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7.	19-08-24	Exp 4: Studies on Heat Transfer through Composite wall.	Uma Kiran 26/08/24.
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9.	26-08-24	Exp 3: Studies on Heat Transfer in - Vertical Condenser.	
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Aim:

i) To plot the variation of temperature along the length of pin under forced convection.

ii) To determine the value of heat transfer coefficient under forced-convection and to find-

a) Theoretical values of temperatures along the length of the fin.

b) Effectiveness and efficiency of the pin fin for insulated end condition.

Theory:

The heat transfer phenomenon from a heated surface to its ambient is given by the relation, $q = hA\Delta T$. In this relation,

$h \rightarrow$ heat transfer coefficient ($W/m^2 \cdot K$)

$A \rightarrow$ The area normal to (m^2)

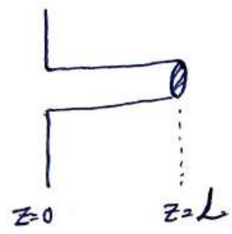
$\Delta T \rightarrow$ temperature difference (K)

the value of heat transfer can be increased easily by increasing the value of A, i.e, increasing the surface area.

The surface area is increased by attaching extra material in the form of rod on the surface whose heat transfer rate is to be increased. This extra material attached is called the '<u>Fin</u>'.

Depending on the shape of the surface the fin is attached to fins can be of many types such as circumferential fin for cylindrical surface.

now for a pin fin with an insulated tip;



$$\left.\frac{d\theta}{dz}\right|_{z=L} = 0.$$

and we know.

$$\theta(z) = C_1 e^{mz} + C_2 e^{-mz}. \quad [\text{by } Q_{cond} = Q_{conv}]$$

now.

$$C_1 + C_2 = \theta_0$$

$$C_1 e^{mL} - C_2 e^{-mL} = 0$$

So, $$C_1 = \frac{\theta_0}{1+e^{2mL}} = \frac{\theta_0 e^{-mL}}{e^{-mL}+e^{mL}}$$

$$C_2 = \frac{\theta_0 e^{mL}}{e^{-mL}+e^{mL}}$$

So. $$\theta(z) = \theta_0 \frac{e^{-m(L-z)} + e^{m(L-z)}}{e^{-mL} + e^{mL}}$$

$$\therefore \frac{\theta(z)}{\theta_0} = \frac{\cosh m(L-z)}{\cosh mL}$$

now,

$$q = \theta_0 \sqrt{h_c P K A} \tanh(mL) = \text{amount of heat flow (W)}$$

where,

$\theta_0 = (T_0 - T_\infty)$ (°C)

$\theta = (T - T_\infty)$ (°C)

T = Temperature at any distance z from the base on the fin (°C)

T_0 = Temperature at $z=0$ (°C)

T_∞ = ambient Temperature (°C)

L = Length of the fin (m)

$$m = \sqrt{\frac{h_c P}{KA}}$$

where,

h_c = convective heat transfer coefficient ($W/m^2 \cdot K$)

P = perimeter (m)

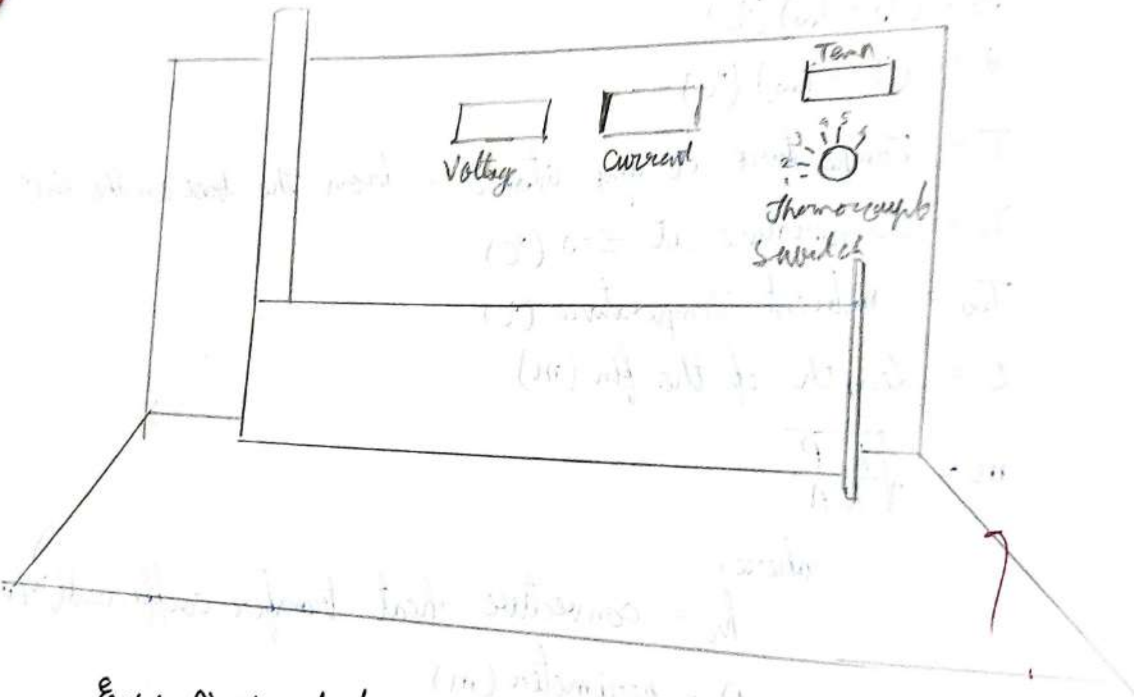
A = Area of fin (m^2)

K = Thermal conductivity of the fin material ($W/m \cdot K$)

So,

$$\text{effectiveness } (\varepsilon) = \sqrt{\frac{PK}{h_c A}} \tanh(mL)$$

$$\text{efficiency } (\eta) = \left[\frac{\tanh(mL)}{mL}\right] \times 100\%$$



Exp: Pin Fin heat exchanger experimental setup.

Procedure:

→ connect the equipment to electric power supply and keep the thermocouple selector switch to zero position.

→ Turn the dimmerstat knob clockwise and adjust the power-input to the heater to the desired value.

→ Set the air flow rate to the desired value by adjusting the difference in water levels in the manometer.

→ Tabulate time vs. temperature for intervals of about 3 minutes. Allow sufficient time to attain steady state.

→ Note down the difference in levels of the manometer.

→ Repeat the experiment for different power inputs to the heater.

Observation:

Length of the fin = 150 mm

diameter of the fin = 12 mm

Thermal conductivity of the material = 110 W/m.K

$d_{orifice} = 0.02\,m$ $\rho_{fluid} = 1000\,kg/m^3$

$w_{duct} = 0.15\,m$

$b_{duct} = 0.1\,m$

coefficient of discharge of the orifice = 0.61

Exp-2 (Pin-fin Expt.)

Voltage	Current	Manometer	T_1	T_2	T_3	T_4	T_5	T_6
40V	0.19 A	Δh = 2cm	60	47.9	45.5	43.2	42.2	36.7
	0.19A	2Cm	61	48.2	45.4	43.6	42	36.6
	0.19A	2Cm	61.8	48.3	45.6	43.6	41.6	37
50V	0.26A	2cm	64	49.5	46.2	44	42.3	37.3
	0.26 A	2Cm	66.4	50.5	47.1	44.7	42.9	37.5
	0.26 A	2Cm	68.4	51.4	47.8	45.3	43.4	37.5

Length of the fin = 150mm

diameter of the fin = 12mm

Thermal conductivity of the material : 110 W/m.K.

$d_{orifice}$ = 0.02 m

W_{duct} = 0.15 m

b_{duct} = 0.1 m.

Saurabh

Coefficient of discharge of the orifice = 0.61.

ρ_{fluid} = 1000 Kg/m³

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Serial No.	Heat Input			Pressure drop of water (cm)	Temperature (°C)					
	volt (V)	Current (A)	Power (W)		T_1	T_2	T_3	T_4	T_5	T_6
1.	40	0.19	7.6	2	60	47.9	45.5	43.2	42.2	36.7
2.	40	0.19	7.6	2	61	48.2	45.4	43.6	42	36.6
3.	40	0.19	7.6	2	61.8	48.9	45.6	43.6	41.6	37.[illegible]
4.	50	0.26	13	2	64	49.5	46.2	44	42.3	37.3
5.	50	0.26	13	2	66.4	50.5	47.1	44.7	42.9	37.5
6.	50	0.26	13	2	68.4	51.4	47.8	45.3	43.4	37.5

• Calculation:

velocity at the orifice: $V_0 = C_d \sqrt{\dfrac{2gh(\rho_f - \rho_a)}{\rho_a(1-\beta^4)}}$

$C_d = 0.61$ $g = 9.81\ m/s^2$ $\Delta h = 0.02\ m$, $\rho_f = 1000\ kg/m^3$

$\rho_a = 1.17\ kg/m^3$, $\beta = \dfrac{d}{D} = \dfrac{0.02}{0.0381} = 0.52$

$V_0 = 0.61 \sqrt{\dfrac{2 \times 9.81(1000 - 1.17) \times 0.02}{1.17(1 - 0.52^4)}}\ m/s = 11.597\ m/s$

now, $A_d V_d = A_o V_o$

so, $V_d = \frac{A_o V_o}{W b}$

$\Rightarrow V_d = \frac{\frac{\pi}{4} \times 0.02^2}{0.15 \times 0.1} \times 11.597$

$\therefore V_d = 0.243$ m/s (velocity in the duct)

now,
$$Re = \frac{\rho V d}{\mu}, \quad T_\infty = 23.5^\circ C$$

as we know.
$$T_{avg\,i} = \left.\frac{\sum_{j=1}^{6} T_j}{6}\right|_{\text{for SL. no } i} \qquad T_m|_i = \frac{T_{avg\,i} + T_\infty}{2}$$

So

SL. no.	Tavg (°C)	T_m (°C)
1	45.9	34.70
2	46.1	34.80
3	46.4	34.95
4	47.2	35.35
5	48.1	35.8
6	49	36.25

Physical Properties of air at, 35°C is as following.

$Pr = 0.712, \quad \rho = 1.145 \, kg/m^3$

$\mu = 1.8915 \times 10^{-5} \, kg/m \cdot s$

$\nu = 1.651 \times 10^{-5} \, m^2/s$.

so.
$$Re = \frac{V d}{\nu} = \frac{0.243 \times 0.012}{1.651 \times 10^{-5}}$$

$\therefore Re = 176.62$

now,
$$Nu = 0.683\,(Re)^{0.466}\,(Pr)^{1/3}$$

$$\Rightarrow Nu = 6.798$$

$$\Rightarrow \frac{h_c d}{k} = 6.798$$

$$\Rightarrow h_c = \frac{6.798 \times 0.0267}{0.012}$$

$$\therefore h_c = 15.12 \ W/m^2 \cdot K$$

now,
$$m = \sqrt{\frac{h_c P}{k A_c}} = \sqrt{\frac{15.12}{110 \times \frac{0.012}{4}}}$$

$$\therefore \boxed{m = 6.769 \ /m}$$

now.
$$\frac{T - T_\infty}{T_0 - T_\infty} = \frac{\cosh(m(L - z))}{\cosh(mL)}$$

$$\therefore T = T_\infty + (T_0 - T_\infty)\frac{\cosh(m(L - z))}{\cosh(mL)}$$

$$\therefore T = 23.5(°C) + (T_0 - 23.5)\frac{\cosh(6.769\,(0.15 - z))}{\cosh(6.789 \times 0.15)}$$

P.T.O.

now lets calculate fin effectiveness.

$$\varepsilon = \sqrt{\frac{PK}{h_c A}} \tanh(mL)$$

$$\varepsilon = \sqrt{\frac{4K}{h_c d}} \tanh(mL)$$

$$\varepsilon = \sqrt{\frac{4 \times 110}{15.12 \times 0.012}} \tanh(0.15 \times 6.769)$$

$$\varepsilon = 37.82$$

and now calculating the efficiency,

$$\eta = \frac{\tanh(mL)}{mL} \times 100\%$$

$$\eta = \frac{\tanh(0.15 \times 6.769)}{0.15 \times 6.769} \times 100\%$$

$$\eta = (0.756 \times 100)\%$$

$$\therefore \eta = 75.6\%$$

So, the fin effectiveness and efficiency are 37.82 and 75.6% respectively.

