AHSANULLAH UNIVERISTY OF SCIENCE AND TECHNOLOGY (AUST)

DEPARTMENT OF MECHANICAL AND PRODUCTION ENGINEERING (MPE)

Heat and Mass Transfer Lab

LAB MANNUAL: ME 3224 (ME-3/2)

UPDATED BY

Prof. Dr. Fazlar Rahman

Dr. Tareq Zobaer

Dr. Qazi Talal

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Experiment 1
Measurement of Thermal Conductivity of a Metal Bar

Introduction

Heat transfer occurs due to a spatial temperature difference. Conduction is a mode of heat transfer from more energetic particles of a substance to adjacent less energetic particles as a result of interactions between them. In gases and liquids, conduction occurs due to molecular collisions and diffusion during random motion. In solids, it is a combination of molecular (or lattice) vibrations and energy transport by free electrons.

The rate of heat conduction through a medium depends on its geometry, thickness, material properties, and the temperature difference across it. Experiments have shown that the rate of heat conduction is proportional to the Temperature difference ($\Delta T = T_2 - T_1$), area (A), and inversely proportional to the thickness (Δx). Therefore,

$$\dot{Q}_{cond,x} \propto A\frac{T_1 - T_2}{\Delta x}$$
$$\Rightarrow \dot{Q}_{cond,x} = -k_x A\frac{\Delta T}{\Delta x}.$$

Here, k_x is the thermal conductivity of the material. It measures the ability of a material to conduct heat. In the limiting case of $\Delta x \rightarrow 0$,

$$\dot{Q}_{cond,x} = -k_x A\frac{dT}{dx}.$$

This is called Fourier's law of heat conduction. Rearranging this equation, an expression of thermal conductivity can be obtained,

$$k_x = -\frac{\dot{Q}_{cond,x}/A}{\dfrac{dT}{dx}}.$$

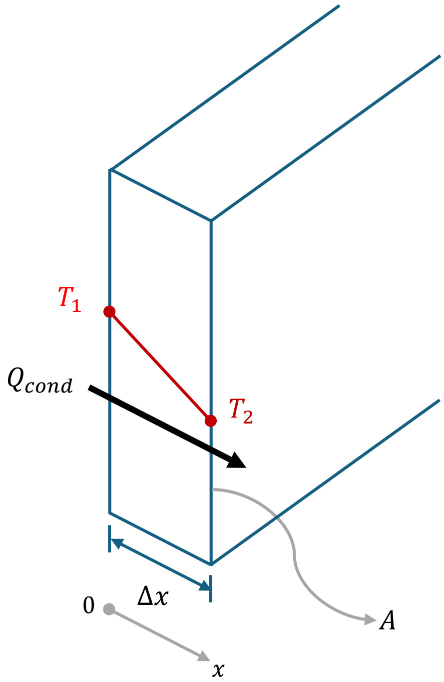


Figure 1: Heat conduction through a medium.

The thermal conductivity of an isotropic material is independent of the direction of transfer, thus $k_x = k_y = k_z = k$. It is to be noted here that, $k_{solid} >> k_{liquid} > k_{gas}$.

Heat conduction occurs in the direction of decreasing temperature. When temperature decreases in the x direction, $\frac{dT}{dx}$ becomes negative. The negative sign in Fourier's law ensures that the heat flux in positive x is positive when heat flows in that direction.

The heat conduction equation

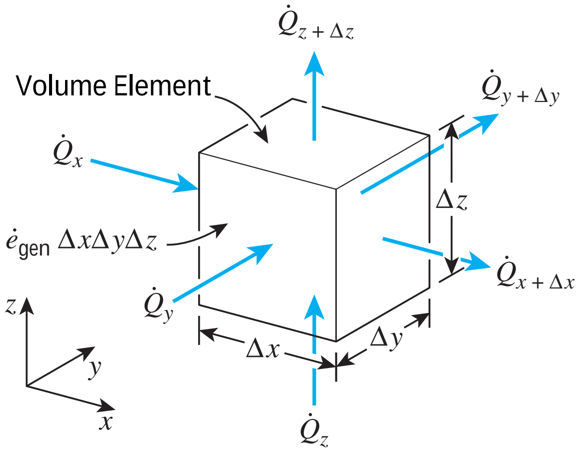


Figure 2: Heat conduction through a three-dimensional rectangular volume element.

From energy balance on the three-dimensional volume element shown in Figure 2 during a small time interval,

$$\begin{pmatrix}\text{Rate of heat}\\ \text{conduction at}\\ x, y, \text{ and } z\end{pmatrix} - \begin{pmatrix}\text{Rate of heat}\\ \text{conduction at}\\ x+\Delta x,\ y+\Delta y,\ \text{and } z+\Delta z\end{pmatrix} + \begin{pmatrix}\text{Rate of heat}\\ \text{generation}\\ \text{inside the element}\end{pmatrix} = \begin{pmatrix}\text{Rate of change}\\ \text{of the energy}\\ \text{content of the element}\end{pmatrix}$$

$$\Rightarrow \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{e}_{\text{gen}}}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t}.$$

Several assumptions can be made simplify this equation.

- If the circumferential surface of the bar is insulated (also neglecting lateral heat transfer), heat conduction is restricted to the x-direction only. Hence,

$$\dot{Q}_y = -k_y A \frac{\partial T}{\partial y}$$
$$\Rightarrow \frac{\partial T}{\partial y} = 0$$
$$\Rightarrow \frac{\partial^2 T}{\partial y^2} = 0.$$

  Similarly, $\dfrac{\partial^2 T}{\partial z^2} = 0.$

- There is no internal heat generation. So, $\dot{e}_{gen} = 0$.

- Steady state is achieved before final data is recorded. Thus, $\dfrac{\partial T}{\partial t} = 0$.

So, the heat conduction equation can be simplified as,

$$\frac{d^2T}{dx^2} = 0$$
$$\Rightarrow \frac{dT}{dx} = C_1 \Rightarrow T = C_1 x + C_2.$$

In other words, ideally, temperature should vary linearly with distance. Using the boundary conditions, at $x = 0 : T = T_0$ and at $x = L : T = T_L$, the following can be obtained,

$$C_2 = T_0 \quad \text{and} \quad C_1 = \frac{T_L - T_0}{L}.$$

Substituting,

$$T = \frac{T_L - T_0}{L} x + T_0.$$

It is evident from the preceding equation that the temperature gradient $\frac{dT}{dx}$ is the slope of the $T - x$ curve.

The objective of this experiment is to determine the thermal conductivity of a metal bar under steady-state, one-dimensional heat conduction. The experimental setup, data collection, and calculation procedure are discussed next.

Experimental setup

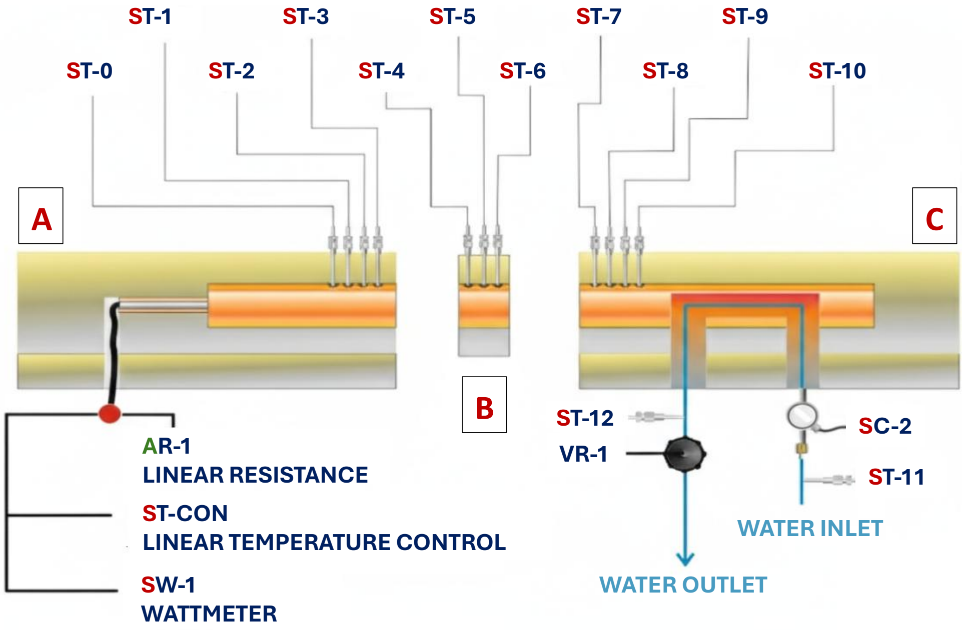


Figure 3: Schematic of the heat conduction module.

The setup consists of three different parts:

- **Input heat section, A:** This section contains a contact resistance heater that supplies heat to the system. Four Type-T thermocouples (ST-0 to ST-3), spaced at 10 mm intervals, measure the axial temperature distribution. An external insulating cover minimizes radial heat losses, promoting one-dimensional (axial) conduction. Electrical power is provided through a digital wattmeter (SW-1) with an output range of 0–150 W.
- **The central section, B:** This section accommodates interchangeable test specimens: a 25 mm diameter brass cylinder, a 25 mm diameter stainless steel cylinder, and a 10 mm diameter brass cylinder. Each specimen is equipped with three Type-T thermocouples (ST-4 to ST-6) positioned at 10 mm intervals to measure the axial temperature gradient.
- **Cooling section, C:** The cooling section is water-cooled at its outer surface. It contains four Type-T thermocouples (ST-7 to ST-10), spaced at 10 mm intervals. By continuously removing heat, the system prevents heat accumulation. This allows the apparatus to reach and maintain steady one-dimensional conduction.

Cooling water inlet and outlet temperatures are measured using two Type-J thermocouples (ST-11 and ST-12). A flow sensor (SC-2) is used to measure the flow rate of the cooling water, with a measurement range of 0.25-6.5 L/min. The flow rate is controlled using the variable regulator (VR-1).

Operational procedure

1. Verify that all the temperature sensors and the heating resistors have been connected, and also that the accessory is aligned with the fixed conduction cylinders.
2. Circulate water through the cooling system at a flow rate of SC-2 = 1.0 L/min.
3. Set the heating resistor to required power level, $\dot{Q}$ W (as indicated on the wattmeter plate SW-1) using the power controller.
4. Wait until the system stabilizes and no significant temperature fluctuations are observed in the sensors.
5. Record the readings of all thermocouples every ten minutes, as well as the water inlet and outlet temperatures, in the provided data table. Continue taking measurements until the system reaches steady state.

Data sheet

ME 3206: Heat and Mass Transfer Sessional

EXPERIMENT 1: MEASUREMENT OF THERMAL CONDUCTIVITY OF A METAL BAR

Group: Student IDs: Date:
Power:

Temp (°C) / Time (mins)	ST0	ST1	ST2	ST3	**ST4**	**ST5**	**ST6**	ST7	ST8	ST9	ST10	*ST11*	*ST12*
0													
10													
20													
30													
40													
50													
60													
70													
80													
90													
100													

____
Signature of the Teacher

Calculation

Analyze the thermal evolution along the conductive bar by treating each section (A, B, and C) independently. The following steps must be completed in order.

1: Plot Temperature vs. Distance

Using the steady-state thermocouple readings from the data sheet, plot the experimentally measured temperature T_e against axial position x for **all three sections** on a single graph. Label each section clearly.

2: Determine dT/dx for the central section (B)

From the T_e–x plot, fit a straight line through the data points belonging to section B (thermocouples ST-4, ST-5, ST-6). The slope of this line is the temperature gradient.

3: Calculate the Heat Removed by the Cooling Water

At steady state, the heat conducted through the bar equals the heat carried away by the cooling water,

$$\dot{Q} = \dot{m}_w \, C_{p_w} \, \Delta T_w,$$

where:

- $\dot{m}_w$ is the mass flow rate of cooling water (kg/s),
- $C_{p_w} = 4120$ J/kg·K is the specific heat of water,
- $\Delta T_w = T_{out} - T_{in}$ is the rise in cooling water temperature (ST-12 − ST-11).

4: Calculate Thermal Conductivity

Using Fourier's law and equating it to the heat removed by the cooling water,

$$\dot{Q} = \dot{m}_w \, C_{p_w} \, \Delta T_w = -k \, A \, \frac{dT}{dx}$$

$$\Rightarrow \quad k = \frac{\dot{m}_w \, C_{p_w} \, \Delta T_w}{-A \, \dfrac{dT}{dx}},$$

where A is the cross-sectional area of the metal bar.

