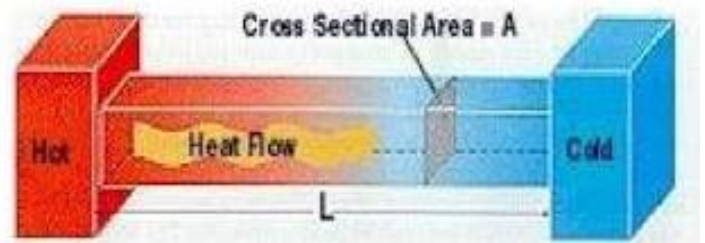
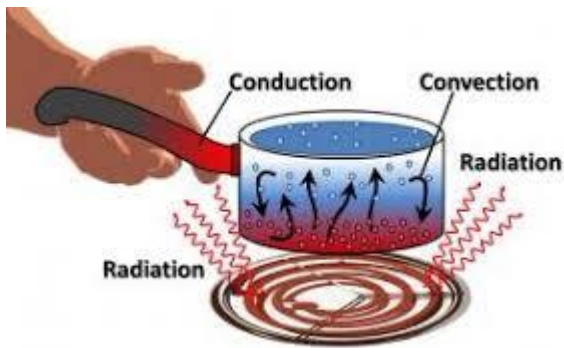


**BMS COLLEGE OF ENGINEERING,
BANGALORE – 19
DEPARTMENT OF MECHANICAL ENGINEERING
ACADEMIC YEAR 2017**



**ACADEMIC YEAR 2017
FUNDAMENTAL HEAT TRANSFER (FHT)
LABORATORY MANUAL,
V Semester Mechanical (V.T.U)**

NAME	
USN	



**BMS COLLEGE OF ENGINEERING,
BANGALORE – 19
DEPARTMENT OF MECHANICAL ENGINEERING**



CERTIFICATE

This is to certify that Ms./Mr. _____ bearing USN _____, a student of V Semester B.E.(Mech.), B.M.S. College of Engineering, affiliated to **Visvesvaraya Technological University, Belgaum** has successfully completed the lab work connected with the Fundamental Heat Transfer Lab as prescribed by the University during the Academic year 2017.

Total Marks obtained

15

Staff Incharge : _____

Date : _____

Prof.& Head
Dept. of Mechanical Engineering.

FUNDAMENTAL HEAT TRANSFER (FHT)

IA Marks: 15

No. of Practical Hrs. / Week: 02

Total No. of Practical Hrs: 26

PART – A

- 1. Thermal conductivity of Metal rod of constant area of cross section.**
- 2. Determination of Overall Heat Transfer Coefficient of a Composite Wall**
- 3. Pin - Fin apparatus**
- 4. Heat Transfer through Natural/Free Convection**
- 5. Heat Transfer through Forced Convection**

PART – B

- 1. Measurement of Surface Emissivity Apparatus**
- 2. Stefan-Boltzman Constant**
- 3. Heat Exchangers**
- 4. Vapour Compression Refrigeration**
- 5. Critical Heat Flux (Boiling Heat Transfer)**

Scheme of Examinations

Record : 5 Marks

Test : 5 marks (Write up + Conducting Experiment)

Viva Voce : 5 Marks

Total: 15 Marks

Please provide the team involved in preparing the Lab Manual.

1 Thermal Conductivity of Metal rod

1.1 Aim:- To determine the thermal conductivity of a given metal rod.

1.2 Apparatus:-

- Metal Rod (Copper)
 - Length of copper rod 200 mm
 - Diameter of the copper rod 50 mm
- Band Heater
- Cooling water jacket with water supply.
- Thermal Insulation Material covered on the circumference of the copper rod to restrict heat loss to atmosphere
- Thermocouples
- Rotameter for measuring the mass flow of the fluid.
- Digital temperature indicators and channel selector.

1.3 Theory:-

Heat transfer occurs in a material medium when ever temperature gradient exists. There are basically 3 modes of heat transfer and conduction is one of them. The amount of heat transfer per unit area is determined by Fourier's law, which states that

"The amount of heat flux (rate of heat transfer per unit area) is proportional to the normal temperature gradient at the surface."

$$q'' \propto \nabla T \quad (1)$$

$$q'' = \frac{\dot{Q}}{A}, \nabla T \cdot \hat{n} = \frac{\partial T}{\partial x_n} \quad (2)$$

$$q'' = -k \frac{\partial T}{\partial x_n} \quad (3)$$

where q'' is the heat flux (rate of heat transfer per unit area), $\dot{Q}$ is the rate of heat transfer, " A " is the area of cross section, ∇T is the normal temperature gradient and k is thermal conductivity of the material under consideration

and $\hat{n}$ represents the direction in which the heat flux transfer takes place. The negative sign in the eq(3) is in accordance to second law of thermodynamics.

The units of heat flux q'' is $\frac{W}{m^2}$, temperature gradient is $\frac{^{\circ}K}{m}$ and hence the units of thermal conductivity is given as $\frac{W}{m^{\circ}K}$.

Thermal Conductivity is a material property which signifies how fast and

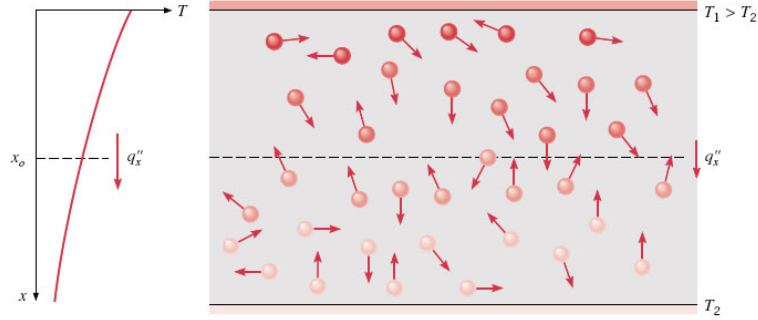


Figure 1: Thermal Conductivity of Metal Rod

efficiently a material can transfer heat. It is a function of temperature for solids and liquids, and of both temperature and pressure for gases. In many engineering applications, variation of thermal conductivity for a particular material over a range of temperature is very small. Hence the accuracy of measurements is not lost by making an approximation of constant thermal conductivity. The usual range of thermal conductivity for metals is in the range of (10-100) and their value is higher for pure metals.

1.4 Experimental Set up:-

Experimental setup consists of a metal rod (copper rod of circular cross section is used in this case). This copper rod is heated from one end with a heater and the opposite end is cooled with water flowing at a particular mass flow rate ($\dot{m}$). The surface of the rod is insulated with insulated material signifying that the heat transfer occurs only along the length of the rod. The set up is given as shown in the figure(??). It can be seen from the figure(??) that four thermocouples are used to measure the temperature on different locations of the copper rod and two thermocouples used for measure the cooling water inlet and exit temperatures. The mass flow rate of the cooling water is measured by rota-meter.

