Experimental Analysis on Pin Fin Heat Exchanger using Different Circular Fin Material

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Abstract— Extended surfaces or fins are used to increase the heat transfer rates from a surface to the surrounding fluid wherever it is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid. Fins are fabricated in variety of forms. Fins around the air cooled engines are a common example. The Pin Fin Apparatus that is Dynamic apparatus consists of a simple pin fin which is fitted in a rectangular duct. The duct is attached to suction end of a blower. One end of fins is heated by an electrical heater. Thermocouples are mounted along the length of fin and a thermocouple notes the duct fluid temperature. When top cover the fin is opened and heating started, performance of fin with natural convection can be evaluated and with top cover closed and blower started, fin can be tested in forced convection. The aim of the present study is to improve the heat transfer characteristics and to investigate the performance of fin efficiency by using fin of different materials in pin fin apparatus. Here the system follows natural and forced convection as the mode of heat transfer and principle used in it. This experiment accomplished by using blower and without blower in the riser tube, which connects to the thermocouples which flows the air to the heater. From the heater the air gets heated and the air transfer to the pin fin in it. This procedure followed for the fin of different materials, Reynold`s number, Nusselt`s number are calculated and heat transfer coefficient and fin efficiencies are analyzed.
Key words: Pin Fin Heat Exchanger, Circular Fin Material

I. Introduction

Now-a-day's residual heat is produced in large number of engineering equipment needed to be dissipated to atmosphere. If the heat is not removed then the equipment tends to fail due to overheating. During the conversion of one form of energy into the other form of energy losses of energy takes place. The large amount of energy is lost in the form of thermal energy i.e. heat. Heat is transient form of energy. Heat transfer is an important parameter for the proper & efficient working of the all the mechanical as well as electronics components. Engines, condensers, transformers, etc. possesses large amount residual heat energy generated during the working of these equipments. This heat generated causes development of stresses in the components & shortens the life of the components. Hence, there is need of increasing the heat transfer rates from the main body to the atmosphere. Now considering the heat transfer equation, which gives the rate of heat transfer from any surface without fins.

$$Q_{conv} = h\ As\ \Delta T$$

From this formula we can extract few ways to increase the heat transfer and those are,

1) Increase the convection heat transfer coefficient h.
2) Increase the surface area 'As'.
3) Increase the temperature difference.

But, increasing 'h' may require the installation of a pump or fan or replacing the existing one with a larger one. For natural convection processes it is not possible to increase the convective heat transfer coefficient. For the given conditions the temperature difference remains constant. The remaining alternative is to increase the surface area by attaching extended surfaces. Hence, heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins and dimples. In most of the cases fins are used to enhance the heat transfer.

A. Design Methodology

There are two main ways to increase the heat transfer rate i.e. either by increasing h or by increasing the surface area. Generally it is not possible to increase the convective heat transfer coefficient. So we are left with the only alternative of increasing the effective surface area by using fins or extended surfaces.

There are large number of fins are available with different geometries. The geometry of the fins required is selected based on the cost, efficiency, performance, weight, availability of space, pressure drop in the heat transfer fluid, etc. Fins are used in a large number of applications to increase the heat transfer from surfaces. Typically, the fin material has a high thermal conductivity. The fin is exposed to a flowing fluid, which cools or heats it, with the high thermal conductivity allowing increased heat being conducted from the wall through the fin. The design of cooling fins is encountered in many situations and we thus examine heat transfer in a fin as a way of defining some criteria for design. Here, we are considering the Pin type fin with the uniform cross sectional area. Practically for adequately long length and thin fin the heat transfer at the tip is negligible so we can consider the tip of the pin fin as insulated. Several assumptions were taken into consideration during the analysis of heat transfer through the pin fin are as follows:

Heat conduction is along the X-direction under steady state

1) Convective heat transfer coefficient over the fin is uniform over the entire surface.
2) There is no heat generation within the fin
3) Material is isotropic i.e. 'K' is constant
4) Base temperature is maintained at constant too.

