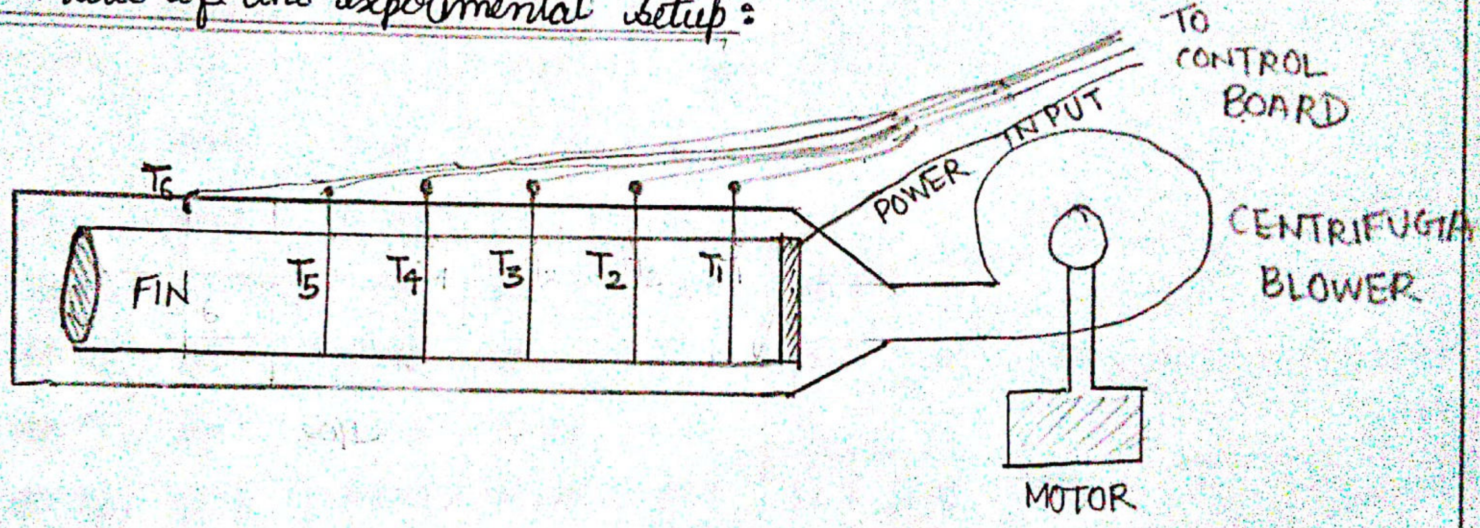
HEAT TRANSFER FROM PIN FIN

- **Objectives:** Determination of temperature distribution along pin fin in natural and forced convection and estimation of the rate of heat transfer from the fin

- **Schematic of the experimental setup:**



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Observations:

Room temperature, $T_0 = 29°C$

least count of the manometer = 1 mm

Run No. 1 (NATURAL CONVECTION MODE)

SL. NO.	TIME (min)	VOLTAGE V (V)	CURRENT I (A)	POWER (W)	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	T_5 (°C)	T_6 (°C)
1	10	79.8	0.342	27.29	45.8	42.4	x	40.9	40.8	31.3
2	20	80.2	0.345	27.67	55.2	52.1	x	50.4	49.7	32.5
3	25	80.9	0.347	28.07	61.2	56.7	x	55.4	54.9	32.7
4	30	80.4	0.345	27.74	63.7	60.6	x	57.8	57.5	33.1
5	35	80.3	0.344	27.62	65.6	60.9	x	60.2	59.9	35.4
6	40	80.9	0.347	28.07	67.1	62.7	x	60.8	60.6	35.4
7	45	81.0	0.347	28.10	68.2	64.4	x	62.2	61.8	33.9
8	50	79.8	0.342	27.29	68.9	65.1	x	63.0	62.2	37.4
9	55	79.5	0.341	27.11	69.7	65.1	x	63.3	62.9	34.6
10	60	80.3	0.344	27.62	70.0	65.1	x	63.8	63.5	35.6
11	65	79.9	0.343	27.41	70.3	65.8	x	64.3	64.0	35.8

Run No. 2 (FORCED CONVECTION MODE)

MANOMETER DIFFERENCE, H (mm)	SL. NO.	TIME (min)	VOLTAGE V (V)	CURRENT I (A)	POWER (W)	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	T_5 (°C)	T_6 (°C)
126	1	0	79.6	0.341	27.14	56.3	50.4	x	49.6	49.1	35.6
124	2	5	81.1	0.346	28.06	53.2	47.8	x	46.8	46.3	35.9
124	3	10	78.1	0.333	26.00	52.4	47.4	x	46.4	46.0	36.1