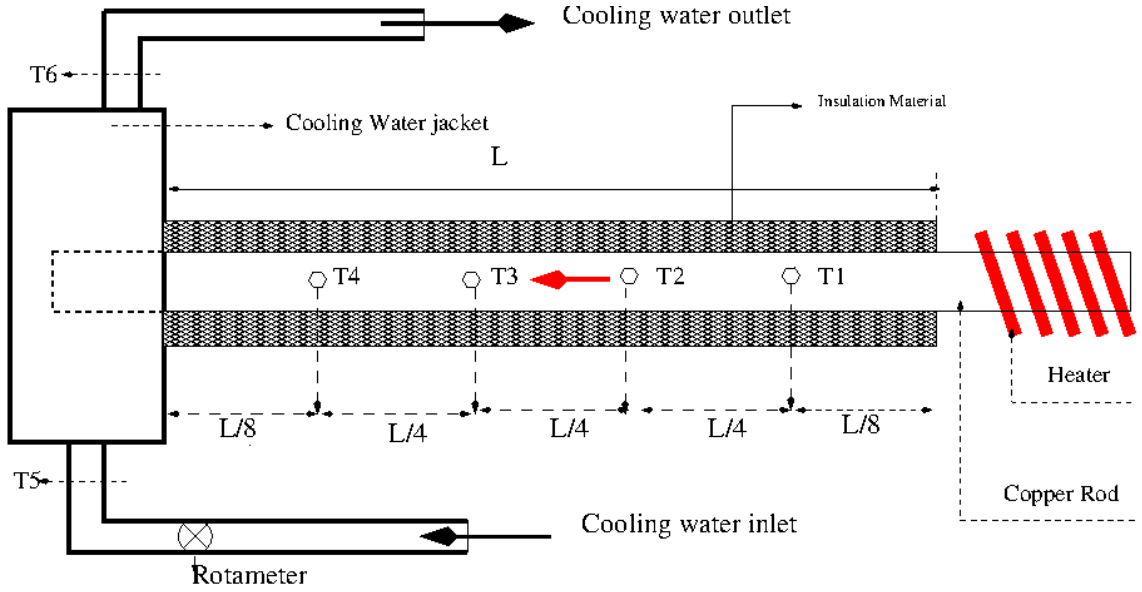


Figure 2: Experimental Setup for Thermal Conductivity of Metal Rod

1.5 Working Principle

Figure(??) depicts the setup to conduct the experiment to determine the thermal conductivity of a copper rod with constant area of cross section. The basic assumptions in this experiment area

1. The copper material is assumed to be pure and homogeneous in nature.
2. Heat flow is in steady state.
3. No loss from the circumference to the ambient atmosphere.

Writing the energy balance equation for the above set up

$$\dot{Q}_{inlet} = \dot{Q}_{outlet} + \dot{Q}_{loss} \quad (4)$$

- $\dot{Q}_{inlet}$ is the rate of heat transfer that is being transferred from the heater end.
- $\dot{Q}_{outlet}$ is the rate of heat that is being lost to the cooling water.
- $\dot{Q}_{loss}$ is the rate of heat lost to the atmosphere.

Assuming that there is no loss of heat to the ambient atmosphere

$$\dot{Q}_{inlet} = \dot{Q}_{outlet} = \dot{m}c_p\Delta T_{water} \quad (5)$$

where

- $\dot{m}$ is the mass flow rate of water into the cooling jacket in $\frac{kg}{s}$
- c_p specific heat of water $4.1789 \frac{kJ}{kg^\circ C}$
- $\Delta T_{water} = T_5 - T_6$, is the temperature difference between the cooling water entering and leaving out of the cooling jacket.
- T_5 & T_6 - temperatures entering and leaving the cooling jacket.

1.5.1 Determination of thermal conductivity along the Copper rod

After measuring the temperatures from the experiment at prescribed locations, the determination of the thermal conductivity is to be carried out. Temperatures are measured along four different locations $T_1 \cdots T_4$ which are located at equidistant from each other, measured along the length of the rod.

$$\nabla T = \hat{i} \frac{\partial T}{\partial x} + \hat{j} \frac{\partial T}{\partial y} + \hat{k} \frac{\partial T}{\partial z} \quad (6)$$

$$\text{in one dimension we can approximate it as } \nabla T = \hat{i} \frac{dT}{dx} \quad (7)$$

$$\text{which can be further approximated as } \frac{dT}{dX} = \lim_{\Delta x \rightarrow 0} \frac{\Delta T}{\Delta X} \quad (8)$$

using the relation in equation(6) we can find the gradient of temperature on the Copper rod. The gradient is approximated as $\frac{\Delta T}{\Delta x}$ where $\Delta T = T_{i+1} - T_i$ where $i = 1, 2, 3, 40$. The heat transfer rate is found using equation(5).

$$\dot{Q} = -kA \frac{\Delta T}{\Delta x} \quad (9)$$

$$A = \frac{\pi}{4} d^2 \quad (10)$$

$$-kA \frac{\Delta T}{\Delta x} = \dot{m} c_p \Delta T_{water} \quad (11)$$

The thermal conductivity can be found out by using equation(11). The typical value of co-efficient of thermal conductivity of Copper is $386 \frac{W}{m^\circ C}$. The measured value can be accepted with in the tolerance of $\pm 10\%$.

1.6 Operational Procedure

1. Switch on the mains and the console.
2. Open the valve at inlet for the cooling water and maintain a constant mass flow rate.

3. Switch on the heater.
4. Adjust the control or regulator of heat input to the required power.
5. wait about 10-15 minutes till the temperature indicators show constant,(Steady state attainment).
6. Note down the temperatures $T_1 \cdots T_4$ on the metal rod as shown in the digital temperature indicator.
7. Note down the cooling water inlet and exit temperatures T_5 & T_6
8. Measure the mass flow rate in the Rotameter.

1.7 Sample Calculation

Measurements :

$\dot{m} =$	$\frac{kg}{s}$	$T_1 =$	$^{\circ}C$
$T_5 =$	$^{\circ}C$	$T_2 =$	$^{\circ}C$
$T_6 =$	$^{\circ}C$	$T_3 =$	$^{\circ}C$
$\Delta T_{water} = T_5 - T_6 =$	$^{\circ}C$	$T_4 =$	$^{\circ}C$

1.7.1 Plots

Plot T_1, T_2, T_3, T_4 vs the distance of thermocouple and use a linear curve fitting for the curve and obtain the slope of the curve.