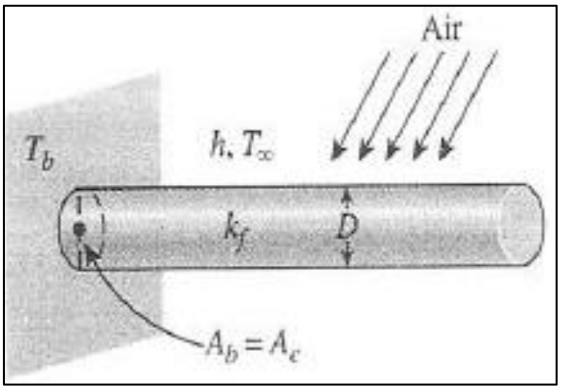


Fig. 1: Geometry of Pin Fin

The rate of heat transfer through the pin fin with insulated tip is given by,

$$Q = \sqrt{(hpka)}\,\Theta_0 \tanh(mL)$$

$$Q\ \dot{\alpha}\ \tanh(mL)$$

From the above equations a graph of 'pin fin efficiency v/s mL' is plotted. As the value of mL or L increases the value of tanh(mL) increases rapidly and then slowly decreases and finally it becomes asymptotic at mL≈3.

Hence from the graph for 100% efficiency of the pin fin, value of mL approaches to 3. By considering the above condition and from the graph we can assume the value for mL.

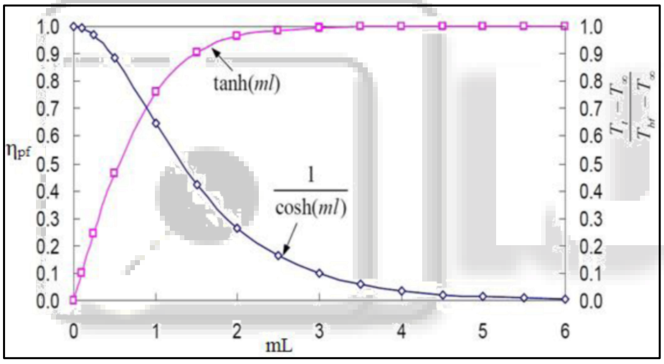


Fig. 2: for Efficiencies Around 70%, Ml = 1.3

Here,

$$m = \sqrt{\frac{hP}{kA}}$$

Where,

h = Convective heat transfer coefficient
P = Perimeter
k = Thermal conductivity of brass
A = Cross section area of pin fin
L = Length of the fin

II. EXPERIMENTAL SETUP

Extended surfaces or fins are used to increase the heat transfer rates from a surface to the surrounding fluid wherever it is not possible to increase the value of the surface heat transfer coefficient or the temperature difference between the surface and the fluid. Fins are fabricated in variety of forms. Fins around the air cooled engines are a common example.

As the fins extend from primary heat transfer surface, the temperature difference with the surrounding fluid diminishes towards the tip of the fin, which plays an important role in fin design.

The Pin Fin Apparatus that is Dynamic apparatus consists of a simple pin fin which is fitted in a rectangular duct. The duct is attached to suction end of a blower. One end of fins is heated by an electrical heater. Thermocouples are mounted along the length of fin and a thermocouple notes the duct fluid temperature. When top cover the fin is opened and heating started, performance of fin with natural convection can be evaluated and with top cover closed and blower started, fin can be tested in forced convection.