now,
for, 40V
as derived earlier,

$$T = 23.5(°C) + (T_0 - 23.5(°C))\ 0.641 \cosh(6.769(0.15-z))$$

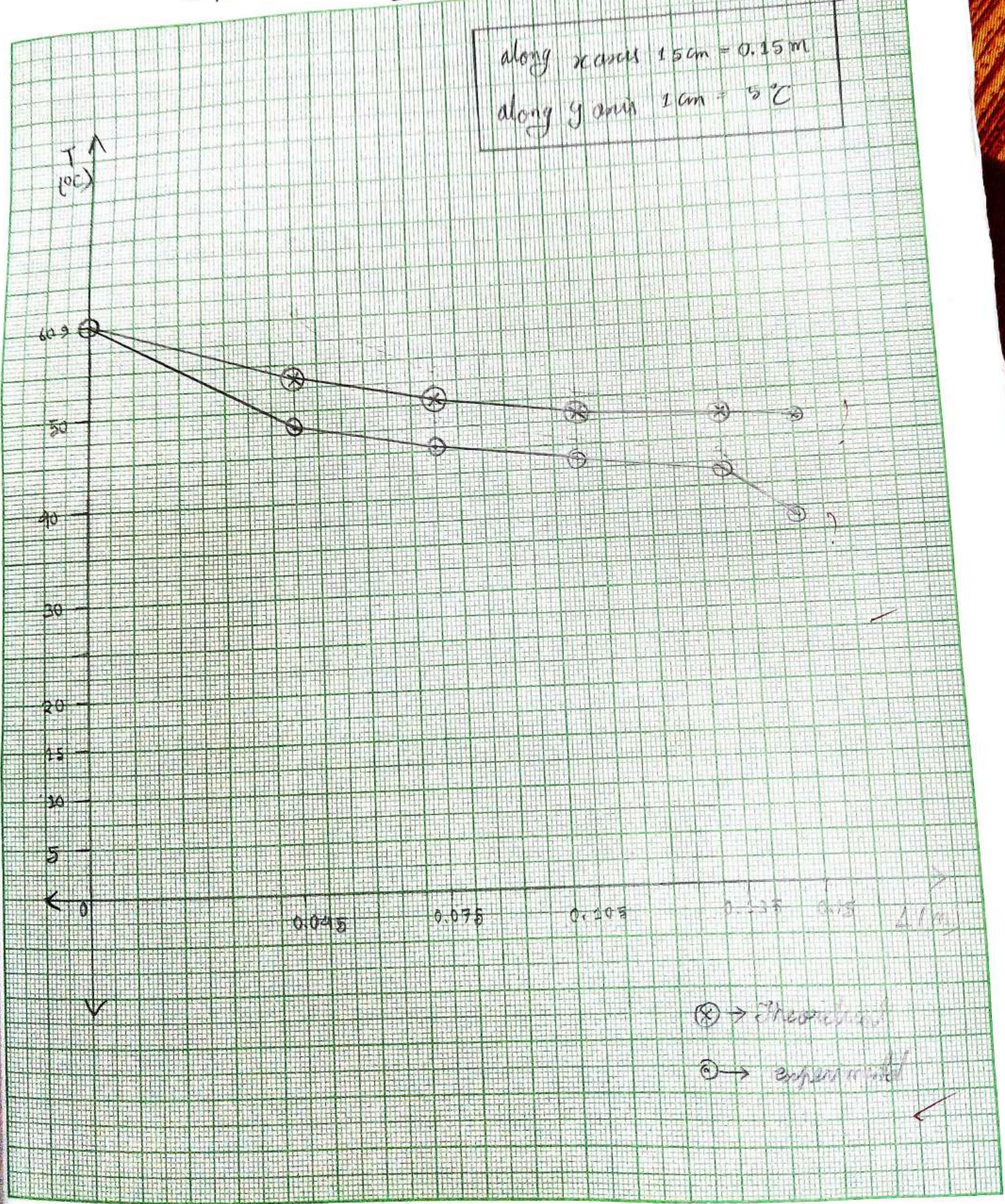
$$T_0 = 60.9°C$$

$$\therefore T = 23.5(°C) + 23.97(°C) \cosh(6.769(0.15-z))$$

now, lets compare,

Distance (m)	Temperature from the experiment (°C)	Temperature from the calculation (°C)
0	60.9	60.90
0.045	48.3	53.78
0.075	45.5	50.63
0.105	43.5	48.59
0.135	41.9	47.59
0.15	36.77	47.47

Comparison between experimental and theoretical data for T vs L Plot
@ 40V



now, for, 50V.

as derived earlier,

for, $T_0 = 66.27^\circ C$.

now,

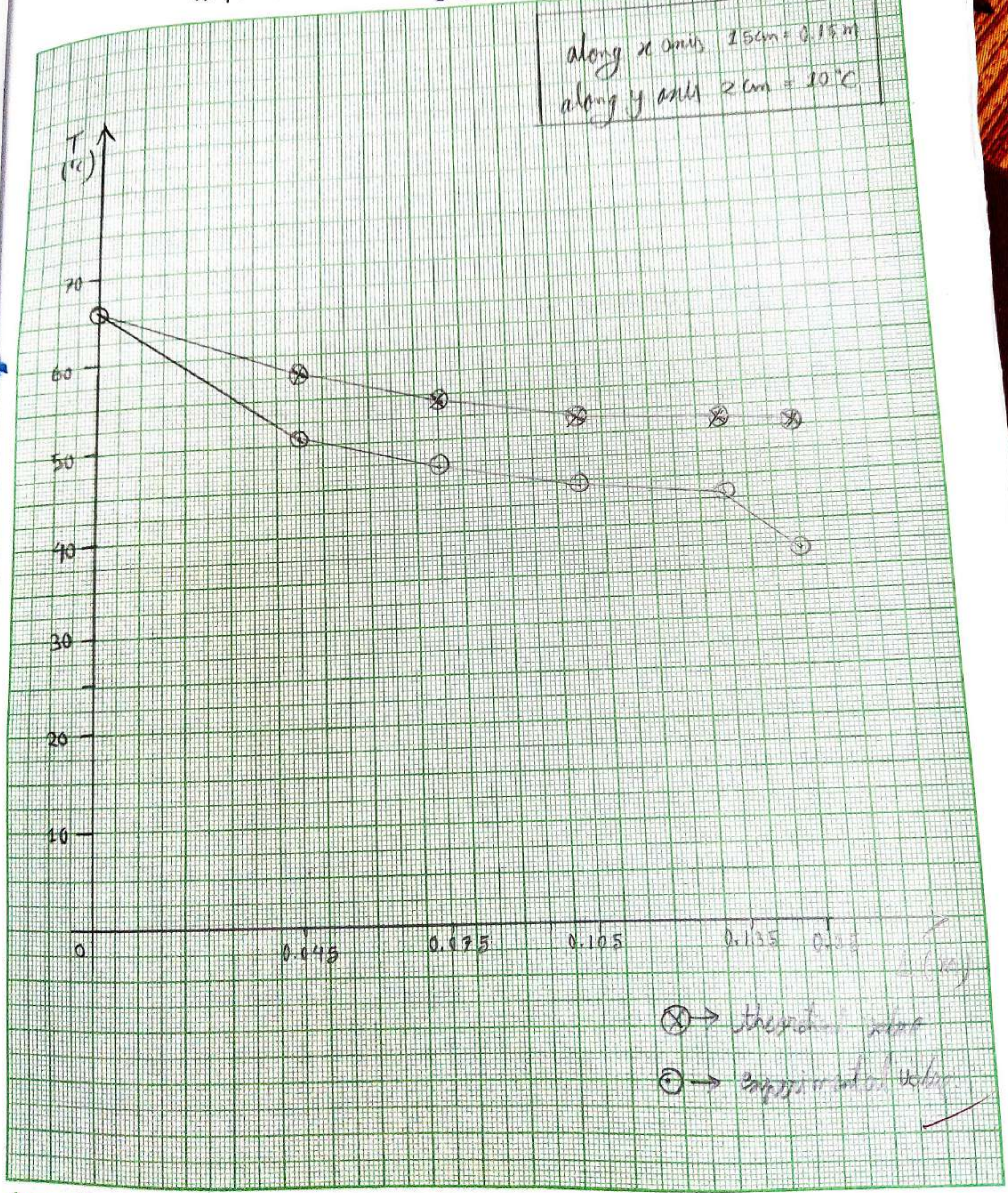
$$T = 23.5(^\circ C) + (66.27 - 23.5)\, 0.641 \cosh(6.769(0.15 - z))$$

$$\therefore T = 23.5(^\circ C) + 27.42(^\circ C) \cosh(6.769(0.15 - z))$$

now lets compare,

Distance (m)	Temperature from the experiment (°C)	Temperature from the Calculation (°C)
0	66.27	66.27
0.045	50.47	58.14
0.075	47.03	54.53
0.105	44.67	52.20
0.135	42.87	51.06
0.15	37.43	50.92

Comparison between theoretical and experimental data for T vs L plot.
@ 50V



• Precautions:

i) Maximum operating voltage should be 100 volts

ii) Maximum manometer difference will be 4 cm.

iii) Need to avoid Paralax error while calculating / taking reading from the manomater.

iv) Need to wait until readings in the digital screen of the thermo couple becomes stable.

• Discussions:

→ Pin fins are used to increase the heat transfer from a heated surface to air.

→ The rate of convection by introduction of airflow increases the rate of convection.

→ For very high temperature radiation becomes also prominent in the fins.

→ Fin is considered ideal if its' temperature is equal to its surface-'s temperature.

→ $\varepsilon|_{ideal\ case} > \varepsilon|_{real\ case}$.

→ Fins are cost effective unlike any other way of increasing the rate of heat transfer.

→ For forced convection experiment, we should fill the monometer with water and close the duct cover, as well as guarantee correct earthing to the unit.

→ Accuracy in the reading of the thermocouple depends on its installation in the system

→ The output voltage of a thermocouple depends on temperature as follows

$$V = AT + \frac{1}{2} BT^2 + \frac{1}{3} CT^3.$$

So. sensitivity is,

$$\frac{\partial V}{\partial T} = S = (A + BT + CT^2)$$

very high variation of will be shown in the readings for small fluctuations in the temperature due to its dependence on 3rd power of temperature

for, 2°C variation we show O(8) variation in the voltage. considering, $V = O(T^3)$

→ Pressure calculation also contains some error,

as, $\Delta h = h_2 - h_1$

the error is of 2 x ~~min~~ Least count.

• Critical comments:

→ Need to be careful with the thermocouple and the manometer reading.

→ To apply adiabatic condition we need to apply the insulation to fins tip.

→ The temperature change theoritically is smaller compared to which occurs in the experiment the reason for this essentially is that no material is 100% insulated.

• Results/Conclusion:

the effectiveness of the fin (ε) = 37.82

the efficiency of the fin under forced convection (η) = 75.6%

as the $\varepsilon \geq 2$, so the application of fin in this situation is perfectly fine.

→ The ~~sudden~~ drop near the the end of the fin is due to non-adiabatic condition, which is assumed otherwise.

UCP 05/8

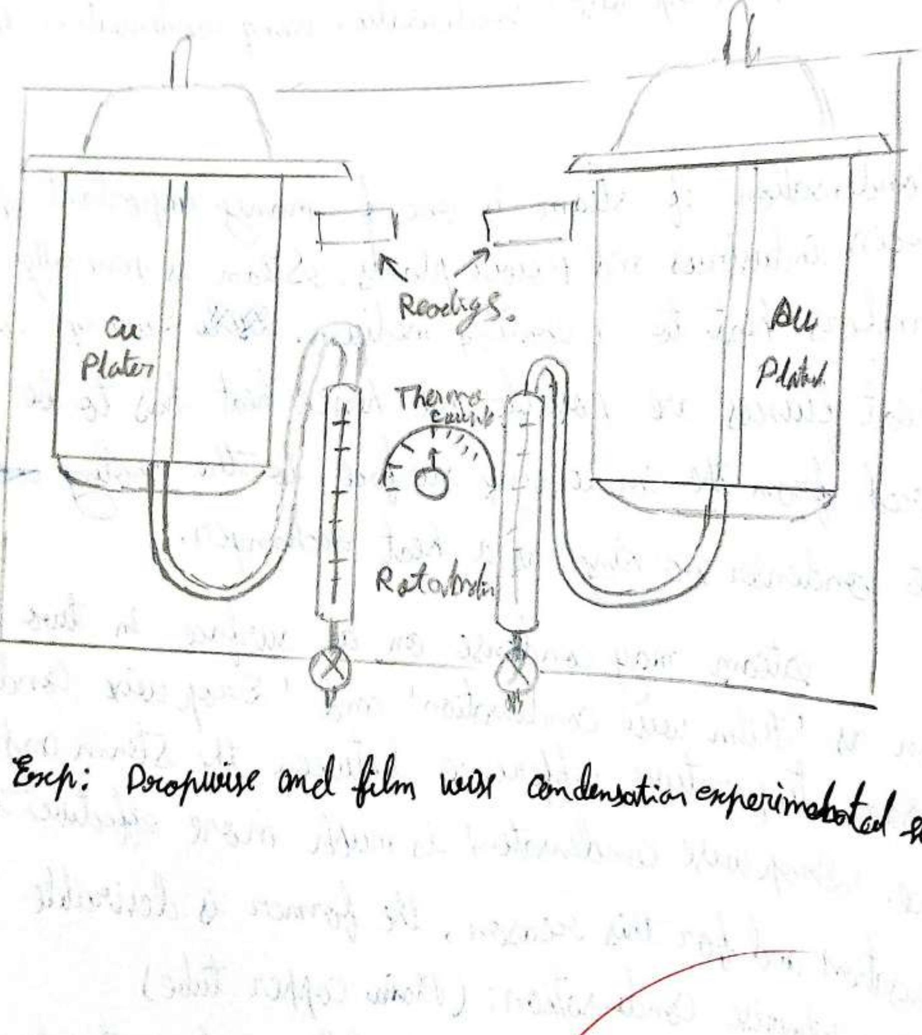
Aim:
To determine the individual Heat transfer coefficient in 'Film wise' condensation or, 'Drop wise' condensation using condensation Apparatus.

Theory:
Condensation of steam is one of many important processes occuring in process industries and Power plants. Steam is generally condensed as it transfers heat to a cooling medium. During condensation very high heat fluxes are possible and hence heat has to be quickly transferred from the condensing surface to the cooling medium in the condenser working as a heat exchanger.

Steam may condense on a surface in two distinct modes, known as 'Film wise condensation' and 'Drop wise Condensation' for the same temperature difference between the steam and the condensing surface. 'Drop wise condensation' is much more effective than 'film wise condensation and for this reason, the former is desirable.

- Filmwise Condensation: (Plain copper tube)

Unless specially treated, most materials are wettable and as condensation occurs a film of condensate spreads over the surface. The thickness of the film depends upon a number of factors such as (i) The rate of condensation (ii) The viscosity of the condensate (iii) orientation of condensing surface.



Exp: Dropwise and film wise condensation experimentated setup.

After awhile the film reaches an equilibrium thickness.

- Dropwise Condensation: (Gold plated copper tube)

By treating the condensing surface specially, such as - plating it with aluminium. It can be made 'non-wettable'. As steam condenses, generally a large number of spherical beads cover the surface. As condensation proceeds, the beads become large, collides and then trickle downwards from the condensing surface. The moving beads gathers on the static beads along its trail. The 'Bare' surface offers very little resistance so the transfer of heat and very high heat fluxes are therefore possible.

• Experimental Setup:

The equipment consists on a metallic container in which steam generation takes place. A suitable electric heater is installed in the lower portion of the container which heats water and facilitates steam generation. To regulate the rate of steam generation the input voltage to the heater can be altered by means of a voltage regulator. An opening is provided in the cover for filling water. The glass ~~cla~~ cylinder houses two water cooled copper condensers, one of which is chromium plated to promote dropwise condensation and the other is in its natural state to give film wise condensation. A pressure gauge is provided to measure the steam-pressure. Separate cooling water connections are provided to the two condensers. A rotameter is provided to measure flow rate of cooling water through the condenser. A multichannel digital temperature indicator is provided to measure temperature of steam, condenser surfaces, condenser cooling water inlet and outlet.

• Specifications:

copper tube condenser for film wise condensation.

inner diameter, $d_i = 11$ mm
outer diameter, $d_o = 12.7$ mm
Length of the tube, $L = 250$ mm

Gold plated condenser for drop wise condensation:

inner diameter, $d_i = 11$ mm
outer diameter, $d_o = 12.7$ mm
Length of the tube, $L = 250$ mm

Procedure:

→ Fill the steam generator with water to little over half, checking the water level in the tank by a level indicator.

