EXPERIMENT 1

HEAT TRANSFER IN NATURAL CONVECTION

CONTENT:

Note:

1. At least one complete calculation process with equation must be shown in the beginning of result and discussion part.
2. Students are requested to write their own conclusion in the lab report.
3. Lab report should be submitted within given time.

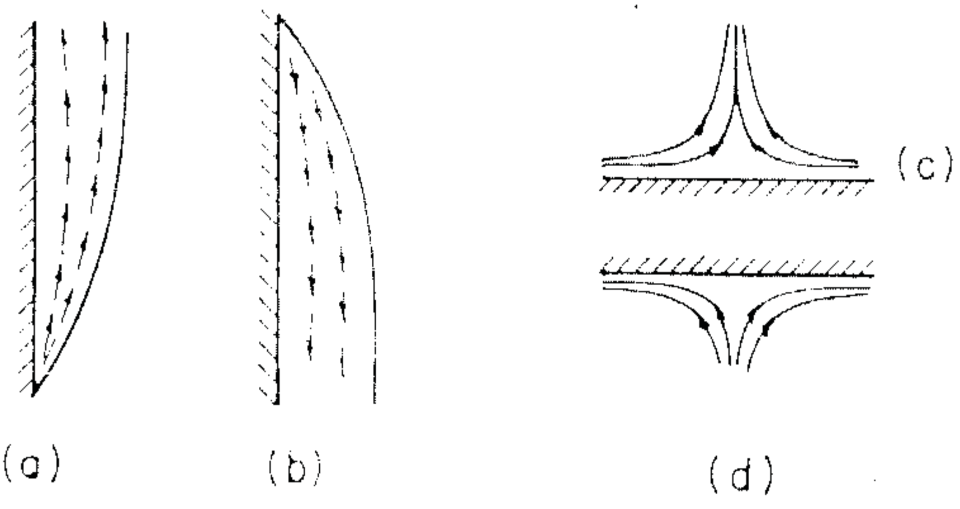
HEAT TRANSFER IN NATURAL CONVECTION

AIM: To determine the natural convection heat transfer coefficient for the vertical cylindrical tube which is exposed to the atmospheric air and losing heat by natural convection.

OBJECTIVE: The purpose of this experiment is to study experimentally the natural convection pipe flows at different heating level.

INTRODUCTION:
There are certain situations in which the fluid motion is produced due to change in density resulting from temperature gradients, which is the heat transfer mechanism called as free or natural convection. Natural convection is the principal mode of heat transfer from pipes, refrigerating coils, hot radiators etc. The movement of fluid in free convection is due to the fact that the fluid particles in the immediate vicinity of the hot object become warmer than the surrounding fluid resulting in a local change of density. The warmer fluid would be replaced by the colder fluid creating convection currents. These currents originate when a body force (gravitational, centrifugal, electrostatic etc.) acts on a fluid in which there are density gradients. The force which induces these convection currents is called a buoyancy force which is due to the presence of a density gradient within the fluid and a body force. Grashoff number (Gr) plays a very important role in natural convection.

In contrast to the forced convection, natural convection phenomenon is due to the temperature difference between the surface and the fluid is not created by any external agency. Natural convection flow pattern for some commonly observed situations is given in Figure 1.



(a) Heated vertical plate
(b) Cooled vertical plate
(c) Upper surface of a heated horizontal plate
(d) Lower surface of a heated horizontal plate

Figure 1 Natural convection flow patterns

The test section is a vertical, open ended cylindrical pipe dissipating heat from the internal surface. The test section is electrically heated imposing the circumferentially and axially constant wall heat flux. As a result of the heat transfer to air from the internal surface of the pipe, the temperature of the air increases. The resulting density non-uniformity causes the air in the pipe to rise. The present experimental setup is designed and fabricated to study the natural convection phenomenon from a vertical cylinder in terms of the variation of the local heat transfer coefficient and its comparison with the value which is obtained by using an appropriate correlation.

THEORY/BACKGROUND:

When a hot body is kept in a still atmosphere, heat is transferred to the surrounding fluid by natural convection. The fluid layer in contact with the hot body gets heated, rises up due to the decrease in its density and the cold surrounding fluid rushes in to take its place. The process is continuous and heat transfer takes place due to the relative motion of hot and cold particles.

