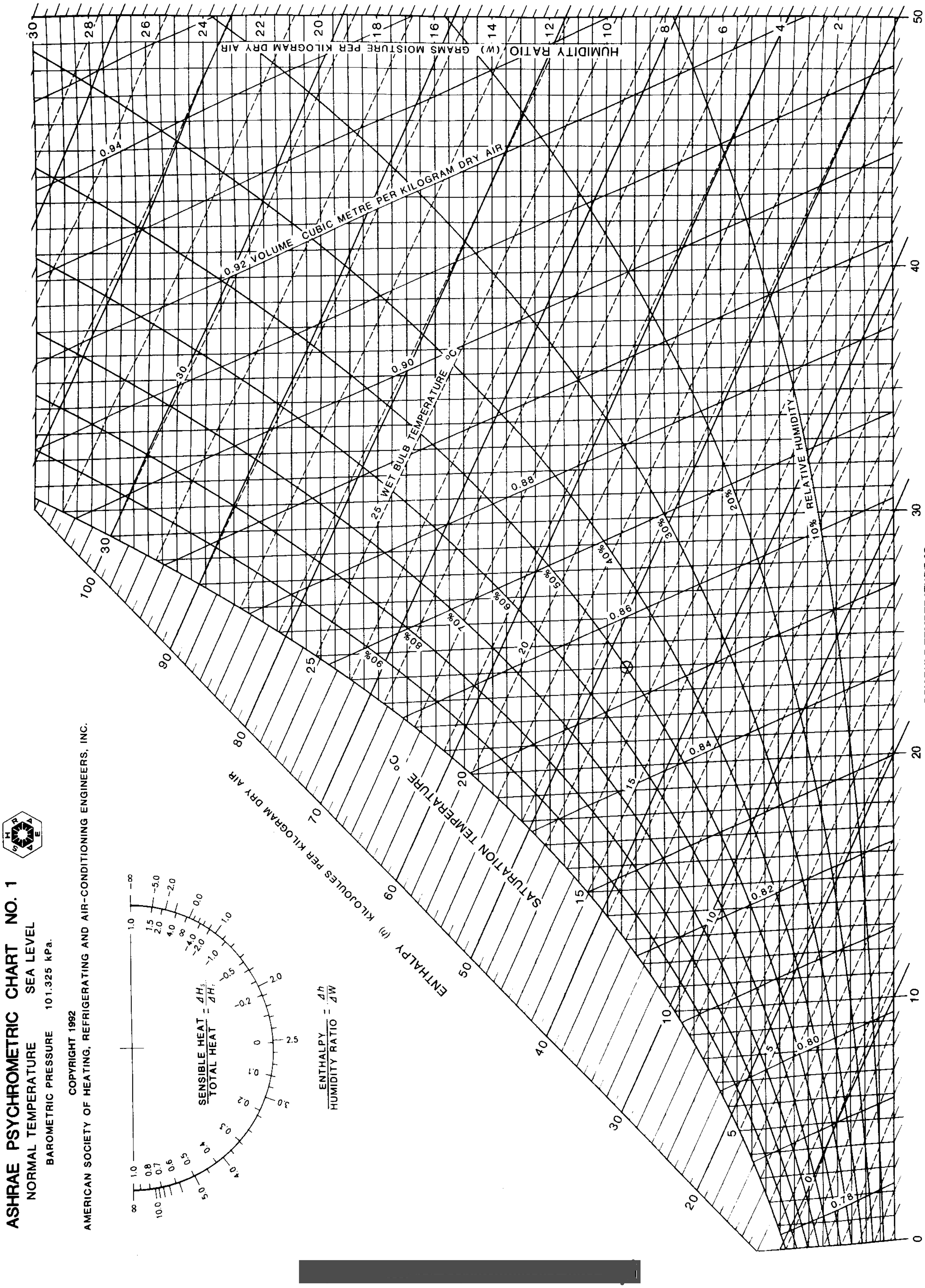
Observations:

Barometric Pressure (kPa):

Outdoors DBT (℃):
Outdoors WBT (℃):

Subject Group	t_a (℃)	WBT (℃)	p_a (kPa)	v_r (m/s)	t_g (℃)
1.					
2.					
3.					