→ Allow the cooling water to flow through one of the condensors, which is selected for the test and note down the water flow-rate from the rotameter.

→ Switch on the heater. Adjust the voltage across heater to be around 150V as indicated by voltmeter. Wait till pressure becomes 0.5 kg/cm²

→ Allow the steam slowly to flow through the glass chamber, as the steam flows over the selected test section it gets condensed and falls to the bottom of the glass container and drains out through the drain valve. Maintain the pressure in the steam generator constant by increasing and decreasing voltage across the heater according to the variation in the pressure. Depending upon the type of the condenser selected drop wise or filmwise condensation can be visualized.

→ Note down the water flow rate, steam pressure, temperatures T_1 to T_9 and enter them to table.

Exp - 9 (Dropwise and Filmwise condensation)

Gold platted surface - Dropwise.
Copper platted surface - Film wise.

Temp (°C)	for, Q = 20cc/s	Q = 30 cc/s	Q = 50cc/s.
T_1	38.7	38.7	38.8
T_2	39	39.2	39.2
T_3	38.2	39.2	38.1
T_4	39.7	39.9	39.8
T_5	41.2	41.2	40.9
T_6	39.5	39.8	40.2
T_7	81.5	89.3	83.6
T_8	41	41.2	41.4
T_9	88.5	91.4°C	91.3

28/7/24

Temperatures (°C),

$T_1 \rightarrow$ water inlet temperature.

$T_{2-3} \rightarrow$ water temperature in copper tube

$T_{4-5} \rightarrow$ water temperature in gold plated tube.

$T_6 \rightarrow$ vapour temperature (left side glass column)

$T_7 \rightarrow$ vapour temperature (right side glass column)

$T_8 \rightarrow$ surface temperature in copper tube.

$T_9 \rightarrow$ surface temperature in gold plated tube.

observations:

Area of tube $= \pi d L = \pi \frac{12.7}{1000} \times \frac{2500}{1000} = 0.003175 m^2$

for dropwise condensation.

	for $Q = 2\times10^{-5} m^3/s$	$Q = 3\times10^{-5} m^3/s$	$Q = 5\times10^{-5} m^3/s$
T_1	38.7	38.7	38.8
T_2	39	39.2	39.2
T_3	38.2	39.2	38.1
T_4	39.7	39.9	39.8
T_5	41.2	41.2	40.9
T_6	39.5	39.8	40.2
T_7	81.5	89.3	83.6

T_8	41	41.2	41.4
T_9	88.5	91.4	91.3

for filmwise condensation.

	$Q = 2 \times 10^{-5} m^3/s$	$Q = 3 \times 10^{-5} m^3/s$
T_1	39.4	40.1
T_2	39.6	41.1
T_3	40.2	41.9
T_4	40.6	40.7
T_5	42.2	42
T_6	53.9	81.5
T_7	47.3	45
T_8	92.4	96.1
T_9	51.2	48.7

calculations:

for $Q = 2\times10^{-5}\ m^3/s$.

for dropwise condensation,

$$\dot{Q} = m C_p (T_5 - T_1)$$

$$\dot{Q} = 2\times10^{-5} \times 4.2\times10^{3} \times (41.2 - 38.7) \times 10^{3}$$

$$\therefore \dot{Q} = 210\,W.$$

and.

$$T_m = \frac{T_5 - T_4}{\ln\left(\frac{T_7 - T_4}{T_7 - T_5}\right)} = \frac{41.2 - 39.7}{\ln\left(\frac{81.5 - 39.7}{81.5 - 41.2}\right)} = 41.05\,°C.$$

So. overall heat transfer coefficient.

$$U = \frac{210}{0.003175 \times \cancel{314.05}\ 41.05}$$

$$\therefore U = \cancel{1611.609\ W/m^2\cdot K}\ 1611.248\ W/m^2\cdot K$$

External surface heat transfer coefficient,

$$h = \frac{Q}{A_o (T_7 - T_4)} = \frac{210}{0.003175 \times 41.8}$$

$$\therefore h = 1582.34\ W/m^2\cdot K$$

Filmwise condensation. ($Q = 2\times10^{-5}\ m^3/s$)

Heat picked up by water, $\dot{Q} = m\,C_p(T_3 - T_2)$

$$= 20\times4.2\times(40.2-39.4)$$

$$= 67.2\,W$$

and,

$$T_m = \frac{T_3 - T_2}{\ln\frac{T_6 - T_2}{T_6 - T_3}} = \frac{40.2 - 39.6}{\ln\frac{53.9-39.6}{53.9-40.2}} = 13.998\,K.$$

So.

$$U = \frac{\dot{Q}}{A_o T_m} = \frac{67.2}{0.003175\times13.998} = 1512.03\ W/m^2\cdot K$$

and,

$$h = \frac{\dot{Q}}{A_o(T_6 - T_2)} = \frac{67.2}{0.003175\times(53.9-39.6)}\ W/m^2\cdot K$$

$\therefore h = 1480.095\ W/m^2\cdot K$

for flow-rate, $Q = 3\times10^{-5}\ m^3/s$.

for dropwise condensation,

$$\dot{Q} = 30\times 4.2(41.2-38.7)$$

$$\therefore \dot{Q} = 315\ W$$

and,

$$T_m = \frac{41.2-38.7}{\ln\left(\frac{84.3-38.7}{84.3-41.2}\right)} = 44.3\ K$$

overall heat transfer coefficient $U = \frac{315}{0.003175\times 44.3}\ W/m^2\cdot K$

$$U = 2239.56\ W/m^2\cdot K$$

and.

$$h = \frac{315}{0.003175\times(84.3-39.9)} = 2234.518\ W/m^2\cdot K$$

So. for film wise condensation,

$$\dot{Q} = 30\times 4.2(41.9-40.1) = 226.8\ W$$

$$T_m = \frac{41.9-41.1}{\ln\left(\frac{81.5-41.1}{81.5-41.9}\right)} = 39.998\ K.$$

So overall heat transfer coefficient $U = \frac{226.8}{0.003175\times 39.998} = 1785.92\ W/m^2\cdot K$

and

$$h = \frac{226.8}{0.003175\times(81.5-41.5)} = 1768\ W/m^2\cdot K$$

for 50 cc/s,

for dropwise condensation,

$$\dot{Q} = 50 \times 4.2 (40.9 - 38.8)$$

$$\therefore \dot{Q} = 441 \text{ W}$$

and.

$$T_m = \frac{40.9 - 39.8}{\ln\left(\frac{83.6 - 39.8}{83.6 - 40.9}\right)} = 43.25 \text{ K}$$

Also.

$$U = \frac{441}{0.003175 \times 43.25} = 3211.51 \text{ W/m}^2\text{-K}$$

and

$$h = \frac{441}{0.003175 \times (83.6 - 39.8)} = 3171.18 \text{ W/m}^2\text{.K.}$$

• Discussion:

→ The overall heat transfer coefficient and surface heat transfer coefficient is higher for dropwise condensation compared to filmwise condensation.

→ Dropwise condensation is much more effective than filmwise condensation.

→ Gold platings' hydrophobic nature is the reason behind drop. formation.

→ The liquid film provides more resistance to heat transfer than droplets.

→ Droplets have more surface area which leads to more rate of heat transfer.

• Critical Comments:

→ If water in the steam generator is not sufficient then it might get burned.

→ As the thermocouple is used here the readings are very sensitive to small fluctuations.

Results / Conclusion:

at $Q = 20cc/s$.

$$U|_{dropwise} = 1611.248 \ W/m^2 \cdot K$$

$$U|_{filmwise} = 1512.03 \ W/m^2 \cdot K.$$

at, $Q = 30cc/s$.

$$U|_{dropwise} = 2239.56 \ W/m^2 \cdot K$$

$$U|_{filmwise} = 1785.92 \ W/m^2 \cdot K.$$

So we can draw a conclusion that,

$$U|_{dropwise} > U|_{filmwise}$$

Aim:
To determine the thermal conductivity of a given copper bar or rod:??

Theory:
The thermal conductivity of a substance is a physical Property defined as the ability of a substance to conduct heat. The thermal conductivity of metal depends on chemical composition, state of matter, the crystalline structure of a Solid, temperature, pressure and whether or not it is a homogenous material. The heater will heat the ~~bar~~ on one of its end and heat will be conducted to the other end. Since the rod is insulated from the outside, it can be safely assumed that the heat transfer along the copper rod is mainly due to axial conduction and steady-state. The Q transfered will be equal to the amount of heat absorbed by the water in the cooling end. The heat conducted by the rod will create a temperature profile along the rod.
which can be assumed to be,

$$T(x) = Ax^2 + Bx + C.$$

now,
heat absorbed by cooling water is,

$$Q = \dot{m} C_p \Delta T$$

where, $\dot{m} \rightarrow$ mass flow rate of water.

$C_p \rightarrow$ Specific heat of water.

$\Delta T \rightarrow$ Temperature rise of cooling water $(T_{11} - T_{12})$

now, along the rod heat conducted is,

$$Q = -kA\frac{dT}{dx}$$

so,

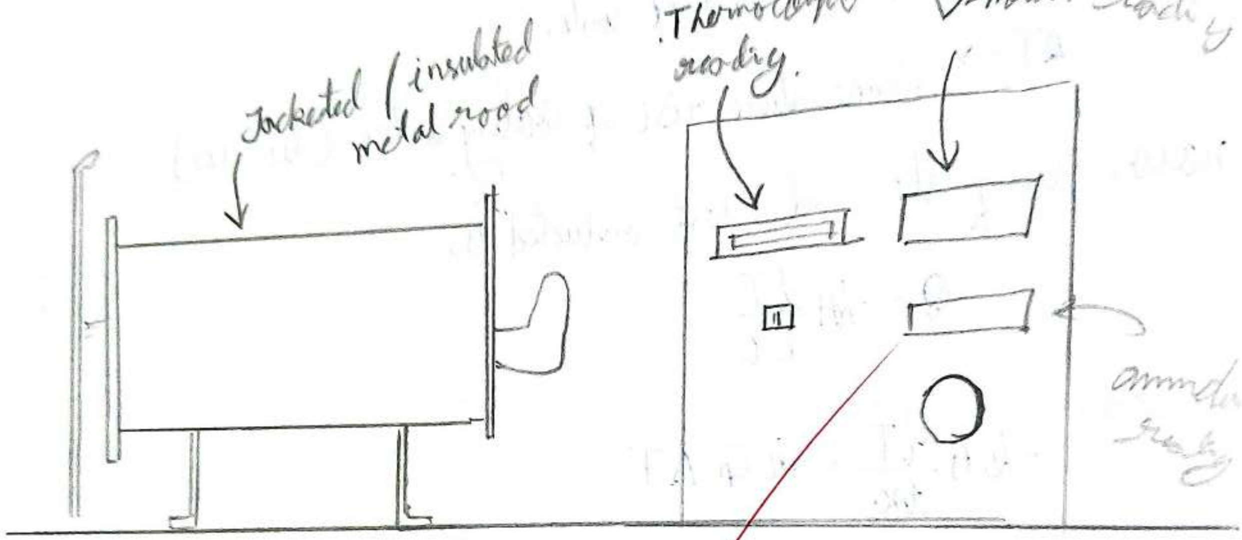
$$-kA\frac{dT}{dx} = \dot{m} C_p \Delta T$$

So.

$$\therefore \quad k = \frac{\dot{m} C_p \Delta T}{-\left(A\frac{dT}{dx}\right)\Big|_{x=L}}$$

at steady state,

The assumption that at a steady state, the heat flow is mainly due to axial conduction can be verified by the readings of temperature sensors fixed in the insulation material around the rod in the radial direction. Less variation in the reading shall confirm the assumption. The value of dT/dx is obtained from the at $\frac{dT}{dx}\Big|_{x=L}$.


Jacketed / insulated metal rood
Thermocouple reading.
V-meter reading
ammeter reading

Exp-1, Setup.

• Experimental Set-up:

The apparatus consists of a metal bar, one end of which is heated by an electric heater while the other end of the bar projects inside the cooling water jacket. The middle of bar ~~of~~ is insulated using asbestos ~~insult~~ insulation powder. The temperature of the bar is measured at different sections while the radial temperature is measured by separate temperature sensors at two different sections of the insulating shell. The heater is ~~mounted~~ provided with a dimmerstat for controlling the heat input. Water under constant head conditions is circulated through the jacket and its flow rate and temperature rise are noted by two temperature sensors provided at the inlet and outlet of the water.

• Specifications:

Length of the metal bar = 450 mm

diameter of the metal bar = 25 mm

Test length of the bar = 235 mm

Total number of temperature sensors = 12

number of temperature sensors mounted on the bar = 6.

number of temperature sensors on the insulation shell = 4.

~~Procedure~~:

Number of temperature sensors mounted on the water jacket = 2 Nos.

Type of temperature sensors = RTD PT-100

Heater = Nichrome heater.

Cooling Jacket diameter = 100 mm.

Length of the cooling Jacket = 75 mm

Dimmerstat for heater coil = 2 Amp. 230 V AC

Digital voltage of voltameter = 0 to 250 V

Digital ammeter = 0 to 2.5 Amps.

Procedure:

→ After cleaning the Apparatus, it was connected with cold water supply.

→ water supply was maintained at a constant head.

→ After ensuring about the switch's position, the main Power supply was turned on.

→ Switches of the panel were used to detect $T_1 - T_{12}$.