The heat transfer coefficient is given by:

$$h = \frac{q}{A_s\left(T_s - T_a\right)} \qquad (1)$$

Here,

h = Average surface heat transfer coefficient.

q = Heat transfer rate.

A_s = Area of heat transferring surface

T_s = Average surface temperature (°C),

where,

$$T_s = \frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7}{7} \qquad (2)$$

T_a = Ambient temperature in the duct (°C) = T_8

The surface heat transfer coefficient of a system transferring heat by natural convection depends on the shape, dimensions and orientation of the body, the temperature difference between the hot body and the surrounding fluid and fluid properties like κ, μ, ρ etc. The dependence of 'h' on all the above mentioned parameters is generally expressed in terms of non-dimensional groups, as follows:

$$\frac{hL}{k} = A\left[\left\{\frac{gL^3\beta\Delta T}{\nu^2}\right\}\left\{\frac{\mu C_p}{k}\right\}\right]^n \qquad (3)$$

Here,

$\frac{hL}{k}$ is called the Nusselt Number (Nu),

$\frac{gL^3\beta\Delta T}{\nu^2}$ is called the Grashoff Number (Gr) and,

$\frac{\mu C_p}{k}$ is called the Prandtl Number

A and n are constants depending on the shape and orientation of the heat transferring surface.
L is a characteristic dimension of the surface,
κ is the thermal conductivity of the fluid,
ν is the kinematic viscosity of the fluid,
μ is the dynamic viscosity of the fluid,
C_p is the specific heat of the fluid,
β is the coefficient of volumetric expansion of the fluid,
g is the acceleration due to gravity at the place of experiment,
$\Delta T = T_s - T_a$

For gases, $\beta = \frac{1}{T_f + 273} K^{-1}$

where, T_f = mean film temperature = $\frac{T_s + T_a}{2}$

For a vertical cylinder losing heat by natural convection, the constants A and n of equation (3) have been determined and the following empirical correlations have been obtained:

$$\mathrm{Nu} = \frac{h_{th}L}{k} = 0.59(\mathrm{Gr.Pr})^{0.25}, \text{ for } 10^4 < \mathrm{Gr.Pr} < 10^9 \tag{4}$$

$$\mathrm{Nu} = \frac{h_{th}L}{k} = 0.59(\mathrm{Gr.Pr})^{1/3}, \text{ for } 10^9 < \mathrm{Gr.Pr} < 10^{12} \tag{5}$$

Here L is the length of cylinder and h_{th} is theoretical heat transfer coefficient.
All the properties of the fluid are evaluated at the mean film temperature (T_f)

APPARATUS:

The apparatus consists of a stainless steel tube fitted in a rectangular duct in a vertical fashion. The control panel for the natural convection apparatus is shown in figure 2. The heat input to the heater is measured by an ammeter and a voltmeter and is varied by a dimmerstat. The temperatures of the vertical tube are measured by seven thermocouples (1 to 7) and are marked on the Temperature Indicator Switch of the instrument panel as shown in Figure 2. One more thermocouple is used to measure ambient temperature. The schematic of the natural convection apparatus is shown in figure 3.