2 Boiling Heat Transfer

2.1 Aim:-

To estimate critical heat flux in Boiling heat transfer experiment's

2.2 Introduction

When heat is added to or absorbed from a solid submerged in a liquid heat transfer takes place to or from the liquid. If the solid surface temperature is higher than the saturation temperature of liquid, change of phase occurs in the liquid. This change of phase is called "Boiling if heat is added to the liquid and Condensation if the heat is absorbed from the liquid". Latent heat effects associated with phase changes are significant in boiling and condensation process. The change from liquid to vapour state due to boiling is sustained by heat transfer from the solid surface. In the process of boiling, motion of fluid occurs. Hence this kind of heat transfer is classified under convective mode of heat transfer. As the phase change occurs, the temperature of bulk of fluid is not affected by the heat transfer process. In both Boiling and Condensation processes large heat transfer rates can occur with small temperature differences. The parameters that are important in boiling heat transfer process are

- Latent Heat, h_{fg}
- Surface Tension, σ
- Density differences between the two phases, $\Delta\rho = \rho_l - \rho_g$

This difference in density results in buoyancy force proportional to $g(\rho_l - \rho_g)$. The combined effect of latent heat and the buoyancy force results in large heat transfer rates and heat transfer coefficients when compared to conventional heat transfer where there is no phase change.

2.2.1 Boiling Modes

Evaporation occurs at the submerged surface with surface temperature T_s exceeding the saturation temperature T_{sat} of the corresponding liquid at corresponding pressure. Heat is transfer from the solid to liquid according to Newton's law of cooling given by

$$q_s'' = h(T_s - T_{sat}) = h\Delta T_e \quad (12)$$

where ΔT_e , is the excess temperature. This process is characterized by the formation of vapor bubbles, which grow subsequently and detach from the surface. The dynamics of the bubble and its growth depends upon the many thermo-physical properties of the fluid. This dynamics of bubble formation and its growth affects the liquid motion near the surface. Based on the liquid motion this boiling can be classified as

- Pool Boiling - The bulk motion of the fluid is not present during the boiling process (Free Convection).
- Flow Boiling - Fluid motion is incorporated using external means.

Boiling may also be classified as

- Sub-cooled boiling
- Saturated boiling

depending on the temperature of the surrounding fluid, whether it is in sub-cooled state or in saturated state.

2.2.2 Pool Boiling or Saturated Pool Boiling

This kind of boiling is studied extensively, though there is steep increase in temperature near to the solid surface the temperature in most of the liquid is at saturated temperature corresponding to saturated pressure of that fluid. Bubbles get generated at the solid liquid interface and rise to and are transported across the liquid-vapor interface. This phenomenon can be well understood by the boiling curve.

2.2.3 Boiling Curve

2.3 Apparatus

The apparatus consists of

- Container - housing the test heater and Heater. The main heater is to heat the water there by controlling the T_{sat} . This heater is connected to the mains (Heater R_2)
- Test heater (Nichrome wire) is connected to the mains via dimmer-stat, ammeter and voltmeter to read current and voltage.
- Micro controller based peak detector - to measure the maximum current during the process. The heater wire can be viewed through a polycarbonate glass window. The schematic of the setup is shown in Figure().

2.4 Experimental Setup

The experimental setup is designed to study pool boiling phenomenon and to find the critical heat flux. The pool boiling over the Nichrome wire (test heater) can be visualized in the different regions up to critical heat flux where burnout of test heater occurs. The heat flux from the wire is slowly increased by increasing the voltage. During this process formation of bubbles, their growth and further the convection of the bubbles can be observed till the burnout occurs. The burnout indicates the occurrence of critical heat flux point.

2.4.1 Specifications of the setup

1. Heater for initial heating - Heater coil = 1.5 kW
2. Test heater (Nichrome wire) diameter = 0.2 mm
3. Test heater Length = 80 mm
4. Dimmer-stat = 1.5 A (Approximately) 260 V
5. Voltmeter = 0 - 200 V
6. Ammeter = 0 - 20 A
7. Thermometer = 0 – 100°C

2.4.2 Procedure

1. Fill the container with sufficient quantity of distilled water (approximately 1.5 lit).
2. Ensure the both the heaters (Main and the test heater) are completely submerged in the water.
3. Connect the spiffed gauge (Nichrome wire) of suitable length across the studs.
4. Switch on the auxiliary heater and maintain the bulk temperature of the water in the container to say ($50^{\circ}C$, $60^{\circ}C$, $70^{\circ}C \cdots 100^{\circ}C$)
5. Initialize the peak indicator by using the soft switch.
6. Switch on the test heater R_2 .

7. Slowly increase the voltage across the Nichrome wire and observe the boiling phenomenon on the wire.
8. Record the voltage and current at various intermediate stages. This can be utilized to find the resistance of the wire at varying temperature. Note down the resistance at room temperature of test wire- Resistance temperature co-efficient for Nichrome wire using this temperature T , from which the surface temperature can be calculated.
9. Increase the voltage till the burnout occurs, and record the voltage and current at this point.
10. Reduce the variac to zero voltage and press the "MAX" soft switch in the peak indicator and note down the maximum current.
11. Repeat the experiment by changing the water.

2.4.3 Precautions

1. Ensure the variac to zero voltage position before starting the experiment.
2. Ensure sufficient amount of distilled water in the container so that both heaters are immersed.
3. Do not touch the water or terminal after switching on the power.
4. Vary the variac in steps and allow sufficient time in between.
5. After attainment of critical heat flux condition decrease slowly the voltage and initialize to zero.

2.4.4 Constants

- Diameter of Nichrome wire 0.2 mm (36 swg)
- Length of Nichrome wire 80mm.
- Resistance of Nichrome wire, R_o at room temperature 4Ω

2.5 Calculations

- Temperature coefficient of resistance of Nichrome wire = 0. 00013
- Resistance of Nichrome wire at room Temperature $R_o = 4\Omega$

2.5.1 Formula

$$\begin{aligned} \text{Critical Heat flux, } q_c &= \frac{q}{\pi dl} \\ q &= V \times I \end{aligned} \quad (13)$$

d = diameter of test wire

l = length of testwire

Theroetical Critical heat flux, q_{ct}

$$q_{ct} = \frac{\pi}{24} h_{fg} \rho_v \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{\frac{1}{4}} \left(1 + \frac{\rho_v}{\rho_l} \right)^{\frac{1}{2}} \quad (14)$$

where

- h_{fg} = Latent heat of evaporation in $\frac{kJ}{kg}$
- ρ_v = Density of the vapor in $\frac{kg}{m^3}$
- ρ_l = Density of the liquid in $\frac{kg}{m^3}$
- σ = surface tension of the vapor-liquid in $\frac{N}{m}$

$$\Delta T = T_s - T_w \quad (15)$$

$$T_s = \frac{R_t - R_o}{R_o} \quad (16)$$

$$R_t = \frac{V}{I} \quad (17)$$

where

- T_s = Wire surface Temperature
- R_o = Resistance at Room Temperature, 4Ω
- T_w = Surrounding Water Temperature

Temperature $T^{\circ}C$	Density of liquid $\rho_l(\frac{kg}{m^3})$	Density of Vapor $\rho_v(\frac{kg}{m^3})$	Surface Tension $\sigma(\frac{N}{m})$	Latent heat of vaporization - $h_{fg}(\frac{kJ}{kg})$

3 Measurement of Surface Emissivity Apparatus

3.1 Aim

To determine the emissivity of the radiating surface.