Discussion

- Explain the experimental temperature distribution (temperature versus distance plot).
- What is the actual value of thermal conductivity of the metal bar at the temperature that was maintained during the experiment? Is there any discrepancy between the actual and the experimental value? If yes, why?
- Explain the variation in the Experimental temperature distribution and theoretical temperature distribution plot, if there is any.

Experiment 2

Study of Heat Transfer by Natural Convection from Pin Fins

Introduction

An extended surface is commonly used in reference to a solid that experiences energy transfer by conduction within its boundaries, as well as energy transfer by convection to its surroundings. The extended surface is most often utilized in quick removal of heat. The rate of heat removal by convection from surfaces is increased by increasing the surface area for heat transfer by using extended surfaces called fins. A fin with a cylindrical shape and high aspect ratio (length/diameter) is called a pin fin.

Fins are often used seen in engine cooling, electrical appliance such as in a computer power supply or substation transformers, etc. The generalized or governing equation for one dimensional heat transfer through extended surfaces is given by,

$$\frac{d^2T}{dx^2} = m^2(T - T_\infty)$$

Where, $m^2 = \frac{hp}{kA}$

T = T(x), Surface temperature of the fin is a function of distance (x) from the base wall.

T_∞ = Temperature of the fluid surrounding of the fin.

h = heat transfer coefficient between the surface of the fin and the surrounding fluid (W/m_2 k)

k = Thermal conductivity of the fluid

A = Cross sectional area of the extended metal bar or fin with the following assumptions:

(i) The fin material having no external heat generation

(ii) One dimensional conduction only along length of the fin

(iii) Steady state condition

(iv) Radiation heat loss is neglected

(v) The base temperature is fixed or constant

(vi) Natural convection conditions are maintained

The solution of the aforementioned differential equation depends on the choice of boundary conditions. Three different boundary conditions are set below will be considered.

Case-1: The fin is very long and the temperature at the end of fin is essentially that of the surrounding fluid

Case-2: The fin is finite length but its tip is insulated.

Case-3: The fin is of finite length and heat loses by convection from its end.

For case 1: the solution becomes,

$$\frac{T - T_\infty}{T_0 - T_\infty} = e^{-mx}$$

For case 2: The solution becomes,

$$\frac{T - T_\infty}{T_0 - T_\infty} = \frac{Cosh\,[m(L-x)]}{\cosh mL}$$

For case 3: The solution becomes,

$$\frac{T - T_\infty}{T_0 - T_\infty} = \frac{Cosh\,[m(L-x)] + (h/mk)\sinh[m(L-x)]}{\cosh mL + (h/mk)\sinh mL}$$

Where,

T_o = Temperature of the base wall (heater plate) of the fin

L = Length of the fin

x = Distance/location on the fin section from the base wall where the temperature is being measured.

The amount of heat transfer involved in these three cases may be given by the following equations,

Case 1: $Q_1 = \sqrt{hpkA}\,(T_o - T_\infty)$

Case 2: $Q_2 = \sqrt{hpkA}(T_o - T_\infty)\tanh mL$

Case 3: $Q_1 = \sqrt{hpkA}(T_o - T_\infty)\frac{\sinh mL + \left(\frac{h}{mk}\right)\cosh mL}{\cosh mL + \left(\frac{h}{mk}\right)\sinh mL}$

To indicate the heat transfer performance of a fin, two parameters are defined as below,

1. **Fin efficiency**: defined as the ratio of actual heat transferred to heat which would be transferred if the entire fin area were at base wall temperature, $\eta_f = \frac{\tanh(mL)}{mL}$

2. **Fin effectiveness:** defined as the ratio of heat transfer from the wall after adding fin to the heat transfer from the wall before adding fin, $\eta_f = \frac{\tanh(mL)}{\sqrt{\frac{h \cdot A}{k \cdot P}}}$

The specific objectives of this experiment are as follow:

- To plot the temperature distribution along the fins.
- To plot $\frac{T-T_\infty}{T_0-T_\infty}$ against $\frac{x}{L}$ to show the temperature distribution along the fins in nondimensional form for both experimental and theoretical considerations using three different boundary conditions as stated before.
- To estimate heat transfer under all conditions by theoretical equations.
- To estimate fin efficiency and fin effectiveness.

Experiment Set-up:

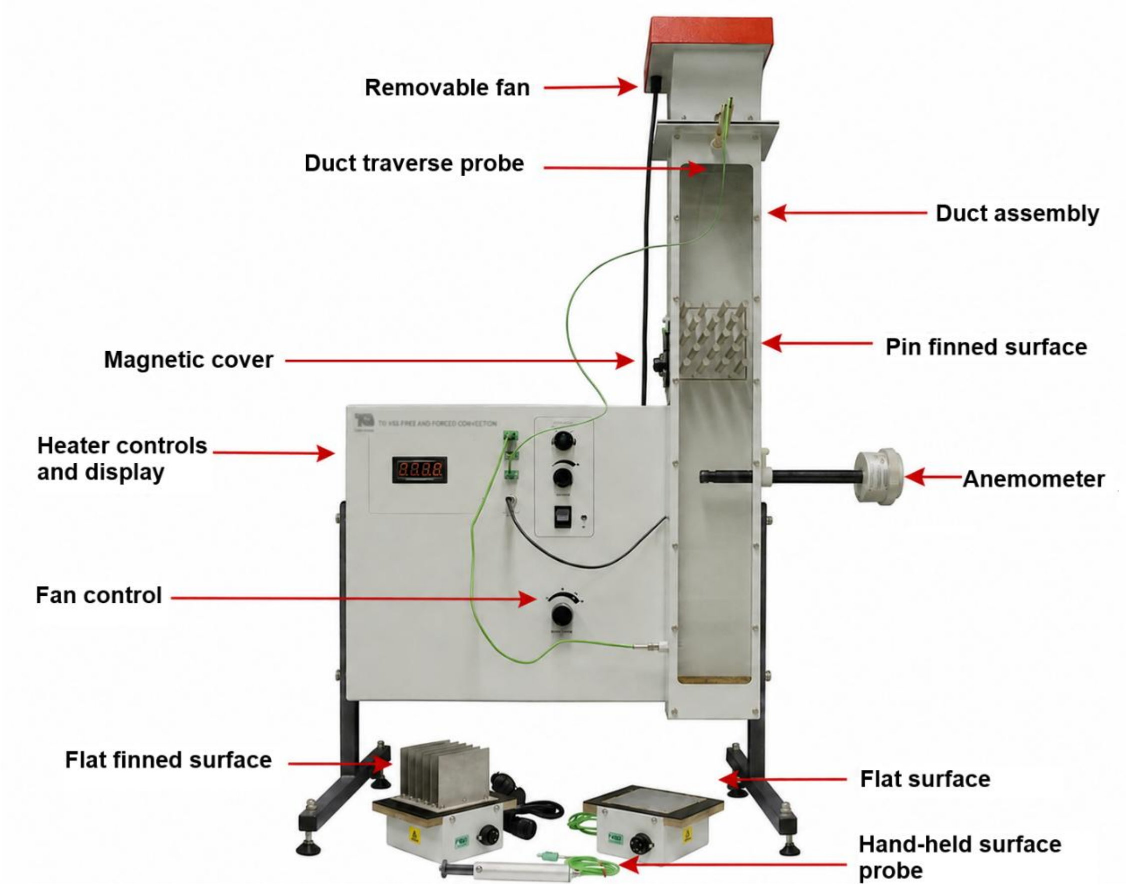


Figure 1: Experimental setup.

- The Main Unit is a compact bench-mounted frame with a vertical duct assembly and a control panel for electrical supply, controls, and displays.
- The duct enables air to flow over the heat transfer surface by free convection or forced convection using a variable-speed fan at the top.
- A fixed thermocouple measures inlet (ambient) air temperature, while a traversing probe records outlet temperature distribution, allowing calculation of the bulk outlet temperature.
- An anemometer measures air velocity.
- Each heat transfer surface has a built-in thermocouple,
- A hand-held probe can be used to measure heat distribution along finned and pinned surfaces. The probe inserts into six equally spaced holes in the duct, with a magnetic cover to seal unused holes and minimize stray convection.

Pin Fin:

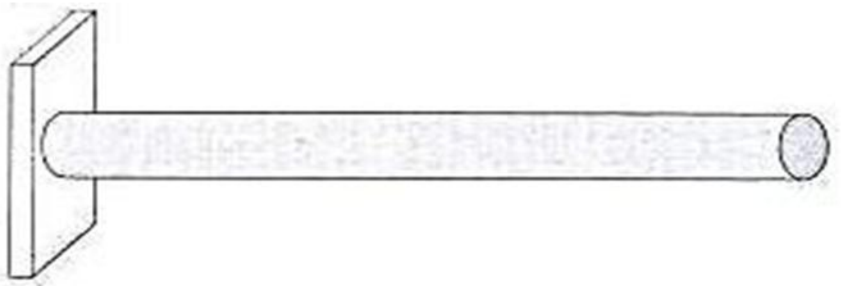


Figure 2: Pin fin

Fin material is stainless steel and its thermal conductivity, $k = 16.26\ \frac{w}{m \cdot k}$

Procedure:

1) Record room temperature and that is the surrounding fluid temperature (T_∞)
2) Maintain natural convection condition as far as possible during the experiment and collecting data
3) Switch on the heater and adjust the watt setting for heating purpose.
4) Take the initial readings at the different positions of pin fin by handheld digital thermocouple.
5) Take the reading of the same position after 10 minutes and check steady state condition.
6) Take temperature reading at different six position of fin after achieving steady state condition.

7) If possible then repeat this same experiment for the second watt setting.

8) Plot the temperature distribution along the fins.

9) Plot $\frac{T-T_\infty}{T_0-T_\infty}$ against $\frac{x}{L}$ to show the temperature distribution along the fins in nondimensional form for both experimental and theoretical considerations.

10) Find the effectiveness and efficiency if this metal bar is considered as fin.

11) Find the experimental errors, find source of errors, and explain how to minimize it.

12) Discuss the nature of true experimental and theoretical results you get from the graph and through calculations.

Data collection table:

Air velocity = m/s; Inlet air temperature, T_∞ = , Outlet air temperature, T_{out} =

Heater plate temperature, T_0 =

Watt (Heater capacity)	Position of thermocouples (mm)	Temperature (°C)	$\frac{T-T_\infty}{T_0-T_\infty}$ (from experiment)	$\frac{T-T_\infty}{T_0-T_\infty}$ from Theory		
				Case-1	Case-2	Case-3
	0					
	7.5					
	19.5					
	31.5					
	43.5					
	55.5					
	67.5					
	0					
	7.5					
	19.5					
	31.5					
	43.5					
	55.5					
	67.5					

Calculations:

Diameter of rod, D = 12 mm

Length of fin, L = 73 mm

Thermal conductivity of fin material (Stainless steel), k = 16.26 W/m-K

Heat transfer coefficient between in surface and surrounding fluid (air), h = 10 W/m^2 K

- Find the value of $\frac{T-T_\infty}{T_0-T_\infty}$ from the experiment and from the theory
- Find the fin efficiency and effectiveness for different cases.

Experiment 3

Free and Forced Convection Through Pin and Flat Fins

Introduction

In practical engineering applications, fins arranged into 'heat sinks' are designed to cool a critical heated components such as engine cylinder heads, power electronics, or microprocessors by maximizing the surface area exposed to the cooling medium (air). This experiment investigates and compares two common heat sink geometries: **flat fins** and **pin fins** (shown in Figure 1), under both free and forced convection conditions.

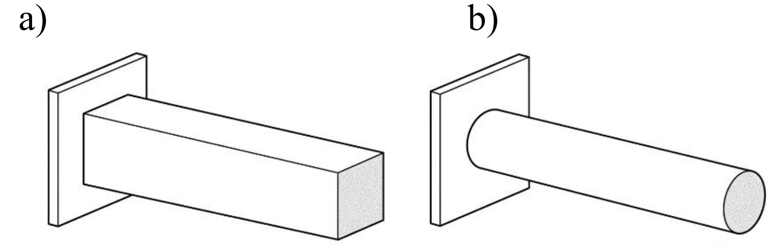


Figure 1: (a) Flat fin and (b) pin fin.

The effectiveness of a heat transfer surface is best evaluated by supplying a fixed input power and observing the resulting surface temperature. For identical power inputs, a heat sink that reaches a lower steady-state temperature is transferring heat more effectively to the surrounding air and is therefore the superior heat sink design.

