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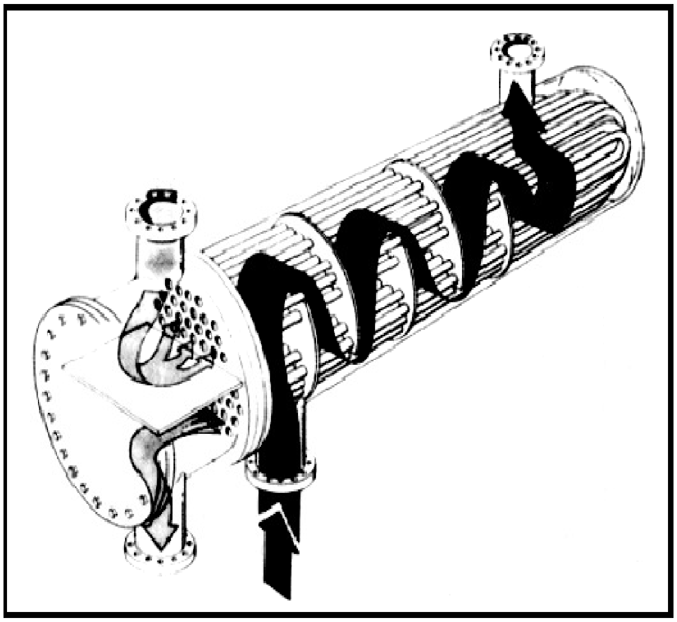
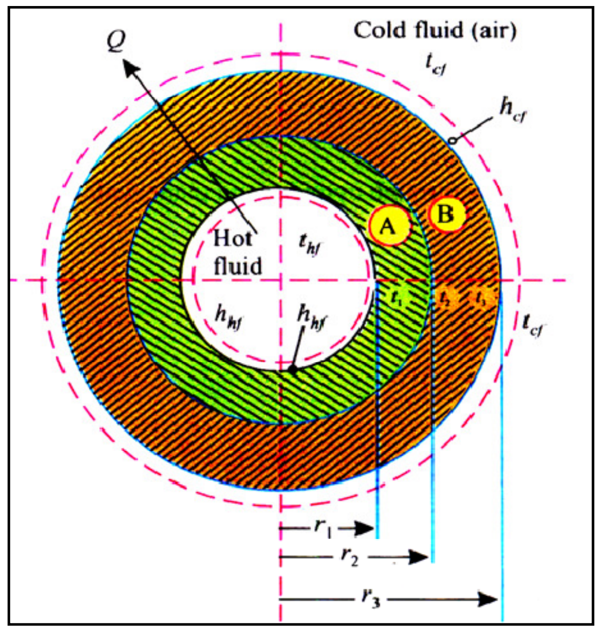
DEPARTMENT OF MECHANICAL ENGINEERING





THERMAL ENGINEERING II LAB MANUAL

REG NO	
NAME	
SUBJECT CODE\TITLE	ME 6512 - THERMAL ENGINEERING LABORATORY – II
BRANCH	MECHANICAL ENGINEERING
YEAR \ SEM	III \ V
REGULATION	2013
ACADEMIC YEAR	2017-2018

GENERAL INSTRUCTION

- ❖ All the students are instructed to wear protective **uniform, shoes** & **identity card** before entering into the laboratory.
- ❖ Before starting the exercise, students should have a clear idea about the principal of that exercise
- ❖ All the students are advised to come with completed record and corrected observation book of previous experiment.
- ❖ Don't operate any instrument without getting concerned staff member's prior permission.
- ❖ The entire instrument is costly. Hence handle them carefully, to avoid fine for any breakage.
- ❖ Utmost care must be taken to avert any possible injury while on laboratory work. In case, anything occurs immediately report to the staff members.
- ❖ One student form each batch should put his/her signature during receiving the instrument in instrument issue register.

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ME6512 THERMAL ENGINEERING LAB - II

LIST OF EXPERIMENTS

HEAT TRANSFER

1. Thermal conductivity measurement using guarded plate apparatus.
2. Thermal conductivity measurement of pipe insulation using lagged pipe apparatus.
3. Determination of heat transfer coefficient under natural convection from a vertical cylinder.
4. Determination of heat transfer coefficient under forced convection from a tube.
5. Determination of Thermal conductivity of composite wall.
6. Determination of Thermal conductivity of insulating powder.
7. Heat transfer from pin-fin apparatus (natural & forced convection modes).
8. Determination of Stefan-Boltzmann constant.
9. Determination of emissivity of a grey surface.
10. Effectiveness of Parallel/counter flow heat exchanger

REFRIGERATION AND AIR CONDITIONING

1. Determination of COP of a refrigeration system
2. Experiments on Psychometric processes.
3. Performance test on a reciprocating air compressor.
4. Performance test in a HC Refrigeration system.
5. Performance test in a fluidized bed cooling tower.

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HEAT TRANSFER LAB

S.No	Date	Name of the Experiment	Staff Signature	Remarks
1		Thermal conductivity measurement using guarded plate apparatus		
2		Thermal conductivity measurement of pipe insulation using lagged pipe apparatus.		
3		Determination of heat transfer coefficient under natural convection from a vertical cylinder.		
4		Determination of heat transfer coefficient under forced convection from a tube.		
5		Heat transfer from pin-fin apparatus (natural & forced convection modes).		
6		Determination of Stefan-Boltzmann constant.		
7		Determination of emissivity of a grey surface.		
8		Effectiveness of Parallel/counter flow heat exchanger		

REFRIGERATION AND AIR CONDITIONING LAB

S.No	Date	Name of the Experiment	Staff Signature	Remarks
1		Determination of COP of a refrigeration system		
2		Experiments on Psychometric processes.		
3		Performance test on a reciprocating air compressor.		

HEAT TRANSFER LAB

THERMAL CONDUCTIVITY – GUARDED HOT PLATE METHOD

Ex. no: 1

Date:

AIM:

To find the thermal conductivity of the specimen by two slabs guarded hot plate method.