3.2 Theory

All surfaces of finite temperature emit energy in the form of electromagnetic waves. This mode of heat transfer is called radiation, and doesn't need any material medium to transfer energy in the form of heat. The rate at which energy is released per unit area ($\frac{W}{m^2}$) is termed the surface emissive power E . The upper limit of surface emissive power is given by Stefan-Boltzmann law

$$E_b = \sigma T_s^4 \quad (18)$$

where T_s is the absolute temperature in $^{\circ}K$ of the surface and σ is Stefan-Boltzmann constant ($\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4}$). Such a surface is called ideal radiator or black-body. The heat flux emitted by a real surface is less than that of a blackbody at the same temperature and is given by

$$E = \epsilon \sigma T_s^4 \quad (19)$$

where ϵ is the radiative property of the surface termed the emissivity. The values of emissivity will be in the range of $0 \leq \epsilon \leq 1$. This property emissivity provides a measure of how efficiently a surface emits energy relative to a black body. It depends on the surface material and surface finish.

The heat flux transferred from a surface is given by

$$q''_{rad} = \frac{q}{A} = \epsilon E_b(T_s) - \alpha G = \epsilon \sigma (T_s^4 - T_{surr}^4) \quad (20)$$

where G is irradiation and T_{surr} represents the surrounding temperature. The equation(20) represents the difference between thermal energy that is released due to radiation emission and that which is gained due to radiation absorption.

3.3 Apparatus

- Black Body - A circular aluminum plate of (diameter $d = 200$ mm) coated with black Teflon, mounted on Asbestos Cement sheet.
- Test Body - A circular plate of diameter $d = 200$ mm with polished surface, mounted on Asbestos sheet.

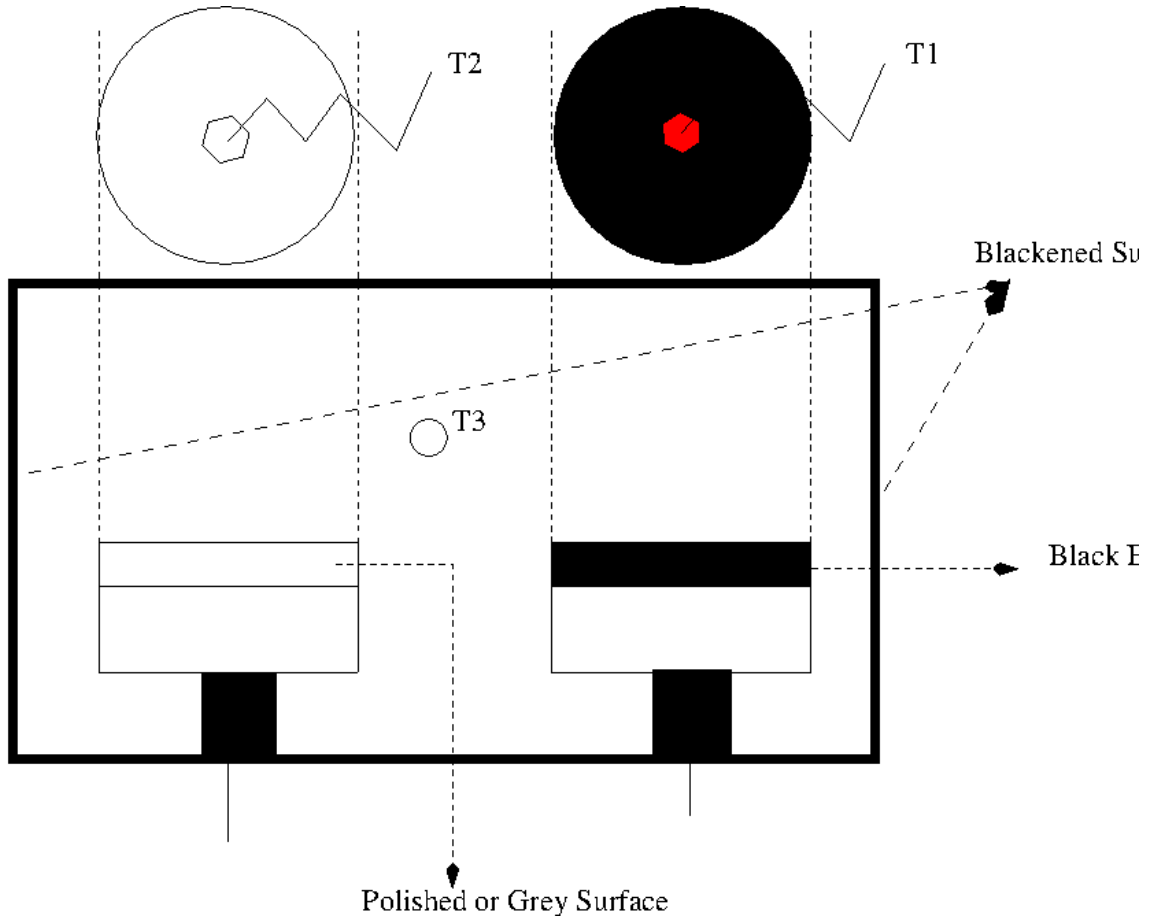


Figure 3: Experimental Setup for Emissivity of Grey Body

- Enclosure - a closed surface blackened to absorb incident radiation from both black body and test plate. The dimension of the enclosure are $580\text{cm} \times 30\text{cm} \times 30\text{cm}$, having mounted on cement sheet and one side of perspex.
- Heaters of same capacity are mounted on Grey and black body to maintain identical temperatures.
- Voltmeter and Ammeter - Digital meters are provided to measure input power from both the bodies.
- Thermocouples - to measure the surface temperatures of Black body T_1 , Test body T_2 and surface temperature T_3 .
- Channel selector for digital temperature display

- Heat controller or Regulator - To vary the input power to the heaters independently.
- Control Panel - To switch On/Off the console and the heater.

3.4 Working principle

The figure shows the schematic of the apparatus used to determine the emissivity of the test surface. The experimental setup is designed in such a way that under steady state conditions, the heat dissipation by conduction and convection, though small, are same for both bodies. The difference in power input to the heaters of black body and grey body which are at same temperatures is due to the difference in power inputs to the heaters and are related to emissivity by

$$W_1 - W_2 = \frac{(E_b - E) \sigma A (T_1^4 - T_2^4)}{0.86} \quad (21)$$

where

- $W_1 = V_1 I_1$ - Heat input to the black surface
- $W_2 = V_2 I_2$ - Heat input to the Grey surface
- V_1 = Voltage across the heater for the Black surface.
- V_2 = Voltage across the heater for the Grey surface.
- I_1 = Current across the heater for the Black surface.
- I_2 = Current across the heater for the Grey surface.
- d = Diameter of the Black surface.
- T_1 = Temperature of the Black surface.
- T_2 = Temperature of the Grey surface.
- E_b = Emissivity of the Black surface.
- E = Emissivity of the Grey Surface.
- $\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 K^4}$ - Stefan-Boltzmann constant.