Signature of instructor
Date:


ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE 101.325 kPa.
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) KILOJOULES PER KILOGRAM DRY AIR
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE °C
0.92 VOLUME CUBIC METRE PER KILOGRAM DRY AIR
HUMIDITY RATIO (W) GRAMS MOISTURE PER KILOGRAM DRY AIR
RELATIVE HUMIDITY
DRY BULB TEMPERATURE °C

Experiment 2: Studies on vapour compression refrigeration test-rig

2.1. 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

2.2. Introduction:

The test-rig consists of a hermetic compressor, an air-cooled condenser, a capillary tube/TEV(Thermo-static Expansion Valve) and a water-cooled evaporator. In addition to these four major components, the test-rig also consists of several other components such as manual shut-off valves, sight glass, filter, dryer, solenoid valve etc. Pressure gauges are installed to measure the condenser and evaporator pressures (in psig). Similarly thermometric wells are provided at the inlet and exit of evaporator, compressor and condenser to facilitate measurement of the refrigerant temperature temperatures at these points using suitable thermometers. A rotameter is installed in a by-pass liquid line to measure the flow rate of the liquid refrigerant by opening the manual shut-off valve. The test-rig uses CFC-12 (di-chloro di-fluoro methane) as refrigerant.

Evaporator:

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 fibreglass 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. As the low pressure, low temperature refrigerant flows through the copper coil; heat is transferred from the water to the refrigerant leading to flow boiling of refrigerant liquid inside the coil. In steady state, the heat input to the water by the heater is approximately equal to the heat absorbed by the refrigerant, which is the desired refrigeration effect. The energy meter connected across the heater measures the heater input. A digital thermometer measures the water temperature.

Compressor:

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 energy input to the compressor can be measured using either the energy meter or from the voltmeter and ammeter readings. Since the electrical load on the compressor is inductive, a suitable power factor (≈ 0.80) should be used while calculating energy input from the product of voltmeter and ammeter readings.

Condenser:

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. The heat rejection rate at condensate is obtained by measuring the airflow rate over the condenser and temperature of air at the inlet and outlet of the condenser. A vane type anemometer is used to measure the average velocity of air at the outlet of condenser. The average velocity is obtained by taking the average of measured air velocities at 5 locations (four corners and center) on the condenser. The mass flow rate of air is obtained from the average air velocity, face area of the condenser and the density of air (obtained from barometric pressure and air temperature).

Expansion device:

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. However, during the measurements only the capillary is to be used.

The total number of fins, fin thickness, width and height are measured. The diameter of condenser tubes and width of the condenser is also noted down. Finally the evaporator tube diameter coil diameter and number of coils are noted down.

2.2. Measurements:

Once steady-state is reached, measure the pressure gage readings, temperature readings of all the thermometers, heater and compressor energy meter (or voltmeter and ammeter) readings, 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 open the valve in the rotameter by-pass line and take the rotameter reading. Then the system is switched-off and measurements of condenser and evaporator dimensions are to be taken. For the condenser, measure the width and height of condenser, number of condenser tubes (no. of rows X no. of tubes per row), number of fins (no. of fins per inch X width of condenser in inches), thickness, width and height of fins and number of tubes passing through the fins. For the evaporator, measure the copper tube diameter, coil diameter and number of coils immersed in water.