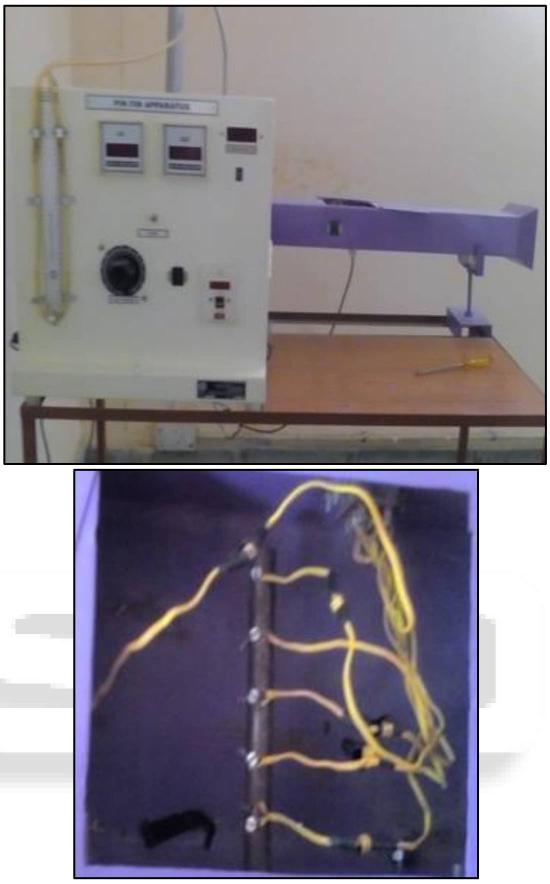


Fig. 3: Pin Fin Heat Exchanger with Fin Arrangement

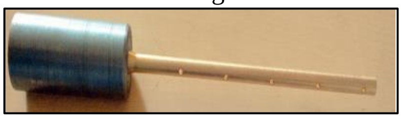


Fig. 4: Mild steel fin

Fig. 5: Brass fin

Fig. 6: Aluminum fin

To measure the temperatures along the length of fin, K - type of thermocouples are used at five equidistant points on the fin surface at a distance OF 2.5cm as shown in fig below,

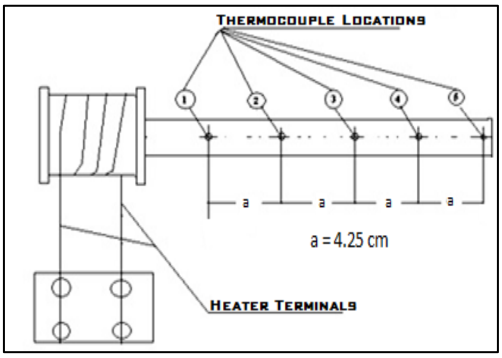


Fig. 7: Heater and Fin Configuration

A. Specification

1) Fins
12mm, outer diameter, effective length 102mm with 5Nos. of thermocouple positions along the length made of brass, mild steel and aluminum- one each.
Fin is screwed in heater block which is heated by a band heater.
2) Duct
150*100mm cross-section 1000mm long connected to suction side of blower.
3) F.H.P.
Centrifugal blower with orifice and flow control valve on discharge side.
4) Orifice
dia. 22 mm. coefficient of discharge Cd= 0.64.
5) Measurements and controls
 – Dimmerstat to control heater input 0-230V, 2amp.
 – Voltmeter 0-250V, for heater supply voltage.
 – Ammeter 0-1amp. For heater current.
 – Multichannel digital temperature indicator.
 – Water manometer connected to orificemeter.

III. EXPERIMENTAL PROCEDURE

A. For Natural Convection

Open the duct cover over the fin. Ensure proper earthing to the unit and switch on the main supply. Adjust dimmerstat so that about 80 volts are supplied to the heater. The fin will start heating. When the temperature remain steady, note down the temperature of the fin and duct fluid temperature. Repeat the experiment at different inputs to heater.

B. For Forced Convection

Close the duct cover over the fin. Start the blower. Adjust the dimmer stat so that about 100-110volts are supplied to the heater. When the temperatures become steady, note down all the temperature and the manometer difference. Repeat the experiment at different inputs and at different air flow rates.

C. Precautions

1) Operate all the switches and controls gently.
2) Do not obstruct the suction of the duct or discharge pipe.
3) Open the duct cover over the fin for natural convection experiment.
4) Fill up water in the manometer and close duct cover for forced convection experiment
5) Proper earthing to the unit is necessary.
6) While replacing the fins, be careful for fixing the thermocouples. Incorrectly fixed thermocouples may show erratic readings.

D. Advantages

Pin fin heat exchangers offer several advantages over the other heat exchangers:

1) Compactness

Large heat transfer surface area per unit volume (Typically 1000 m2/m3), is usually provided by the plate fin heat exchanger. This in turn produces a high overall heat transfer coefficient due to the heat transfer associated with the narrow passages and corrugated surfaces.

2) Effectiveness

Very high thermal effectiveness more than 95% can be obtained.

3) Temperature Control

The plate heat exchanger can operate with relatively small temperature differences. A close temperature approach (Temperature approach as low as 3K between single phase fluid streams and 1K between boiling and condensing fluids is 5 fairly common.),This is an advantage when high temperatures must be avoided. Local overheating and possibility of stagnant zones can also be reduced by the form of the flow passage.

4) Flexibility

Changes can be made to heat exchanger performance by utilizing a wide range of fluids and conditions that can be modified to adapt to the various design specifications. Multi steam operation is possible upto 10 streams.

5) True Counter

Flow operation (Unlike the shell and tube heat exchanger, where the shell side flow is usually a mixture of cross and counter flow.).

The main disadvantages of a pin fin heat exchanger are:
1) Limited range of temperature and pressure.
2) Difficulty in cleaning of passages, which limits its application to clean and relatively noncorrosive fluids, and Difficulty of repair in case of failure or leakage between passages.