SPECIFICATIONS:

- Specimen material = Asbestos
- Thickness of the specimen L = 24 mm= 12+12=24 mm
- Diameter of cylinder D = 150 mm

FORMULAE:

1. **HEAT INPUT:**

The power input to heater

$$\mathbf{Q = V \times I} \text{ in } \textit{Watts}$$

Where,

Q = heat input
V = volts
I = current in amps

2. **THERMAL CONDUCTIVITY(K):**

$$\mathbf{K} = \frac{(q \times L)}{A \times \Delta T} \quad \text{in } W/m\,K$$

- ★ Two specimen pieces, so one at the top and another one at the bottom.
- ★ Thermal conductivity of specimen $K = \frac{(K_1 + K_2)}{2}$

Where,

$$K_1 = \frac{(q \times L_1)}{A \times \Delta T_1} \qquad K_2 = \frac{(q \times L_2)}{A \times \Delta T_2}$$

- q= heat input in *watts*
- L= thickness of the specimen = 76.20 *mm*
- L_1 = lower specimen =12 *mm*
- L_2 = upper specimen =12 *mm*
- A = area of the specimen

$$\Delta T_1 = \frac{(T_2 + T_3)}{2} - T_7 \quad \text{in } K \text{ (LOWER SIDE)}$$

$$\Delta T_2 = \frac{(T_5 + T_6)}{2} - T_8 \quad \text{in } K \text{ (UPPER SIDE)}$$

PROCEDURE:

1. Switch on the unit, allows the unit to stabilize for about 15 to 25 minutes.
2. Now vary the voltmeter reading and note down the temperature T_1 to T_2 ammeter reading.
3. The average temperature of each cylinder is taken for calculation. The temperature is measured by thermocouples with input multipoint digital temperature indicator.

TABULATION:

THERMAL CONDUCTIVITY – GUARDED HOT PLATE METHOD

S.No	Heat Input 'W'			Main Heater ^0C	Ring Heater ^0C	Bottom Specimen Temperature ^0C		Top specimen Temperature ^0C		Water Outlet Temperature ^0C		Water Inlet Temperature ^0C	Thermal Conductivity K *(W/mK)*
	V	I	Q =V X I	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	

RESULT:

The thermal conductivity of the specimen is found to be **(K) = __________***W/mK***.**

THERMAL CONDUCTIVITY - LAGGED PIPE METHOD

Ex. no: 2

Date:

AIM:

To find the thermal conductivity of the specimen by lagged pipe method.

DESCRIPTION OF APPARATUS:

The apparatus consists of a guarded hot pipe and cold pipe. A specimen whose thermal conductivity is to be measured is saw dust between the hot and cold pipe thermocouple are attached to measure temperature in between the hot pipe and specimen pipe.

A multi point digital temperature with indicator selector switch is provided to not the temperature at different locators. An electric regulators is provided to not and vary the input energy to the heater.

The whole assembly in kept in an enclose with insulating material field all around to minimum to the heat loss

FORMULAE:

1. **HEAT INPUT:**

The power input to heater

$$\mathbf{Q = V \times I} \text{ in } \textit{Watts}$$

Where,

Q = heat input
V = volts
I = current in *amps*

2. **THERMAL CONDUCTIVITY(K):**

$$\mathbf{K} = \frac{\left(q \times \ln\left(\frac{r_2}{r_1}\right)\right)}{2\pi \times L \times \Delta T} \quad \text{in } W/m^2\,k$$

Where,

- q = Heat input supply in watts
- K = Thermal conductivity *W/m k*
- r_1= Radius of inner pipe = 25.40 *mm*
- r_2= Radius of outer pipe = 76.20 *mm*
- L = Length of the pipe =500 *mm*
- ΔT=Average outside temperature inner pipe- Average in side temperature outer pipe

$$\Delta T= \frac{(T_1+T_2+T_3)}{3} - \frac{(T_4+T_5+T_6)}{3} \quad \text{in } K$$

Where,

- T1,T2,T3=Outside temperature inner pipe
- T4,T5,T6=Inside temperature outer pipe

PROCEDURE:

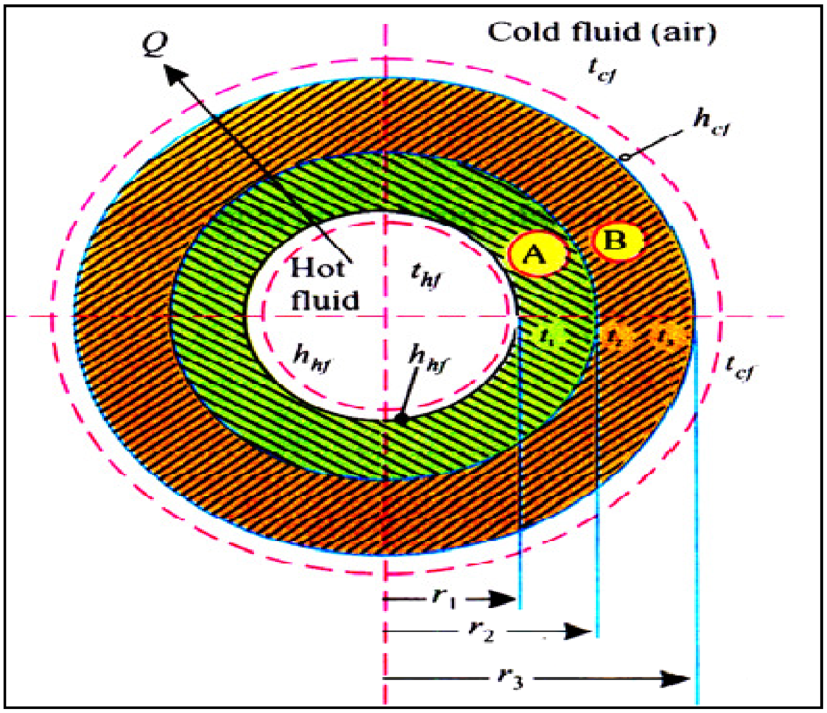
1. Switch on the unit, allows the unit to stabilize for about 15 to 25 minutes.
2. Now vary the voltmeter reading and note down the temperature T_1 to T_2 ammeter reading.
3. The average temperature of each cylinder is taken for calculation. The temperature is measured by thermocouples with input multipoint digital temperature indicator.

TABULATION:

THERMAL CONDUCTIVITY - LAGGED PIPE METHOD

Sl.No	Heat Input *'W'*			Outside Temperature Of Inner Pipe ^{0}C				Inside Temperature Of Inner Pipe ^{0}C				Thermal Conductivity(K) *'W/mK'*
	V	I	Q=V x I	T_1	T_2	T_3	AVG	T_4	T_5	T_6	AVG	

DIAGRAM:



RESULT:

The thermal conductivity of the specimen is found to be **(K) =** ___________*W/mK.*

DETERMINATION OF HEAT TRANSFER COEFFICIENT UNDER NATURAL CONVECTION FROM A VERTICAL CYLINDER

Ex. no: 3

Date:

AIM:

To determine the convective heat transfer co-efficient for heated vertical cylinder losing heat to the ambient by free or natural convection

DESCRIPTION OF APPARATUS:

Convection is a made of heat transfer where by a moving fluid transfer heat from a surface when the fluid movement is caused by density differences in the fluid due to temperature variation. It is called **FREE or NATURAL CONVECTION**.