Free Convection (Natural Convection)

Free convection arises without any externally imposed fluid motion. When a surface is heated, the air immediately adjacent to it gains thermal energy, causing a local decrease in air density. This less-dense, more buoyant air rises along the heated surface under the influence of gravity, drawing cooler ambient air upward from below to replace it and establishing a continuous circulation pattern that carries heat away from the surface. The rate of heat transfer in free convection depends strongly on the temperature difference between the surface and the ambient air, as well as the geometry and orientation of the surface. Because these buoyancy-driven fluid velocities are relatively low, fluid mixing is limited.

Forced Convection

Forced convection occurs when an external agent such as a fan or blower drives fluid over the heated surface. The externally imposed velocity field continuously sweeps heated air away from the surface, replacing it with cooler ambient air. Because the rate of heat removal is strongly coupled to air velocity, forced convection generally achieves significantly higher heat transfer rates than free convection for the same surface geometry and power input.

Thermal Mass and Transient Behavior

Before steady state is achieved, the thermal mass (or thermal inertia) of the fin plays an important role. The energy stored in a body relative to its surrounding is given by,

$$E_{st} = m \cdot C_p \cdot (\mathrm{T} - \mathrm{T}_\infty)$$

Where m is the mass, C_p is the specific heat capacity, T is the fin temperature, and T_∞ is the ambient air temperature. A fin with greater mass requires more energy to reach a given temperature and takes longer to cool. This is directly analogous to mechanical inertia in a flywheel: a large thermal mass buffers and dampens transient fluctuations in temperature or heat supply. For this reason, it is essential to wait for full steady-state conditions (where the rate of energy storage, $\dot{E}_{st} = 0$) before recording experimental data. This ensures that all transient effects have subsided and the readings truly reflect the fin's convective performance.

Comparison between Pin Finned and Flat Finned Surfaces

Although both fin geometries increase the surface area available for heat transfer, the way each interacts with the surrounding airflow differs significantly, as shown in Figure 2. The flat finned surface sits directly in the main body of airflow and creates channel-like flow paths between adjacent fins, producing a relatively uniform airflow through and out of the fin array with a modest pressure drop across the surface. In contrast, the pin finned surface causes turbulent airflow and varied velocities in and around its pins, creating a larger pressure difference between the upstream and downstream sides of the array. While the airflow approaching the pin fin may be uniform, it becomes turbulent immediately downstream. This greater disruption to airflow enhances mixing of the surrounding air but also causes a larger pressure drop compared to the flat finned surface.

These aerodynamic differences mean that the relative thermal performance of the two geometries is not fixed; it depends on whether free or forced convection is the dominant heat transfer mode. Identifying which geometry performs better under each condition is one of the central objectives of this experiment.

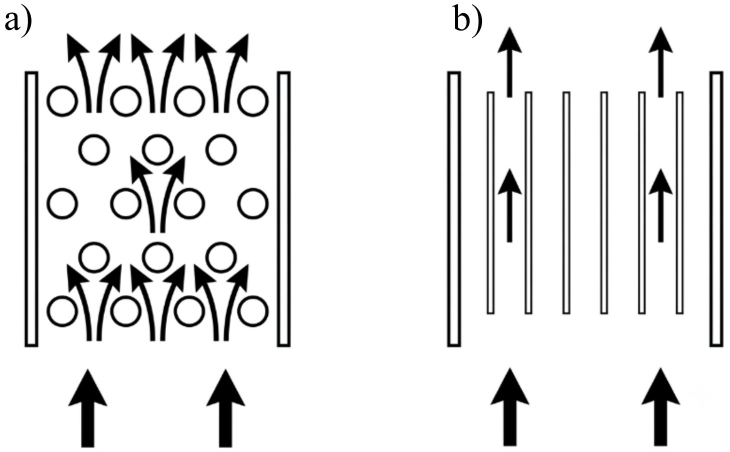


Figure 2: Air flow over (a) pinned and (b) flat finned surface.

Objective:

- To compare the surface temperature distribution along a pin finned and flat finned surface for a given heater power under free and forced convection.
- To evaluate which fin geometry is more effective as a heat transfer surface under each convection mode.

Experimental setup:

Same compact bench-mounted setup that was used in Experiment 2 is used here

Procedure:

1. Fit the chosen heat transfer surface (pin fin or flat fin) onto the apparatus.
2. For free convection, turn off the fan. For forced convection, ensure the fan is fitted and operational.
3. Set the heater to the desired power output and adjust the fan speed accordingly (2 m/s for forced convection; zero for free convection).
4. Allow the system to reach steady-state conditions before proceeding. This is indicated when the recorded temperatures remain stable over time.
5. Once steady state is achieved, record the following: duct inlet (ambient) temperature (T_∞), heater plate temperature (T_s), downstream air temperature (T_{out}), air velocity, and heater power.
6. Record the temperature at multiple locations along the fin surface using the hand-held probe. Adjust the sliding magnetic cover to uncover individual measurement holes, collecting data from the region nearest the heating surface (0 mm) to the farthest point (67.5 mm). Enter this data into Data Table 2.
7. Repeat Steps 1 through 6 for the other heat transfer surface (pin fin or flat fin) and for the other convection mode.
8. Repeat the full procedure for additional heater power settings and air velocities as required.

Data Table 1:

Free convection:

Heater Power = W			
Heat Transfer Surface	**Heater Plate Temperature T_s (°C)**	**Duct Inlet (ambient) temperature T_∞ (°C)**	**Difference ($T_s - T_\infty$) (°C)**
Flat Fin			
Pin Fin			

Forced convection:

Heater Power = W Fan velocity = m/s			
Heat Transfer Surface	**Heater Plate Temperature, T_s (℃)**	**Duct Inlet (ambient) temperature, T_∞ (℃)**	**Difference ($T_s - T_\infty$) (℃)**
Flat Fin			
Pin Fin			

Data Table-2:

Air velocity = m/s, Inlet air temperature, T_∞ = , Outlet air temperature, T_{out}=

Heater plate temperature, T_s =

Heater capacity (W)	Position of thermocouples, x (mm)	x/L	Probe Temperature, T_p (℃)	$\frac{T_p - T_\infty}{T_s - T_\infty}$
	0			
	7.5			
	19.5			
	31.5			
	43.5			
	55.5			
	67.5			
	0			
	7.5			
	19.5			
	31.5			
	43.5			
	55.5			
	67.5			

Graphs and Calculation:

1. Plot $\frac{T_p - T_\infty}{T_s - T_\infty}$ against $\frac{x}{L}$ for force and free convection of an identical Fins (Flat/Flat and Pin/Pin) in the same graph.
2. Plot $\frac{T_p - T_\infty}{T_s - T_\infty}$ against $\frac{x}{L}$ graph for Pin and Flat Fins for an identical air velocity and heater power, then discuss the differences.
3. Calculate air density (ρ_{air}) at room temperature by using equation of state.
4. Calculate mass flow rate of air in the duct, $\dot{m} = A_{duct} \cdot \rho_{air} \cdot v_{air}$ ($A_{duct} = 128\,mm\ \times 75\,mm$)
5. Calculate amount of heat taken out by air, $Q_{air} = \dot{m} \cdot C_p \cdot \Delta T = \dot{m} \cdot C_p \cdot (T_{out} - T_{in})$

Analysis of Results

- Which fin surface recorded a lower steady-state temperature at identical air velocity and heater power? What does this indicate about its heat dissipation effectiveness?
- Based on the data, which mode of convection heat transfer appears to be more effective?
- For which fin type did the surface temperature change more significantly between free and forced convection?
- Discuss the results obtained for free and forced convection.

Dimension of Flat Fins:

No. of Fin = 6

Size of Fin = 90 mm (height) and Length = 73 mm

Thickness of each Fin = 1.5 mm

Total surface area of Fin = 0.092 m^2 (including tip area)

Experiment 4

Heat Transfer Coefficient and Nusselt Number

Introduction

The earlier experiments compared the performance of heat transfer surfaces under free and forced convection by simply measuring surface temperature. While useful for direct comparison, this approach is qualitative, since it only reveals which surface stays cooler, but not how effectively heat is being transferred. A more rigorous evaluation requires quantifying the heat transfer process. This experiment introduces two such parameters, the convective heat transfer coefficient (h_c) and the Nusselt number (Nu), to characterize the performance of a flat plate under both free and forced convection.

Heat transfer coefficient

The convective heat transfer coefficient quantifies the rate of heat exchange between a solid surface and the moving fluid (air) in contact with it. A higher value of h_c indicates more effective convective heat transfer. It depends on fluid properties, flow conditions (velocity, laminar vs. turbulent), and surface geometry. Typical values for air range from 5 to 25 W/m²K for free convection and 10 to 200 W/m²K for forced convection. Using Newton's Law of Cooling adapted for bulk fluid temperature changes, the heat transfer coefficient is calculated as,

$$h_c = \frac{Q}{A_s \cdot T_m}$$

where Q is the heat flow from the surface to the air, A_s is the surface area, and T_m is the logarithmic mean temperature difference. The logarithmic mean is used (rather than a simple arithmetic mean) because the temperature difference between the surface and the air varies non-linearly as the air moves along the heated surface. It is defined as,

$$T_m = \frac{T_{out} - T_\infty}{\ln \dfrac{T_s - T_\infty}{T_s - T_{out}}}$$

Here, T_s is the temperature of the heated surface, T_∞ and T_{out} are the temperature of air at the inlet and outlet, respectively.

Nusselt Number (Nu)

The Nusselt number is a dimensionless quantity expressing the ratio of convective to conductive heat transfer across the fluid boundary layer,

$$Nu = \frac{h_c \cdot L}{k_{air}}$$

where L is the characteristic length (over which the air flows) and k_{air} is the thermal conductivity of air evaluated at the average air temperature. A Nusselt number close to 1 indicates that the fluid layer behaves almost like a stagnant solid (dominant conduction), whereas high Nusselt numbers indicate active fluid motion driving aggressive convective heat transport.

Thermal Conductivity of Air ($\boldsymbol{k_{air}}$)

The thermal conductivity of a material quantifies its intrinsic ability to conduct heat, that is, the rate at which thermal energy propagates through a unit length of material per unit cross-sectional area per unit temperature difference. For air, thermal conductivity increases approximately linearly with temperature over the range 0 to 100 °C (as shown in the Figure 1), which has a measurable influence on calculated heat transfer coefficients and the overall thermal performance of a fin at elevated fin temperatures.

Objectives

- To demonstrate how to determine the convective heat transfer coefficient and the Nusselt number for a finned surface in a duct, under both free and forced convection.
- Provide a quantitative confirmation of the superior performance of forced convection observed in earlier experiments.

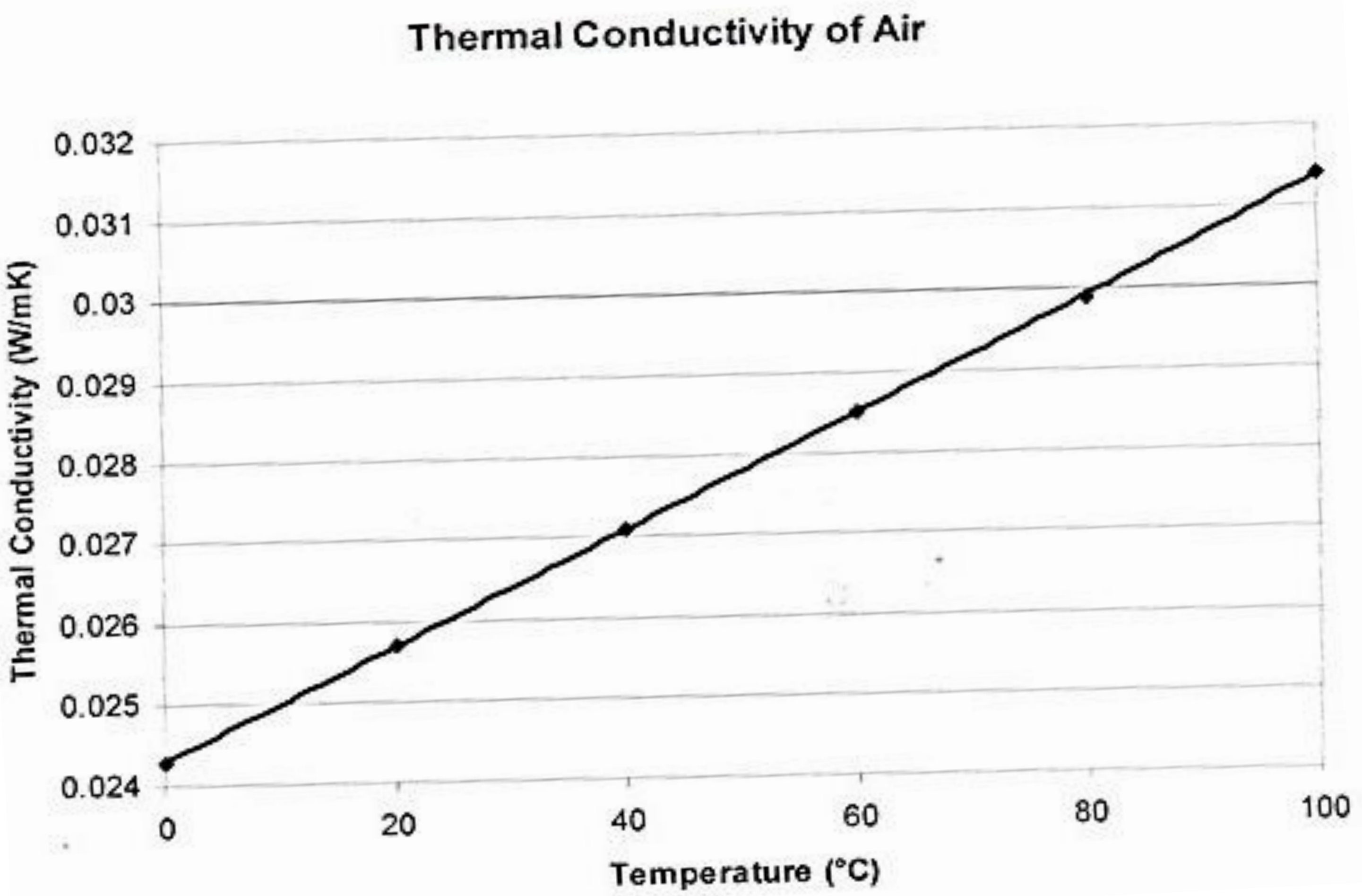


Figure 1: Thermal conductivity of air versus temperature.