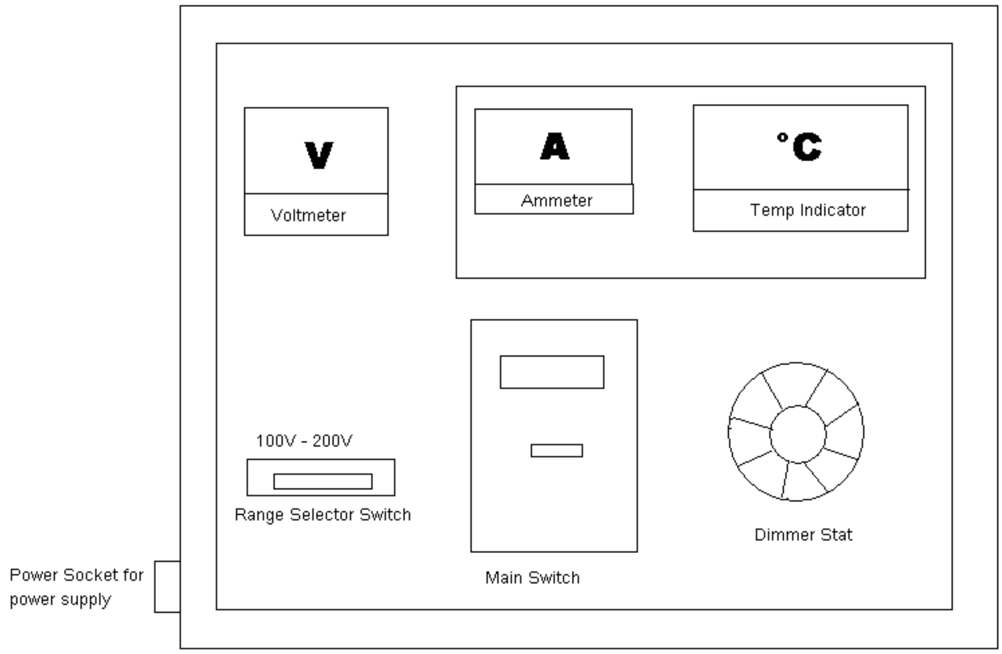


Figure 2 Control panel for natural convection apparatus

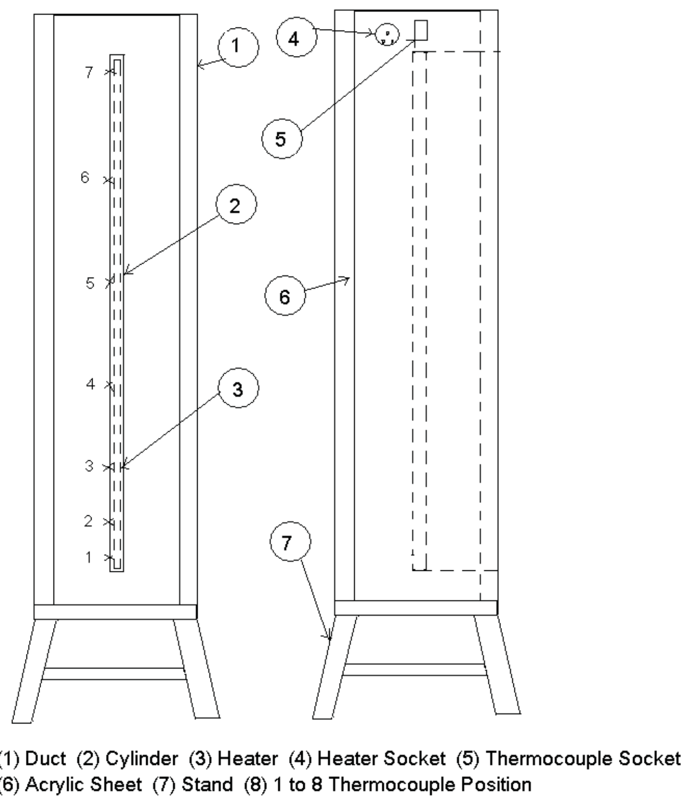


Figure 3. Schematic diagram of natural convection apparatus

The duct is open at the top and the bottom forms an enclosure which serves the purpose of undisturbed surroundings. One side of the duct is made up of perspex for visualization. An electric heating element is kept in the vertical tube which internally heats the tube surface. The heat is lost from the tube to the surrounding air by natural convection. The vertical cylinder with the thermocouple positions is shown in Figure 4.