2.3. Observations:

Condenser pressure, p_c	= (+ 14.7) /14.5 bar
Evaporator pressure, p_e	= (+ 14.7)/14.5 bar
Condenser inlet temperature, t_{ci}	= ^{o} C
Condenser outlet temperature, t_{co}	= ^{o} C
Evaporator inlet temperature, t_{ei}	= ^{o} C
Evaporator outlet temperature, t_{eo}	= ^{o} C
Compressor energy meter reading, W	= revolutions in minutes
Heater energy meter reading, Q_e	= revolutions in minutes
Number of fins, N_f	=
Fin thickness, e	= m
Fin width, b	= m
Fin height, h	= m
Condenser width, L	= m
Number of tubes in condenser, N_{tc}	=
Diameter of condenser tubes, d_c	= m
Average air velocity through condenser, V	= (+ + + +) /5 fpm
	= m/s
Air temperature at condenser inlet, $T_{air,in}$	= ^{o} C
Air temperature at condenser outlet, $T_{air,out}$	= ^{o} C
Temperature of water in evaporator, T_{water}	= ^{o} C
Evaporator coil diameter, D_e	= m
Evaporator tube diameter, d_e	= m
Number of tubes in the coil, N_{te}	=
Room temperature	= ^{o} C
Room pressure	= Pa
Displacement of compressor,PD	= 2.15 m^3/h

2.4. Calculations:

<u>i) COPs</u>

$$COP_{Carnot} = \frac{T_{evp}}{T_{cond} - T_{evp}}$$

where T_{evp} and T_{cond} are the saturation temperatures (K) corresponding to evaporator pressure and condenser pressure (to be obtained from Thermodynamic Tables or p-h charts of R-12)

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

where h_1,h_2 and h_3 are the enthalpy of refrigerant vapor at evaporator outlet, enthalpy of refrigerant vapor at compressor outlet and enthalpy of refrigerant liquid at condenser outlet respectively (to be obtained from tables or p-h charts from the temperature and pressure values).

$$COP_{actual} = \frac{Actual\ refrigetaion\ effect}{Actual\ energy\ input\ to\ compressor} = \frac{Energy\ meter\ reading\ of\ heater}{Energy\ meter\ reading\ of\ compressor}$$

the actual energy input to heater/compressor are obtained by multiplying the energy meter reading (rpm) with the energy meter constant

ii) Overall heat transfer coefficients of evaporator and condenser (U_{evp}, U_{cond})

$$U_{evp} = \frac{Actual\,refrigeration\,effect}{A_{coil}.(T_{water} - T_e)}$$

where A_{coil} is the area of the evaporator coil (to be obtained from the evaporator tube diameter, coil diameter and number of coils immersed in water), T_{water} and T_e are the water and refrigerant temperatures in the evaporator.

$$U_{cond} = \frac{Condenser\,heat\,rejection\,rate}{A_{cond}.LMTD_{cond}} = \frac{(m.c_p)_{air}.(T_{air,out} - T_{air,in})}{A_{cond}.LMTD_{cond}}$$

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

$T_{air,in}$ and $T_{air,out}$ are the inlet and outlet temperatures of air, T_{cond} is the saturation temperature of refrigerant corresponding to condenser pressure.

The condenser area, A_{cond} is the sum total of the bare tube area (primary area) and the fin area (secondary area). These areas have to be obtained from the condenser dimensions and geometry, number of fins, fin dimensions etc.

iii) Overall volumetric efficiency of compressor, η_{ov}:

$$\eta_{ov} = \frac{Actual\,refrigerant\,mass\ flow\,rate}{Maximum\ possible\,mass\ flow\,rate} = \frac{Q_{liq}/\upsilon_{liq}}{PD/\upsilon_{suction}}$$

where Q_{liq} is the volumetric flow rate of liquid refrigerant as indicated by the rotameter (m^3/s), υ_{liq} is the specific volume of liquid refrigerant at condenser outlet (m^3/kg), $\upsilon_{suction}$ is the specific volume of refrigerant vapor at compressor inlet (m^3/kg). PD is the piston displacement of the compressor in m^3/s. Refrigerant specific volume is to be obtained from thermodynamic property tables or p-h chart.