→ After 30 minutes of starting, temperature readings were taken for the $T_1 - T_{12}$, in every 10 minutes interval till the steady state arrives

Exp-1 Thermal conductivity of metal rod.

voltage = 91V , current = 0.68A.

t = 0	t = 10 min	t = 20 min	t = 30 min	t = 40 min
T_1 = 86.1	114.7	95.1	87.4	95
T_2 = 80.2	108.7	89.6	81.8	89.5
T_3 = 73.5	102.2	83.2	75.5	83.1
T_4 = 67.1	96	77.2.	69.6	77.2
T_5 = 60.8	89.8	71.2.	63.7	71.3
T_6 = 55	84.3	65.8	58.4	66.2
T_7 = 46.5.	75.6	57.1	49.4	56.7
T_8 = 46.5	75.6	57.1	49.4	56.8
T_9 = 40.7	62.8	51.1.	42.3	50.5
T_{10} = 38.8	68	43.1	41.7	49
T_{11} = 44	40.6	46.7	45.8	46.9
T_{12} = 44.9	50.2	47.6.	66.4	47.2

observations:

Temperature(°C)	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{11}	T_{12}
	86.1	80.2	73.5	67.1	60.8	55	46.5	40.7	38.8	44	44.9
	95.1	89.6	83.2	77.2	71.2	65.8	57.1	51.1	43.1	46.7	47.6
	87.4	81.8	75.5	69.6	63.7	58.4	49.4	49.4	43.3	45.8	66.4
	95	89.5	83.1	77.2	71.3	66.2	56.7	56.8	50.5	46.9	47.2.
SS values	95	89.5	83.1	77.2	71.3	66.2	56.7	56.8	50.5	46.9	47.2

calculations:

On Regression (Polynomial)

$$T(x) = (44.6x^2 - 158x + 101)\,°C.$$

now: @ $L = 0.45\,m$

$$\left.\frac{dT}{dx}\right|_{x=L} = 89.2 \times 0.45 - 158 = -117.86.$$

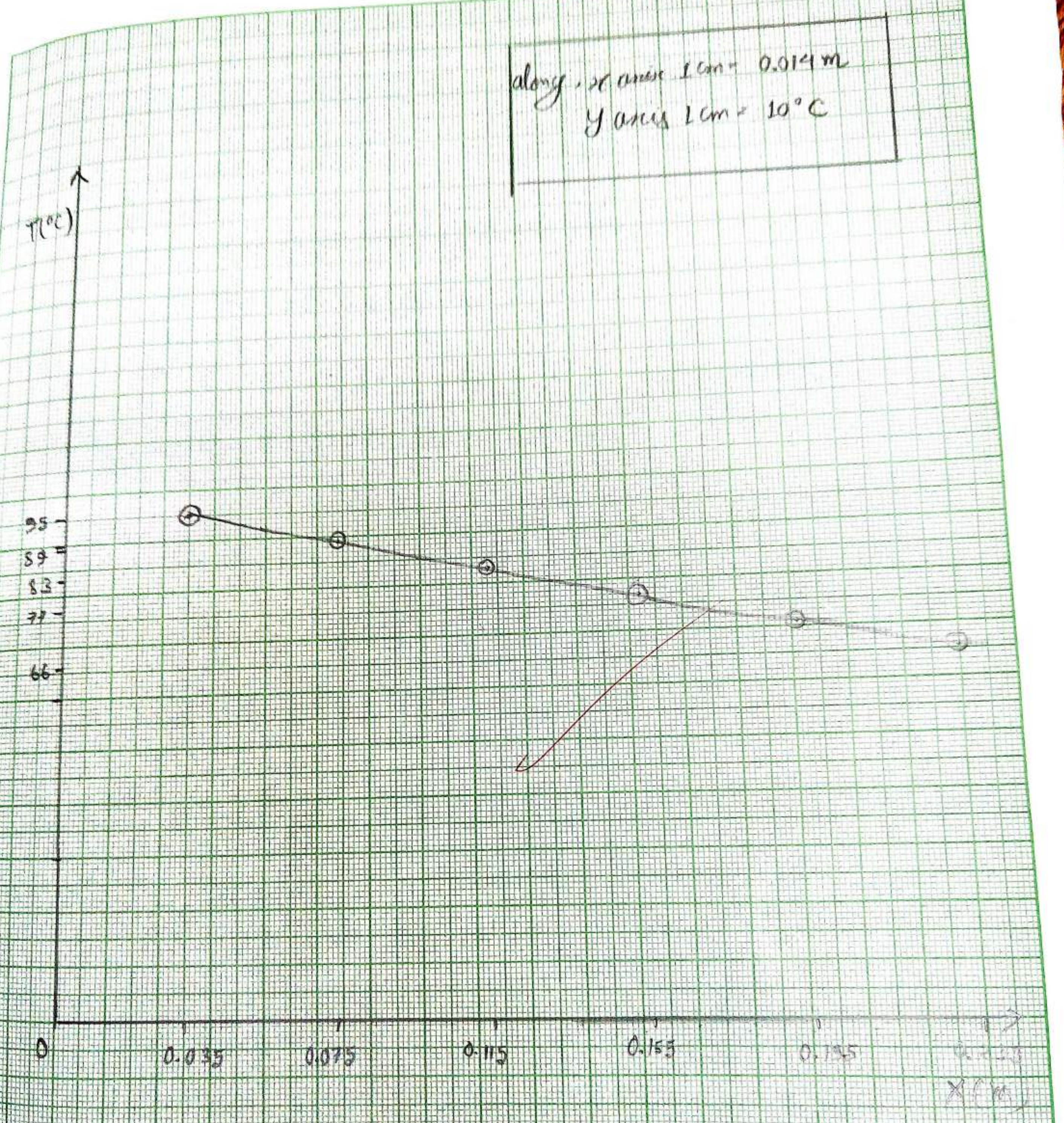
and,

$$\Delta T = (47.2 - 46.9)\,°C = 0.3\,°C$$

So

$$Q = \dot{m}\, C_p\, \Delta T$$

T vs, x plot (Parabolic)



happy hour

1 MM DIVISION

for water,

$$C_p = 0.998 \text{ kcal / kg °C.}$$

$$\dot{m} = 58.0836 \text{ kg/hr.}$$

So $Q = \dot{m} C_p \Delta T$

$$\therefore \boxed{Q = 17.390229 \text{ Kcal/hr}}$$

and,

$$A = \frac{\pi}{4} D^2$$

where $D = 0.0254$ m.

$$\therefore \boxed{A = 0.00050671 \text{ m}^2}$$

So.
$$k = \frac{Q}{-A \frac{dT}{dx}\Big|_{x=L}}$$

$$\therefore \boxed{\hat{k} = 291.1931 \text{ Kcal /hr-m°C}}$$

from literature, $k_{cu} = 332$ Kcal /hr-m°C.

So.
$$|E| = \left(\frac{k_{cu} - \hat{k}}{k_{cu}} \times 100 \right) \% = \underline{12.29\%}$$

Result:

Thermal conductivity of the metal bar (k) = 292.1931 kcal/hr.m°C.

Error in the results = 12.29%

Precautions:

→ The AC single Phase supply should be used only.

→ voltage should be slowly increased.

→ The assembly should not be disturbed.

→ Power supply must be between 180 V to 240 V.

→ Selector switch must be handled gently.

→ The apparatus should be kept dust free.

Discussion:

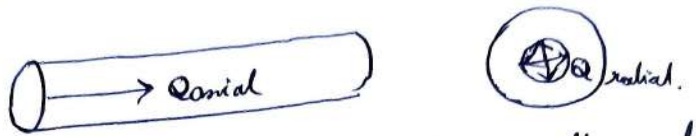
→ we've assumed that the convection is zero but which is not the case in reality as nothing can be 100% insulated

→ actually,

$$Q_{Total} = Q_{cond} + Q_{conv}.$$

neglecting $Q_{conv.}$ reduces the value of calculated Thermal conductivity.

→ The heatflow is assumed to be 1-Din nature in this case but which is not completely accurate.



there will be some value of Qradial which further get lost into the environment.

→ There exists some random error in the measurement as the measurements were taken only 4 times.

→ Taking multiple measurements after reaching steady state is the only way to reduce random error.

critical comments:

→ Taking convection and radial flow of heat into account can give better results.

→ Changing of thermocouples frequently can cause high fluctuation which must be done with Patiance.

• Conclusion:

The heat loss to the ambient in the experiment lowers the value - of thermal conductivity (k) of copper. Confirming axial heat flow - assumptions and using the correct length are crucial. Proper precautions ensure accurate and safe results.

Aim:
To find the thermal conductivity of the liquid.

Theory:
The heat transferred by conduction through the liquid layer is given by -

$$Q = -A\frac{dT}{dx}$$

10

also,

$$Q = \dot{m}\, C_p\, \Delta T \quad \text{(for temperature change of coolant.)}$$

for, cylinder,

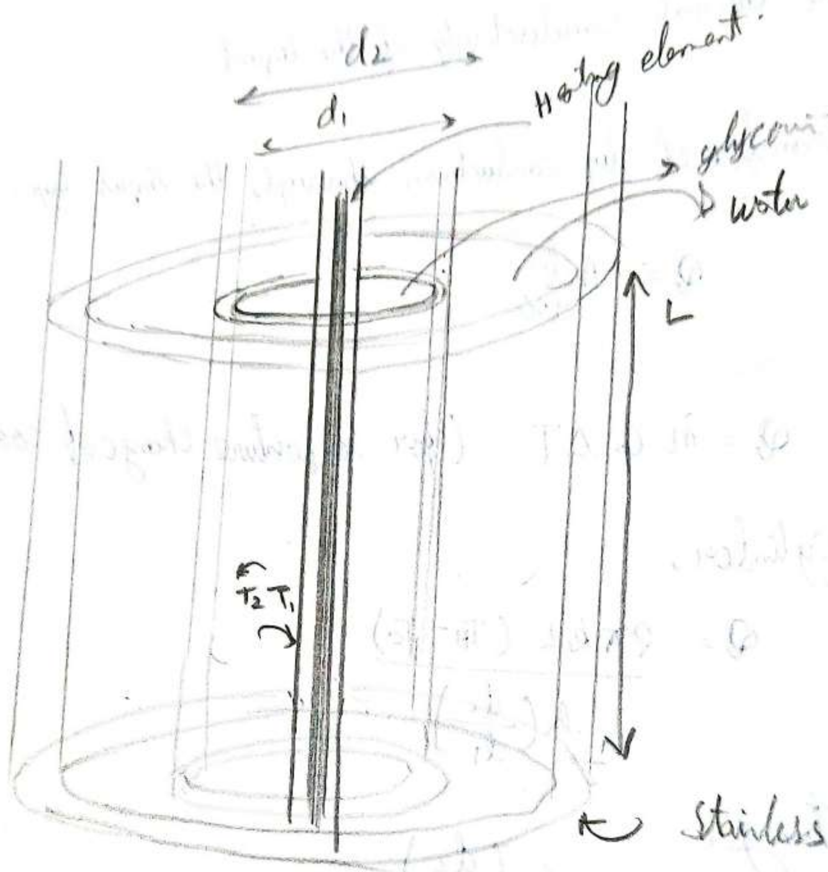
$$Q = \frac{2\pi k L\,(T_H - T_C)}{\ln\left(\frac{d_2}{d_1}\right)}$$

On equating,

$$k = \frac{Q \ln\left(\frac{d_2}{d_1}\right)}{2\pi L\,(T_H - T_C)}$$

where,

$\Delta T \rightarrow$ temperature change of the coolant.
$\dot{m} \rightarrow$ mass flow rate of coolant.
$C_p \rightarrow$ Heat capacity of the coolant.
$d_1, d_2 \rightarrow$ inner and outer diameters of the cylinder.
$L \rightarrow$ height of the cylinder.
$k \rightarrow$ Thermal conductivity of the liquid.



Exp-10. Setup, for the liq and cylinder.

Experimental Setup:

The apparatus consists of 3 concentric cyliders, in the inner heater is fitted, In the middle cylinder liquid is filled and the outer cylinder acts as a water cooling jacket, in which water flows in and out continuously. Energy meter is installed to measure the Power.

Specifications:

→ Electricity Supply: single phase, 220V AC, 50Hz, 5-15 Amp combined socket with earth connectia.

→ water supply: continuously @ 2LPM at Bar / can be adjusted

→ Floor area: 1m × 1m

→ $d_1 = 0.05$ m

→ $d_2 = 0.065$ m.

→ $C_p = 4186$ J/Kg°C.

→ mass flow rate of water = 0.003667 Kg/s.

Experimental Procedure:

→ The water supply was connected to the inlet of the water-chamber.

→ connected outlet of the chamber to the drain.

→ glycerine was filled using a funnel.

→ Electric supply was connected.

→ Flow-rate was adjusted to 220 mL/min.

→ PID controller was set in the range 50 to 70°C.

→ After 30 minutes when the steady state was achieved readings were taken in every 5 mins interval.

→ Took readings for 20 minutes in this way.

→ calculated the value of k for each reading.

→ Averaged the value of k found.

Exp - 10

Title?

$Q = 220\ mL/min$

F (mL/min)	time (after SS) (min)	T_1	T_2	T_3	T_4
220	5	59.9	38.6	33.7	32
	10	59.9	38.8	33.9	32.2
	15	59.6	38.9	39	32.2
	20	59.7	38.9	34°C	32.3

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DATE

Observations:

time (min)	T_1	T_2	T_3	T_4	Q (W)	k (W/m°C)	$\bar{k}$ (W/m°C)
5	59.9	38.6	33.7	32	26.0928	0.3410	
10	59.8	38.8	33.9	32.2	26.0928	0.3459	0.3519
15	59.6	38.9	34	32.2	27.6276	0.3715	
20	59.7	38.9	34	32.3	26.0928	0.3492	

Sample calculations:

$$Q = \dot{m} C_p \Delta T = 0.003667 \times 4186 \times (33.7 - 32)$$

$$\therefore Q = 26.0928 \text{ W}$$

and,

$$k = \frac{Q \ln\left(\frac{d_2}{d_1}\right)}{2\pi L (T_H - T_C)}$$

$$\Rightarrow k = \frac{26.0928 \ln\left(\frac{0.065}{0.05}\right)}{2 \times \pi \times 0.15 \times (59.9 - 38.6)}$$

$$\therefore \boxed{k = 0.34102 \text{ W/m°C}}$$

Results:

Thermal conductivity of the glycerine in the operational condition was found to be = 0.352 W/m°C.

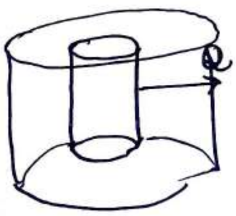
Precautions:

→ The apparatus should be operated only when $V \in (180, 230)$ V.
→ Assembly should not be disturbed.
→ Apparatus should be free of dust.

Detailed

Discussions:

→ The assumption of 1D flow (radial) might cause error in the value of k calculated.



Q (direction of heat flow assumed.

→ Although the assumption is acceptable for very low $\frac{D}{L}$ ratio. as D will be negligible. ~~also~~

→ for large D there will be a temperature gradiant in the length's direction.

→ During the experiment the flow-rate of the coolant must be kept constant.

→ Only after SS is achieved the values / readings should be taken.

• Conclusion:

Finally we can conclude that the usage of a ~~short~~ long highted cylinder is recommended instead of a long one to ensure 1-D flow in the system.

⇝ which will help us with infinite rod assumption.

DATE 12-08-24 | Exp 5: Studies on Heat transfer By Natural - Convection. | SHEET NO. 1.