Experimental setup

Same compact bench-mounted setup that was used in Experiment 2 and 3 is used here. Two measurement concepts of the setup pertinent to this experiment are discussed below.

Measurement of velocity

The setup uses an anemometer to measure air velocity in the middle section of the duct. In free convection, although air near the heated surface has a finite velocity due to buoyancy, the probe positioned at the duct center records nearly zero velocity. In contrast, under forced convection, the probe detects the maximum air velocity driven by the fan, since the flow is assumed to be fully developed across the duct. This is illustrated in Figure 2.

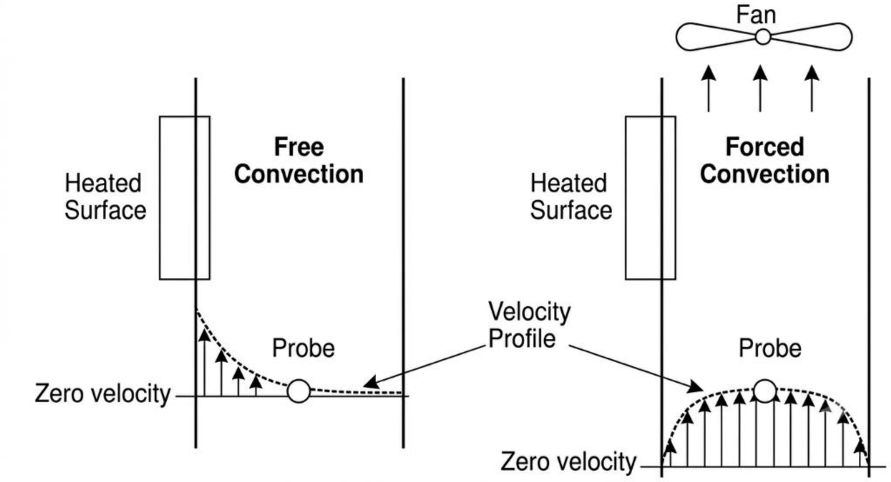


Figure 2: Velocity measurement in free and forced convection.

Measurement of bulk downstream air temperature (T_{out})

Due to the heat source on one side, the air temperature in the duct downstream of the heated surface may not be even across the duct, giving a temperature profile across the duct and a variation in T_{out}. This can be seen in Figure 3. Thus, a single point measurement of downstream temperature may not give an accurate value for the bulk temperature of the air. To address this, the setup includes an upper probe that traverses the duct for temperature measurements at multiple positions. To calculate bulk downstream temperature, T_{out}, these readings can be simply averaged. They can also be numerically integrated for improved accuracy.

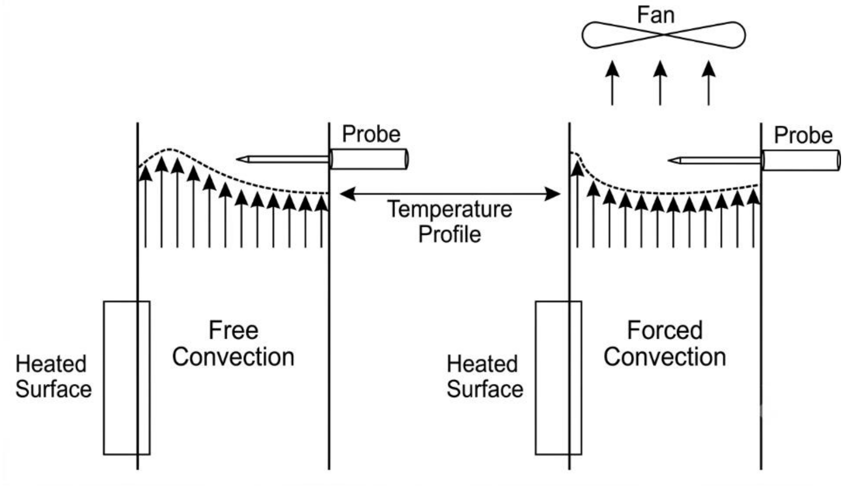


Figure 3: Temperature measurement in free and forced convection.

Procedure

1. Turn off the fan from the top of the duct for free convection
2. Fit the finned heat transfer surface.
3. Set the heater to 20 Watts.
4. Move the duct traverse probe so it reads 0 (zero) so that its tip touches the opposite wall of the duct at this position. Now move it to the 1 mm position.
5. Wait for the temperatures to stabilize and then take readings of the surface, inlet and outlet (duct probe) temperatures.
6. Either choosing to move in equal steps (if time permits) or larger steps, take readings of the temperatures across the duct using the traverse. Stop when you reach 74 mm (the tip is almost fully retracted into the near sidewall of the duct at this point). Recheck the inlet and surface temperatures as you do this.
7. For forced convection, repeat the experiment with an airflow of 3 m/s.

Data Table:

For Free Convection

Duct Traverse Probe Position, x (mm)	Ambient Temperature, T_∞ (°C)	Heat Transfer Surface Temperature, T_s (°C)	Duct Traverse Probe Temperature, T_p (°C)	$T_s - T_\infty$ (°C)	$T_p - T_\infty$ (°C)

For Forced Convection

Duct Traverse Probe Position, x (mm)	Ambient Temperature, T_∞ (°C)	Heat Transfer Surface Temperature, T_s (°C)	Duct Traverse Probe Temperature, T_p (°C)	$T_s - T_\infty$ (°C)	$T_p - T_\infty$ (°C)

Calculation and Analysis of Results

For each set of results (free and forced):

1. Produce a plot of $(T_p - T_\infty)$ vs position to visualize the outlet temperature profile with respect to inlet (this allows for changes in ambient temperature).
2. Determine T_{out} using simple arithmetic averaging.
3. Calculate average values for the other temperature readings.
4. Use T_{out} and the average readings to find the logarithmic mean temperature difference, T_m.
5. Use T_m to find the heat transfer coefficient (h_c), assuming that heat transferred to the air is equal to the applied electrical power.
6. Find the thermal conductivity (k_{air}) for air at the average inlet temperature from Figure 1.
7. Use your values of h_c and k_{air} to find the Nusselt number, Nu.
8. Compare the experimentally obtained values with those predicted by theory.

Experiment 5

The Effect of Varying Flow Rate in parallel and Counter flow in Concentric Shell and Tube Heat Exchanger

Introduction

The heat exchanger is a simple shell and tube type heat exchanger. It has two tubes one inside the other. The outer tube is shell. The inner tube carries the water from the hot circuit of the service module; the outer tube carries the water from the cold circuit. Heat transfer between the two tubes. Parallel and Counter flow are possible in this heat exchanger module but not the cross flow

This heat exchanger is in two equal parts with extra thermocouples at the midpoint.

Parallel flow:

When the direction of flow for the both hot water and cold water is same then it is called parallel flow

Counter flow:

When the direction of flow for the hot water is just opposite of cold fluid then it is called counter flow

Some common terminology:

The mean temperature efficiency and heat transfer coefficient give more useful results for comparison between heat exchangers.

The ***temperature efficiency*** of the ***hot*** circuit of the Heat Exchanger is the ratio of the temperature change in the hot circuit, divided by the difference between the maximum and minimum temperatures of the hot and cold circuits:

$$\eta_H = \frac{T_{H1} - T_{H2}}{T_{H1} - T_{C1}} \times 100\ \%$$

The ***temperature efficiency*** of the ***cold*** circuit of the Heat Exchanger is the ratio of the temperature change in the cold circuit, divided by the difference between the maximum and maximum and minimum temperatures of the hot and cold circuits:

$$\eta_C = \frac{T_{C2} - T_{C1}}{T_{H1} - T_{C1}} \times 100\ \%$$

The ***mean temperature efficiency*** of the two circuits is the average efficiency of them both:

$$\eta = \frac{\eta_H + \eta_C}{2}$$

Logarithmic Mean Temperature Difference (LMTD)

This is a measure of the heat driving force that creates the heat transfer. It is a logarithmic average of the temperature difference between the hot and cold circuits at each end of the heat exchanger.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

For Parallel Flow:

$$\Delta T_1 = T_{H1} - T_{C1}$$

$$\Delta T_2 = T_{H2} - T_{C2}$$

For Counter Flow:

$$\Delta T_1 = T_{H1} - T_{C2}$$

$$\Delta T_2 = T_{H2} - T_{C1}$$

Heat Transfer Coefficient (U)

This is the overall heat transfer coefficient for the wall and boundary layers. It is a measure of how well the heat exchanger works. A good heat exchanger will give a high coefficient; therefore, this value is important to engineers.

$$U = \frac{Q_e}{A \times LMTD}$$

Mean heat transfer area A = 0.02 m^2

Heat Transfer and Energy balance (Q, Q_e and Q_a)

The subscript 'e' represents here emission and 'a' represents here absorption. Here the heat is emitted from the hot water, and the cold water is absorbing heat energy.

Commonly we know that $\dot{Q} = m^{\cdot} \times C_p \times \Delta T$

And mass flow rate, $\dot{m} = \rho \times \dot{V}$

ρ = density and $\dot{V}$ = volumetric flow rate

Heat emitted by the hot water is $\dot{Q}_e = \rho_H \times \dot{V}_H \times C_{pH} \times \Delta T_H$

And heat absorbed by the cold water is $\dot{Q}_a = \rho_C \times \dot{V}_C \times C_{pC} \times \Delta T_C$

Density and specific heat must be measured at the average temperature of inlet and outlet.

In ideal heat exchanger, the heat emitted by the hot water must be equal to the heat absorbed by the cold water but practically it is not the case. There are some losses in the surroundings.

$Q_e = Q_a$ ± Losses or gain from surroundings

Heat balance Coefficient, $\boldsymbol{C_{EB} = \frac{Q_a}{Q_e}}$

If there is gain from the surrounding, then $Q_a > Q_e$. In this case, the energy balance coefficient may be greater than 1.

Objective

- To show how different cold flow rates affect the performance of the heat exchanger in both parallel flow and counter flow connection (hot flow rate and heater temperature are fixed).
- Temperature vs position graphs for both the parallel flow and counter flow.
- Calculation of power emitted, power absorbed, mean temperature efficiencies and energy balance for parallel and counter flow
- Find LMTD and overall heat transfer coefficient (U) for each flow rate.

Procedure:

1) Connect and set up your heat exchanger
2) Press the solenoid valve at the hot water system for filling water inside the tank. Stop pressing when the full green lamp is on.
3) Switch off the inlet regulator for hot circuit.
4) Start heater and set the heater tank temperature at 60 °C
5) After achieving that temperature stop the heater and open the inlet regulator (hand operated flow control valve). Start the pump immediately. Set the flow of the hot water circuit 3 L/min and the cold-water circuit 3 L/min.
6) Allow at least five minutes for the heat exchanger temperatures to stabilize. Generally, the temperature at the inlet of the hot water circuit is low and then start increasing and after sometime the temperature will begin to fall. It is because the thermocouple at the inlet needs some time and also continuous flow for stabilization. So, the temperatureT_{H1} at the beginning of experiment start

rising but as the cold water is also taking some heat so after giving a peak temperature it will begin to fall again.