IV. OBSERVATION

A. Natural Convection: For Mild Steel Fin

S. N.	INPUT		FIN TEMPERATURE (°C)					DUCT FLUID TEMPERATURE°C
	V	I	T_1	T_2	T_3	T_4	T_5	T_6
1	65	0.601	85	77	67	63	58	32
2	75	0.716	105	97	91	85	80	32
3	85	0.801	125	100	87	80	67	32
4	95	0.845	145	120	110	95	82	33
5	105	0.946	165	132	118	98	89	32

Table 1: Similary for Aluminium and Brass fin observation considered

B. Forced Convection: For Mild Steel Fin

S NO	MANOMETER DIFFERENCE	FIN TEMPERATURE (^0C)					DUCT FLUID TEMPERATURE (^0C)
	H(m of water)	T_1	T_2	T_3	T_4	T_5	T_6
1	20-10.5	62	60	58	57	55	31
2	20-10.5	65	63	61	57	56	31
3	20-10.5	67	65	64	61	59	31
4	20-10.5	69	67	66	63	61	31
5	20-10.5	71	70	68	66	64	32

Table 2: Similary for Aluminium and Brass fin observation considered

V. NOMENCLATURE FOR CALCULATION AND FORMULAS

A	=	Cross-sectional area of fin, m^2.
P	=	Circumference of the fin, m.
L	=	Length of the fin= 0.102m.
T_1	=	Base temperature of the fin.
T_f	=	Duct fluid temperature (channel No. 6 of temperature indicator=T_6).
Ø	=	Temperature difference of fin and fluid temperature= (T_1-T_f).
h	=	Heat transfer coefficient, $w/m^{2\,0}C$
K_f	=	Thermal conductivity of fin material =110w/mK for brass =46w/mK for mild steel. =232w/mK for aluminium.
T_m	=	Average fin temperature= (T_1+T_2+T_3+T_4+T_5)/5
ΔT	=	T_m-T_f
T_{mf}	=	Mean film temperature= (T_m+T_f)/2
$\acute{\rho}_a$	=	Density of air, kgm/m^3.
$\acute{\rho}_w$	=	Density of water kgm/m^3= $1000kgm/m^3$.
D	=	Diameter of pin fin =$12*10^{-3}$m
d	=	Diameter of orifice = $22*10^{-3}$m.
C_d	=	Coefficient of discharge of orifice=0.64
μ	=	Dynamic viscosity of air, N-s/m^2.
C_p	=	Specific heat of air, KJ/kg^{0}C
ν	=	Kinematic viscosity, m^2/s.
K_{air}	=	Thermal conductivity of air, w/m^{0}C.
B	=	Volume expansion coefficient= 1/(T_{mf}+273).
H	=	Manometer difference, m of water.
V	=	Velocity of air in duct m/sec.
Q	=	Volume of air at mean film temperature.
V_{tmf}	=	Velocity of air at mean film temperature.

VI. CALCULATION PROCEDURE

A. For Natural Convection

The fin under consideration is horizontal cylinder loosing heat by natural convection. For horizontal cylinder, Nusselt number.

Nu= 1.10 $(Gr.Pr)^{1/6}$-----------for 10^{-1}< Gr.Pr<10^4
Nu= 0.53$(Gr.Pr)^{1/4}$-----------for 10^4< Gr.Pr<10^9
Nu= 1.10 $(Gr.Pr)^{1/6}$-----------for 10^9< Gr.Pr<10^{12}

Where ,
Gr.= Grashof number= $g\beta D^3\Delta T/\nu^2$
Pr.= Prandtl number= $C_p\mu/k_{air}$
Determine Nusselt number
Now, Nu= $(hD)/k_{air}$

Then fin the heat transfer coefficient. Using h and m, determine temperature distribution in the fin which formula is

$ø/ø_1$= (T-T_f)/(T_0-T_f) = coshm(L-x)/coshm. L the rate of heat transfer from the fin can be calculated as

q= $\sqrt{(hpk_{fin}A)}$* (T_m-T_f)tanhml and efficiency of the fin can be calculated as,

η= tanhmL/mL

B. For Forced Convection

As in natural convection for horizontal cylinder loosing heat by forced convection.