Objective:

Study of convection heat transfer in natural convection.
To find out the heat transfer coefficient of vertical cylinder in natural convection.

Theory:

Natural convection phenomenon is due to the temperature - between the surface and the fluid and is not created by any external agency. The setup is designed and fabricated to study the natural convection phenomenon from a vertical cylinder in terms of average heat transfer coefficient.

The heat transfer coefficient is given by,

$$h = \frac{Q}{A(T_s - T_a)}$$

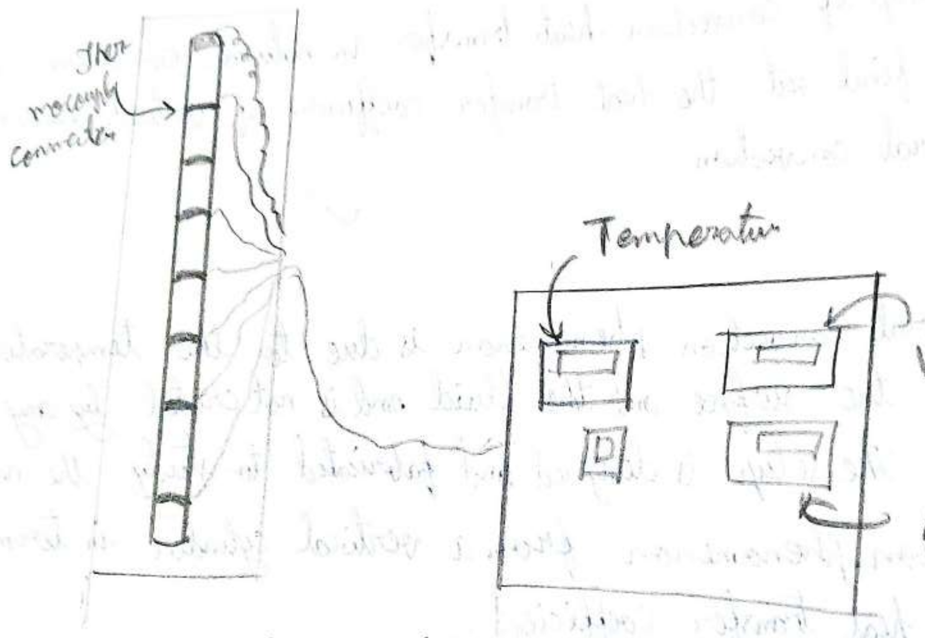
where,

$Q \rightarrow$ Amount of heat transfer.

$T_s \rightarrow$ Surface temperature.

$T_a \rightarrow$ Ambient temperature.

$h \rightarrow$ heat transfer coefficient



Experimental Setup.

Apparatus setup:

The apparatus consists of a brass tube fitted in a rectangular duct in a vertical fashion. The duct is open at the top and bottom and forms an enclosure and serves the purpose of undisturbed surrounding. One side of it is made up of glass/acrylic for visualization. A heating element is kept in the vertical tube, which heats the tube surface. The heat is lost from the tube to the surrounding air by natural convection. Digital temperature indicator measures the temperature at the different points with the help of seven temperature sensors. The heat input to the heater is measured by digital ammeter and digital voltmeter and can be varied by dimmer stat.

Experimental Procedure:

i) The apparatus was cleaned to make is dust free.

ii) The zero error of instruments was taken care of.

iii) After 30 minutes of turning on the heating element the readings were taken at 5 minutes interval.

iv) After steady state was achieved the average surface temperature was calculated and from that h was calculated.

Exo - 5 Heat Transfer in Natural Convection:

V	I	Time (min)	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
45V	0.25A	10	42.3	43.3	43.9	44.4	44.1	42.6	41.8	31.7
		15	42.6	43.5	44.1	44.6	44.4	42.8	42.1	31.9
		20	42.8	43.8	44.4	45	44.8	43.1	42.5	32.2
		25	43.2	44.1	44.7	45.2	45.1	43.5	42.6	32.3
40V	0.22A	10	43.4	44.2	44.8	45.2	44.8	43.4	42.6	32.7
		15	43.4	44.1	44.7	45	44.6	43.3	42.6	32.7
		20	43.3	44.1	44.6	45.0	44.6	43.2	42.5	32.5
		~~25~~								

Anup 12/8/24

So.

$Q = hA(T - T_\infty)$

$\therefore T = 44.06\,^\circ C$, $T_\infty = 32.3\,^\circ C$.

$\therefore h = \dfrac{11.25}{\pi d L \times 11.76} = 16.024$

observations: $d = 0.038\ m$, $L = 0.5m$

Volt(curr)	time(min)	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
45V (0.25A)	10	42.3	43.3	43.9	44.4	44.1	42.6	41.8	31.7
	15	42.6	43.5	44.1	44.6	44.4	42.8	42.1	31.9
	20	42.8	43.8	44.4	45.0	44.8	43.1	42.5	32.2
	25	43.2	44.1	44.7	45.2	45.1	43.5	42.6	32.3
40 (0.22A)	10	43.4	44.2	44.8	45.2	44.8	43.4	42.6	32.7
	15	43.4	44.1	44.7	45.0	44.6	43.3	42.6	32.7
	20	43.3	44.1	44.6	45.0	44.6	43.2	42.5	32.4

calculations:

for 45V,

$$T_s = \frac{43.2+44.1+44.7+45.2+45.1+43.5+42.6}{7} = 44.06^\circ C$$

$$T_\infty = 32.3^\circ C$$

now,

$$h_1 = \frac{IV}{\pi d L (T_s - T_\infty)}$$

$$\therefore h_1 = 16.024\ W/m^2 °C$$

@ 40V,

$$T_s = \frac{43.3 + 44.1 + 44.6 + 45.0 + 44.6 + 43.2 + 42.5}{7} = 43.9°C$$

$$T_\infty = 32.5°C$$

So,

$$h_2 = \frac{IV}{\pi d L (T_s - T_\infty)}$$

$$\Rightarrow h_2 = \frac{40 \times 0.22}{\pi \times 0.5 \times 0.038 (43.9 - 32.5)}$$

$$\therefore h_2 = 12.932\ W/m^2 °C.$$

Results:

So finally,

@ 45V, $h = 16.024\ W/m^2 °C$

@ 40V, $h = 12.932\ W/m^2 °C$

Precautions:

→ The apparatus should not be run with very high power supply.

→ The temperature indicator should be handled gently.

→ The apparatus should be kept dust free.

Discussion:

→ Heat transfer by convection is proportional to the surface area of the object. Therefore, the heat transfer coefficient (h) will vary depending on how the plate is positioned.

→ In this experiment, the characteristic length (L) is taken to be the length of the metal cylinder as the focus is on temperature variation along its length.

→ Thermocouples are strategically placed along the rod to measure temperature variations. Additional thermocouples are placed within the setup to measure the surrounding temperature.

→ The setup allows hot air to move upward due to the density difference and Buoyancy force, facilitated by an opening at the top.

Critical Comment:

→ The open duct design can lead to airflow disturbances, affecting temperature readings and causing error in the experiment.

→ Heat may escape through the brass tube walls, leading to uneven heating and inaccuracies in temperature measurement

Conclusion:

Finally we can conclude that the heat transfer coefficient increased with the voltage which is due to the changed temperature gradient which causes stronger convection currents.

Objective:

→ Calculating the critical heat flux for a material in different T_∞.

→ Beyond critical heat flux system failure increases and overheating is caused.

Theory:

Boiling is a fundamental heat transfer process that occurs when a liquid is heated to its saturation temperature, causing it to undergo a phase change into vapor. This process is best understood by examining the boiling curve, which plots the heat flux against the temperature difference between the heated surface and the liquid. The boiling curve is typically divided into three distinct regions, each characterized by different heat transfer mechanisms and behaviors:

i) Natural Convection Region:

In this initial region, the temperature difference between the heated surface and the liquid (ΔT) is relatively small typically less than 10°C. The heat transfer occurs primarily through natural convection, where the warmer-

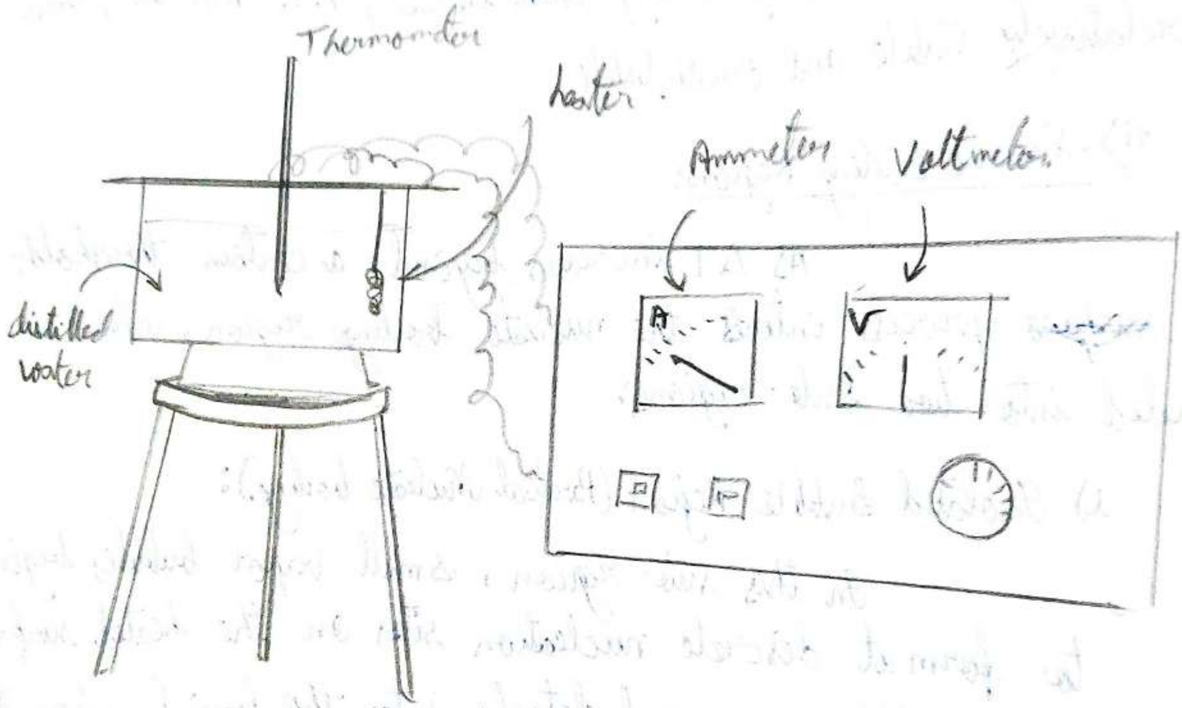
, less dense liquid near the surface rises, and cooler, denser liquid descends to replace it. There is no bubble formation in this region. The heat flux increases gradually with increasing ΔT, and the process is relatively stable and predictable.

ii) Nucleate Boiling Region:

As ΔT increases beyond a certain threshold; The ~~surface~~ process enters the nucleate boiling region, which is divided into two sub-regions:

a) Isolated Bubble Region (Partial Nucleate boiling):

In this sub-region, small vapor bubbles begin to form at discrete nucleation sites on the heated surface. These bubbles grow and detach into the liquid, where they collapse. The formation and collapse of these bubbles significantly enhance heat transfer by disrupting the thermal boundary layer and promoting fluid mixing. The heat flux increases with increasing ΔT.



Experiment Setup.

(b) Fully developed Nucleate boiling region:

with a further increase in ΔT, bubble formation becomes more vigorous and widespread across the heating surface. Bubbles coalesce, and the agitation in the liquid leads to extremely efficient heat transfer. The region exhibits the highest heat transfer coefficients, and the heat flux reaches its maximum value at the critical heat flux point or Leidenfrost point. Operating near this point is desirable for applications requiring efficient heat removal, such as in boiling water reactors and industrial boilers.

iii) Film Boiling Region:

If ΔT increases beyond the critical heat flux point, The process transitions into the film boiling region.

(a) Transition Boiling:

This is an unstable region where patches of vapour film intermittently cover the heating surface.

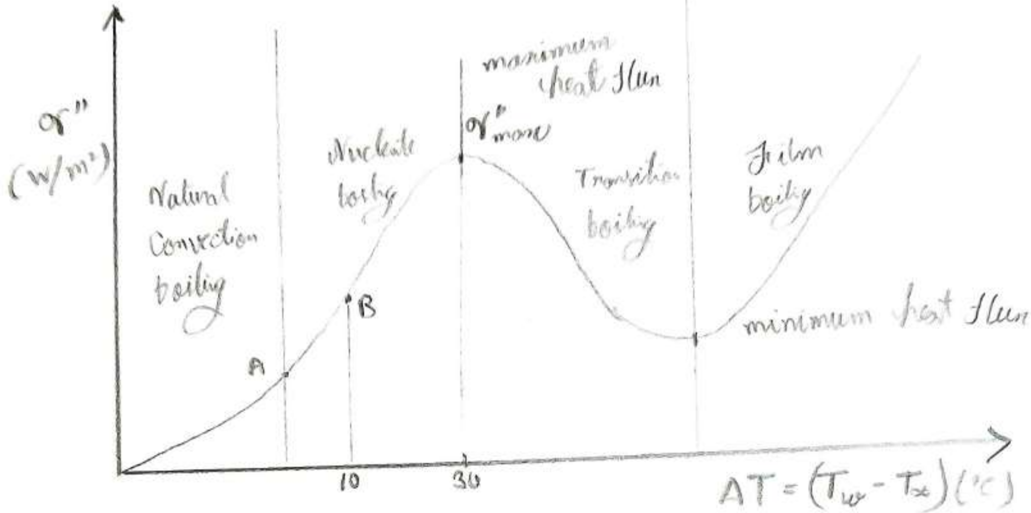
(b) Stable film Boiling:

At even higher ΔT, a stable and continuous vapour film envelops the entire heating surface.