MANOMETRIC DIFFERENTIAL

⦿ SAMPLE CALCULATION – FORCED CONVECTION

Density of water, $\rho_w = 1000\ kg/m^3$

Gravitational acceleration, $g = 9.81\ m/s^2$

Power Input to the heater $= VI = 26\ W$ (78.1×0.333)

Average Fin Temperature, $T_m = (T_1+T_2+T_3+T_4+T_5)/5 = 48.05\ °C$

Base Temperature of fin, $T_{base} = T_1 = 52.4°C$

Duct fluid Temperature, $T_f = T_6 = 36.1°C$

Mean film temperature, $T_{mf} = (T_m + T_f)/2 = 42.075°C$

Density of air at temperature T_f and atmospheric pressure,

$$\rho_{air} = \frac{P_{atm}}{R T_f} = 1.142\ kg/m3$$

Volume flow rate of air through orifice, q_{flow}

$$= C_d \left(\frac{\pi d^2}{4}\right)\left[2gH\left(\rho_w/\rho_{air} - 1\right)\right]^{1/2}$$

$$= 3.623 \times 10^{-3}\ m^3/s$$

Area of duct, $A_{duct} =$ $0.015 m^2$

Velocity of air at duct fluid temperature (T_f), $V_f = \frac{q_{flow}}{A_{duct}}$

$$= 0.242\ m/s$$

Ratio of densities of air at T_f and T_{mf}, $\frac{\rho_{air}}{\rho} = \frac{T_{mf} + 273.15}{T_f + 273.15}$

$$= 1.019$$

velocity of air at duct mean film temperature (T_{mf})

$$V = \frac{\rho_{air}}{\rho} V_f = 0.247\ m/s$$

Reynolds Number, $Re = \frac{\rho V D}{\mu} = 179.4$

Coefficient of thermal expansion, $\beta = \frac{1}{T_{mf} + 273.15} = 3.172 \times 10^{-3}\ K^{-1}$

Characteristic temperature difference, $\Delta T = T_m - T_f = 11.95°C$

Grashof number, $Gr = \frac{g\beta \Delta T D^3}{\nu^2} = 2452$

Prandtl number, $Pr = \frac{\mu C_p}{K_{air}} = 0.724$

Rayleigh Number, $Ra = Gr Pr = 1775.25$

$\therefore$ Nusselt Number, $Nu = 0.615 Re^{0.466}$ $40 < Re < 4000$

$= 6.905$

Heat Transfer coefficient, $h = \frac{Nu K_{air}}{D}$

$= 14.79 \ \frac{W}{m^2K}$

cross-sectional area of fin, $A = \frac{\pi D^2}{4} = 1.227 \times 10^{-4} m^2$

Circumference of fin, $C = \pi D = 0.03927 \ m$

Now, fin efficiency (adiabatic tip), $\eta = \frac{\tanh(mL)}{mL}$

where $m = \sqrt{\frac{hC}{KA}}$

$= 6.55 \ m^{-1}$ $\therefore mL = 0.9839$

$\therefore \eta = \frac{\tanh(mL)}{mL}$ $(L = 150 mm)$

$(k = 110 W/mk)$

$\Rightarrow \boxed{\eta = 0.767}$

Rate of heat transfer from the fin (adiabatic tip), $\phi = \sqrt{hCkA}(T_{base} - T_f) \times \tanh mL$

$= 1.089 \ W$

Fin effectiveness, $\epsilon = \frac{A_{fin}}{A}\eta$ $A_{fin} = CL$

$= 5.8905 \times 10^{-3} m^2$

$\boxed{\epsilon = 36.82}$

FOR NATURAL CONVECTION :

Average Fin Temperature $= (T_1+T_2+T_3+T_4+T_5)/5 = T_m$
$= 66.1°C$

Base temperature of fin, $T_{Base} = T_1 = 70.3°C$

Duct fluid temperature, $T_f = T_6 = 35.8°C$

Mean film temperature, $T_{mf} = (T_m + T_f)/2 = 50.95°C$

Coefficient of thermal expansion, $\beta = \frac{1}{T_{mf}+273.15} = 3.0869\times10^{-3}\,K$

Characteristic temperature difference, $\Delta T = T_m - T_f = 30.3°C$

Grashof Number, $Gr = \frac{g\beta\Delta T D^3}{\nu^2} = 5488$

Prandtl Number, $Pr = 0.7219$

Rayleigh Number, $Ra = Gr.Pr = 3961.78$

Nusselt Number, $Nu = 1.1\,Ra^{1/6} = 4.375$

$\therefore$ Heat transfer coefficient, $h = \frac{Nu\,k_{air}}{D} = 9.6\ \frac{W}{m^2k}$