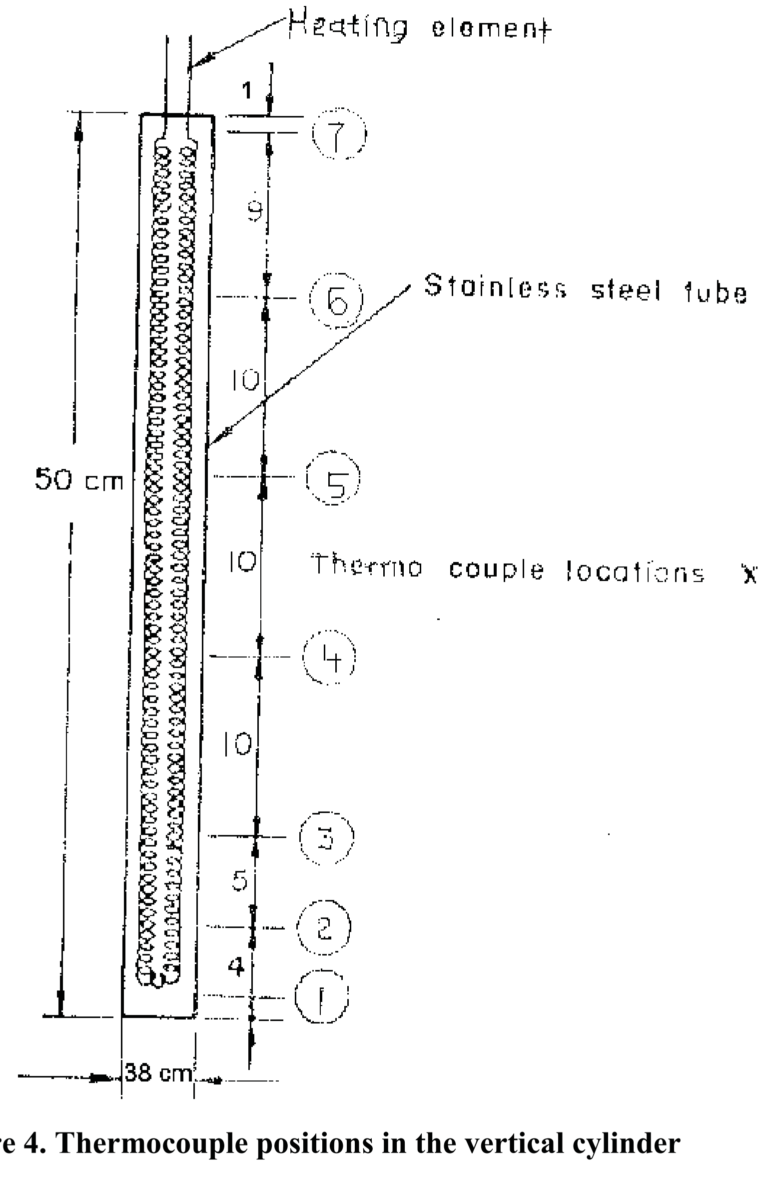


Figure 4. Thermocouple positions in the vertical cylinder

.

While the possible flow pattern and the expected variation of local heat transfer coefficient are shown in Figure 3. The tube has been polished to minimize the radiation losses.

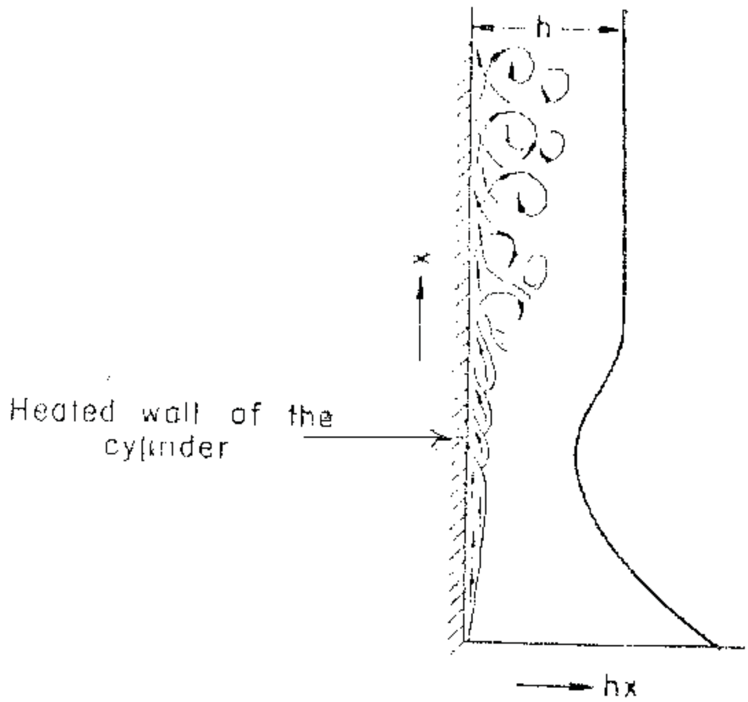


Figure 5. Variation of the heat transfer coefficient along the height of the tube in free air flow and dependence of this variation on the nature of flow