Exp-6 Determination of critical heat flux.

d = 0.0076 in, L = 100 mm = 0.1 m
= 0.000193

T	V	A	Q.	ϕ (W/m²)
55°C	18	7	126	2078085.27
65°C	17.5	6.5	113.75	1876049.199
75°C	16	6	96	1583303.06



: Boiling curve :

Observations :

$d = 0.0076\ in = 0.000193\ m$

$L = 100\ mm = 0.1\ m.$

T(°C)	Voltage (V)	Current (A)	heat (W)	flux (W/m²)
55	18.0	7.0	126.00	2.078×10^6
65	17.5	6.5	113.75	1.876×10^6
75	16.0	6.0	96.00	1.583×10^6

Sample calculations:

$$\phi = \frac{q}{\pi d L}$$

$$q = VI = 18 \times 7 = 126\ W$$

$$\text{So. } \phi = \frac{126}{\pi \times 0.000193 \times 0.1}$$

$$\therefore \phi = 2.078 \times 10^{6}\ W/m^2.$$

Discussion:

→ Essentially we took different temperatures which are 55°C, 65°C and 75°C, as we can consider that, q''_{max} is a function of T_∞ or the ambient temperature as well [from the values]

→ As ΔT decreases with increasing T_w the critical heat flux also decreases.

→ For a constant area the value of flux only varies with amount of heat supplied, which is $q \propto \Delta T$.

→ The wire broke from the side where the temperature was maximum as the current was flowing through it and was getting dissipated into the distilled water through the whole surface,

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DATE

Conclusion:

With more temperature of the 'Distilled water' the amount of possible critical heat flux due to the negative correlation between the heat flux and the temperature difference (could have been better).

Critical comments:

→ Although the water was distilled, there might exits some impurities which can cause error.

→ The rust in the instrument will cause loses which are not taken into account is also a source of error.

→ The heaters must be completely immersed in the distilled-water and the reading of the ammeter must be taken with maximum care.

→ The rusty parts can be replaced with new parts which will give more accurate results and the usage of digital meters will also reduce human errors.

→ As the experiment is highly sensitive to environmental condn, such as room temperature or, air currents, which are difficult to control and those introduce variability in the results.

P.R.E.

Aim: obj

→ Study of conduction heat transfer through composite wall.

→ To determine total thermal resistance and thermal conductivity of composite wall.

→ To Plot the temperature profile along the composite wall.

Theory:

when a temperature gradient exists in a body, there is an energy transfer from high temperature to the low temperature region. Energy is transferred by conduction and heat transfer rate per unit area is proportional to the normal temperature gradient.

$$q = -k A \frac{dT}{dx}$$

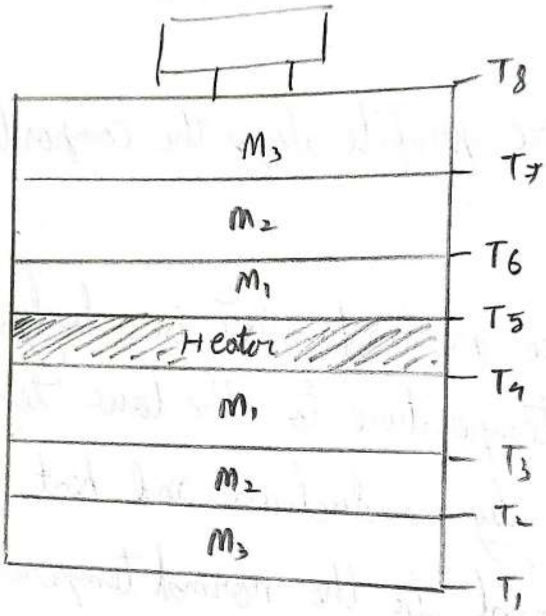
where,

q is the rate of heat transfer rate.

$\frac{dT}{dx}$ is the temperature gradient in the direction of heat flow.

A- crossectional area normal to the heatflow direction.

k- thermal conductivity.



Schematic Diagram of composite wall.

Diagram?

A direction of Fourier's law is the plane wall, Fourier's equation,

$$q = -kA \frac{(T_2 - T_1)}{\Delta x}$$

where, the thermal conductivity is considered constant. The wall thickness is Δx, and T_1 and T_2 are surface temperatures. The temperature gradients in the three materials ($M_1, M_2 M_3$). the heat flow may be written as.

$$-k_1 A \Delta T_1/\Delta x_1 = -k_2 A \Delta T_2/\Delta x_2 = -k_3 A \Delta T_3/\Delta x_3$$

Experimental Procedure:

→ The main Power supply of 220AC single Phase 50Hz, was turned on.

→ The input to heater by the dimmerstat was slowly increased from zero volt position.

→ The heat input by voltmeter and ammeter was adjusted.

→ Temperature sensors readings were taken at frequent intervals till the steady state arrived.

→ Based on steady state data k_1, k_2, k_3 were calculated

Exp-4 : Studies on Heat transfer through Composite walls.

t	V	I(A)	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
10	60V	0.45	48.8	40.5	37.8	38	34.4	34.3	34.2	32.2
20			39.3	40.5	38.7	39	35.1	35.6	33.7	32.5
30			40.3	41.5	40.3	40	35.8	36.2	33.7	33.2
40			41.3	43.3	41	41.1	36.8	37.2	34.1	336
50			42.3	43.5	41.7	42.3	37.7	382	34.7	34
			T_4	T_5	T_3	T_6	T_2	T_7	T_1	T_8

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SHEET NO. 3

observation:

t (min)	V (Volt)	I (Amp)	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8
10	60	0.45	48.8	40.5	37.8	38	34.4	34.3	34.2	32.2
20			39.3	40.5	38.7	39	35.1	35.6	33.7	32.5
30			40.3	41.5	40.3	40	35.8	36.2	33.7	33.2
40			41.3	43.3	41	41.1	36.8	37.2	34.1	33.6
50			42.3	43.5	41.7	42.3	37.7	38.2	34.7	34

change the nomenclature and do the calculations accordingly

calculation:

$T_1' = \frac{T_1 + T_8}{2} = 38.15°C$, $T_2' = \frac{T_2 + T_7}{2} = 39.10°C$, $T_3' = \frac{T_3 + T_6}{2} = 39.95°C$

$T_4' = \frac{T_4 + T_5}{2} = 40°C$ and, $x_1 = 0.02\,m$, $x_2 = 0.015\,m$, $x_3 = 0.012\,m$

$d = 0.25\,m$ $A = \frac{\pi d^2}{4} = 0.0491\,m^2$

so, $q = \frac{VI}{2} = 13.5$ watts.

now $K_{m_1} = \frac{q x_1}{A(T_4' - T_3')} = 109.979\,W/m°C$.

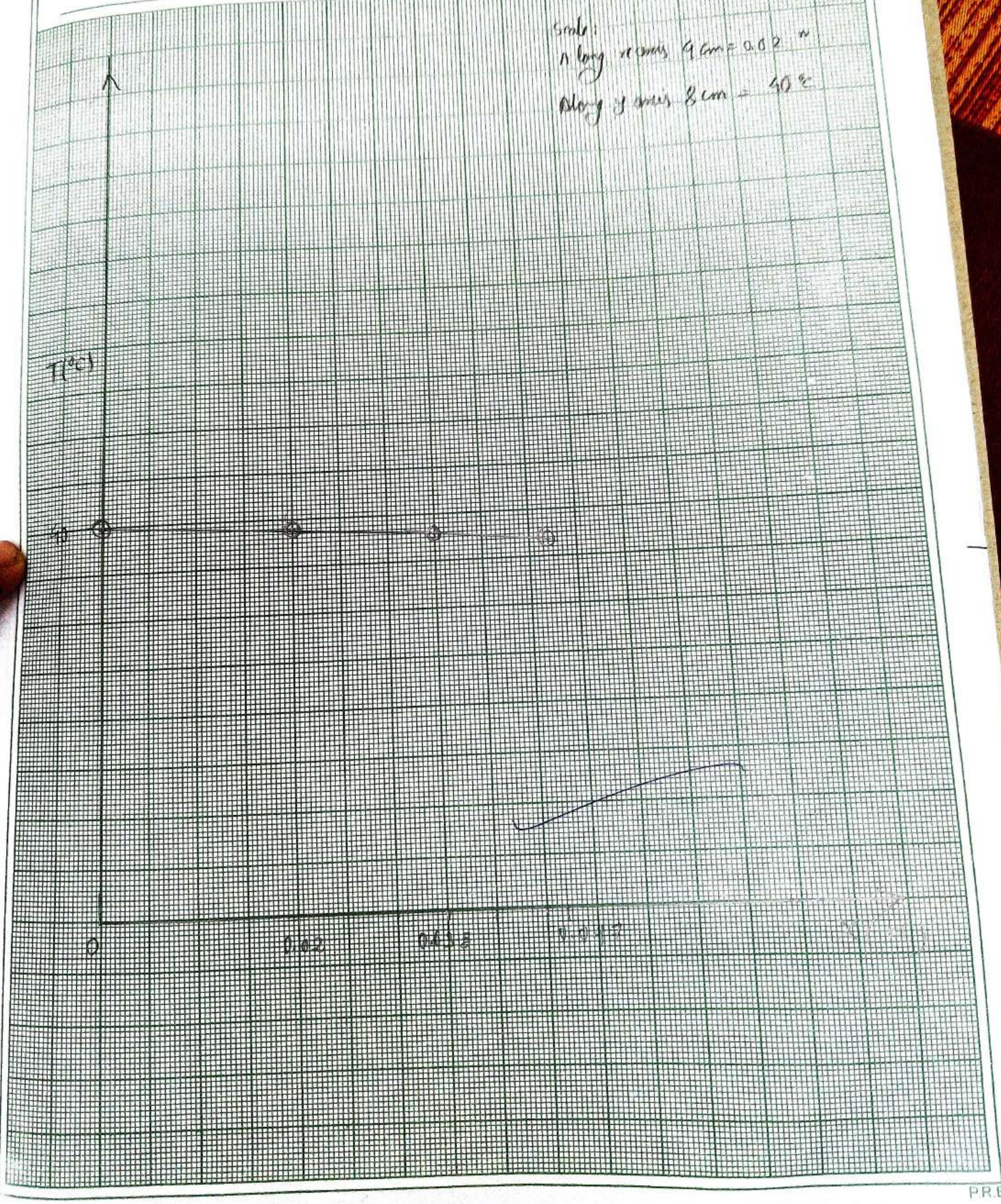
$K_{m_2} = \frac{q x_2}{A(T_3' - T_2')} = 4.852\,W/m°C$.

$K_{m_3} = \frac{q x_3}{A(T_2' - T_1')} = 3.473\,W/m°C$

$K = \frac{q x}{A(T_4' - T_1')}$

$= 6.985\,W/m°C$

T(°C) vs. X(m)



Results:

$K_{m_1} = 109.979\ W/m°C$

$K_{m_2} = 4.852\ W/m°C$ and $K = 6.985\ W/m°C$.

$K_{m_3} = 3.473\ W/m°C$

Total thermal conductivity??
Total thermal resistance??

Discussion:

→ Temperature profiles are assumed to be linear along the axis due to the absence of heat generation within the materials, and temperature conventions are assigned starting from the material closest to the heater.

→ A symmetric approach is used to analyze the composite material, with average temperatures and heat flux divided equally across both sides assuming these materials on either side of the heater are identical.

→ The experimental setup involves tightly pressing the materials together to reduce contact resistance, ensuring more accurate temp measurements.

Critical Comments:

→ The experiments' accuracy heavily depends on ensuring steady state conditions, particularly at higher power ratings, which may be difficult to maintain without proper temperature control.

→ The assumption that the materials on either side of the heater are perfectly identical may not hold in practical setups, potentially introducing systematic errors in the k-composite calculations.

→ The method for assigning temperature conventions from thermocouples could introduce errors.

Conclusion:

The experiment effectively measures the thermal conductivities of different materials and their composite at various power ratings. However, significant errors arise at higher power ratings due to inadequate temperature control, emphasizing the importance of maintaining steady state conditions.

Aim :

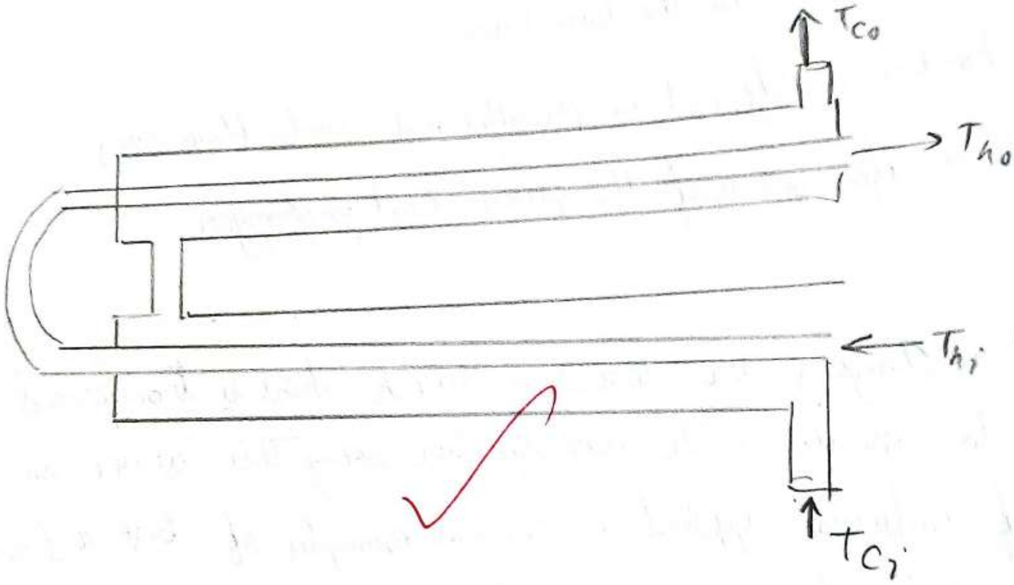
→ Temperature distributions in parallel flow and counter flow heat exchanger.
→ Heat transfer rate in the two runs.
→ Heat transfer coefficient in Parallel and counter flow runs.
→ To obtain effectiveness of the given heat exchanger.

Theory:

Heat exchangers are devices in which heat is transferred from one fluid to another. The necessity for doing this arises in a multitude of industrial applications. Common examples of best heat exchanger are the radiator of a car, the condenser at the back of a domestic refrigerator and the steam boiler of a thermal power plant.

HE are classified into three categories-

i) Transfer type.
ii) Storage type
iii) Direct contact type.



Parallel flow heat exchanger.

A transfer type of Heat Exchanger is one in which both fluids pass simultaneously through the device and heat is transferred through separating walls. In practice most of the Heat Exchangers used are transferred types.

The transferred type Exchangers are further classified according to flow arrangements as.

(1) Parallel Flow.

(2) Counter Flow.

(3) Cross Flow.

Experimental Set-up:

→ Simple example of a transferred type of Heat exchanger can be the form of a ~~pipe~~ tube-in tube arrangement.

→ A fluid flows through the inner tube and another fluid flows through the annular space and heat transfer takes place through the walls of the inner tube.