The constant in the denominator of eq(21) (0.86) takes into account various factors such as Radiation shape factor, effect of conductance and free convection losses. It also takes into account for non-uniformities in enclosure temperature, which causes deviations from the typical heat transfer experiment. It should be made sure that the Grey body temperature and the blackbody temperature are the same.

3.5 Operational Procedure

1. Switch on the mains and console.
2. Switch on the heater of Black Body and adjust the power input to the desired value using the regulator.
3. Switch on the heater of Grey/Polished body and adjust the power input by the regulator. Ensure that the grey body power input is less than that of the black body.
4. Observe temperatures of the black body and test surface at closed intervals and adjust power input to the test body heater such that both the bodies have same temperatures.
5. After attainment of steady state record the input powers ($V_1 I_1$) and ($V_2 I_2$), temperatures T_1, T_2 & T_{surr} of Black body, test surface and the surrounding respectively.
6. Calculate the emissivity of the test surface using the formulations.

Precautions

- use stabilized power supply.
- Gradually increase the power input to the heaters from zero.

3.6 Calculations

S.No	W_1 Watts	W_2 Watts	T_1 $^{\circ}C$	T_2 $^{\circ}C$	T_3 $^{\circ}C$	E

$$V_1 = \quad Volts, V_2 = \quad Volts$$

$$I_1 = \quad Amps, I_2 = \quad Amps$$

$$W_1 = V_1 I_1 = \quad Watts$$

$$W_2 = V_2 I_2 = \quad Watts$$

$$A = \quad m^2, \sigma = 5.67 \times 10^{-8} \frac{W}{m^2 K^4}$$

4 Stefan-Boltzmann Apparatus

4.1 Aim

To determine the Stefan-Boltzmann constant for the given material.

4.2 Apparatus/ Specifications

- Hemisphere enclosure of diameter - 200 mm
- Water Jacket of diameter - 230 mm
- Water Jacket height - 125 mm
- Base plate made of Bakelite of diameter = 300 mm
- Diameter of test disc = 20 mm
- thickness of test disk = 2 mm.
- Thermocouples and their locations
 - T_1 - heating equipment.
 - $T_2, T_3 \& T_4$ attached to copper hemisphere.
 - T_5 attached to copper specimen.

4.3 Theory

All bodies at some temperature greater than the surrounding emits radiation.

4.4 Operational Procedure

- Fill the water tank with distilled water.
- Switch on the immersion heater coil till it reaches 85°C .
- Remove the copper disk before allowing the water into lower tank.
- Switch of the heater and allow hot water into lower tank.
- Allow the system to reach steady state.
- Note down the temperature readings $T_2 \cdots T_4$.
- Insert the copper disk and immediately note down the temperature T_5 at every 5 seconds interval starting from zero.
- Plot the graph of Temperature v/s time.
- Tabulate the results

4.5 Calculations

Thermocouple Number	Temperature $^{\circ}C$
T_2	
T_3	
T_4	
Average value	

Time in Seconds, t	Temperature $T_5^{\circ}C$
0	
5	
10	
15	
20	
25	
30	
35	
40	
45	

Plot the graph of T vs t from the above table and obtain the slope $\frac{dT}{dt}$

$$\sigma = \frac{MS \frac{dT}{dt}}{A_d (T_s^4 - T_{s,mean}^4)} \quad (22)$$

where

- Surface area of the disc $A_d = \frac{\pi}{4}d^2$
- Volume of the specimen, $V = A_d \times t$
- Mass of the disc, $M = \rho_{copper} \times V$
- $\rho_{copper} = 8954 \frac{kg}{m^3}$
- Specific heat of copper $S = 381 \frac{J}{kg^{\circ}K}$
- Surface Temperature, $T_m = \frac{T_2+T_3+T_4}{3}$
- Specimen Temperature, $T_{sp} = T_5$

5 Free Convection

5.1 Aim

To determine the natural heat transfer coefficient from the surface of the tube in both Vertical and horizontal position.

5.2 Apparatus:

Chromium plated Copper Tube, Thermocouples, Temperature indicators Control Panel

5.3 Theory:

In Free convection the motion of the fluid is due to Buoyancy forces. Buoyancy is due to the combined presence of a fluid density gradient and a body force that is proportional to density. Gravitational force is the usual force considered in the experiments conducted in the lab although the centrifugal force in rotating fluid machinery acts as body force. Out of many ways by which the density gradients can be setup the most common way of establishing a density gradient is due to the existence of temperature gradient.

Free convection flows may be classified according to whether the flow is bounded by a surface. In the absence of the adjoining surface, free boundary flows occur, in the form of plume or bouyant jet.

5.3.1 Similarity parameters

The dimension less parameters that govern the free convection flow are

- Grashof Number - Buoyancy force is the main cause for this number to exist. It is equal to the square of the Reynolds Number $Re = Gr^{\frac{1}{2}}$. It plays the same role as Reynolds number in the forced convection. It gives the measure of buoyancy forces to viscous forces acting on the fluid.
- The product of Grashof number and Prandtl number is called as Rayleigh number $Ra = GrPr$

It can be assumed that the correlations used for free convection takes the form $Nu_L = f(Gr_L, Pr)$

5.4 Long Horizontal cylinder

For an Iso thermal cylinder, Morgan suggested that $\bar{Nu}_D = \frac{\bar{h}D}{K} = C Ra_D^n$, the values of C and n are given below.

In contrast Churchill and Chu have recommended a single correlation for a wide range of Rayleigh number

$$Nu_D = \left[0.60 + \frac{0.387 Ra_D^{\frac{1}{6}}}{\left[1 + \left(\frac{0.599}{Pr} \right)^{\frac{9}{16}} \right]^{\frac{8}{27}}} \right]^2, Ra \lesssim 10^{12} \quad (23)$$

5.5 Experimental Setup

A typical experimental setup consists of a metallic tube of diameter "D" and length "L". Heat is supplied by the heater embedded into the tube. The amount of heat is varied using either dimmerstat or a power controlling regulator. Four thermocouples are placed on the surface at different points on the surface to measure the surface temperature.

5.6 Operational Procedure

1. Keep the tube in Vertical Position or Horizontal position
2. Switch on the main and the console switches
3. Set the regulator to adjust the Heat input to approximately 25 watts
4. Wait for sufficient time to allow temperature to reach steady state.
5. Note down the Temperature T_1 to T_5 using Channel Selector & Digital temperature Indicator.
6. Note down the Ammeter and Voltmeter readings.
7. Tabulate the heat Input and measured temperatures.
8. Calculate the convection heat transfer co-efficient below
9. Repeat the experiment by changing the tube to horizontal position with different heat input.