7) Record the hot water inlet temperature T_{H1}, Hot water outlet temperature $T_{H2}$and a middle hot circuit temperature T_{H3}. Also record the readings of cold-water inlet temperature T_{C1} and cold-water outlet temperature T_{C2} and the cold-water circuit middle temperature T_{C3}. Take all the temperature at the peak of T_{H1}
8) Follow the procedure for different flow rate in case of cold-water flow.
9) Apply the same procedure for counter flow.

Experimental set up:

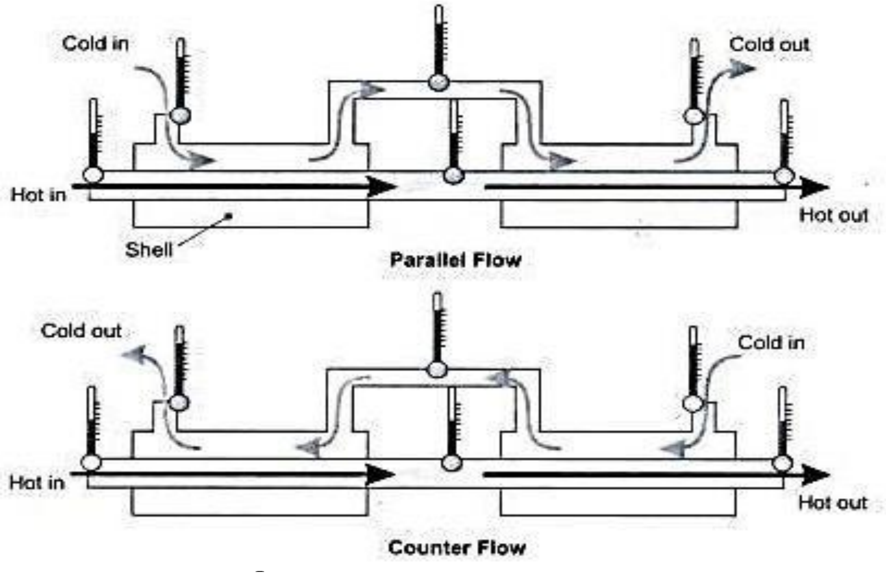


Fig 1: Parallel flow and Counter flow for concentric shell and tube heat exchanger.

Data Sheet:

Data Table 1:

Parallel flow connection

Ambient tank temperature:

Heater tank temperature:

Test	Hot Flow L/min	**Cold flow L/min**	**T_{H1}**	**T_{H2}**	**ΔT_H**	**Average T_H**	**T_{H3}**	**T_{C1}**	**T_{C2}**	**ΔT_C**	**Average T_C**	**T_{C3}**
1												
2												
3												
4												

Data Table 2:

Counter flow connection

Ambient tank temperature:

Heater tank temperature:

Test	Hot Flow L/min	Cold flow L/min	T_{H1}	T_{H2}	ΔT_H	Average T_H	T_{H3}	T_{C1}	T_{C2}	ΔT_C	Average T_C	T_{C3}
1												
2												
3												
4												

Calculation Table:

Table 1:

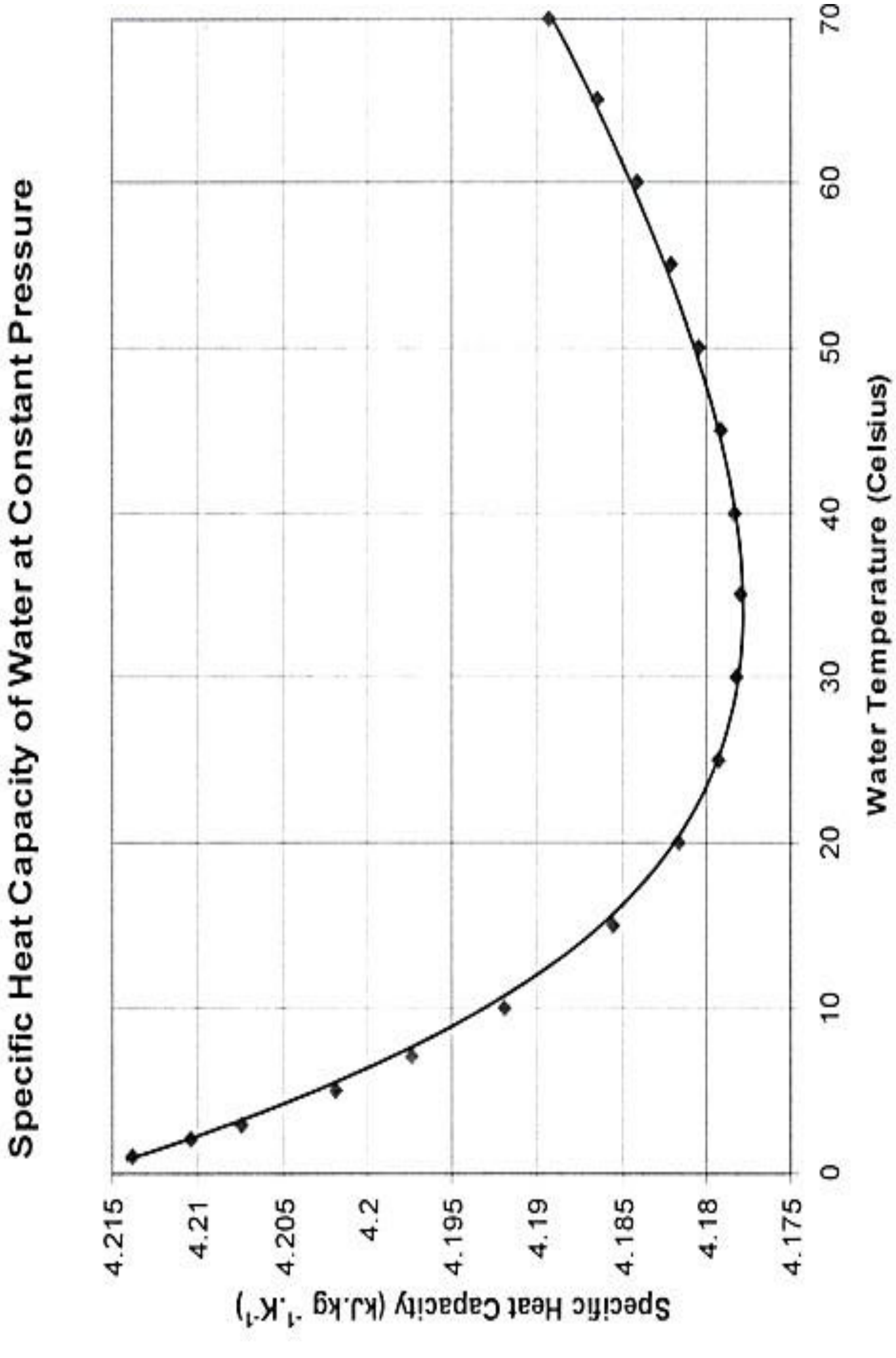
Parallel Flow												
Test	η_H	η_C	η	ρ_H	ρ_C	C_{PH}	C_{PC}	Q_e	Q_a	C_{EB}	LMTD	U
1												
2												
3												
4												

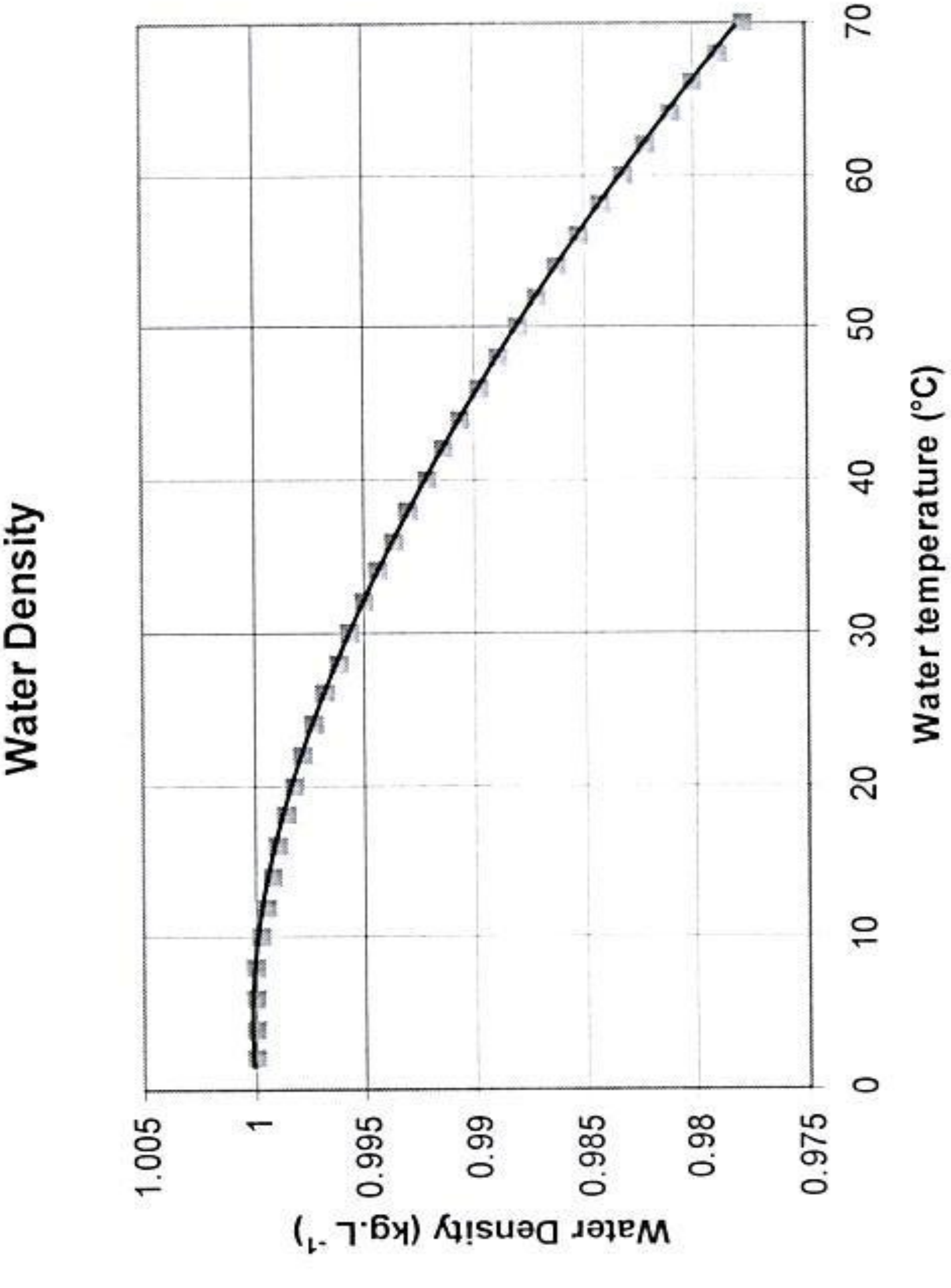
Table 2:

Counter Flow												
Test	η_H	η_C	η	ρ_H	ρ_C	C_{PH}	C_{PC}	Q_e	Q_a	C_{EB}	LMTD	U
1												
2												
3												
4												

Result Analysis:

- Discuss the deviation in results of counter and parallel flow.
- Write about the energy losses to the surroundings.


Specific Heat Capacity of Water at Constant Pressure
Specific Heat Capacity (kJ.kg -1.K-1)
4.215
4.21
4.205
4.2
4.195
4.19
4.185
4.18
4.175
0
10
20
30
40
50
60
70
Water Temperature (Celsius)


Water Density
Water Density (kg.L⁻¹)
1.005
1
0.995
0.99
0.985
0.98
0.975
0
10
20
30
40
50
60
70
Water temperature (°C)

Experiment 6

The Effect of Varying Temperature in parallel and Counter flow in Concentric Shell and Tube Heat Exchanger

Objective

- To show how different temperatures affect the performance of the heat exchanger in both parallel flow and counter flow connection (hot flow rate and cold flow rate are fixed).
- Temperature vs position graphs for both the parallel flow and counter flow for different temperatures.
- Calculation of power emitted, power absorbed, mean temperature efficiencies and energy balance for parallel and counter flow.
- Find LMTD and overall heat transfer coefficient (U) for each temperature.