→ Heat transfer resistance, $R_T = \frac{1}{UA} = \frac{1}{A_i h_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi K L} + \frac{1}{A_o h_o}$

Exe-7

Counter flow:

t	Flow (LPH) F_h	F_c	T_1 (hot)	T_2 (hot)	T_3 (cold)	T_4 (cold)
0 min	51	114	50.2	42.4	34.1	31.8
5 min	51	119	49.6	42.1	34.2	31.9

Parallel flow:

t	F_h (LPH)	F_c (LPH)	T_1	T_2	T_3	T_4
0 min	50	103	49.0	42.1	31.4	34.4
5 min	50	103	49.8	40.5	31.5	34.6

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Observations:

units?

Counter flow

t(min)	Fn (LPH)	Fc (LPH)	T_1	T_2	t_1	t_2
0	51	114	50.2	42.4	34.1	31.8
5	51	114	49.6	42.1	34.2	31.9

Parallel flow.

t(min)	Fn (LPH)	Fc (LPH)	T_1	T_2	t_1	t_2
0	50	103	49.0	42.1	31.4	34.4
5	50	103	49.8	40.5	31.5	34.6

Calculations:

for counter flow:

$$\dot{q}_h = \dot{m}_h C_p (T_1 - T_2) = \frac{51}{3600} \times 4184 \times (49.6 - 42.1)$$

$$= 444.55\ W$$

$$\dot{q}_c = \frac{114}{3600} \times 4184 \times (34.2 - 31.9) = 304.73\ W$$

$$\varepsilon = 0.685.$$

$$\Delta T_{LMTD} = \frac{(49.6 - 34.2) - (42.1 - 31.9)}{\ln\left(\frac{49.6 - 34.2}{42.1 - 31.9}\right)}$$

$$= 12.62^\circ C.$$

So. $q = \frac{q_h + q_c}{2} = 374.64\,W,$ $\quad A_o = 0.084\,m^2$, $A_i = 0.051\,m^2$

now,

$$U_i = \frac{q}{\pi d_i L \, \Delta T_{LMTD}} = 582.08\ W/m^2\,^\circ C.$$

$$U_o = \frac{q}{\pi d_o L \, \Delta T_{LMTD}} = 353.41\ W/m^2\,^\circ C$$

for Parallel flow:

$$\dot{q}_h = \frac{50}{3600} \times 4184 \times (49.8 - 40.5) = 540.93\ W$$

$$\dot{q}_c = \frac{103}{3600} \times 4184 \times (34.6 - 31.5) = 371.09\ W$$

$\varepsilon = 0.687$, $q = 455.76\ W$,

now. $$\Delta T_{LMTD} = \frac{18.3 - 5.9}{\ln\left(\frac{18.3}{5.9}\right)} = 10.95^\circ C$$

$U_i = 816.12\ W/m^2\ ^\circ C.$

$U_o = 495.49\ W/m^2\ ^\circ C.$

Results :

For the given setup.

Parallel flow: $U_i = 816.12\ W/m^2\ ^\circ C$, $U_o = 495.49\ W/m^2\ ^\circ C$.

$\varepsilon = 0.687$.

Counter flow:

$U_i = 582.08\ W/m^2\ ^\circ C$, $U_o = 353.41\ W/m^2\ ^\circ C$.

$\varepsilon = 0.685$.

Discussion :

→ The difference in U-values is attributed to different flow rates. used in the calculations, making direct comparison misleading.

→ The placement of fluids in a heat exchanger should be determined based on factors like the corrosiveness or fouling tendencies of the fluid.

→ Counterflow is typically used in heat exchanger design as they require smaller surface area to achieve same heat transfer.

Critical Comments:

→ The comparison of U-values between parallel and counterflow setups is invalidated due to different flow rates, which makes the statement potentially misleading without proper clarification.

→ Statements emphasizes theory but lacks practical examples or specific case studies to support the decisions made about fluid placement and flow configuration.

Conclusion:

The comparison of heat transfer coefficients between parallel and counterflow heat exchangers highlights the importance of consistent flow conditions in such evaluations. Practical considerations like fluid fouling and corrosion should guide the choice of heat exchanger design.

26/08/24

incomplete

AIM:

I) To find out the overall heat transfer co-efficient

II) Steam side coefficient.

III) water side coefficient.

IV) Draw wilson Plot.

Theory:

Special Heat transfer devices used to liquefy vapour by removing their latent heat are called condensers. The latent heat is removed by absorbing it in a cooler liquid, called the coolant, since the temperature of the coolant obviously is increased in a condenser, the unit also acts as a heater by functionally it is the condensing action that is important and the name reflects this fact.

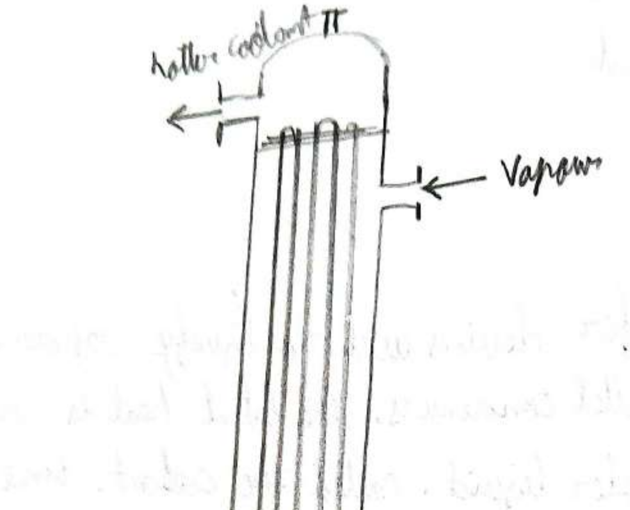
Different type of condensers:

→ Air cooled condernser: Natural and forced convection.

→ water cooled condenser: Double tube, shell and coil condenser, shell and tube condenser.

→ Evaporative condenser.

To obtain higher heat transfer coefficient larger velocities and shorter tubes should be used. The multipass principle used in heat-exchangers may also be used for the coolant in a condenser.



Vertical Condenser.

Apparatus:

The vertical condenser is mounted on a square tube frame. One side of the condenser is connected by a monoblock pump with a bypass valve. An acrylic rotameter is provided to note the flow rate of water allowed to the condenser. A steam boiler with accessories is provided with a condensate measuring tank to collect the condensate. Panel board is mounted on the front with a temperature indicator and selector switch, D.P. switches for heater and for the pumps.

Specifications:

1. Shell length : 1000 mm
   Shell diameter : 150 mm
   Material : M.S.
   Insulation : Asbestos cloth
2. Tube length : 1020mm (Approx)
   Tube diameter : 12mm
   Material : M.S
   No of Tubes : 14.
3. Boiler height : 450mm
   Boiler Diameter : 350 mm
   Material : M.S
   Heaters : 2Kw/2Nos
4. Pump. H.P : 0.5 HP Monoblock.
5. Sump tank capacity : 36 Lt
   Material : S.S.
   Size : 300 x 360 x 400 mm
6. Digital temperature indicator :
   No. of Digits : 4
   Range : 400°C
   Resolution : 1°C
7. Thermocouple types : Cr-Al
   No. of Thermocouples : 4
8. Condensate Tank/Jar : 36 Lt.
   Material : SS. , Size : 300X300X400 mm

vertical condenser. 14 tubes are there

water temp. Inlet T_3	water temp. outlet T_4	Steam temp T_1	Condensate temp T_2	steam Pr.	Cond. Quality Q_c	Cond. Quality t_c	Water flow rate Q_w	Water flow rate t_w
40.2	85.0	108.6	41.7		30cc	2min	30cc	1sec
40.1	76.8	108.6	41.3	1bar	55cc	2min	40cc	1sec
40.1	70.2	108.4	41.3		60cc	2min	50cc	1sec
40.1	~~77.2~~	~~108.3~~	40.8				~~30cc~~	~~1sec~~

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Observations:

Steam Pressure (bar)	Water temperature (°C)		Steam temp(T_3) (°C)	Condensate temp(T_2) (°C)	Condensate flow rate (cc/s)	Water flow rate (cc/s)
	Inlet(T_5) (°C)	Outlet(T_4) (°C)				
1	40.2	85	108.6	41.7	0.250	30
1	40.1	76.8	108.6	41.3	0.458	40
1	40.1	70.2	108.4	41.3	0.500	50

Calculations:

$\rho_w = 997\ kg/m^3$. $\rho_c = 0.579\ kg/m^3$

Latent heat of vaporisation, $\lambda = 2260\ KJ/kg$.

Thermal heat conductivity, $k = 0.604\ W/mK$

viscosity, $\mu = 0.5465\ cp$.

i) water Quality.

a) $Q_w = 30 \times 10^{-6}\ m^3/s \Rightarrow m_w = 0.02991\ kg/s$

b) $Q_w = 40 \times 10^{-6}\ m^3/s \rightarrow m_w = 0.03988\ kg/s$

c) $Q_w = 50 \times 10^{-6}\ m^3/s \Rightarrow m_w = 0.04985\ kg/s$.

ii) Heat transfer to water

a) $m_w = 0.02991\ (kg/s)$ $\quad C_{pw} = 4187\ J/kg\,K$

$q_w = m_w C_{pw} (T_4 - T_3)\ (J/s)$

$\therefore q_w = 5610.45\ (J/s)$.

b) $m_w = 0.03988\ (kg/s)$

$\therefore q_w = 6128.08\ (J/s)$

c) $m_w = 0.04985\ (kg/s) \Rightarrow q_w = 6282.53\ (J/s)$

iii) Heat given out by stream:

a) $q_c = Q_c \rho_c \lambda = 0.327\ (J/s)$

b) $q_c = 0.599\ (J/s)$

c) $q_c = 0.654\ (J/s)$

iv) LMTD.

a) $\Delta T_{LMTD} = \dfrac{85 - 40.2}{\ln\left(\dfrac{108.6 - 40.2}{108.6 - 85}\right)} = 42.10^\circ C$

b) $\Delta T_{LMTD} = \dfrac{76.8 - 40.1}{\ln\left(\dfrac{108.6 - 40.1}{108.6 - 76.8}\right)} = 47.83^\circ C.$

c) $\Delta T_{LMTD} = \dfrac{70.2 - 40.1}{\ln\left(\dfrac{108.4 - 40.1}{108.4 - 70.2}\right)} = 51.8^\circ C,$

v) Surface area,

$A = \pi d_o L N \ m^2, \quad d_o = 12 \times 10^{-3} \ m, \ L = 1 \ m, \ N = 14$

$\therefore A = 0.527787 \ m^2$

vi) overall heat transfer coefficient.

a) $U_{exp} = \dfrac{q_w}{A \, \Delta T_{LMTD}} = 252.497 \ W/m^2K.$

b) $U_{exp} = \dfrac{q_w}{A \, \Delta T_{LMTD}} = 242.753 \ W/m^2K$

c) $U_{exp} = \dfrac{q_w}{A \, \Delta T_{LMTD}} = 229.798 \ W/m^2K$

vii) cooling water velocity through tubes.

a) $v = 0.0189$ m/s.

b) $v = 0.0252$ m/s.

c) $v = 0.0315$ m/s.

viii) water side heat transfer coefficient,

for, $Q_w = 30$ cc/s.

$Re = 415, \quad Pr = 3.57$.

$$h_i = 1.86 \left(Re \times Pr \times \frac{d_i}{L} \right)^{1/3} \times \left(\frac{k}{d_i} \right) = 259 \, W/m^2K.$$

ix) Shell side heat transfer coefficient.

$$h_o = 0.943 \left(\frac{g k^3 \lambda \rho_w^2}{\mu \cdot L (T_s - T_w)} \right)^{0.25}$$

$h_o = 4366.15$ W/m²·K.

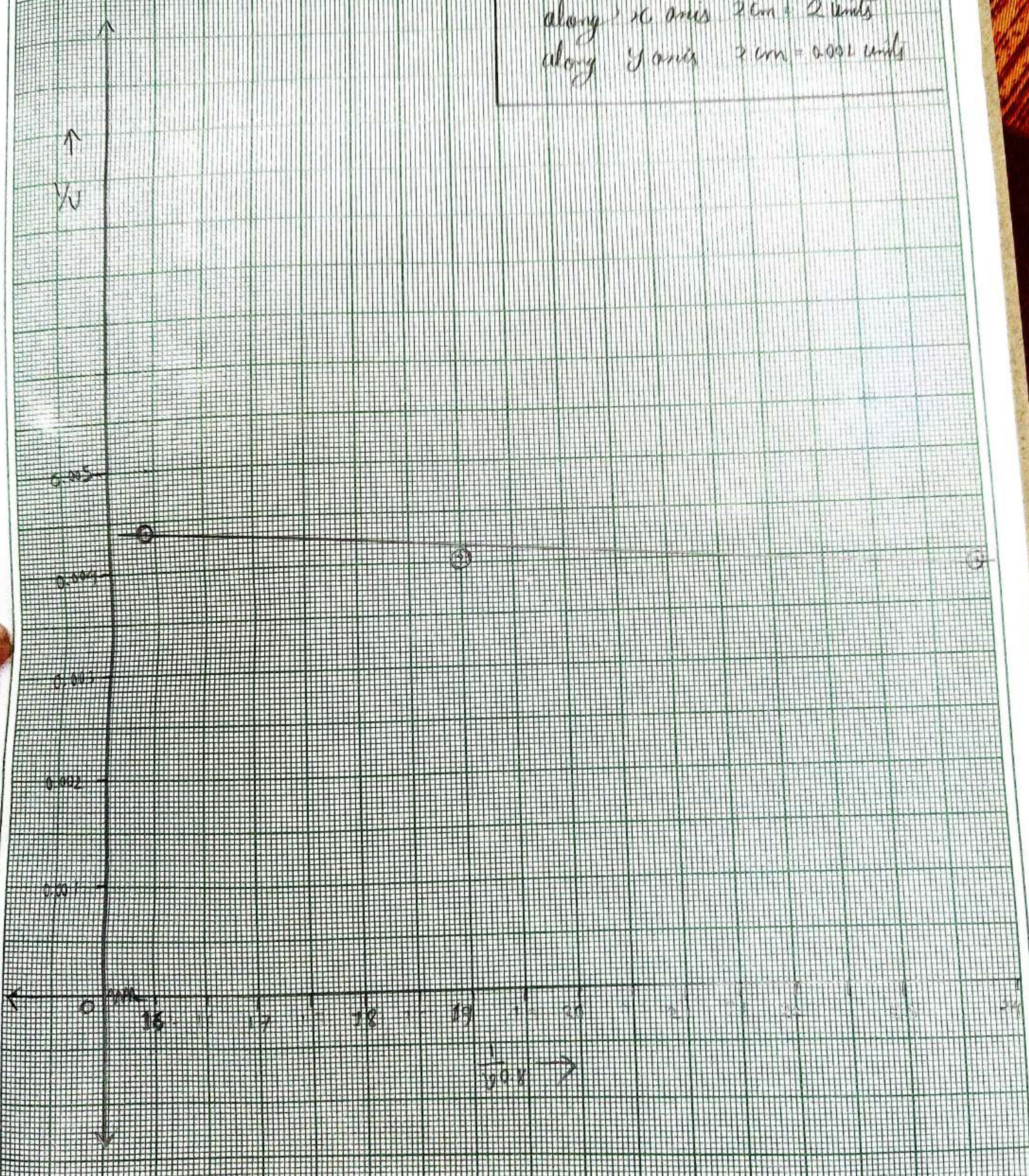
P.T.O

$\frac{1}{U}$	$\frac{1}{v^{0.8}}$
0.00396	23.92
0.00412	19.01
0.00435	15.89.