At the end of your study, you must try to answer the following questions:

1. What are the errors in measurement of refrigerant temperature? Suggest a better method of measuring the temperature.
2. What are the errors in energy meter reading?
3. What are the assumptions made in calculating the Carnot, cycle and actual COPs?
4. 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?
5. Why is thermostat used in the water tank?
6. What are the safety devices used in the set-up?
7. The experiment gives COP at a fixed temperature. Suggest a method to find the COP at different condenser and evaporator temperatures.
8. What are the assumptions made in calculating LMTD for condenser?
9. What is the significance of volumetric efficiency of compressor?
10. Carry out an error analysis of your experimental results.

EXPERIMENT 3: Studies on a small cold storage

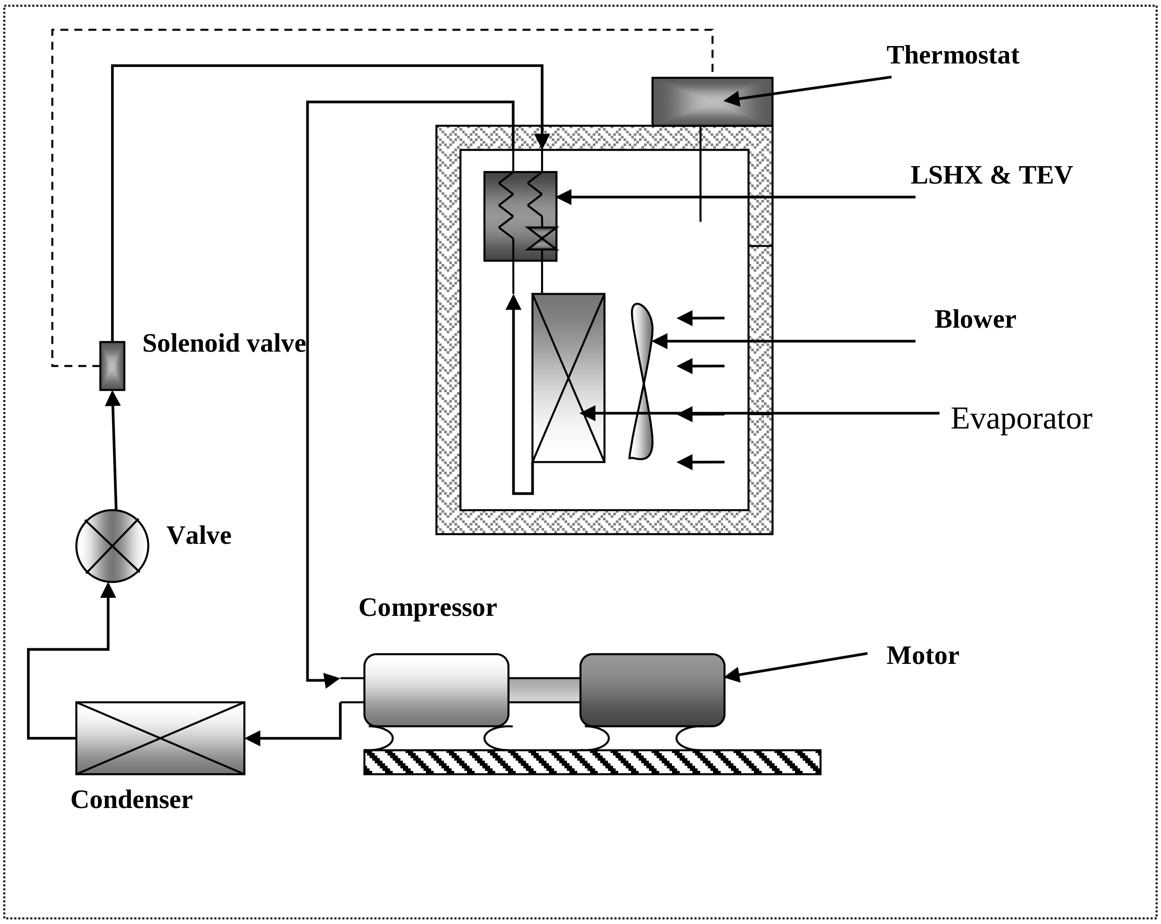
3.1. Objective:

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

3.2. 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 domestic refrigerator may be considered as a small cold storage. Commercial cold storages are very large with thousands of tons of storage capacity. 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 cut-out 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.