Data Sheet:

Data Table 1:

Parallel flow connection

Hot water flow rate:2.86 L/min

Cold water flow rate:1.43 L/min

Ambient tank temperature:

Heater tank temperature:

Test	Heater set Temperature	T_{H1}	T_{H2}	ΔT_H	Average T_H	T_{H3}	T_{C1}	T_{C2}	ΔT_C	Average T_C	T_{C3}
1											
2											
3											
4											

Data Table 2:

Counter flow connection

Hot water flow rate:

Cold water flow rate:

Ambient tank temperature:

Heater tank temperature:

Test	Heater Set Temperature	T_{H1}	T_{H2}	ΔT_H	Average T_H	T_{H3}	T_{C1}	T_{C2}	ΔT_C	Average T_C	T_{C3}
1											
2											
3											
4											

Calculation Table:

Table 1:

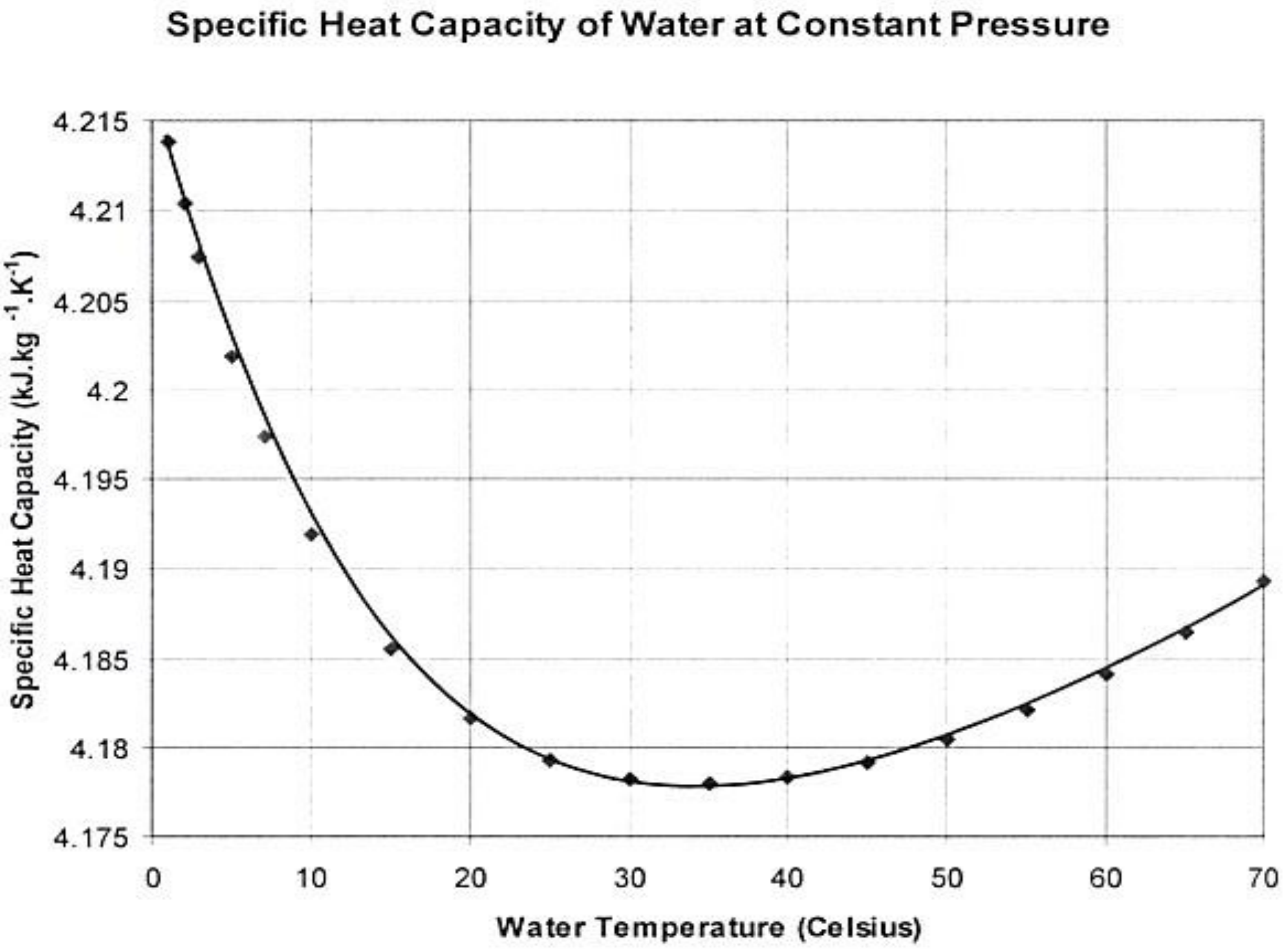
Parallel Flow												
Test	η_H	η_C	η	ρ_H	ρ_C	C_{PH}	C_{PC}	Q_e	Q_a	C_{EB}	LMTD	U
1												
2												
3												
4												

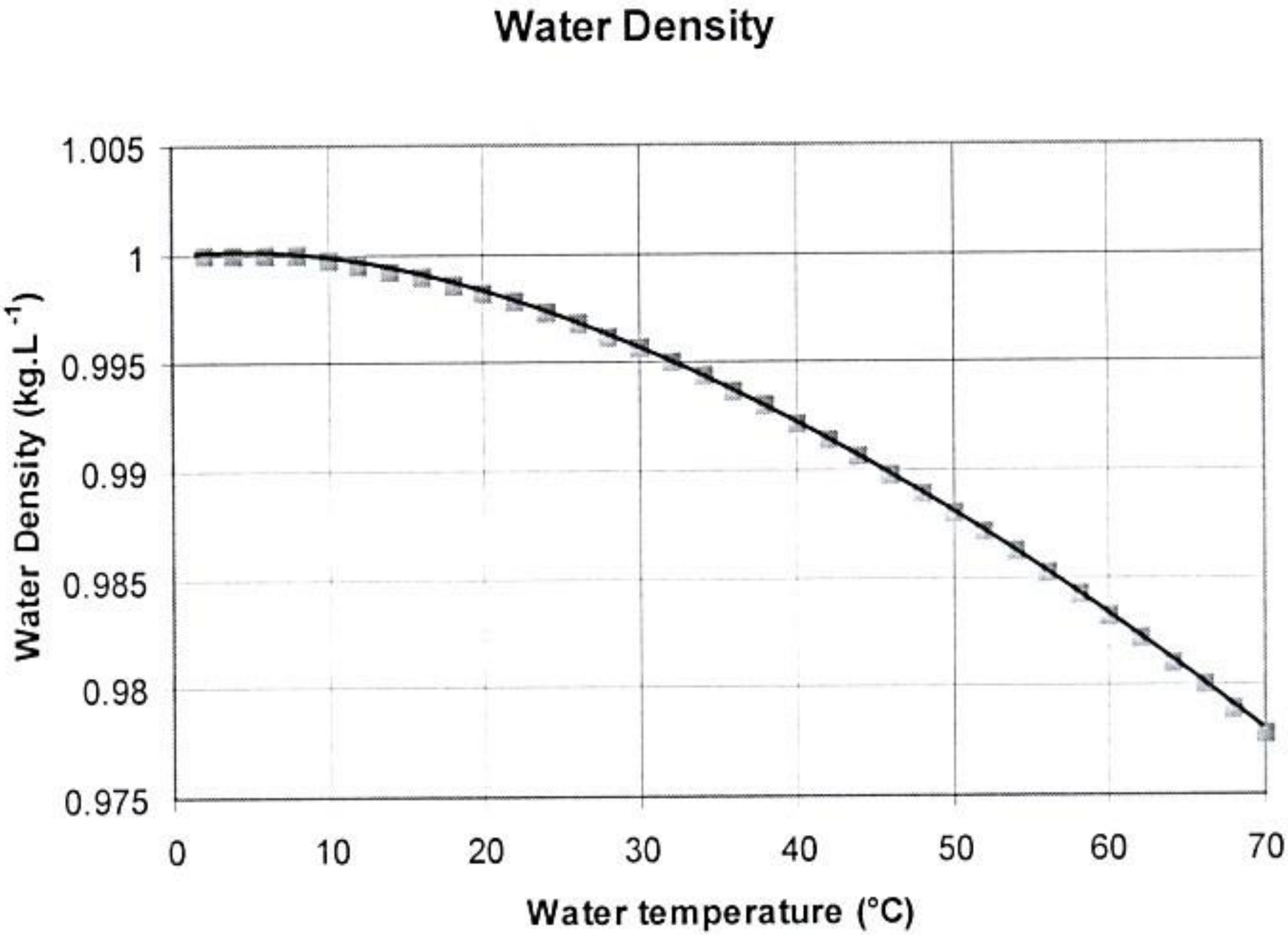
Table 2:

Counter Flow												
Test	η_H	η_C	η	ρ_H	ρ_C	C_{PH}	C_{PC}	Q_e	Q_a	C_{EB}	LMTD	U
1												
2												
3												
4												

Result Analysis:

- Discuss the deviation in results of counter and parallel flow.
- Write about the losses in surroundings.


Specific Heat Capacity of Water at Constant Pressure
Specific Heat Capacity (kJ.kg $^{-1}$.K^{-1})
4.215
4.21
4.205
4.2
4.195
4.19
4.185
4.18
4.175
0
10
20
30
40
50
60
70
Water Temperature (Celsius)


Water Density

Experiment 7

Transient Heat Transfer Analysis

Objectives

The objectives of this experiment are

1) To observe unsteady state conduction of the centre of a solid shape when a step change is applied to the temperature at the surface of the shape.
2) To Investigate the Lumped Thermal Capacitance method of transient temperature analysis.

Description

The Unsteady State Heat Transfer unit H111G is designed for the investigation of the temperature variation with time and heat flow within solid shapes that are subjected to sudden heating, Figures 1 and 2.

The unit comprises a stainless-steel water bath and integral flow duct with external water circulating pump. The bath has a capacity of approximately 30 liters and is heated by a thermostatically controlled *3* kW electric heating element in the base of the bath. Adjustment of the thermostat allows the bath to be set to a nominally constant temperature before beginning the experimental procedure. The heating element incorporates internal thermal protection so that power is switched off in the event of the heater being turned on when not covered by water. However, repeatedly allowing the heater to overheat in this wax should be avoided as it will eventually fail.

The bath, circulating pump, miniature circuit breakers and residual current circuit breaker are all mounted on a plastic base board for stability.

Seven simple shapes of different materials (Stainless steel and brass) are provided as shown in Figure 3 and each is fitted with a thermocouple well at its geometric centre to allow measurement of its core temperature. A carrier allows each of the shapes to be attached and a 350 mm duplex thermocouple **(T3)** to pass through the centre. A 250 mm thermocouple (T2) also fits into the carrier to sense water temperature adjacent to the shape.

The carrier is designed to centralize the shape under test in a 70mm diameter flow duct fitted centrally in the removable lid of the bath. Water from the tank is taken from a combined inlet and discharge fitting and circulated by the pump into the base of the flow duct. The water flows upwards past the shape under test and returns to the tank through overflow holes in the duct. The pump has a 3-speed selector switch. Changing the pump speed selector may vary the velocity of the water flow past the shapes and accordingly, vary the convection heat transfer coefficient.

Due to the small mass of the shapes relative to the large volume of water contained in the bath the bulk temperature of the bath remains essentially constant during the experimental period. This is confirmed by a fixed thermocouple **(Tl)** located in the flow duct and emerging from the lid.

Figure 1: Unsteady heat transfer experimental unit

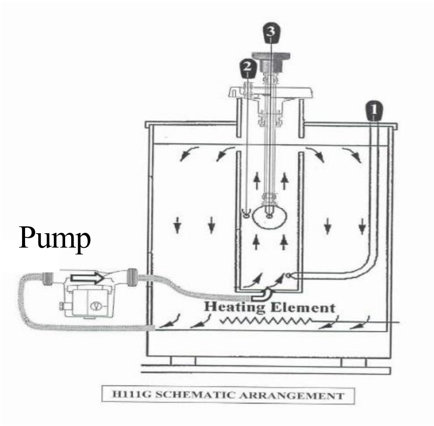


Figure 2: Schematic layout of the experimental unit

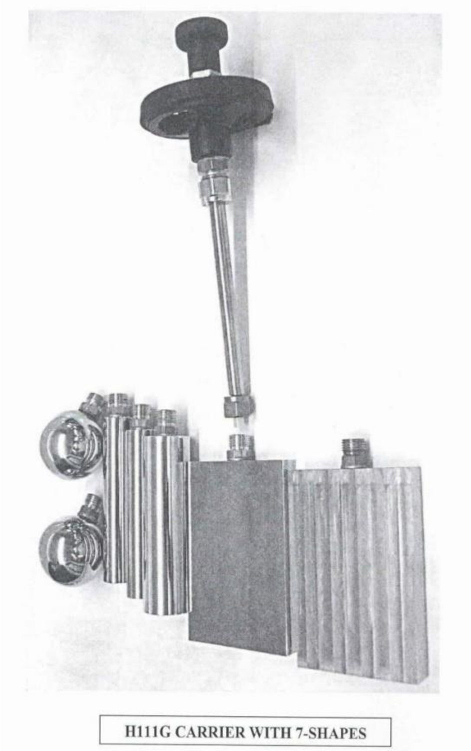


Figure 3: The seven shapes with their carrier

Filling the bath is most simply achieved using a hose and draining is achieved by connecting a hose to the drain point and opening the drain tap.