Data for the graph.

willson Plot ($\frac{1}{U}$ vs. $\frac{1}{v^{0.8}}$)

along x axis 2 cm = 2 units
along y axis 2 cm = 0.001 units



Results:

i) Overall heat transfer coefficient is = 241.68 W/m^2K.

ii) Tube side heat transfer coefficient = 259 W/m^2K

iii) Shell side heat transfer coefficient = 4366.15 W/m^2K

Discussion:

→ To get the film heat transfer coefficient we can utilise wilson plot.

→ The heat transfer coefficient of a horizontal condenser is higher due to drop formation, whereas the heat transfer coefficient of a vertical condenser is lower due to film formation.

→ In vertical condenser, the pumping cost higher than in a horizontal condenser.

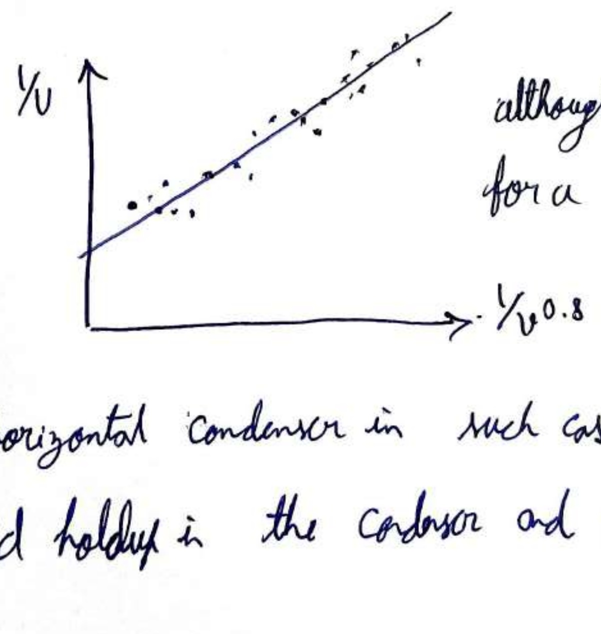
→ we used a vertical condenser in our experiment because they take up less storage space and have less fouling, so we can ignore the dirt problem.

→ vertical condensors are mostly utilised in the food processing industry.

Critical comments:

→ As we can clearly see in the data that the amount of condensate formed increases with flow rate as the amount of heat transferred from the steam to the vapour increases.

→ We took the data only for 3 flow rates which were close to each other and due to the very small size of sample readings the data is not sufficient to draw a proper regression plot for Wilson's plot as the statistical avg of the multiple one will give a good result.



although the points decreases for a

→ We can't use horizontal condenser in such case as that would cause flooding / liquid holdup in the condenser and finally no pump would work.

→ The values of thermocouple readings might have some calibration error which causes the error in the temperature reading, Conclusion:

The steam side coefficient is significantly higher than the water-side. This shows that the heat transfer on the steam side is more efficient, likely due to the condensation of steam releasing latent heat more effectively. The use of various condensers - air-cooled, and evaporative - emphasizes flexibility depending on the application's needs. To enhance heat transfer coefficient we can increase water flow which will also - increase the rate of condensate formation.

Aim: To determine the value of Stefan-Boltzmann constant for radiation heat transfer.

Theory:

Stefan-Boltzmann law states that the emissive Power of a perfect black body is directly proportional to the fourth Power of the absolute temperature.

$$Q = \sigma \varepsilon A T^4.$$

where, $\sigma \rightarrow$ Stefan Boltzmann Constant (W/m^2K^4)

$E_a (W/m^2) \rightarrow$ total emissive Power $= \int_0^\infty E_{a\lambda} \, dA = \sigma \varepsilon T^4$

whereas $E_{a\lambda} (W/m^2 \mu m)$ is the amount of radiant energy emitted by surface a per unit area per unit time per unit wavelength

$$\varepsilon = \frac{\text{energy radiated by a body.}}{\text{energy radiated by blackbody.}}$$

for blackbody,

$$\varepsilon = 1,$$

Stefan - Botlzmann Apparatus.

Apparatus:

The apparatus consists of a flanged copper hemisphere fixed on a flat non-conducting plate. A test disc made of copper is fixed to the plate. Thus test disc is completely enclosed by the hemisphere. The outer surface of the hemisphere is enclosed in a vertical water jacket used to heat the hemisphere to a suitable constant temperature. Three Cr-Al thermo-couples are attached at three strategic places on the surface of the hemisphere to obtain the temperature. The disc is mounted on a Bakelite sleeve which is fitted in a hole drilled at the centre of the base plate. other Cr-Al thermocouples are fixed to the disc to record its temperature

Specification:

Diameter of the disc(d) = 20 mm

Thickness of the disc = 1.5 mm

Mass of the disc (m) = 5g

Specific heat of the test disc (C_p) = 380 J/kg K

Inner diameter of the hemispherical surface = 200 mm

Procedure:

→ Filled the water in the stainless steel container with immersion - heater kept on top of the panel

→ Test disc was removed before starting the experiment

→ The water in the stainless steel container was heater to its boiling - point.

→ The boiling water was then allowed into the container kept into the bottom containing copper hemisphere until it is full - allow sufficient time to attain thermal equilibrium which is indicated by the three thermocouple provided on the hemisphere.

→ Inserted the test disc fixed on the Bakelite sleeve fully in and locked it and started the clock simultaneously.

→ Temperature was noted down at 10 sec interval for 10 - minutes.

Exp-8

$t_1 = 79.5^{\circ}C$ (wall temp)

$t_2 = 82.7^{\circ}C$ (")

$t_4 = 97.5^{\circ}C$ (hot water temp)

no time) 10 sec intervals

$t(^{\circ}C)$		
	60.8	65
53.5	61.1	65.2
53.9	60.8	65.0
54.6	61.3	65.8
54.7	62.2	65.9
55.3	61.7	66
56.2	62.4	65.9
56.5	62.7	66.2
56.8	62.8	66.1
57.3	63	66.2
58.1	63.1	66.1
57.9	63.2	66.5
57.8	63.9	66.[illegible]
58.7	63.8	66.9
59.1	64	67
59.4	64.9	66.8
59.6	64.4	67.[illegible]
59.9	64.7	66.8
60.4	65.9	67.2
60.6	65.3	67.3
	64.7	67.2
		67.2

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SHEET NO. 4.

Observation:

T_1 (wall temperature) = 79.5°C T_4 (hot water temperature) = 97.5°C

T_2 (wall temperature) = 82.7°C

T_3 : Disc temperature .

time (s)	T_3 (°C)	T_3 (K)	time (s)	T_3 (°C)	T_3 (K)
10	53.5	326.5	250	61.7	334.7
20	53.9	326.9	260	62.4	335.4
30	54.6	327.6	270	62.7	335.7
40	54.7	327.7	280	62.8	335.8
50	55.3	328.3	290	63	336
60	56.2	329.2	300	63.1	336.1
70	56.5	329.5	310	63.2	336.2
80	56.8	329.8	320	63.9	336.9
90	57.3	330.3	330	63.8	336.8
100	58.1	331.1	340	64	337
110	57.9	330.9	350	64.9	337.9
120	57.8	330.8	360	64.4	337.4
130	58.7	331.7	370	64.7	337.7
140	59.1	332.1	380	65.9	338.9
150	59.4	332.4	390	65.3	338.3
160	59.6	332.6	400	64.7	337.7
170	59.9	332.9	410	65	338
180	60.4	333.4	420	65.2	338.2
190	60.6	333.6	430	65	338
200	60.8	333.8	440	65.8	338.8
210	61.1	334.1	450	65.9	338.9
220	60.8	333.8	460	66	339
230	61.3	334.3	470	65.9	338.9
240	62.2	335.2	480	66.2	339.2

time(s)	T_3(°C)	T_3(K)	time (s)	T_3(°C)	T_3(K)
490	65.9	338.9	550	67	340
500	66.2	339.2	560	66.8	339.8
510	66.1	339.1	570	67.2	340.2
520	66.5	339.5	580	67.3	340.3
530	66.9	339.9	590	67.2	340.2
540	66.9	339.9	600	67.2	340.2

• Calculations:

from the plot, $\frac{dT}{dt} = 0.0326\ K/s$.

$A_D = 4.084 \times 10^{-4}\ m^2$

$T_D = 326.5K$. , $T_{avg} = \left(\frac{79.5 + 82.7}{2}\right) = 354.6\ K$

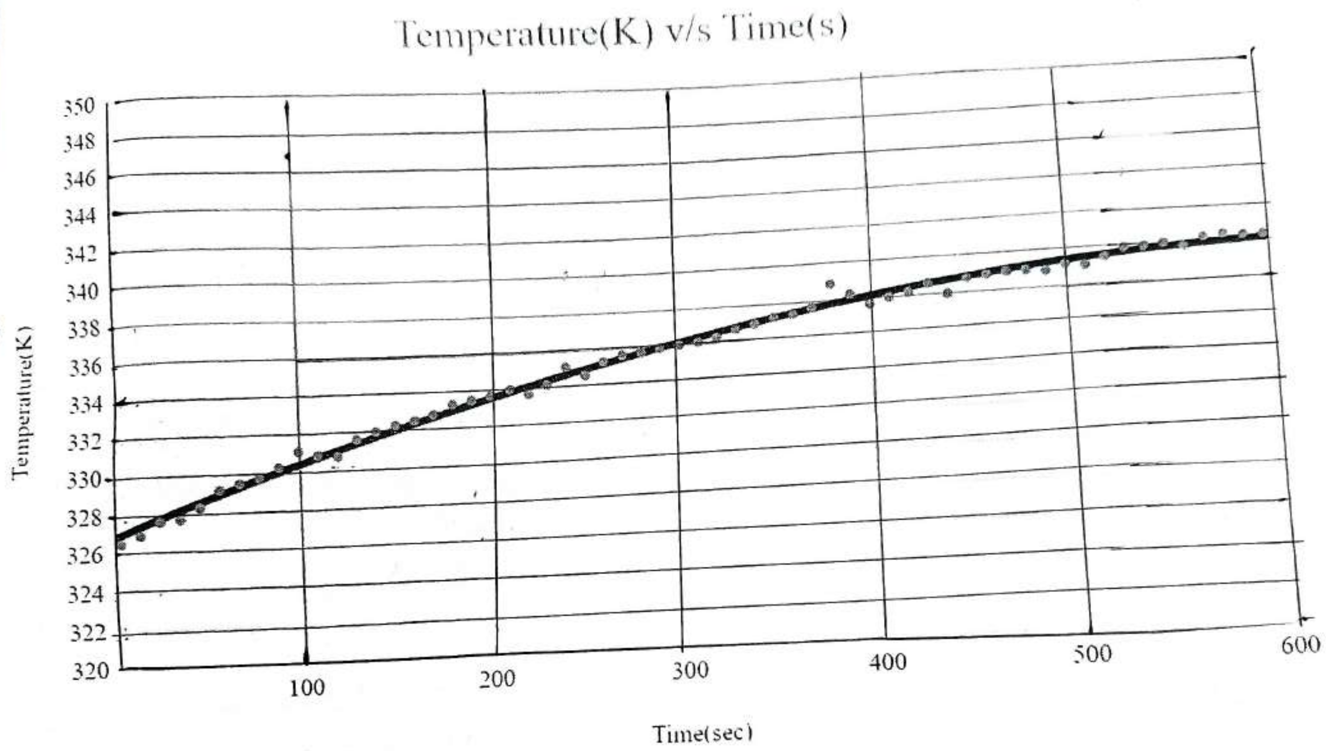
now by heat balance,

$$mC_p \frac{dT}{dt} = \sigma A_D (T_{avg}^4 - T_D^4)$$

$$\Rightarrow \sigma = \frac{5 \times 10^{-3} \times 380 \times 0.0326}{4.084 \times 10^{-4} (354.6^4 - 326.5^4)}$$

$$\therefore \sigma = 3.411 \times 10^{-8}\ W/m^2 K^4.$$

along x axis 1 unit = 200 sec
along y axis 1 unit = 2 K.



Results :

Stefan - Baltzmann constant for radiation heat transfer is.

$3.411 \times 10^{-8}\ W/m^2K^4$.

Discussion :

→ we know that the theoretical value of stefan boltzman constant is $5.67 \times 10^{-8}\ W/m^2K^4$ but we got $3.411 \times 10^{-8}\ W/m^2K^4$ as we've used $\varepsilon = 1$ for heat balance.

→ The emissivity of the disc was less than 1 more precisely 0.602.

→ The temperature of the disca increases with time almost linearly for a while but platoses after some time.

→ As the temperature of the water is very high in the experiment we can consider that the radiation prevalent here.

→ The water temperature and the disc temperature was checked using thermocouple and there were slight fluctuation.

P.T.O

Critical comment:

→ The asbestos cladding was unable to provide 100% insulation which caused some heat loss to the environment.

→ Usage of some different fluid with high boiling point would make calculations more accurate as the radiation dominates at high temperature.

→ Assumption of $\epsilon = 1$ will make sense if we used something like lamp black which can be assumed to be an ideal black body.

Conclusion:

The experiment yielded an experimental value of Stefan Boltzmann constant less than the actual (theoretical) value of it. This difference is due to the fact that the $\epsilon < 1$ but we took $\epsilon = 1$ in our calculation. Although the overall data trend validates our theoretical knowledge.