Schematic of the cold storage with refrigeration plant

In small refrigeration systems, which do not have a reservoir, a thermostat is connected directly to the compressor motor, which cuts out power supply to the compressor motor when the desired temperature is achieved in the evaporator. During the off period, the condenser is initially at higher pressure than the evaporator. As a result, liquid refrigerant flows from the condenser to the evaporator and equalization of pressure takes place. This might cause flooding of the evaporator (filling up of evaporator with liquid refrigerant) and subsequent slugging of the compressor (entry of liquid refrigerant). The slugging of the compressor can cause valve damage. Hence all small systems use critical charge, that is, the total quantity of refrigerant in the system is such that in the eventuality of all of it going to evaporator, it does not fill the evaporator. These systems are safe from the point of view of flooding of evaporator and subsequent slugging of the compressor. These systems use hermetic compressors, which are leak-proof.

In the present cold storage set-up the air temperature inside the cold storage is the controlled parameter. The system uses an open type of compressor with refrigerant R-22 as working fluid. Since the compressor is open type, there will be a continuous leakage of refrigerant from the compressor shaft. In addition, the required refrigerant mass flow rate may also vary depending upon the operating conditions. Hence a refrigerant reservoir is used in these systems that will take care of small leakages and variable mass flow rate requirements. The reservoir also acts as a liquid seal for the expansion valve so that the vapor from the condenser does not enter the expansion valve. The reservoir is also used to store the refrigerant during overhauling of compressor. In a reservoir-based system, if the thermostat is connected to the compressor, the power supply to the compressor motor will be cut-off when the desired temperature is achieved in the evaporator. The liquid refrigerant in the reservoir being at higher condenser pressure will flow to the evaporator, flood it and then slug the compressor. The only way to avoid this is to put a valve in the liquid line between reservoir and the expansion valve, so that when the desired temperature is reached, the valve closes. A solenoid valve connected to the thermostat serves this purpose. Then advantage is taken of the HP/LP Cut-out switch to cut-out the power supply to the compressor.

3.3. 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. *The cut out of a thermostatic expansion valve will be shown and its working principle will be demonstrated*.

3.4. Experimental procedure:

- Note down the room temperature and temperature set on the thermostat.
- Start the system and by closing and opening the hand valve, note down the cut-out and cut-in pressures.
- Record the time, temperature, pressure, and solenoid valve condition and compressor condition at solenoid closing, compressor closing, solenoid opening and compressor re-starting. Tabulate the observations as shown below:

Clock Time (hh:min:sec)	Temperature	Evaporator Pressure	Condenser Pressure	Solenoid Condition	Compressor condition
				Open	On
				Closed	On
				Closed	Off
				Open	Off
				Open	On
				Closed	On
				Closed	Off
				Open	Off
				Open	On
				Closed	On
				Closed	Off
				Open	Off
				Open	On

Data Reduction:

Draw a plot of temperature vs. time indicating the solenoid and compressor condition on it. Determine the on time and off time for the first four cycles, The on time of the first cycle (which is equal to the pull-down time) is an indication of the total thermal capacity of the cold storage and the cooling capacity of the refrigeration system. For a given refrigeration system, the pull-down time will be more for a fully loaded cold storage compared to a partly loaded cold storage. For an empty cold storage this is proportional to the thermal capacity of the cold storage materials (walls, door etc). The on time for the subsequent cycles indicated the effectiveness of the thermal insulation. The sensitivity of the thermostat is the temperature difference between cut-out and cut-in.

At the end of your study, you must try to answer the following questions:

1. Describe various methods of capacity control of refrigeration systems.
2. Describe a method for precise control of temperature.
3. Describe methods for humidity control.
4. 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?
5. 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?
6. What is the advantage of buying a condensing unit rather buying separate components and assembling them?
7. Where and why is TEV with external pressure equalizer used?
8. Why are cross charge, liquid charge and vapor charge used in TEV.
9. Describe pull down characteristics of a compressor and outline the procedure to bypass the power peak while starting a compressor.
10. What decides the cut-in pressure?
11. What happens if in stead of using the LP cut-out/cut-in, the thermostat itself simultaneously closes the solenoid valve and switches off compressor?
12. 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)?