The three thermocouples (T1, T2, and T3) connect to the Heat Transfer Service unit H111 using miniature thermocouple plugs and matching sockets. Each may be selected using the selector switch and displayed to a resolution of 0.1°C.

Duplex thermocouples terminate with a screw connector for use by the Data Acquisition Upgrade.

For details of the solid shapes provided refer to Figure 3 and the following list.

- Cylinder 20mm diameter, 100mm long BRASS
- Cylinder 20mm diameter, 100mm long STAINLESS STEEL
  Cylinder 30mm diameter, 100mm long BRASS
- Slab 70mm x 15mm x 76mm BRASS
- Slab 70mm x 15mm x 76mm STAINLESS
  STEEL Sphere 45mm diameter BRASS
- Sphere 45mm diameter STAINLESS STEEL

Operating Procedure

1) Ensure the residual current circuit breaker (RCCB) located behind the unit is open- circuit. Ensure also that the drain valve adjacent to the circulating pump is in the closed position and half-fill the water bath with clean water

2) Pump bleeding - Switch on the RCCB/MCB to cause the pump to run. Incline the pump by lifting the baseboard from the front to allow air to escape. Noise from the pump is a sign of trapped air. A Bleed screw is fitted to the head of the pump.

3) Continue filling the water bath until the water level is at mid height of the holes in the duct. If the local water contains a large quantity of dissolved solids, that normally result in scale build up then it is recommended that the bath is filled with de-ionized or de- mineralized water. Ensure that the thermostat has been turned fully anti-clockwise and is in the off position.

4) Ensure that the H111 unit main switch is in the off position (None of the three digital displays should be illuminated). Ensure that the residual current circuit breaker on the rear panel is in the ON position. Ensure that the residual current circuit breaker on the base board is in the ON position.

5) Turn on the power supply to the Unsteady State Heat Transfer Unit and turn on the 16A bleater miniature circuit breaker (MCB). Ensure that the red power indicator adjacent to the thermostat is illuminated. Turn the thermostat to position 6 for fastest heating. The water will take approximately *30* minutes to heat from cold. At this setting the water will boil, if left unattended. While the water bath is heating the following may be done.

6) Attach the required test shape to the carrier by tightening the coupling nut (finger-tight).

7) Insert the T3 probe to engage fully into the centre of the shape.

8) Insert the T2 probe to sense water temperature adjacent to the shape.

9) Avoid touching the shape by hand to reduce thermal effects and place the shape on the bench to reach ambient temperature.

10) Turn on the power supply to the Heat Transfer Service Unit and turn on the main switch and the three digital displays should illuminate. Set the temperature selector switch to T1 to indicate the temperature of the water bath. Observe T1 to confirm that it is slowly increasing as the bath is heated.

11) Observe the temperature T1. If the bath temperature exceeds that is required, reduce the thermostat setting to OFF and wait for the water to cool. The water bath temperature T1 should not be allowed to exceed 85-90^{0}C as the pump will cavitate.

12) Having achieved the desired temperature, say 85C, reduce the thermostat setting to position 2. It will cycle ON/OFF to maintain the existing temperature.

13) If the Data Acquisition upgrade is being utilized, start recording data just before the test shape is plunged into the flow duct to ensure that all three temperatures are recording correctly and to obtain the start conditions. If readings are to be taken manually then it is necessary to use more than one operator. For manual operation it is sufficient to record only T1 at the specified time interval. The bath temperature TI and the flow duct temperature T2 are only recorded at the start (after plunging the test shape in the bath) and at the end of the test period, when the shape has reached the bath temperature T1.

14) When all experimental procedures that require the hot bath have been completed, then turn

off the RCCB and mains supply to the Unsteady State Heat Transfer Unit. Turn off the main switch on the Heat Transfer Service Unit Hill and turn off the mains supply.

15) If it is necessary to drain the bath allow the bath to cool completely.

Useful Data

Unsteady State Heat Transfer Unit H111G

Brass Test Shapes

Property	Symbol	value
Thermal conductivity	k	121 W/mK
Specific heat	c	385 J/kg
Density	ρ	7930 kg/m^3
Thermal diffusivity	α	3.7×10^{-5} m^2/s

Stainless steel Test Shapes

Property	Symbol	value
Thermal conductivity	k	16.3 W/mK
Specific heat	c	460 J/kg
Density	ρ	8500 kg/m^3
Thermal diffusivity	α	0.45×10^{-5} m^2/s

Experiment 7a:

Observation of Unsteady-State Heat Conduction to the Center of a Solid Subjected to a Step Change in Surface Temperature

The following procedure describes operation with the 30mm diameter brass test shape. However, the procedure is identical for any of the other available shapes and may be repeated on completion of the test using any of the other shapes for direct comparison.

Procedure

1) Follow the OPERATING PROCEDURE mentioned previously in order to establish the following operating conditions:
2) Install the 30mm diameter brass cylinder in the shape carrier G1(6).
3) The water bath temperature T1 should stabilize at approximately 80 to 90 ^{0}C.
4) Set the circulating pump to speed 3 and therefore the water flow velocity in the flow duct.
5) Record the starting condition temperatures and then plunge the shape in the flow duct. Then record temperatures and time as detailed in the Operating Procedure.
6) Repeat the procedure for the other brass cylinder (20 mm). To do that, take the carrier out, put it in a flask of cold water to cool it down to room temperature, replace the current brass cylinder (30 mm diameter) with the other brass (20 mm diameter) cylinder.
7) Repeat the experiment again using the stainless-steel cylinder (20 mm) and record the temperature- time history.

Data Sheet

Material	**Brass**	**Brass**	**Stainless Steel**
Diameter, **D** (mm)	20	30	20
Thermal Conductivity, **k** (W/m.K)	121	121	16.3
Thermal Diffusivity, **α** (m^2/s)	3.7×10^{-5}	3.7×10^{-5}	0.45×10^{-5}
Specific Heat, **c** (J/kgK)	385	385	460
Density, **ρ** (kg/m^3)	7930	7930	8500

Specimen: *20mm* diameter Brass Cylinder

Time, t (s)	***T_1 (ºC)***	***T_3 (ºC)***	***θ***	***Fo***	***1/Bi***	***Bi***	***h***
0 (initial)							
10							
20							
30							
40							
50							
60							
70							
80							
90							

Specimen: *30mm* diameter Brass Cylinder

Time, t (s)	***T_1 (ºC)***	***T_3 (ºC)***	***θ***	***Fo***	***1/Bi***	***Bi***	***h***
0 (initial)							
10							
20							
30							
40							
50							
60							
70							
80							
90							

Specimen: *20mm* diameter Stainless Steel Cylinder

Time, t (s)	***T_1 (ºC)***	***T_3 (ºC)***	***θ***	***Fo***	***1/Bi***	***Bi***	***h***
0 (initial)							
10							
20							
30							
40							
50							
60							
70							
80							
90							

Plot the Dimensionless temperature θ Vs time for the data recorded above to demonstrate effect of geometry and material.

The graph will show that the 20mm diameter brass cylinder has a faster response to immersion in the hot water bath than the 30mm diameter cylinder. Both are of the same material. The slower rate of response of the 30mm diameter cylinder is due to the additional 5mm annulus of material that heat has to pass through to reach the core of the specimen.

Though the 20mm diameter stainless steel cylinder has the same dimensions as the 20mm diameter brass cylinder its response is slower due to its lower thermal conductivity, k.

Calculations

In this section, we will use the data collected in the previous section to calculate the convective heat transfer coefficient for the three different cylindrical samples.

1) Consider any of the reading in the previous table (time, center line temperature T3 and bath temperature T1)
2) Calculate a dimensionless temperature θ. Where: Ti is the initial temperature of the specimen before we immerse it in the hot bath.
   (i) $\theta = \frac{T_3 - T_1}{T_i - T_1}$
3) Calculate the Fourier number (dimensionless quantity that represents time)
   (i) $Fo = \frac{\alpha t}{r^2}\ and\ \alpha = \frac{k}{\rho c}$
   (ii) Where: α: thermal diffusivity and r is the radius of the cylinder.
   (iii) Use the Values of θ_o and Fo in the Heisler Chart given later to get the value of Biot number, Bi. Then,

$$Bi = \frac{hr}{k}$$

4) The thermal conductivity is known, and the radius is also known. Therefore, we can estimate the convective heat transfer coefficient between the circulating water in the bath and the cylindrical sample.

Verify the use of the Heisler chart for the calculations performed by checking the value of Bi. If Bi is greater than 0.1, the use of the chart is justified. Otherwise, the lumped capacitance method can be used as illustrated in experiment 7b. The Heisler chart for cylindrical sample:

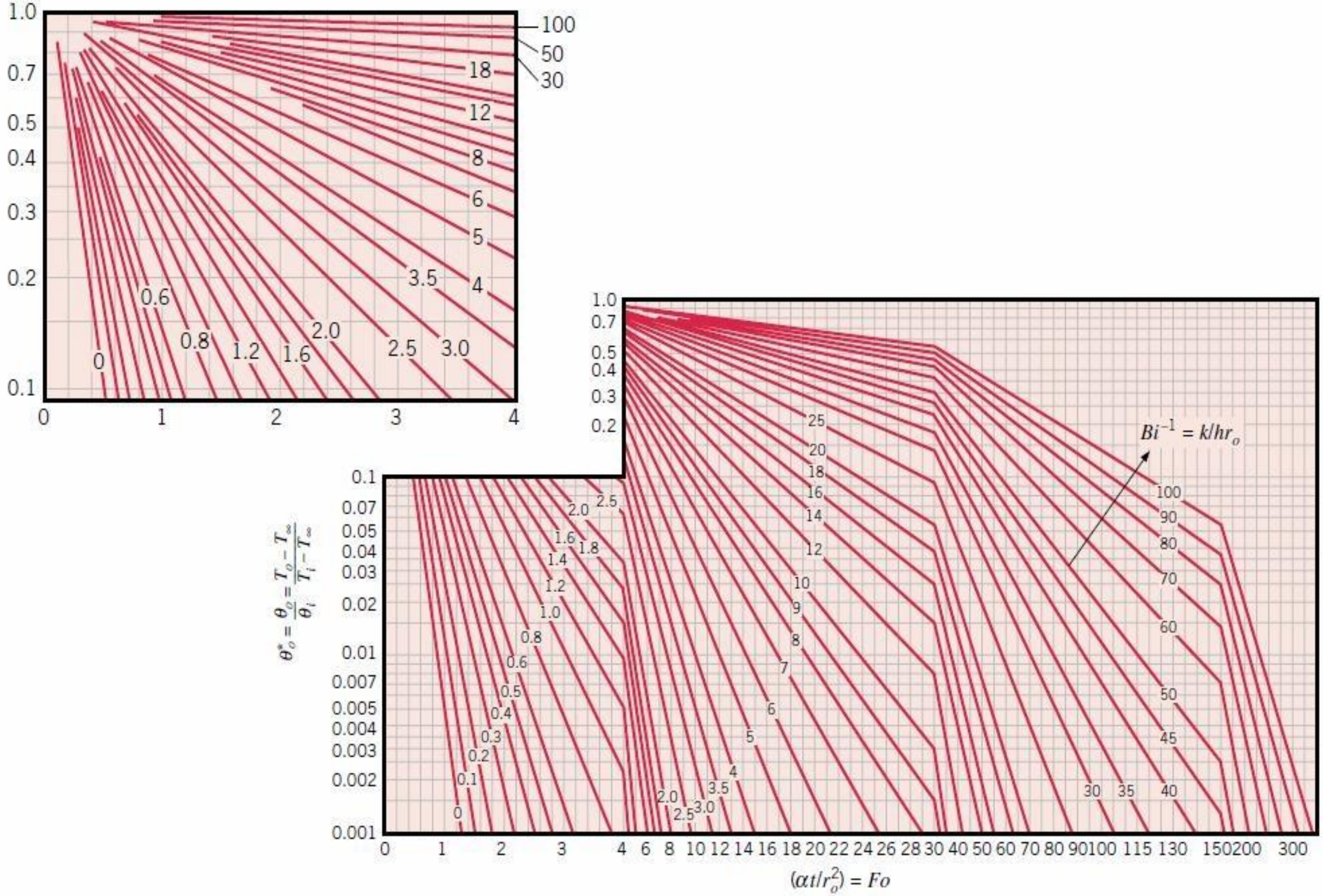


FIGURE 5S.4 Centerline temperature as a function of time for an infinite cylinder of radius r_o [1]. Used with permission.