S.No 1	Temperature $T^{\circ}C$	h Practical Vertical $h(\frac{W}{m^2K})$	h practical Horizontal $h(\frac{W}{m^2K})$	h Theroetical Vertical $h(\frac{W}{m^2K})$	h th

5.7 Calculations

- Heat input rate $Q = VI(W)$
- Mean surface temperature, $T_s = \frac{T_1+T_2+T_3+T_4}{4} ^\circ C$
- Ambient Temperature $T_\infty = T_5$
- Surface area, $A_s = \pi DL$
- Calculate the experimental heat transfer coefficient, $\bar{h}_e = \frac{Q}{A_s(T_s-T_\infty)}$
- The heat transfer coefficient as calculated from the correlations is given by
 - $\bar{Nu} = 0.59 (Gr \cdot Pr)^{\frac{1}{4}}$ for $Gr < 10^9$
 - $\bar{Nu} = 0.10 (Gr \cdot Pr)^{\frac{1}{4}}$ for $Gr > 10^9$

where $\bar{Nu} = \frac{\bar{h}L}{K}$ is the average Nusselt Number, K is the thermal conductivity of air in $\frac{W}{m^\circ C}$,

- $Gr = \frac{\beta g L^3 (T_s - T_\infty)}{\nu^2}$ Grashof number
- $\beta = \frac{1}{(T_f + 273.16)}$ volumetric thermal expansion coefficient
- ν - Kinematic viscosity of air in $\frac{m}{s}$
- $Pr = \frac{\mu}{c_p K}$ is the Prandtl Number
- c_p specific heat of air in $\frac{J}{kg^\circ C}$
- μ dynamic viscosity of air
- The properties of air are to be evaluated at the $T_f = \frac{T_s + T_\infty}{2}$

6 Forced Convection (Inside a circular pipe)

6.1 Aim

To determine the convective heat transfer co-efficient in Forced convection in a circular pipe.

6.2 Apparatus/ Specifications

1. Pipe with circular cross section: the tube is thermally insulated outside to prevent heat transfer losses to the atmosphere.
 - diameter d ,
 - Length L ,
2. Band heater: 500 W (approx)
3. Regulator to control the power input to the heater.
4. Thermocouples T_1 and T_5 to measure the air temperature at the inlet and outlet of the duct, T_2, T_3 & T_4 to measure the heater surface temperatures.
5. Channel Selector
6. Digital temperature Indicator
7. Blower to blow air through heat exchanger
8. Orifice meter with manometer to measure air flow rate from the blower.
 - diameter of the orifice d_o
9. Control Panel to house the whole instrumentation.

6.3 Theory

The rate heat transfer from an heated object is faster when air is blown on to the object with an external aid like a fan, when compared to the object exposed to still air. The heat gets convected away and this process is known as “Heat transfer by Forced Convection”. If the flow takes place over an object it is considered as external flow, otherwise around if the flow takes place in closed enclosures then it can be classified as internal flow. Flow through a pipe or channel is an example of internal flow. As the fluid is flowing relative to the surface boundary layers develop and they are constrained in the internal flow.

The Newtons Law of Cooling in convective heat transfer is given by

$$q = \bar{h}A\Delta T \quad (24)$$

where,

- "q" is Heat transfer rate, Watts
- "A" Surface Area of Heat Flow, m^2 ,
- " $\Delta T = T_s - T_m$ " is Overall Temperature Difference between the wall and Fluid (0^C)
 - T_s is the surface temperature of the pipe
 - T_m - is the mean temperature of the fluid
- " $\bar{h}$ ", Local Convective Heat Transfer Co-efficient ($\frac{W}{m^2C}$)

The convective heat transfer co-efficient given in eq(24) represents the Local heat transfer coefficient. The actual heat transfer coefficient is function of fluid properties (like viscosity μ , its density (ρ)) and also on the dynamics of the fluid, whether the flow is laminar or turbulent based on Reynolds number (Re), in addition to its dependence on the thermal properties of the fluid (thermal conductivity, specific heat etc). If the flow is internal flow, apart from the existence of boundary layers it is also required to know whether the flow is in entrance or fully developed regions.

If the fluid enters the tube at a uniform temperature $T(r, 0)$, which is less than the surface temperature, convection heat transfer occurs and thermal boundary layer begins to develop. If the tube surface condition is fixed by imposing either a uniform temperature (T_s is constant) or a uniform heat flux q'' , a thermally fully developed condition is eventually reached. The shape of the fully developed temperature profile depends upon whether uniform surface temperature is maintained or uniform heat flux is supplied. If the flow is laminar the thermal entry length is given by

$$\left(\frac{x_{fd,t}}{D}\right)_{laminar} \approx 0.005 Re_D Pr \quad (25)$$

The development of the hydrodynamic boundary layer when compared to thermal boundary layer depends upon whether the Prandtl Number $Pr > 1$ or $Pr < 1$. In contrast for turbulent flow, conditions are nearly independent of Prandtl number, and to a first approximation the entry length is assumed as

$$\frac{x_{fd,t}}{D} = 10 \quad (26)$$

For flow in a circular tube with constant ρ and c_p , the mean temperature is given by

$$T_m = \frac{2}{u_m r_o^2} \int_0^{r_m} u T r dr \quad (27)$$

6.3.1 Convection Correlations : Laminar flow in Circular Tubes

6.3.2 Convection Correlations : Turbulent flow in Circular Tubes

The expressions for computing local Nusselt number for fully developed (both hydrodynamically and thermally) turbulent flow in a smooth circular tube are developed by Colburn.

$$\frac{C_f}{2} = \frac{f}{8} = St Pr^{\frac{2}{3}} = \frac{Nu_D}{Re_D Pr} Pr^{\frac{2}{3}} \quad (28)$$

The Colburn equation for Nusselt Number is given as

$$Nu_D = 0.023 Re_D^{\frac{4}{5}} Pr^{\frac{1}{3}} \quad (29)$$

Dittus-Boelter equation has slight modification to the Colburn equation given as

$$Nu_D = 0.023 Re_D^{\frac{4}{5}} Pr^n \quad (30)$$

where $n = 0.4$ for heating $T_s > T_m$ and $n = 0.3$ for cooling $T_s < T_m$. The above equations are confirmed experimentally for the range of conditions

$$\begin{aligned} 0.7 &\leq Pr \leq 160 \\ Re_D &\gtrsim 10000 \\ \frac{L}{D} &\gtrsim 10 \end{aligned} \quad (31)$$

These equations may be used for small to moderated temperature differences $(T_s - T_m)$, with all the properties evaluated at T_m .

Note: To a good approximation, the foregoing correlations may be applied for both the uniform surface temperature and heat flux conditions.