Experiment 4:Studies on a summer air conditioning system

4.1. 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

4.2. Description:

The experimental set up consists of a fully instrumented air-conditioning system. It has a hermetic compressor of 1 TR capacity with a fan to cool its body externally, an air-cooled condenser, finned tube direct expansion coil type evaporator, capillary tube/thermostatic expansion valve and a blower to blow the air that is to be conditioned. 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.

4.3. Principle and procedure:

The COP of the air conditioning system is given by:

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

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 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 ($\mathbf{\Delta h_c}$). 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.

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

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

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 processes 1, 4 & 7.

Student's name :- ___________________________________

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) [^{0}C]									
Air at fan inlet WBT (T2) [^{0}C]									
Air after pre-heater DBT (T3) [^{0}C]									
Air after pre-heater WBT (T4) [^{0}C]									
Air after cooling / dehumidification DBT (T5) [^{0}C]									
Air after cooling / dehumidification WBT (T6) [^{0}C]									
Air after reheating DBT (T7) [^{0}C]									
Temperature at evaporator outlet (T13) [^{0}C]									
Temperature at condenser inlet (T14) [^{0}C]									
Temperature at condenser outlet (T15) [^{0}C]									
Evaporator outlet pressure (P1) [kN m^{-2}]									
Condenser inlet pressure (P2) [kN m^{-2}]									
Condenser outlet pressure (P3) [kN m^{-2}]									
Duct differential pressure (Z) [mm of H_2O]									
Condensate collected ($\dot{m}_c$) [ml]									
Time interval for condensate collected (x) [s]									
R134a Mass flow rate ($\dot{m}_{ref}$) [g s^{-1}]									
Ambient Temperature [^{0}C]									
Ambient WBT [^{0}C]									

Sign of supervisor
Date:-

The above performance parameters have to be obtained for different sets of operating conditions by setting the air flow rate and pre-heater capacity as shown in the observation table.

At the end of your study, you must be able to answer the following questions:

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

2. What are the effects of fin spacing and number of rows on bypass factor?
3. What is the relationship between by-pass factor and the performance of the system?
4. How is the relative humidity controlled in A/C systems?
5. Describe methods for independent control of DBT and RH.
6. What are the errors in the measurements of condensate rate?
7. What precautions should be taken in measurement of WBT?
8. How do you design the thermometer wells used for measurement of temperature?
9. Suggest alternate methods (other than measuring WBT) for finding state points of moist air?
10. Suggest methods for estimating the system performance more accurately.
11. Carry out error analysis of your results.

General guidelines

1. For experiments 1(b), 2, 3 and 4, sample calculations must be carried out immediately after taking all the measurements. You **must not leave the laboratory without verifying your observations through sample calculations**.

2. Report on **experiment 2 must contain uncertainty (error) analysis**

3. Reports must be submitted individually before the final viva-voce. Reports need not be very descriptive but must contain briefly the objectives, schematics of test set-up etc, experimental observations, results, discussion of the results and error analyses (wherever applicable). It must also contain answers to the questions posed above.

Books for Reference:

1. Refrigeration & air conditioning by W.F. Stoecker and J.P. Jones, McGraw Hill, 1982.

2. Refrigeration & air conditioning by R.C. Arora, PHI, 2010.

3. Refrigeration & air conditioning by C.P. Arora, Tata-McGraw Hill, 2012.

4. Modern Refrigeration and Air Conditioning by Andrew D. Althouse, Carl H. Turnquist and Alfred F. Bracciano, Goodheart-Willcox Company Inc., 1985.

5. Experimental Methods for Engineers by Jack P. Holman, McGraw Hill, 1999.