Nu= 0.615$(Re)^{0.466}$--------------for 40<Re<4000
Nu= 0.174$(Re)^{0.618}$--------------for 4000<Re<40000

Where
Re= V_{tmf}.D/ν
V_{tmf} = V(T_{mf}+273)/(T_f+273)

Velocity of air is determined from air volume flow.

Q= $C_d\,\pi\,d^2\sqrt{(2gH(\acute{\rho}_w/\acute{\rho}_a))}$ m^3/sec.
V= Q/duct cross-sectional area
= Q/(0.15*0.1)m/s

From Nusselt number, find out h from h find "m" Now temperature distribution, heat transfer rate and effectiveness of the fin can be calculated using equations 4,5 and 6 respectively. And some other parameter and procedure used from reference papers.

VII. CALCULATION

A. For Natural Convection: For Mild Steel

1	Average surface temperature (T_s)=	(85+77+67+63+58)/5=70^{0}C
2	Mean film temperature (T_{mf})=	(70+32)/2= 51^{0}C
3	At T_{mf} temperature from properties of dry air table μ= $19.7*10^6$Nsec/m^2 C_p= 1.005 KJ/kgK K= 0.0284 W/m-K ν = $17.97*10^{-6}m^2$/sec Pr.= 0.697 Now ΔT= 70-32= 38^{0}C	
4	Coefficient of volumetric expansion(β)=	1/(51+273)= $3.08641*10^{-3}K^{-1}$
5	Grashof number (Gr)	$9.81*3.08641*10^{-3}*38*\{12*10^{-3}\}^2/(17.9*10^{-6})^2$= 6156.79
6	Rayleigh number (Ra)=	(Gr.Pr.)= 6156.79*0.697= 4291.28

7	Nusselt number (Nu)=	$C(Ra)^n = 1.020(4291.28)^{0.148} = 3.5174$
8	Heat transfer coefficient of air of pin(h)	Nu*k/D= 3.5174*0.0284/12*10^{-3}= 8.3245w/mK
9	Fin parameter(m)=	√(8.3245*0.03768/1.1304 *10^{-4})= 7.7650
10	Heat flow rate through fin(Q)=	46*1.1309*10^{-4}*7.7650*(85-32)tanh(7.7650*0.102)= 1.412W
11	Fin efficiency(η)=	1.41/(8.3245*0.03768*0.102*53)=83.156%
12	Effectiveness of fin(Ɛ)=	1.412/(8.3245*1.1309*10^{-4}*53)= 22.88
13	For theoretical temperature At x=2.5cm T-32/{85-32}= cosh7.7650(0.102-2.5*10^{-3})/cosh(7.7650*0.102)=43.97= T At x=5cm, 7.5,10 Same as above T= 42.01, 41.8,40.2 respectively	

Table 3: Similary for Aluminium and Brass Fin Calculation Will Be Done

B. For Forced Convection: For Mild Steel

1	Average surface temperature (T_s)=	(65+63+61+57+56)/5=58.4^{0}C
2	Mean film temperature (T_{mf})=	58.4+31/2=44.7^{0}C
3	At T_{mf} temperature from properties of dry air table K= 0.0278W/m-K ν = 17.42*10^{-6} m^2/sec ῤ= 1.109kg/m^3 Now ΔT= 58.4-31=27.4^{0C}	
4.	Reynolds number (Re)=	(0.6945*12*10^{-3}/17.42*10^{-6})= 478.43
5.	V_{tmf}=	0.66458(44.7+273)/(31+273)= 0.6945

6	Heat transfer flow rate Q=	0.64*3.14*{22*10^{-3}}^{2}√(2*9.81*9.5*10^{-2}{1000/1/109})= 9.96*$10^{-3}$$m^3$/sec
7	V=	9.96*10^{-3}/(0.15*0.1)=0.66458
8	Nusselt no.(Nu.)=	$0.615(478.43)^{0.466}$= 10.90
9	Heat transfer coefficient(h) =	10.90*12*10^{-3}/0.0278=25.25
10	Fin parameter(m) =	√(25.25*0.3768/46*1.1309*10^{-4})= 13.52
11	Fin efficiency(η) =	Tanh(13.52*0.102/)/(13.52*0.102) *100= 63.86%

Table 4: Similary for Aluminium and Brass Fin Calculation Will Be Done.