The apparatus provides students with a sound introduction to the features of free convection heat transfer from a heated vertical rod. A vertical duct is fitted with a heated vertical placed cylinder. Around this cylinder air gets heated and becomes less dense causing in to rise. This turn gives to a continuous flow of air upwards in the duct. The instrumentation provides give the heat input and the temperature at different points on the heated cylinder.

SPECIFICATIONS:

- Length of cylinder L = 450 mm
- Diameter of cylinder D = 48 mm

FORMULA USED:

1. THEORETICAL HEAT TRANSFER CO-EFFICIENT (h_{the}):

$$h_{the} = \frac{(Nu.K)}{L} \quad \text{in } W/m^2\,K$$

Where,

- Nu = Nusselt number
- K = Thermal conductivity of air in $W/m\,K$
- L = Characteristics Length is height of the cylinder in mm

A. Nusselt number (Nu):

$$Nu = 0.53(Gr.Pr)^{1/4} \text{ for } GrPr < 10^5$$

$$Nu = 0.56(Gr.Pr)^{1/4} \text{ for } < 10^5 GrPr < 10^5$$

$$Nu = 0.13(Gr.Pr)^{1/3} \text{ for } < 10^8 GrPr < 10^{12}$$

Where,

- h = heat transfer co-efficient
- L = Characteristics Length is height of the cylinder in mm
- Gr = Grashoft number
- Pr = prandtl number of air

B. Grashoft number (Gr):

$$Gr = \frac{(L^3 \times \beta \times g \times \Delta T)}{V^2}$$

Where,

- h= heat transfer co-efficient
- L = Characteristics Length is height of the cylinder in mm
- g = Acceleration due to earth's gravity
- $\Delta T = T_s - T_a$ in K
- T_s=Average surface temperature in K
- T_a=Average ambient temperature in K
- $\beta = 1/T_f$ in K
- V^2=Kinematic viscosity of air at film temperature

C. Film temperature (T_f):

$$T_f = \frac{(Ts - Ta)}{2} \text{ in K}$$

Where,

- T_f = Film temperature in K
- T_s=Average surface temperature in K
- T_a=Average ambient temperature in K

*****NOTE*****

The following air properties data should be taken from the HMT Data book for film temperature (T_f)

Air properties

- Pr =Prandtl number
- K = Thermal conductivity of air in *W/m K*
- ʋ = Kinematic viscosity of air in
- ρ_a = Density of air

2. EXPERIMENTAL HEAT TRANSFER CO-EFFICIENT (h_{exp}):

The power input to heater

$$\mathbf{q = V \times I} \text{ in Watts}$$

Where,

Q =heat input

V =volts

I = current in amps

$$h_{exp} = \frac{q}{A \times \Delta T}$$

Where,

A = Area of pipe

ΔT = Ts-Ta in K

ΔT = Tube temperature -Air temperature in K

TABULATION:

DETERMINATION OF HEAT TRANSFER COEFFICIENT UNDER NATURAL CONVECTION FROM A VERTICAL CYLINDER

Sl.No	Heat Input 'W'			Surface Temperature °C								Ambient Temperature °C	Heat Transfer Co-Efficient Of Theoretical (h_{the})	Heat Transfer Co-Efficient Of Experimental (h_{exp})
	V	I	Q=V*I	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_S	T_8		

RESULT:

1. The theoretical heat transfer co-efficient is found to be $\mathbf{h_{the}}$= ____________ W/m^2K.

2. The experimental heat transfer co-efficient is found to be $\mathbf{h_{exp}}$= ____________ W/m^2K.

DETERMINATION OF HEAT TRANSFER COEFFICIENT UNDER FORCED CONVECTION FROM A TUBE.

Ex. no: 4

Date:

AIM:

1. To determine the convective heat transfer co-efficient for a horizontal pipe through which air flow under forced convection
2. To find the theoretical heat transfer co-efficient for the above condition and to compare with the experimental value.

SPCIFICATION:

- Inside diameter of the pipe (D)=25 mm
- Orifice diameter (d_o) =20 mm
- Length of the pipe (L) =400 mm

PROCEDURE:

1. Switch on the main and on the blower.
2. Adjust the regulator to any desired power into input to heater.
3. Adjust the position of the valve to any desired flow rate of air.
4. Wait till steady state temperature is reached.
5. Note down the manometer reading h_1, h_2 and temperatures T_1, T_2, T_3, T_4, T_5, T_6 and T_7.
6. Take the voltmeter and ammeter reading.
7. Adjust the position of the valve and vary the flow rate of air and repeat the experiment.
8. For various valve openings and for various power inputs the readings may be taken to repeat the experiments.

FORMULA USED:

1. THEORETICAL HEAT TRANSFER CO-EFFICIENT (h_{the}):

$$h_{the} = \frac{(N_u.K)}{D} \quad \text{in } W/m^2\,K$$

Where,

Nu = Nusselt number

K = Thermal conductivity of air in *W/m K*

D = Diameter of the tube in *mm*

A. REYNOLDS NUMBER (Re):

$$Re = \frac{(UD)}{\gamma}$$

Where,

U = velocity of flow in *m/s*

D =Diameter of the specimen =25 *mm*

B. NUSSELT NUMBER (Nu):

$$Nu = C.Re^n.Pr^{1/3}$$

Where,

Re =Reynolds number

Pr =Prandtl number

For,

Re = 0.4 to 4.0	C = 0.989 & n = 0.33
Re = 4 to 40	C = 0.911 & n = 0.385
Re = 40 to 4000	C = 0.683 & n = 0.466
Re = 4000 to 40000	C = 0.293 & n = 0.618
Re = 40000 to 400000	C = 0.27 & n = 0.805

C. VELOCITY OF FLOW (U):

$$U = (Q/A) \quad \text{in } m^3/sec$$

Where,

Q = Discharge of air m^3/sec

A = Area of pipe = ΠDL

D. DISCHARGE OF AIR (Q):

$$Q = C_d \times a_o \times \sqrt{(2g.H_{air})} \quad \text{in } m^3/sec$$

Where,

C_d = Co-efficient of discharge = 0.62

a_o = Area of orifice = $\pi/4 \times d_0^2$

H_{air} = Heat of air = $\left(\frac{\rho_w - \rho_{air}}{\rho_{air}}\right) \times H_m$

*****NOTE*****

The following air properties data should be taken from the HMT Data book for mean temperature (T_m).