Specifications:

1. Outer Diameter of the tube (d) = 38 mm
2. Length of the tube (L) = 500 mm
3. Duct size = 20cm × 20cm × 1m length
4. Number of the thermocouples = 8
5. Thermocouple number 8 reads the Ambient Temperature and is kept in the duct.
6. Temperature Indicator 0-300 ^oC. Multi-channel type calibrated from iron constantan thermocouples with compensation of ambient from 0-50 ^oC.
7. Ammeter
8. Voltmeter
9. Dimmerstat

EXPERIMENTAL PROCEDURE:

1. Switch on the supply and adjust the dimmerstat to obtain the required heat input (say 40 W, 60 W, 70 W).
2. Monitor the temperature T_1 to T_8 every five minutes till steady state is reached.
3. Wait till the steady state is reached. This is confirmed from temperature readings (T_1 to

T_7). If they remain steady and do not register a change of more than 1 ^{o}C per hour.
4. Measure the surface temperature at various points (T_1 to T_7).
5. Note the ambient temperature, T_8.
6. Repeat the experiment for different heat inputs (say 40 W, 60 W, 70 W) by varying dimmerstat position.

PRECAUTIONS:

1. Switch off the ceiling fan before giving supply to set-up. This is to ensure the natural convection heat transfer environment.
2. Adjust the temperature indicator to ambient level by using compensation screw before starting the experiment (if needed).
3. Keep dimmerstat to zero volt position and increase it slowly.
4. Use proper range of Ammeter and Voltmeter.
5. Operate the change over switch of temperature indicator gently from one position to other, i.e. from position 1 to 8 position.
6. Never exceed 80 W power.

CALCULATIONS:

1. Calculate the value of average surface heat transfer coefficient neglecting radiation losses by experimental method.

   Average heat transfer coefficient, $h_{avg} = \dfrac{q}{A_s\left(T_s - T_a\right)}$

   where,

   q = rate of heating = $V \times I$ (watts)

   A_s = surface area of vertical cylinder rod = $\pi \times d \times l$ (m^2)

   T_s = Average surface temperature $= \dfrac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7}{7}$ (°C)

   T_a = ambient temperature = T_8 (°C)

2. Calculate and plot (Figure 4) the variation of the local heat transfer coefficient along the height of the tube using temperature T = T_1 to T_7 and equation (1).

3. Compare the experimentally obtained values with the theoretically predictions of the correlations (4) and (5). All the fluid properties are to be evaluated at mean film temperature.

UNCERTAINTY AND ERROR ANALYSIS:

The uncertainty analysis is a long and iterative process that takes the errors in the measured quantities to determine the uncertainty in the computed quantities. For this experiment all the temperatures represent the measured values. The thermocouples had a resolution of 0.1 °C.

These quantities are used to calculate the uncertainties related to these experiments. To obtain the uncertainty from the accuracy of the instruments a mathematical formula is used to calculate how these individual errors compound to give the net error in a calculation. The equation used for this purpose is,

$$\left[\varepsilon(F)\right]^2 = \left[\frac{\partial F}{\partial x_1}\varepsilon(x_1)\right]^2 + \left[\frac{\partial F}{\partial x_2}\varepsilon(x_2)\right]^2 + ...+ \left[\frac{\partial F}{\partial x_i}\varepsilon(x_i)\right]^2 \tag{6}$$

where F is the calculated quantity, ε is the absolute error and x_1, x_2, etc are the measured variables.

Also, this experiment is based on the "lumped capacitance" assumption. The experiment has been designed such that the resistance to heat conduction inside each specimen is much smaller than the external convective resistance. So, the internal temperature variation inside the solid test specimen will be small. Let T_i represent the instantaneous temperature of the specimen at time't'. Then, the instantaneous heat transfer rate from the specimen can be expressed as:

$$\dot{Q}_{total} = -mC_p \frac{dT_i}{dt} = \dot{Q}_{conv} + \dot{Q}_{rad} \quad = \bar{h}\,A\left(T_i - T_a\right) + \in \sigma A\left(T_i^4 - T_a{}^4\right) \tag{7}$$

where m is the mass of the specimen, C_p is the specific heat of stainless steel, A is the surface area of the specimen. The specimen cools by convection and radiation. Thus, in Eq. (7) the total heat transfer rate from the specimen is set equal to the sum of the convective and radiative heat transfer rates at the surface. Equation (7) can be solved for the average convective heat transfer coefficient, $\bar{h}$:

$$\bar{h} = \frac{-mC_p \dfrac{dT_i}{dt} - \in \sigma A\left(T_i^4 - T_a{}^4\right)}{A\left(T_i - T_a\right)} \tag{8}$$