VIII. Result and Comparision

We have determined all the parameter of circular fin with different materials in natural and forced convection such as mild steel, brass and aluminum which is as follows in case of natural convection:

Material	Heat transfer coefficient(h)	Nusselt Number (Nu)	Rayleigh number	Fin parameter	Efficiency
Mild steel	8.3245w/mk	3.5174	4291.28	7.7650	83.1%
Brass	8.7172w/mk	3.5341	4431.1392	5.138	91.73 %
Aluminum	6.9414	2.933	4313.6	3.15	96.47 %

Table 5: In case of Forced Convection

Material	Heat transfer coefficient(h)	Nusselt Number (Nu)	Reynolds number (Re)	Fin parameter	Efficiency
Mild steel	25.25w/mk	10.90	478.43	13.52	63.86 %
Brass	25.22w/mk	10.89	476.86	8.74	79.88 %
Aluminum	26.06w/mk	11.25	511.50	6.11	88.88 %

From above result we will consider the aluminum type fin which have maximum efficiency.

If we compare the parameters with other than we can see that in performance curve some errors will be developed and result will also vary. There are following curves are

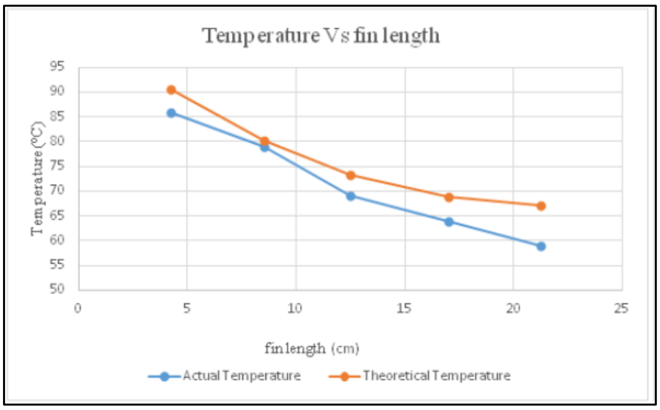


Fig. 8:

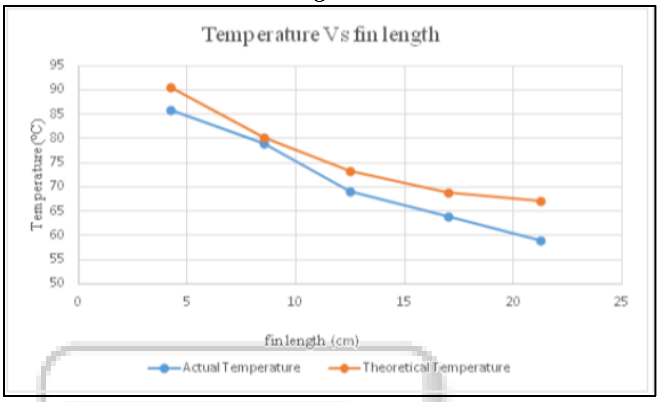


Fig.9

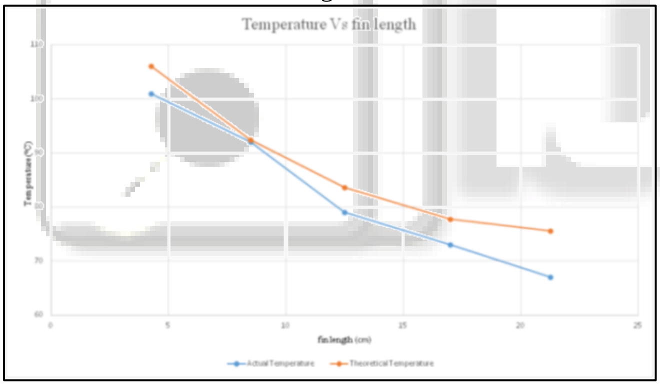


Fig.10

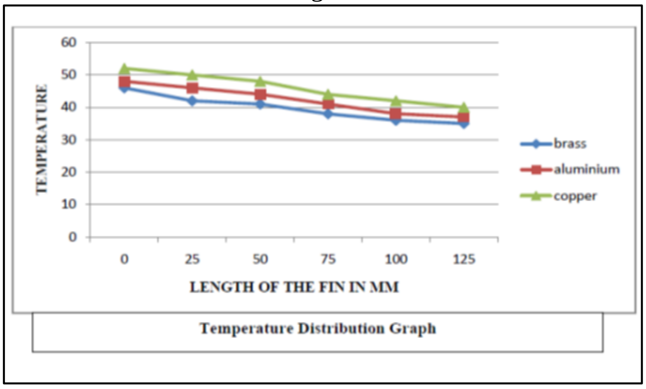


Fig.11

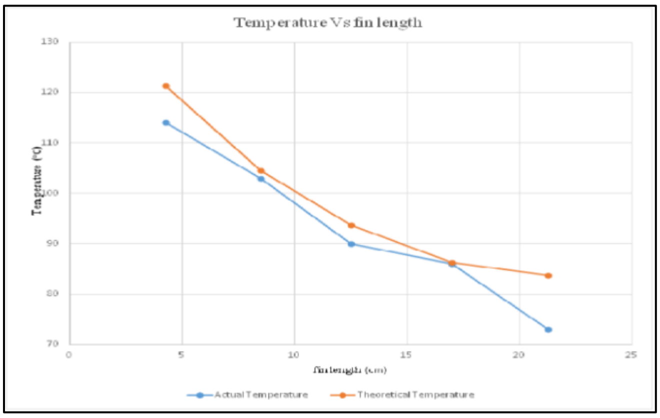


Fig.12

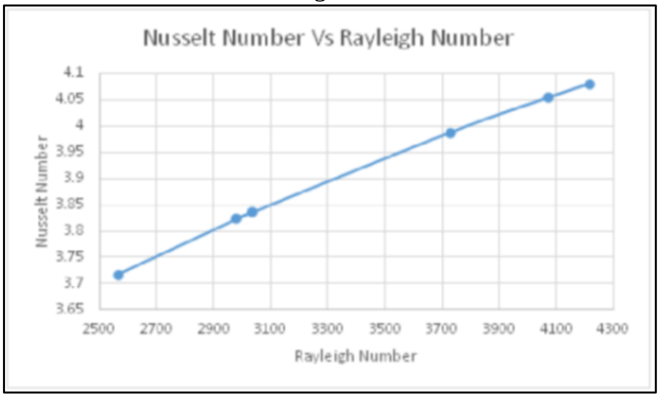
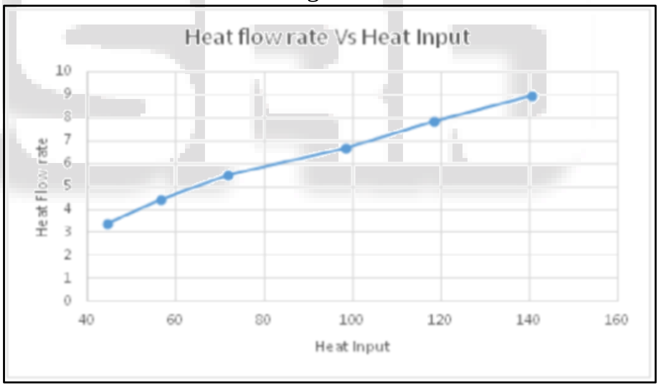


Fig. 13:



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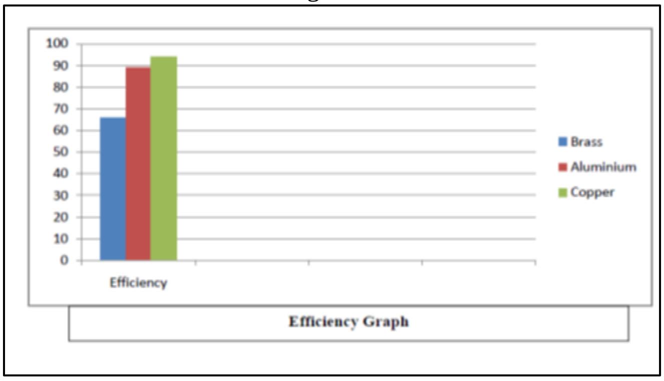

Fig. 15:

IX. CONCLUSION

On the observed temperature distribution and calculation by theory it is expected that observed temperature should be slightly less than their calculated values because of radiation and non-insulated tip.

The graphs of temperature distribution in both natural and forced convection are plotted.

In present work, experimental results gives efficiency of aluminum is greater than brass and mild steel fin.

From the experimental analysis in this thesis heat transfer of fin for different materials is analyzed and it can be improved.

X. Future Scope

Different types of fin with different materials will also be tested and investigated using the set-up constructed. Based on the experimental results obtained from the set-up and the computational fluid dynamics analysis of the same cases, new correlations can be found for the different plate geometries to be tested and analyzed. With the result of new experiments the selection program can also be extended for new type of different fin materials

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