$\therefore m = \sqrt{\frac{hC}{kA}} = 5.325\,m^{-1}$

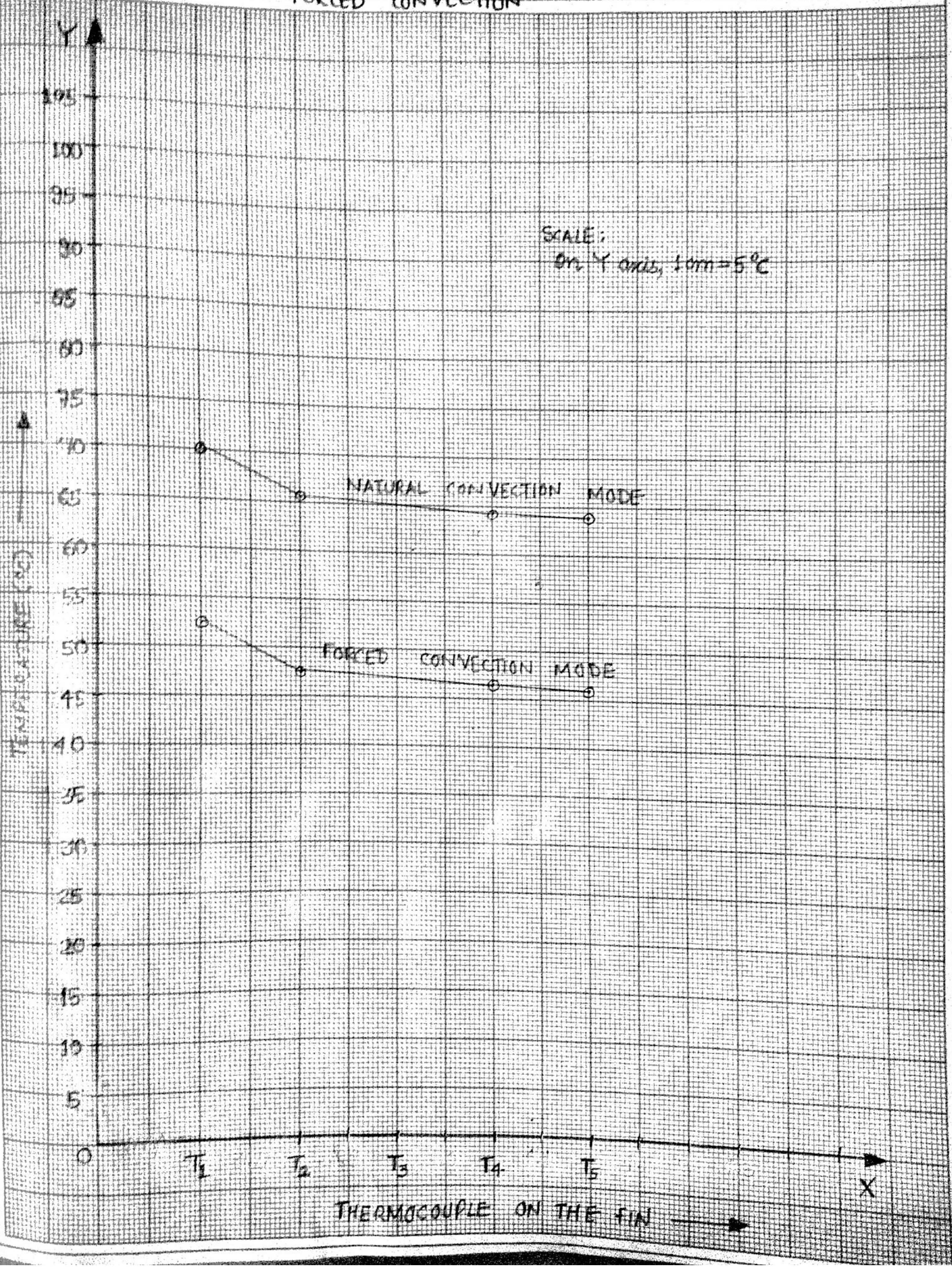
$\therefore mL = 0.7988$

(Efficiency) $\therefore \eta = \frac{\tanh(mL)}{mL} \Rightarrow \boxed{\eta = 0.83}$

(Effectiveness), $\boxed{\epsilon = 39.84}$

(Rate of Heat Transfer), $Q = 1.6325\ W$

● PLOT OF TEMPERATURE DISTRIBUTION ALONG THE FIN IN NATURAL AND FORCED CONVECTION



Discussions:

After performing the experiment, we obtained the values and plotted the wall temperatures of the fin against the sequential thermocouples. It can be observed that the temperature falls from first thermocouple to the last thermocouple. Also, the rate of fall of temperature is larger initially (magnitude wise) and it decreases subsequently.

It is also observed that for the same amount of power input, forced convection is more effective in cooling than free convection

Sources of Errors:

- Lack of insulation.