Over each time interval, the cylinder cools from temperature T_i to temperature T_{i+1}. Using the measured temperature at these time intervals, the cooling rate can be approximated as:

$$\frac{dT_i}{dt} = \frac{T_{i+1} - T_i}{\Delta t} \tag{9}$$

Over this time interval, the average specimen temperature is taken to be $\overline{T_i} = \left(T_i + T_{i+1}\right)/2$. Using this average temperature and the cooling rate from Eq. (6), the actual average convective heat transfer coefficient can be calculated using Eq. (5) as:

$$\bar{h} = \frac{-mC_p\left(\dfrac{T_{i+1} - T_i}{\Delta t}\right) - \in \sigma A\left(\overline{T_i^4} - T_a{}^4\right)}{A\left(\overline{T_i} - T_a\right)} \tag{10}$$

where, $\overline{T_i} = \left(T_i + T_{i+1}\right)/2$ is the average surface temperature of the time interval.

Equation (10) gives the value of measured heat transfer coefficient including radiatiative heat transfer coefficient. The error can be calculated by taking difference of measured values and the predicted or theoretical calculated values.

RESULTS AND DISCUSSION:

Some typical experimental results are shown in Figure 4 for two different heater inputs.

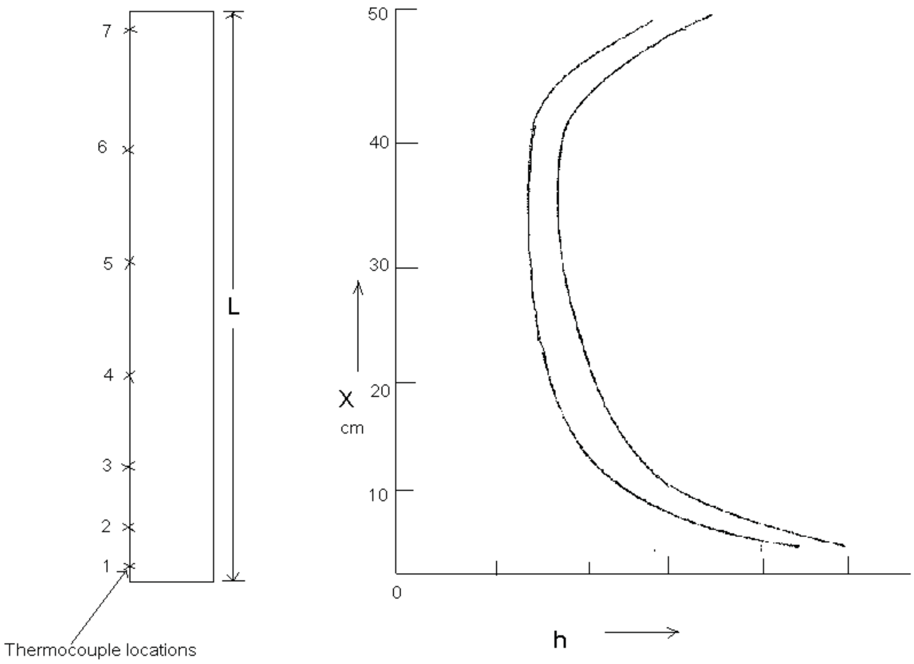


Figure 6. Some typical experimental results

The heat transfer coefficient is having a maximum value at the bottom of the vertical cylinder as expected because of the just starting of the building of the boundary layer and it decreases as expected in the upward direction due to thickening of the boundary layer which is a laminar one. This trend is maintained up to half the height of the cylinder and beyond that there is little variation in the value of the local heat transfer coefficient because of the formation of transition and turbulent boundary layers. The last point shows somewhat increase in the value of 'h' which is attributed to end-loss causing a temperature drop.