Experiment 7b

Investigation of the Lumped Thermal Capacitance method of transient temperature analysis.

Experimental Procedure

The following procedure describes operation with the 45 mm diameter brass sphere.

1) Follow the OPERATING PROCEDURE on section 8.2 onwards in order to establish the following operating conditions:
2) Install the 45 mm diameter brass sphere in the shape carrier.
3) The water bath temperature T1 should be stabilized at approximately 80 to 90 ^{0}C.
4) Set the circulating pump to speed 3 and therefore the water flow velocity in the flow duct.
5) Place the shape in the flow duct and allow several minutes to reach a stable temperature. This should be close to the bath temperature T1.

6) Then remove the shape from the bath and place in still air in the carrier and record temperatures T2 and T3 at 30 second intervals until the shape has cooled close to the ambient temperature which will be indicated by T2.

Data Sheet for Lumped Thermal Capacitance Method*:*

Air temperature (T_∞) =   (Approximately 20-25° C)

Time (s)	$T_{3,exp}$ **(°C)**	θ	**ln**θ	h **(W/m^2-K)**	**Time constant Γ, (s)**	Bi	$T_{3,cal}$ **(°C)**
0							
30							
60							
90							
120							
150							
180							
210							
240							
270							
300							
330							
360							
390							
420							
450							
480							
510							
540							
570							
600							

Where : Diameter of the brass sphere, D = 45 mm

$$\theta = \frac{T_{3,exp} - T_\infty}{T_i - T_\infty}\ ; \qquad h = -\frac{\rho c D\ \text{In}\theta}{6t}\ ; \qquad Bi = \frac{hD}{6k}\ ; \qquad \Gamma = \frac{\rho c D}{6h}$$

$$T_{3,cal} = T_\infty + (T_i - T_\infty)\exp\left(\frac{-t}{\Gamma}\right)$$

Data Sheet for Lumped Thermal Capacitance Method*:*

Air temperature (T_∞) = (Approximately 20-25° C)

Time (s)	$T_{3,\,exp}$ (°C)	θ	lnθ	h (W/m^2-K)	Time constant Γ, (s)	Bi	$T_{3,\,cal}$ (°C)
0							
30							
60							
90							
120							
150							
180							
210							
240							
270							
300							
330							
360							
390							
420							
450							
480							
510							
540							
570							
600							

Where : Diameter of the steel sphere, D = 45 mm

$$\theta = \frac{T_{3,exp} - T_\infty}{T_i - T_\infty} \; ; \qquad h = -\frac{\rho c D \,\mathrm{In}\theta}{6t} \; ; \qquad Bi = \frac{hD}{6k} \; ; \qquad \Gamma = \frac{\rho c D}{6h}$$

$$T_{3,cal} = T_\infty + (T_i - T_\infty) \exp\left(\frac{-t}{\Gamma}\right)$$

Calculations

In this section, we will use the data collected in the previous section to calculate the heat transfer coefficient of the air in which the specimen is cooled.

We will also determine the time constant and use it to deduce the temperature profile as a function of time. Then it can be plotted and compared with experimental readings.

It is important to note that for this experiment, we assume that the Lumped Capacitance method applies, that is, $Bi < 0.1$. (this means that the conduction resistance within the specimen is less than the convection resistance around it since Bi represents the ratio between conduction and convection resistances)

1) Consider any of the reading in the previous table
2) Calculate a dimensionless temperature
   i. $\theta = \frac{T_{3,exp} - T_\infty}{T_i - T_\infty}$

   Where: T_i is the initial temperature of the specimen before cooling begins (after we take it out).

3) The following equations are used to find the heat transfer coefficient and time constant:

$$h = -\frac{\rho c D \, \mathrm{In}\theta}{6t} \; and \; \Gamma = \frac{\rho c D}{6h}$$

Where Γ is the time constant.

4) Calculate the volume and surface area of the specimen (for spherical shapes, $V = 4\pi r^3/3$ and $A_s = 4\pi r^2$). Note also that the specimen properties are known (section 8.3) Verify the assumption of the use of Lumped capacitance method through calculating the Biot number $Bi = hl/k$. where l is the characteristic length (V/A_s). For s sphere, $l = D/6 = r/3$.

$$Bi = \frac{hD}{6k} = \frac{hr}{3k}$$

5) Thus, we can obtain the calculated temperature by using the Lumped capacitance method:

$$T_{3,cal} = T_\infty + (T_i - T_\infty) \exp\left(\frac{-t}{\Gamma}\right)$$

6) Plot the calculated temperature profile ($T_{3,\ cal}$ versus t) in a graph sheet along with the measured values of temperature ($T_{3,\ exp}$). Is there any discrepancy, why?

7) Plot ($T_{3,\ exp}$) Brass and ($T_{3,\ exp}$) Steel, Versus time. Comment and discuss the results comparing Brass and Steel for the Lumped Capacitance method.

Write a technical report to show your experiment steps, calculations, discussion of your results and graphs, conclusion and references. You may also suggest ways to improve this experiment.

Experiment 8

Determination of mass diffusion coefficient of NaCl in water

Introduction

Diffusion

Diffusion is the net movement of substance (Example: atom, ions, and molecules) from a region of higher concentration to a region of lower concentration. Diffusion is driven by a concentration gradient of the object in the solution.

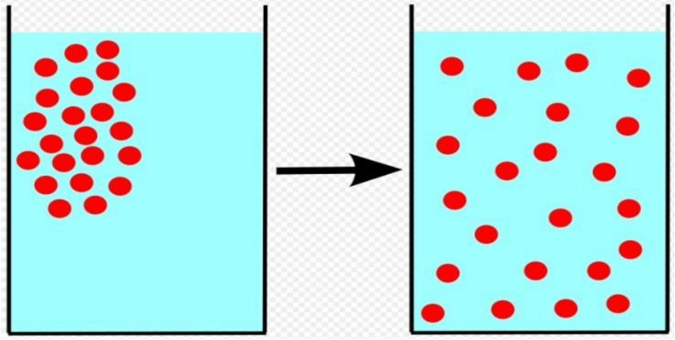


Figure 1: Diffusion *of molecules of a substance in a solution.*

Fick’s law

Fick's Law essentially states the rate of diffusion is proportional to both the surface area and concentration difference and is inversely proportional to the thickness of the membrane.

$$J = D_A \cdot A \cdot \frac{dC}{dx} \qquad \textbf{(1)}$$

Here,

J = mass transfer flux, which is defined as the transfer of the total no. of mole of the solute from one place to another at a unit time. It is denoted by $\frac{dmole}{dt}$.

D_A=Mass transfer coefficient
A=Area of one capillary
dC= Concentration gradient
x= distance

Now, Equation (1) can be rewritten as:

$$\frac{dmole}{dt} = D_A \cdot \left(\frac{\pi}{4} D^2 \cdot N\right) \cdot \frac{M}{x} \qquad \textbf{(2)} \text{ and } \qquad \frac{dmole}{dt} = V \cdot \frac{dM}{dt}$$

Where,

N= Total number of capillaries = 317

D= Diameter of capillary = 1.0 mm

X= Thickness of membrane = 5.0 mm

M= Molarity of NaCl solution in the diffusion cell =2 M

V= Amount of water in the solution = 60 mL

Now equation (2) can be written as:

$$V \cdot \left(\frac{dM}{dt}\right) = D_A \cdot \left(\frac{\pi D^2}{4} \cdot N\right) \cdot \frac{M}{x}$$

$$\frac{dM}{dt} = \frac{D_A \cdot \left(\frac{\pi D^2}{4} \cdot N\right) \cdot \frac{M}{x}}{V} \qquad \textbf{(3)}$$

Again, $C_M = \frac{dk}{dM}$ i.e. $dM = \frac{dk}{C_M}$ where, C_M is the rate of change of electrical conductivity (k) with change in molarity. Substituting $dM = \frac{dk}{C_M}$ in the equation (3),

$$\frac{1}{C_M} \cdot \left(\frac{dk}{dt}\right) = \frac{D_A \cdot \left(\frac{\pi D^2}{4} \cdot N\right) \cdot \frac{M}{x}}{V}$$

$$D_A = \frac{\frac{1}{C_M} \cdot \left(\frac{dk}{dt}\right) \cdot V}{\left(\frac{\pi D^2}{4} \cdot N\right) \cdot \frac{M}{x}} = \frac{\left(\frac{dk}{dt}\right) \cdot V \cdot 4 \cdot x}{\pi \cdot N \cdot D^2 \cdot M \cdot C_M} \qquad \textbf{(4)}$$

Thus, we can obtain the mass transfer coefficient from equation (4) once we obtain the two parameters $\frac{dk}{dt}$ and C_M by carrying out the experiment.

Objective

1. To plot electrical conductivity (k) Vs molarity (M) in a graph to observe the rate of change of electrical conductivity with a gradual increase in molarity.
2. To plot electrical conductivity (k) Vs time (t) graph to observe the rate of change of electrical conductivity with time in case of specific molar solution
3. To determine the mass transfer coefficient (D_A)

Apparatus

- Diffusion cell
- Capillaries
- Conductivity Probe and Conductivity meter
- Diffusion vessel
- Temperature Sensor
- Heating element
- NaCl salt (15 grams)
- Distilled water

Experimental Setup

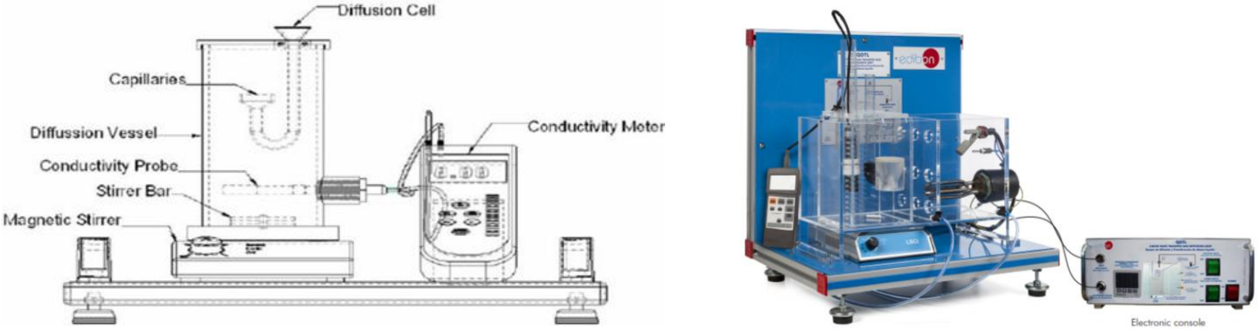


Fig 1. (a) Schematic diagram of mass diffusion experimental setup and (b) Picture of the setup.

Because of the variation in concentration, diffusion of NaCl occurs. The experiment is to be conducted at a fixed temperature (35˚C) as electrical conductivity (k) changes with temperature.

During the experiment, the mass required to prepare solution of NaCl is given by:

$$mass\ of\ NaCl\ (grams) = \frac{58.5 \cdot V \cdot M}{1000} \qquad \textbf{(5)}$$

(Note: Molecular weight of NaCl is 58.44 g/mol.)

Data Sheet:

Table 1: Data table for determination of C_M

(Note: Data will be used to generate graph and evaluate the slope dk/dM)

Amount of NaCl (grams)	**Molarity (M)**	**Electrical conductivity k (Siemens/m or S/m)**

Table 2: Data table to determine dk/dt

Time, t (seconds)	**Electrical conductivity, k (siemens/m or S/m)**
0	
10	
20	
30	
40	
50	
60	
70	
80	
90	
100	
110	
120	
130	
140	
150	
160	

Report

- Objectives
- Theory
- Experimental Setup
- Procedure
- Data Table
- Graphs
- Sample Calculation
- Discussion
- Conclusions