Another correlation valid over a large Reynolds Numbers range including the transition region is given by Gnielinski

$$Nu_D = \frac{(\frac{f}{8})(Re_D - 1000)Pr}{1 + 12.7(\frac{f}{8})^{\frac{1}{2}}(Pr^{\frac{2}{3}} - 1)} \quad (32)$$

where the friction factor f is obtained from Moody diagram or for smooth tubes it is given by Petukhov's relation $f = (0.790 \ln(Re_D) - 1.64)^{-2}$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$. The above correlation is valid for $0.5 \lesssim Pr \lesssim 2000$ and $3000 \lesssim Re_D \lesssim 5 \times 10^6$. In using eq(32) which applies for both uniform surface heat flux and temperature, properties should be evaluated at meat temperature T_m .

6.4 Operational Procedure

1. Switch On the mains and console

2. Start the blower first
3. Control blower flow rate to a suitable value
4. Measure the pressure drop across the orifice meter and calculate air mass flow rate.
5. Switch On the heater and adjust the power input to the heater to a suitable value using regulator.
6. Wait for reasonable time to allow temperatures to reach steady value.
7. Note temperatures T_1 to T_5 using channel selector and digital temperature indicator.
 - T_1 , temperature of air at heat exchanger inlet 0^C
 - T_5 , temperature of air at heat exchanger outlet 0^C
 - $T_2, T_3 \& T_4$, Surface temperatures at three locations on the heater 0^C
8. Measure the power input (P, watts) to the heater.
9. Tabulate the measured temperatures and power input to the heater.
10. Calculate the convective heat transfer co-efficient using the procedure given.
11. Repeat the experiment for different values of power input to the heater and blower air flow rates.

6.5 Calculations

7 Heat Exchangers

7.1 Aim

To study the effectiveness of parallel flow and counter flow heat exchanger using LMTD.

7.2 Apparatus/ Specifications

The apparatus consists of two co-axial tubes in which hot water passes through the core tube made up of copper and the cold water passes through the annulus. Based on the direction of the flow of cold water the experiment is classified as Parallel flow and Counter flow. Hot water is supplied by geyser, the mass flow rate of hot and cold waters are measured with the rotameters, and thermocouples are used to measure the inlet hot and cold water temperatures.

7.3 Theory

Heat exchangers are typically classified according to the flow arrangement and type of construction. The simplest way of constructing a heat exchanger is that of concentric tube also known as double pipe. one way of classifying such heat exchanger is based on the direction of the fluid flow

- Parallel Flow - Both hot and cold fluid flow in same direction.
- Counter Flow - Both hot and cold fluid flow in opposite direction.

Apart from the above two directions fluid can flow in cross flow i.e., perpendicular to each other. In the cross flow heat exchangers fluids crossing can mix or unmix. The basic parameter and often most uncertain part of any heat exchanger analysis is the determination of the overall heat transfer coefficient. This over all heat transfer coefficient is defined in terms of the thermal resistances of all the process that occur in the composite structure of the heat exchanger.

To design or to predict the performance of a heat exchanger, it is essential to relate the total heat transfer rate to the quantities such as inlet and outlet fluid temperatures, the over all heat transfer coefficient, and the total surface area for the heat transfer. Two such relations may readily be obtained by applying over all energy balances to the hot and cold fluids and there is negligible heat transfer between the exchanger and its surroundings, as well as negligible potential and kinetic energies, application of steady flow energy equation results in

$$q = \dot{m}_h(h_{hot,i} - h_{hot,o}) \quad (33)$$

$$q = \dot{m}_c(h_{cold,i} - h_{cold,o}) \quad (34)$$

where $\dot{m}$ represents the mass flow rate and h represents the enthalpy of the fluid. If there is no phase change of the fluids then constant specific heats can be assumed for the analysis which results in

$$q = \dot{m}_h c_{p,hot} (T_{hot,i} - T_{hot,o}) \quad (35)$$

$$q = \dot{m}_c c_{p,cold} (T_{cold,i} - T_{cold,o}) \quad (36)$$

where the temperatures T_{hot} & T_{cold} refer to the mean temperatures at those locations. If $\Delta T = T_{hot} - T_{cold}$ is the temperature difference between the cold and hot fluid then with Newton's law of cooling we have

$$q = UA\Delta T_m \quad (37)$$

where ΔT_m is appropriate mean temperature difference. Equations eq(37) in conjunction with eq(35) and eq(36) are the basic equations applied for Heat exchanger analysis.

7.3.1 Parallel Flow Heat exchanger

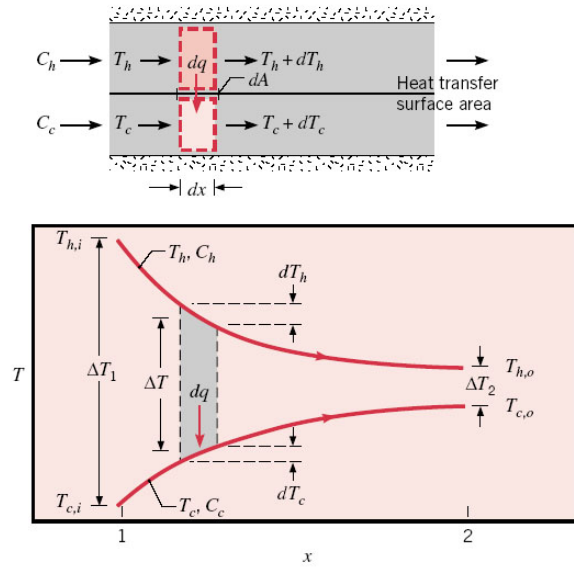


Figure 4: Temperature distribution in Parallel flow Heat exchanger

The hot and cold fluid temperature distributions associated with a parallel-flow heat exchanger are shown in Fig(??). The temperature difference initially is very large and decays with increasing x , approaching zero asymptotically. The outlet temperature of cold fluid for such heat exchanger never exceeds that of the hot fluid. The subscripts in Fig(??) designate opposite ends of the heat exchanger. The following assumptions are used in the analysis

1. The heat exchanger is insulated from its surroundings, in which case the only heat exchange is between the hot and cold fluids
2. Axial conduction along the tubes is negligible
3. Potential and Kinetic energies change are negligible.
4. The Fluid specific heats are constants.
5. The Overall heat transfer coefficient is constant.