The comparison of average heat transfer coefficient is also made with predicted values by using correlation (4) and (5). It is found that the predicted values are somewhat less than the experimental values due to heat loss by radiation.

Transfer of energy can occur by three different modes: conduction, convection and radiation. Furthermore, there are two subtypes of convection: forced and natural convection. Forced convection refers to convection in a system with bulk flow while natural convection describes a system where the motion of the fluid arises primarily from naturally occurring density gradients. The rate of heat transfer between a solid surface and the fluid in convection is given by the Newton's rate equation:

$$q_{conv} = h \times A \times ((T_{heater} - T_a) \quad (11)$$

where 'q_{conv}' is the rate of convective heat transfer, 'A' is the area normal to direction of heat flow, 'h' is the convective heat transfer coefficient. Radiation heat transfer is the transfer of heat by electromagnetic radiation. Radiant heat transfer differs from conduction and convection in that no medium is required for its propagation. Energy transfer by radiation is at a maximum when the two surfaces exchanging energy are separated by a vacuum. The basic equation for heat transfer by radiation is:

$$q_{rad} = A \times \varepsilon \times \sigma \times T^4 \tag{12}$$

where 'q_{rad}' is the rate of radiant heat transfer, 'A' is the area of the radiant body, 'σ' is the Stefan-Boltzmann constant which is 5.676×10^{-8} W/m^{2}K, 'ε' is the emissivity, and 'T' is the temperature of the heat absorbing body.

When thermal radiation falls upon a body, part is absorbed by the body in the form of heat, part is reflected back into space, and part may be transmitted through the body. A black body is defined as an object that absorbs all radiant energy and reflects none. The ratio of the emissive power of a surface to that of a black body is called emissivity and for black body, ε =1.0. A radiation heat transfer coefficient, 'h_r' is analogous to the convective heat transfer coefficient, is given as:

$$q_{rad} = h_1 \times A_1 \times (T_1 - T_2) \tag{13}$$

where 'q_{rad}' is the rate of heat transfer by radiation,'A_1' is the surface area of the radiant body, 'T_1' is the temperature of the radiant body and 'T_2' is the temperature of the heat absorbing body. To obtain an expression for 'h_r' we equate equations, and obtain the following equation:

$$h_r = \frac{\varepsilon \times \left(T_1^4 - T_2^4\right)}{T_1 - T_2} \tag{14}$$

$$q = q_{conv} + q_{rad} \tag{15}$$

The convective heat transfer coefficient is generally between 0-25 W/m^{2}K and for forced convection 25-500 W/m^{2}K. In calculating the convective heat transfer coefficient, the average temperature was used, but the air temperature varies in the tube. As a result, the convective heat transfer coefficient calculated is artificially too low. A better approach would be to take the log-mean temperature r to integrate along the length of the tube to find the convective heat transfer coefficient.

REFERENCES:

1. Sukhatme, Dr. S.P., A textbook of Heat Transfer, Universities Press
2. Holman, J.P., Heat transfer, McGraw Hill publication
3. Cengel, Y.A., Heat transfer a practical approach, McGraw Hill publication

4. Incropera, F.P., and Dewitt., D. P., Fundamentals of Heat and Mass Transfer, John Wiley & Sons, Inc.

DATA SHEET

OBSERVATIONS:

1. Outer diameter of the cylinder (d) = 38 mm.
2. Length of the cylinder (l) = 500 mm.
3. Voltage = V = (Volts), Current = I = (Amperes)
4. Input to the heater (q) = V × I

Date of Experiment:

Name and Roll no. of students:

Set#1

V = I = q =

Time(min)	**Surface Temperature (°C)**							**Ambient Temp** (°C)
	T_1	**T_2**	**T_3**	**T_4**	**T_5**	**T_6**	**T_7**	**T_8**

Set#2

V = I = q =

Time(min)	Surface Temperature (°C)							Ambient Temp (°C)
	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8

Set#3

V = I = q =

Time(min)	Surface Temperature (°C)							Ambient Temp (°C)
	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8

Summary of results:

Time step	Measured Average Heat Transfer Coeff.	Predicted Average Heat Transfer Coeff.	Difference between measured & predicted $\overline{h}$

i	$\overline{h}$ **(W/m^{2}K)**	$\overline{h}$ **(W/m^{2}K)**	**(%)**