Air properties

Pr =Prandtl number

K = Thermal conductivity of air in W/m K

υ = Kinematic viscosity of air

ρ_a = Density of air

E. MEAN TEMPERATURE:

$$T_m = \frac{(T_s + T_a)}{2} \quad \text{in } °C$$

Where,

T_s – Surface temperature of tube °C

T_a – Temperature of air °C

F. TEMPERATURE OF SURFACE OF THE PIPE (Ts):

$$T_s = \frac{(T_2 + T_3 + T_4 + T_5 + T_6)}{5} \quad \text{in } K$$

G. AIR TEMPERATURE (Ta):

$$T_a = \frac{(T_1 + T_7)}{2} \quad \text{in } K$$

2. EXPERIMENTAL HEAT TRANSFER CO-EFFICIENT (h_{exp}):

The power input to heater

$$\mathbf{q = V \times I} \text{ in Watts}$$

Where,

q=heat input

V=volts

I= current in amps

$$\mathbf{h_{exp} = \frac{q}{A \times \Delta T}}$$

Where,

A = Area of pipe, $\Delta T = T_a - T_s$ in K

ΔT = Air Temperature-Tube temperature in K

TABULATION:

Determination of heat transfer coefficient under forced convection from a tube.

Sl.No	Heat Input *'W'*			Manometer Reading *'m'*			Air Temperature *°C*		Tube Temperature *°C*						Heat Transfer Co-Efficient Of Theoretical (h_{the})	Heat Transfer Co-Efficient Of Experimental (h_{exp})
	V	I	Q	h_1	h_2	h	T_1	T_7	T_2	T_3	T_4	T_5	T_6	T_S		

RESULT:

Thus the convective heat transfer co-efficient for convection

1. Theoretical heat transfer co-efficient is $\mathbf{h_{the}}$ **=** ___________ W/m^2K.

2. Experimental heat transfer co-efficient is $\mathbf{h_{exp}}$ **=** ___________ W/m^2K.

HEAT TRANSFER FROM A PIN- FIN APPARATUS

Ex. no: 5

Date:

AIM:

To calculate the value of heat transfer coefficient from the fin for forced convection.

INTRODUCTION:

Extended surfaces of fins are used to increase the heat transfer rate from a surface to a fluid wherever it is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid.

The use of this is variety of shapes. Circumferential fins around the cylinder of a motor cycle engine and fins attached to condenser tubes of a refrigerator are a few familiar examples.

It is obvious that a fin surface sticks out from the primary heat transfer surface. The temperature difference with surrounding fluid will steadily diminish as one move out along the fin. The design of the fins therefore required knowledge of the temperature distribution in the fin. The main objective of this experimental set up is to study temperature distribution in a simple pin fin.

APPARATUS:

A brass fin of circular cross section in fitted across a long rectangular duct. The other end of the duct is connected to the suction side of a blower and the air floes past the fin perpendicular to the axis. One end of the fin projects outside the duct and is heated by a heater. Temperature at five points along the Length of the fin. The air flow rate is measured by an orifice meter fitted on the delivery side of the blower.

EXPERIMENTAL PROCEDURE:

To study the temperature distribution along the length of a pin fin natural and forced convection, the procedure is as under

FORCED CONVECTION**:**

1. Stat heating the fin by switching ON the heater and adjust dimmer stat voltage 80 to 100 volts.
2. Start the blower and adjust the difference of level in the manometer with the help of gate valve.
3. Note down the thermocouple reading (1) to (5) at a time interval of 5 minutes.
4. When the steady state is reached, record the final reading (1) to (5) and also record the ambient temperature reading (6).

5. Repeat the experiment with different manometer readings.

RESULT FROM EXPERIMENTAL:

FORCED CONVECTION:

1. Plot the temperature distribution along the length of the fin from observed readings
2. Calculate the value of m and obtain the temperature at various locations along the length of fin by using equation and plot them.
3. Calculate Re and Pr and obtain Nu from equation
4. Calculate the value of heat transfer rate from the fin and fin effectiveness by using equation.
5. Repeat the same procedure for all other sets of observations.

Specification:

- fin material = brass
- Length of the fin (L_f) =150mm = 0.15m
- diameter of the fin (d_f) =12mm = 0.012m
- diameter of the pipe (d_p) =38mm =0.038m
- diameter of the orifice (d_o) =20mm = 20mm
- with of the duct (w) = 150mm =0.15m
- breath of the duct (b) = 100mm =0.1m
- co- efficient of discharge (c_d) = 0.62
- density of water (ρ_w) =1000 Kg/m^3
- density of Air (ρ_a) =1.165 Kg/m^3

1. HEAT CONVECTIVE TRANSFER CO-EFFICIENT (h_c):

$$h_c = \frac{Nu.k}{D} \quad \text{W/m}^2\,\text{K}$$

Where,

Nu =Nusselt number

K = thermal conductivity of air in W/mK

D = diameter of the fin in m

A. NUSSELT NUMBER (Nu):

$$\mathbf{Nu = C.Re^{n}.Pr^{1/3}}$$

Where

Re = 0.4 to 4.0	C = 0.989 & n = 0.33
Re = 4 to 40	C = 0.911 & n = 0.385
Re = 40 to 4000	C = 0.683 & n = 0.466
Re = 4000 to 40000	C = 0.293 & n = 0.618
Re = 40000 to 400000	C = 0.27 & n = 0.805

B. **REYNOLDS NUMBER (Re):**

$$Re = \frac{V_a\, d_f}{v}$$

Where,

V_a = velocity of air in duct in m/s

d_f = diameter of fin in m

v = kinematic viscosity m^2/s

C. **Velocity of air in duct (V_a):**

$$V_a = \frac{V_o \times \frac{\pi}{4} \times d_o}{w \times b}\ m/s$$

Where,

V_o=velocity at orifice

d_o = dia of orifice

D. **Velocity at orifice (V_0):**

$$V_o = c_d \times \sqrt{2gh}\left(\frac{\rho_w - \rho_a}{\rho_a}\right) \times \left(\frac{1}{\sqrt{1} - \beta_4}\right) m/s$$

Where,

$$\beta = \frac{d_o}{d_p} = \frac{\text{dia of orifice}}{\text{dia of pipe}}$$

E. **Mean temperature T_m:**

$$T_m = \frac{T_s + T_a}{2}\ {}^\circ C$$

F. **surface temperature T_S:**

$$T_s = \frac{T_1 + T_2 + T_3 + T_4 + T_5}{5}\ {}^\circ C$$

NOTE

The following air properties data should be taken from the HMT Data book for surface temperature (T_s)