Applying the energy balance to each of the differential elements in the Fig(??)

$$dq = -\dot{m}_{hot}c_{p,hot}dT_{hot} = C_{hot}dT_{hot} \quad (38)$$

$$dq = \dot{m}_{cold}c_{p,cold}dT_{cold} = C_{cold}dT_{cold} \quad (39)$$

$$(40)$$

where C_{hot} & C_{cold} are heat capacity rates of hot and cold fluid respectively. The heat transfer across the surface area dA can be expressed as

$$dq = U\Delta T dA \quad (41)$$

substituting eq(38) and eq(39) in $\Delta T = T_{hot} - T_{cold}$ results in a differential equation when integrated from inlet to outlet of the heat exchanger results in

$$\ln\left(\frac{\Delta T_1}{\Delta T_2}\right) = -UA((T_{hot,i} - T_{cold,i}) - (T_{hot,o} - T_{cold,o})) \quad (42)$$

For parallel flow heat exchanger $\Delta T_1 = (T_{hot,i} - T_{cold,i})$ and $\Delta T_2 = (T_{hot,o} - T_{cold,o})$ the heat transfer rate is obtained as

$$q = UA \frac{(\Delta T_2 - \Delta T_1)}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \quad (43)$$

$$q = UA\Delta T_{lm} \quad (44)$$

comparing eq(43) and eq(44) the average temperature difference is a log mean temperature difference. For parallel flow heat exchanger

$$\begin{aligned} \Delta T_1 &= (T_{hot,1} - T_{cold,1}) = (T_{hot,i} - T_{cold,i}) \\ \Delta T_2 &= (T_{hot,2} - T_{cold,2}) = (T_{hot,o} - T_{cold,o}) \end{aligned}$$

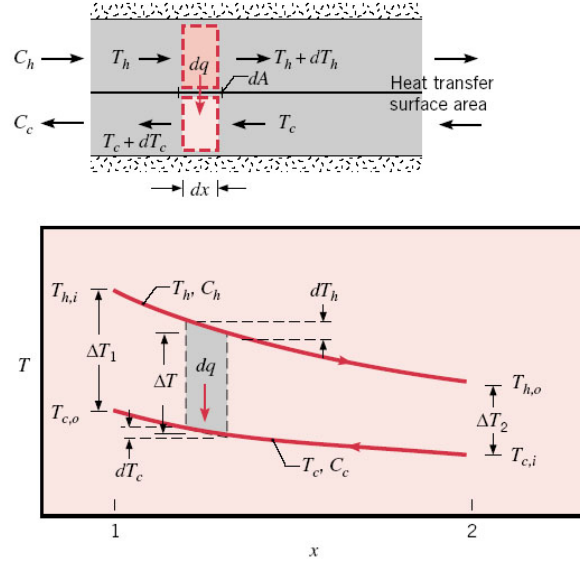


Figure 5: Temperature distribution in Counter flow heat exchanger

7.3.2 Counter Flow heat exchanger

Similarly the derivation of LMTD for counter flow heat exchanger can be carried out with a simple difference given by

$$\begin{aligned}\Delta T_1 &= (T_{hot,1} - T_{cold,1}) = (T_{hot,i} - T_{cold,o}) \\ \Delta T_2 &= (T_{hot,2} - T_{cold,2}) = (T_{hot,o} - T_{cold,i})\end{aligned}$$

For the same inlet and outlet temperatures, the log mean temperature difference for counterflow exceeds that of a parallel flow $\Delta T_{lm,CF} > \Delta T_{lm,PF}$. Hence the surface area required to effect a prescribed heat transfer rate, "q", is smaller for the counter flow than the parallel flow arrangement for the same value of U. Also the $T_{cold,o}$ can exceed the $T_{hot,o}$.

7.4 Operational Procedure

7.5 Calculations

8 Pin Fin Apparatus

8.1 Aim

- To determine the heat transfer coefficient under forced convection and to find the temperature distribution along the length of the pin fin.
- To find the effectiveness and efficiency of pin fin for insulated boundary condition

8.2 Apparatus/ Specifications

1. Pin fin apparatus with one end of the fin insulated
2. Blower for inducing forced convection on the fin.
3. Manometer
4. Orifice meter for measuring mass flow rate.
5. Thermocouples for measuring the temperature along the length of the fin and the temperature of the atmosphere

8.3 Theory

8.4 Operational Procedure

1. Switch on the main electrical power supply.
2. Ensure that the temperature selector of the thermocouples set to zero position.
3. Using Dimmer-stat knob adjust the power input to the heater to the desired value.
4. Switch on the Blower
5. Set the air flow rate to the desired value by adjusting the control valve.
6. Allow sufficient amount of time to reach steady state condition.
7. Note down all the readings from the thermocouples from $T_1 \cdots T_6$.
8. Repeat the experiment by changing the power to the heater of by adjusting the mass flow rate of the air.

8.5 Calculations

- Velocity of air through orifice meter $V_o = c_d \sqrt{\frac{2\vec{g}h(\rho_m - \rho_a)}{\rho_a}} \times \frac{1}{\sqrt{1-\beta^4}}$ where
 - c_d - is the coefficient of discharge
 - $\vec{g}$ - acceleration due to gravity.
 - h - Manometer difference
 - ρ_m - density of manometer fluid
 - ρ_a - density of air
 - $\beta = \frac{d_o}{d_{pipe}}$ d_o - diameter of the orifice, d_{pipe} - diameter of the pipe.
 - Velocity of air in the duct, $V_a = \frac{V_o A_o}{A_{duct}} = \frac{V_o \times \frac{\pi}{4} d_o^2}{W \times B}$
 - V_a - Velocity of air in the duct.
 - A_o - Cross section area of the orifice.
 - A_{duct} - Cross section area of the duct.
 - W - width of the duct.
 - B - Breadth of the duct.
 - Average surface temperature of the fin, $T_{Sm} = \frac{T_1 + T_2 + T_3 + T_4 + T_5}{5}$
 - Ambient Temperature, $T_\infty = T_6$
 - Average temperature, $T_m = \frac{T_{Sm} + T_\infty}{2}$.
 - Properties of air at mean temperature T_m
 - Reynolds Number $Re = \frac{V_a d_f}{\nu}$, where d_f is diameter of fin and ν - Kinematic viscosity of air at T_m
 - Nusselt Number $Nu = CR_e^n Pr^{\frac{1}{3}}$
 - $Nu = \frac{h_c L}{K} \Rightarrow h_c = \frac{Nu K}{L}$
 - Temperature distribution along the length of the fin,
 - $m = \sqrt{\frac{h P_f}{K_f A_f}} = \sqrt{\frac{4h}{K_f d_f}}$
 - $P_f = 2\pi r_f$ perimeter of the fin.
 - $A_f = \frac{\pi d_f^2}{4}$
- Temperature distribution = $\frac{T - T_\infty}{T_0 - T_\infty} = \frac{\cosh(m(L-x))}{\cosh(mL)}$
- Effectiveness of the fin $\epsilon = \frac{\text{Heat loss with fin}}{\text{Heat loss with out fin}} = \sqrt{\frac{PK_f}{h_c A_f}} \tanh(mL)$

8.5.1 Tabular column

S.No	Voltage V volts	Current I Amps	Pressure drop of Mercury, h cm	T_1	T_2	T_3	T_4	T_5	T_6

8.6 Calculations

9 Vapour Compression Refrigeration Test Rig

9.1 Aim

9.2 Apparatus/ Specifications

9.3 Theory

9.4 Operational Procedure

9.5 Calculations