Pr = Prandtl number of air

K = Thermal conductivity of air
N = Kinematic viscosity
ρ = density of air

1. **To find m**

$$m=\sqrt{\frac{h_c\times p}{K\times A}}$$ from HMT D.B.Pg.No:50

Where,

h_c= convective heat transfer co-efficient in W/m^2 K
p =perimeter = $\pi\times d_f$
k=110.7 W/m^2 K (brass)
A= cross section area of fin = $\pi/4 \times d_f^2$

2. **Effectiveness of fin (Ɛ)**

$$\mathcal{E}=\sqrt{\frac{K\times P}{h_c\times A}}\tan h\ mL$$

3. **Efficiency of fin (η)**

$$\eta=\frac{\tanh\ (m(L-x)}{K\times A}\times 100\ \%$$

4. **Temperature distribution:**

$$\frac{T-T_a}{T_b-T_a}=\frac{\cosh\ (m(L-x)}{\cosh\ (mxL)}$$

Where,

Ta- Ambient Temperature °C
T_b- base temperature °C

TEMPERATURE DISTRIBUTION:

Given thermocouple distance:

S.NO	Experimental temperature °C	Calculated Temperature °C	Distance of the thermo couple 'm'
1	T_1=		0.02
2	T_2=		0.05
3	T_3=		0.08
4	T_4=		0.11
5	T_5=		0.14

TABULATION:

HEAT TRANSFER FROM A PIN- FIN APPARATUS

Sl.No	Heat Input ‘W’			Fin Temperature °C						Manometer Reading ‘m’			Ambient Temperature °C	Effectiveness (E)	Efficiency (η)
	V	I	Q	T_1	T_2	T_3	T_4	T_5	T_{AVG}	h_1	h_2	H	T_a		

RESULT:

Heat transfer co-efficient, effectiveness and efficiency are calculated

1. Heat transfer co-efficient ________________________ $W/m^2\ K$
2. Effectiveness of the fin __________________.
3. Efficiency of the fin _________________%

STEFAN – BOLTZMANN APPARATUS

Ex. no: 6

Date:

AIM:

To find value of Stefan – Boltzmann constant for radiation heat transfer.

STEFAN – BOLTZMANN LAW:

Stefan – Boltzmann law state that the total emissive power of a perfect black body is proportional to fourth power of the absolute temperature.

$$E_b = \sigma T^4$$

- σ - Stefan – Boltzmann constant

SPECIFICATIONS:

- Material of the disc & hemisphere = Copper
- Diameter of the disc = 20 mm
- Mass of the disc = 5 grams $=5\times10^{-3}$ Kg
- Specific heat capacity of the copper = 383 J/ Kg K

PROCEDURE:

1. Switch on the heater; heat the water in the tank about 80 °C.
2. Allow the hot water to flow through the hemisphere and allow the hemisphere to reach a steady temperature.
3. Note down the temperature T_1 and T_2. Average of these temperatures is the hemisphere temperature (T_{avg})
4. Refit the disc at the bottom of the hemisphere and start the stop clock.
5. The raise in temperature T_3 with respect to time is noted.

FORMULAE:

Heat Equation:

Rate of Change of Heat Capacity of the Disc = Net Energy Radiated on the Disc

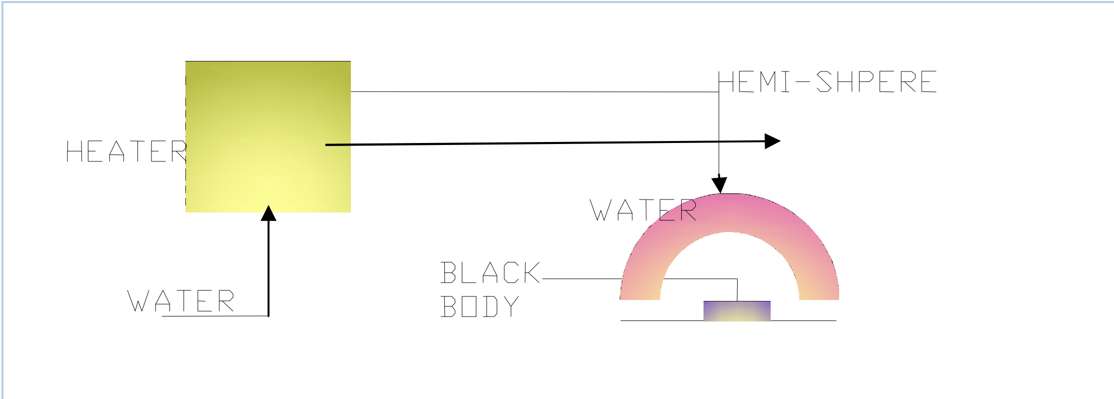
1. $m \times C_p \frac{dT}{dt} = \sigma A_D (T_{avg}^4 - T_D^4)$

2. $\sigma = \frac{m \times Cp \frac{dT}{dt}}{A_D \left(T_{avg}^4 - T_D^4\right)}$ $\mathbf{\left(W/m^2 K^4\right)}$

3. $T_{avg} = (T_1+T_2+T_3) / 3$ **(K)**

Specification:

- σ - Stefan – Boltzmann constant
- m - Mass of the disc in kg
- C_P - Specific heat capacity of the copper = 383 J/ Kg K
- dT -Change in Temperature in (K)
- dt - Change in Temperature in seconds
- A_D - Area of the disc
- T_{avg} - Average Temperature
- T_D - Temperature of the disc before inserting into the plate



TABULATION:

STEFAN – BOLTZMANN APPARATUS

Temperature of the disc before inserting into the plate T_D =

S.no	Hemisphere (Left side) (T_1) °C	Hemisphere (Right side) (T_2) °C	hot water Temperature (T_4) °C	Avg. temperature of hemisphere (T_{avg}) K	Stefan – Boltzmann constant $W/m^2 K^4$

Temperature Time Responses:

Time (sec) **t**	Temperature of the disc ($\mathbf{T_3}$) **°C**	Temperature in **K**
0		
20		
30		
40		
60		
80		
100		

GRAPH:

dT vs dt

RESULT:

Stefan – Boltzmann constant is found to be ----------------------- $W/m^2\ K^4$.

DETERMINATION OF EMISSIVITY OF TEST SURFACE

Ex. no: 7

Date:

AIM:

To measure the emissivity of the test plate surface

DESCRIPTION OF APPARATUS:

An ideal block surface is one, which absorbs the radiation falling on it. Its reflectivity and transivity is zero. The radiation emitted per unit time per unit area from the surface of the body is called emissive power.

The emissive power of a body to the emissive power of black body at the same temperature is known as emissivity of that body.

For a black body absorbvity is 1, emissivity depends on the surface temperature and the nature of the surface.

The experimental set up consists of two circular aluminum plates identical in size and is provided with heating coils at the bottom. The plates are mounted on thick asbestos sheet and kept in an enclosure so as to provide undisturbed natural convection surrounding. The heat input to the heaters is varied by two regulators and is measured by an ammeter and voltmeter. The temperatures of the plates are measured by thermocouples. Each plate is having three thermocouples; hence an average temperature may be taken. One thermocouple is kept in the enclosure to read the chamber temperature.

One plate is blackened by a layer of enamel of black paint to from the idealized black surface whereas the other plate is the test plate. The heat dissipation by conduction is same in both cases.

SPECIFICATION:

- Diameter of test plate and black surface = 150mm

PROCEDURE:

1. Connect the unit to the supply and switch on the unit.
2. Keep the thermocouple selector switch in first position.
3. Keep the toggle switch in position (1.**power will be feed to block plate &** position 2. **power will be feed to test surface plate) allow** the unit to stabilize. Ascertain the power inputs to the block and test surfaces are at set values i.e. equal.
4. Turn the thermocouples selector switch clockwise step by step note down the temperatures indicated by the temperature indicator from channel 1 to 7.
5. Tabulate the readings for various power inputs repeat the experiment.
6. After the experiment is over turn off both the energy regulations 1&2.

FORMULAE:

1. **HEAT INPUT**:

$$q = V \times I \text{ in watts}$$

Where,

Voltmeter = V volts
Ammeter = I amps

2. **AVERAGE BLACK BODY TEMPERATURE:**

$$T_b = \frac{T1+T2+T3}{3}\ °C$$

3. **AVERAGE TEST SURFACE TEMPERATURE:**

$$T_t = \frac{T4+T5+T6}{3}\ °C$$

4. **EMISSIVITY OF TEST SURFACE:**

Heat input to block surface=heat input to test surface

$$q = E_b \times A_b \times (T_b^4 - T_a^4) = E_t \times A_t \times (T_t^4 - T_a^4)$$

Since the power input is same for both block and test surface is also same, knowing the Ɛb=1

$$E_t = E_b \frac{(T_b^4 - T_a^4)}{(T_t^4 - T_a^4)}$$

Where,

- ε_t =emissivity of block surface
- ε_b =emissivity of block surface=1
- T_b = Average block body temperature in K
- T_t = Average test surface temperature in K

TABULATION:

DETERMINATION OF EMISSIVITY OF TEST SURFACE

Sl.No	Heat Input 'W'			Black Body Temperature °C				Test Body Temperature °C				Ambient Temperature °C	Emissivity Of Test Surface
	V	I	Q	T_1	T_2	T_3	T_B	T_4	T_5	T_6	T_T	Ta	E_t

RESULT:

The emissivity of a test surface is --------------------------------------.

EFFECTIVENESS OF PARALLEL AND COUNTER FLOW HEAT EXCHANGER

Ex. no: 8

Date:

AIM:

To determine LMTD, the effectiveness and the overall heat transfer co-efficient for parallel and counter flow heat exchange.

Apparatus required:

- Heat exchange test rig
- Supply of hot and cold water
- Stop watch
- Measuring jar

Specification:

1. Inner tube material – copper
   Inner diameter, d_i = 9.5mm
   Outer diameter, d_o = 12.5mm
2. Outer tube material – galvanized iron
   Inner diameter, D_i = 28.5mm
   Outer diameter, D_o = 32.5mm
3. Length of heat exchanger L = 1500mm

Formula:

1. **HEAT TRANSFER FROM HOT WATER:**

$$q_b = m_h \times C_{ph} \times (T_{hi} - T_{ho}) \text{ in W}$$

Where,

- M_h = mass flow rate of hot water
- C_{ph} = specific heat of water = 4187 J/kg K
- T_{ho} = hot water outlet temperature K
- T_{hi} = hot water inlet temperature K

2. HEAT GAINED BY COLD WATER:

$$q_c = m_c \times C_{pc} \times (T_{co} - T_{ci}) \text{ in W}$$

Where,

- **Mc** = mass flow rate of cold water
- C_{ph} = specific heat of water = 4187 J/kg K
- T_{co} = temperature of cold water outlet in K
- T_{ci} = temperature of cold water inlet in K

3. AVERAGE HEAT TRANSFER(Q_{AVG}):

$$q_{avg} = (q_c + q_h)/2 \quad \text{in W}$$

4. LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD):

$$\text{LMTD} = \frac{(\emptyset_1 - \emptyset_2)}{\ln(\frac{\emptyset_1}{\emptyset_2})} \quad \text{in K}$$

Where,

$\emptyset_1 = T_{hi} - T_{ci}$, $\emptyset_2 = T_{ho} - T_{co}$, **for parallel flow.**

$\emptyset_1 = T_{hi} - T_{co}$, $\emptyset_2 = T_{ho} - T_{ci}$, **for counter flow**.

5. OVER ALL HEAT TRANSFER CO-EFFICIENT BASED ON OUTSIDE SURFACE AREA OF INNER TUBE:

$$U_O = \frac{(q_{avg})}{(A_O \times LMTD)} \text{ in } (W/m^2K)$$

Where,

$$A_O = \pi d_o l \quad \text{in } m^2$$

6. EFFECTIVENESS:

a) $$E = \left(\frac{(m_h \times C_h)}{C_{min}}\right) \left(\frac{(T_{hi} - T_{ho})}{(T_{hi} - T_{ci})}\right)$$

For $m_h \times C_h = C_{min}$

b) $$E = \left(\frac{(m_c \times C_c)}{C_{min}}\right) \left(\frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})}\right)$$

For $m_c \times C_c = C_{min}$

Tabulation I: Parallel Flow

S.No	Hot water collect for 20 sec 'ml'	Cold water collect for 20 sec 'ml'	Temperature of hot Water ° C		Temperature of cold Water ° C		Logarithmic mean Temperature Difference (LMTD) (K)	Over all heat transfer co-efficient (W/m^2K)	Effectiveness
			Inlet T_{hi} (T_1) °C	Outlet T_{ho} (T_2) °C	Inlet T_{ci} (T_3) °C	Outlet T_{co} (T_4) °C			

Tabulation II: Counter Flow

S.No	Hot water collect for 20 sec 'ml'	Cold water collect for 20 sec 'ml'	Temperature of hot Water ° C		Temperature of cold Water ° C		Logarithmic mean Temperature Difference (LMTD) (K)	Over all heat transfer co-efficient (W/m^2K)	Effectiveness
			Inlet T_{hi} (T_1) °C	Outlet T_{ho} (T_2) °C	Inlet T_{ci} (T_4) °C	Outlet T_{co} (T_3) °C			

*****NOTE*****

'a' can be used $(m_h \times C_h) < (m_c \times C_c)$
'b' can be used $(m_h \times C_h) > (m_c \times C_c)$

<u>Result:</u>

LMTD, Effectiveness and the overall heat transfer co-efficient of parallel & counter flow are calculated.

Flow type	Logarithmic mean temperature difference (LMTD) '(K)'	Over all heat transfer co-efficient based on outside surface area of inner tube '(W/m^{2}K)'	Effectiveness
Parallel flow			
Counter flow			

REFRIGERATION

AND

AIR CONDITIONING LAB

EXPERMENTS ON REFRIGERATION SYSTEM

Ex. no: 9

Date:

Aim:

To determine the (i) Experimental COP, (ii) Carnot COP, (iii) Relative COP of a refrigeration system.

Apparatus required:

1. Refrigeration test rig
2. stop watch

Procedure:

1. Switch on the mains and switch on the fan motor and then compressor motor.
2. Allow the plant to run to reach steady conditions. Take readings for every 5 minutes to know the steady state.
3. Observe the readings in compressor motor energy meter. Pressure gauges and thermocouple and record it is tubular form.
4. Switch off the plant after experiment is over by switching off the compressor motor first. Allow the fan motors to run for 10 minutes and then switch off.

Abbreviation and notation:

P_1= pressure of the refrigerant before the compressor.
P_2= pressure of the refrigerant after the compressor.
P_3= pressure of the refrigerant before the expansion valve.
P_4= pressure of the refrigerant after the expansion valve.

T_1=temperature of the refrigerant before compression.
T_2=temperature of the refrigerant after compression.
T_3=temperature of the refrigerant before expansion.
T_4=temperature of the refrigerant after expansion.

Conversion:

Convert all the pressure in PSIG to bar (multiply the value in PSIG by 0.06894 and add 1.013 to convert to bar abs)

FOR EXAMPLES:

P1 = (25×0.06894) +1.0134 =2.736 bar
P2 = (195×0.06894) +1.0134 =14.456 bar
P3 = (150×0.06894) +1.0134 =11.354 bar
P4 = (20×0.06894) +1.0134 =2.391 bar

FORMULA USED:

1. **Experimental COP**:

$$\textbf{Experimental COP:} = \frac{\textbf{Actual Refrigeration effect}}{\textbf{work done}}$$

A. **Actual Refrigeration effect (RE)** = $m_w \times C_p \times \Delta T / \Delta t$ in KW

Where,

- m_w =mass of water in kg
- C_p =specific heat of water =4.186 KJ/ kg K
- ΔT =Temperature drop in the water
- Δt = Time for fall in temperature of water 5 minutes (or)water after decreasing 5°C
- Work done = Energy consumed by the compressor motor to be found out from the energy meter

B. Input energy (or) work done:

$$\textbf{Work done} = \frac{N \times 3600}{t \times x} \text{ KW}$$

Where,

X=energy meter constant=3200 impulse/KW hr.

t= time taken in sec. for 10 flickering of energy meter reading

2. CARNOT COP

$$\textbf{Carnot COP} = \frac{T_L}{T_H - T_L}$$

Where,

T_L=Lower temperature to be maintained in the evaporator in absolute units °k

$T_L = p_{min} = (P_1+P_4)/2$;

T_H=Higher temperature to be maintained in the Condenser in absolute units °k

$T_H = p_{max} = (P_2+P_3)/2$;

3. Relative COP

$$\text{Relative COP} = \frac{\text{Actual COP}}{\text{Carnot COP}}$$

EXPERMENTS ON REFRIGERATION SYSTEM

TABULATION I:

S.No	Quantity of Water in Tank ‘Kg’	Initial Temperature of Water T_{5i} ^{O}C	Final Temperature of Water T_{5f} ^{O}C	Temperature ^{O}C				Pressure PSI				Time taken for 5^{o} falling Temperature T ^{O}C	Number of flickering in Energy meter light ‘N’	Time taken for N flickering t_f
				T_1	T_2	T_3	T_4	P_1	P_2	P_3	P_4			

TABULATION II:

Actual COP	Carnot COP	Relative COP

Result:

The COP of the Refrigeration system are determined and tabulated.

1. Experimental (Actual) COP = ________________________.
2. Relative COP = _________________________.
3. Carnot COP =________________________

DETERMINATION OF COP OF AIR CONDITIONING SYSTEM

Ex. no: 10

Date:

Aim:

To conduct performance test on Air conditioning test rig to determine the co-efficient of performance.

Apparatus required:

1. Air conditioning test rig
2. Stop watch

Specification:

- Orifice diameter = 50mm
- Refrigerant R =22
- Energy meter constant = 3200 impulse/KW hr
- Density of air = 1.184 at 25°C

Procedure:

1. Switch on the mains.
2. Switch on the conditioning unit.

Note down the following:

a) Pressure p_1, p_2, p_3 and p_4 from the respective pressure gauge.
b) Note the corresponding temperature T_1, T_2, T_3, and T_4 at the respective state points.
c) Monometer readings.
d) DBT and WBT of atmosphere air.
e) DBT and WBT of the conditioned air.

Abbreviation and notation:

P_1= pressure of the refrigerant before the compressor.
P_2= pressure of the refrigerant after the compressor.
P_3= pressure of the refrigerant before the expansion valve.
P_4= pressure of the refrigerant after the expansion valve.

T_1=temperature of the refrigerant before compression.
T_2=temperature of the refrigerant after compression.
T_3=temperature of the refrigerant before expansion.
T_4=temperature of the refrigerant after expansion.

DBT = Dry bulb temperature
WBT=Wet bulb temperature

FORMULA USED:

1.COP OF AIR CONDITIONER:

$$\text{Cop of air conditioner} = \frac{\text{Refrigeration effect}}{\text{input energy}}$$

A. Refrigeration effect by Air Conditioner (RE):

$$(RE) = m\times(h_1-h_2) \text{ in KW}$$

Where,

- h_1= enthalpy of air at ambient condition
- h_2= enthalpy of conditioned air
- h_1&h_2 are calculated using DBT. WBT in psychometric chart
- m- Mass flow rate of air

B. Mass flow rate of air:

$$m=C_d\times\rho\times Q \qquad \text{‘kg/sec’}$$

Where,

- Q=volume flow rate of air = A×V m^3/sec
- ρ = density of air = 1.162 kg/m^3
- C_d= 0.65

C. Volume flow rate of air:

$$Q=A\times V$$

Where,

A- Area of orifice = $\pi/4 \times d_o^2$

v- Air velocity =$\sqrt{2gH_a}$

$H_a=\left(\frac{\rho_w-\rho_a}{\rho_a}\right)\times H_m$

ρ_{air}=1.165
refer HMT Data Book

H_m=Manometer pressure difference

D. Input energy or work done by the compressor:

$$\text{Input energy}=\frac{N\times3600}{t\times x} \text{ kW}$$

Where,

- X=energy meter constant=3200 impulse/ kW hr.
- t= time taken in sec for 10 revolutions of energy meter reading

2. CAPACITY OF THE AIR CONDITIONER

Capacity=refrigeration effect/3.5

3. CARNOT COP

$$\textbf{Carnot COP} = \frac{T_L}{T_H - T_L}$$

Where,

- T_L=Lower temperature to be maintained in the evaporator in absolute units °K
- T_L=p_{min}= $(P_1+P_4)/2$;
- T_H=Higher temperature to be maintained in the Condenser in absolute units °K
- T_H=p_{max}= $(P_2+P_3)/2$;

Tabulation:
Expansion Valve:

S.No	Pressure PSI				Manometer Reading ‘mm’			Atmospheric Air ^{O}C		Conditional Air ^{O}C		Time take for 10 Impulse in Energy meter ‘t’	COP	
	P_1	P_2	P_3	P_4	h_1	h_2	H	$T_{1\,D}$ DBT	$T_{1\,W}$ WBT	$T_{2\,D}$ DBT	$T_{2\,W}$ WBT		ACTUAL	CARNOT

Capillary tube:

S.No	Pressure PSI				Manometer Reading ‘mm’			Atmospheric Air ^{O}C		Conditional Air ^{O}C		Time take for 10 Impulse in Energy meter ‘t’	COP	
	P_1	P_2	P_3	P_4	h_1	h_2	H	$T_{1\,D}$ DBT	$T_{1\,W}$ WBT	$T_{2\,D}$ DBT	$T_{2\,W}$ WBT		ACTUAL	CARNOT

Calculations:

Result:

The COP of the Air Conditioning system are determined and tabulated.

EXPANSION VALVE:

1. Experimental (Actual) COP ________________________.
2. Capacity Of the Air Conditioner____________________ tone
3. Carnot COP ________________________.

CAPILLARY TUBE:

1. Experimental (Actual) COP ________________________.
2. Capacity Of the Air Conditioner____________________ tone
3. Carnot COP ________________________.

TEST ON RECIPROCATING AIR COMPRESSOR

Ex. no: 11

Date:

Aim:

To conduct performance test on a two stage reciprocating air compressor to determine the volumetric and isothermal efficiency.

Apparatus required:

The test unit consisting of an air reservoir on air intake tank with an orifice and a **U** tube manometer, the compressor having pressure gauge.

Specification:

Compressor Modal: 2 stage reciprocating

Diameter of low pressure cylinder D_L	=101.6 mm
Diameter of high pressure cylinder D_H	=63.5 mm
Stroke length L	=69.85 mm
Speed of the compressor	=65 rpm
Diameter of orifice	=8.5 mm
Co-efficient of discharge of orifice (C_d)	=0.65
Tank capacity	=250 lit
Motor capacity	=3 HP

Procedure:

1. Close the outlet valve.
2. Fill up the manometer with water up to half level.
3. Start the compressor and observe the pressure developing slowly.
4. At a particular test, pressure outlet valve is opened slowly and adjuster so that pressure in tank and maintained constant.
5. Note down the reading as the observation table.

Formula used:

1. Volumetric efficiency

$$\eta_{vol} = \frac{V_a}{V_t} \times 100 \quad \%$$

Where,

V_a=actual volume of air compressed

V_t= Theoretical volume of air compressed

A. Actual volume of air compressed (V_a):

$$V_a = C_d \times A \times \sqrt{2gH} \quad m^3/sec$$

Where,

C_d= Co-efficient of discharge of orifice =0.65

A = orifice Area in m^2= $(\pi/4) \times d^2$

H= Air head causing flow

B. Air head causing flow (H):

$$H = h \times \left(\frac{\rho_w - \rho_a}{\rho_a}\right) \quad \text{in m}$$

Where,

h = head of water $=h_1-h_2$ in m

ρ_w =density of water =1000 kg/m^3

ρ_a =density of air =1.165 kg/m^3

C. Theoretical volume of air compressed (V_T):

$$V_T = \frac{\pi D_h^2 LN}{4 \times 60} \quad m^3/sec$$

Where,

D_h =Diameter of high pressure cylinder =63.5 m^3/sec

L = Stroke length =69.85mm

N = Speed of the compressor =65 rpm

2. Isothermal efficiency:

$$\eta_{Isothermal} = \frac{\text{Isothermal workdone}}{\text{Actual workdone}} \times 100 \quad \%$$

D. Iso. thermal workdone

$$= P_a \times V_a \times \ln(r) \quad \text{in Nm/sec (or) Watts}$$

Where,

P_a =Atmospheric pressure =1.01325×10^5 N/m^2

V_a = actual volume of air compressed in m^3/sec

$$r = \frac{P_a + P_g}{P_a}$$

P_g =delivery pressure (available in kg/cm^2 should be converted in to N/m^2)

E. **Actual workdone** =HP of the motor=3 HP

NOTE

$1 Kg/cm^2 = 0.9814$ bar

1 bar $= 1 \times 10^5$ N/m^2

Kg/cm^2 to N/m^2

$1 Kg/cm^2 = 98 \times 10^5$ N/m^2

1HP =745.699watts take it as 1HP=746watts

Tabulation:

TEST ON RECIPROCATING AIR COMPRESSOR

S.No	**Delivery Pressure or Gauge Pressure P_g** Kg/m^3	**Tank Pressure P_r** Kg/m^3	**U-tube manometer reading**				**Volumetric efficiency** η_{vol}	**Isothermal efficiency** $\eta_{isothermal}$
			h_1 'cm'	**h_2** 'cm'	**$h = (h_1-h_2)$** 'cm'	**$h = (h_1-h_2)$** 'm'		

RESULT:

Thus performance test on a two stage reciprocating air compressor is conducted

1. Volumetric efficiency ________________________%
2. Isothermal efficiency ________________________%