Experimental investigation of heat transfer in plain and tapered cylindrical fins

M. Krishna Prasanna#1, B V S Raghu Vamsi2*, T. Siva Krishna3

1-Assistant Professor, Department of Mechanical Engineering, Gudlavalleru Engineering College, India

2-Sr. Gr. Assistant Professor, Department of Mechanical Engineering, Gudlavalleru Engineering College, India

3- Assistant Professor, Department of Mechanical Engineering, Gudlavalleru Engineering College, India

Abstract *- In this work, the performance of plain circular and tapered circular fins made of mild steel are compared in free and forced convections for the same and different heat inputs. The experiments were conducted by varying the convective area as change in taper angle for various heat inputs. The Reynolds number is also varied in forced convection to study the effect of velocity of circulating cold fluid on the heat transfer characteristics of both the fins. The results conclude that the variable heat transfer coefficient obtained in tapered fin has a strong influence over the fin efficiency. It has been observed that increase of taper angle resulted in shifting of convective boundary condition to the small end to the insulated tip boundary condition due to decrease in local heat transfer coefficient.*

Keywords*: Fins, convection, Reynolds number, nusselt number, grashoff number, heat transfer coefficient*

I. INTRODUCTION

Rate of heat transfer from a hot surface to the surroundings can be increased in many ways. This can be achieved by convection in a more effective way than radiation. Heat transfer by convection is primarily affected by the convected area, type of cold fluid circulating around the solid and its properties, temperature difference etc. Lot of research has been carried out in improving the heat transfer by convection. Paisarn Naphon et. al[1] conducted experiment on in line and staggered taper pin fin heat sinks under constant heat flux conditions are presented. An experimental apparatus is set up to analyze the problem. The tapered pin fin heat sink is fabricated from square aluminium. A finite volume method with an unstructured non-uniform grid system is employed for solving the model. The predicted results are validated by comparing with measured data. The predicted results are in reasonable agreement with the experiments.

Sampath S. S [2] reported that extended tapered surfaces are provided on the element in order to enhance better convection. B. Kundu and P.K.Das [3] addressed the thermal analysis and optimization of straight taper fins.. From the observations they concluded that the variable heat transfer coefficient has a strong influence over the fin efficiency. Balaram Kundu [4] determined temperature distribution and heat transfer of the fully wet fins. He calculated fin effectiveness and fin efficiency for various thermo-geometric parameters. Jitamitra Swain et. al[5] investigated the efficiency and performance parameters of straight triangular fins and porous pin fins in natural convection. Their study is based on a straight triangular fin and a general porous pin fin profile.

II. EXPERIMENTAL SETUP

The experimental setup is as shown in the fig. 1. Straight plain fins and tapered fins are considered in this analysis, which are made of mild steel material. Thermal conductivity of the mild steel is taken as 46 W/m-K. The dimensions of the fins are considered as given below:

- A plain fin of 12 mm and length 102 mm.
- Three tapered fins of big end diameter 12 mm and short end diameter is varied from 4 mm – 8 mm with a span of 2 mm.

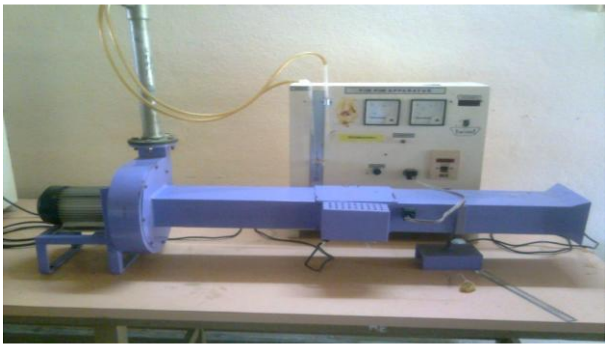


Fig. 1 Experimental Setup

The specification of the setup is given below:

- A dimmer stat to control heater input
- Voltmeter 0-250V, for heater supply voltage.
- Ammeter 0-2A, for heater current.
- Multi-channel digital temperature indicator, to note the thermocouple readings.
- Blower input – 0.5 HP
- Duct: of 150 x 100 mm cross section area and 1000mm length with one end connected to the suction side of the blower

- Water manometer connected to the orifice meter

Fig. 2 Tapered Fins

The big end of the fin is connected to an electrical heater while the other end is left to the free stream of air. It is enclosed in a duct system having rectangular section of 150 mm x 100 mm as shown. One end of the duct is connected to a blower which will suck the air from the surroundings and the duct passage provides the path for the flow of air over the hot surface of the fin. By gaining the heat from the fin, the hot air comes out from the other end of the duct which is kept open to atmosphere. The Reynolds number is varied by adjusting the flow regulating valve fitted to the suction pipe.

An Orifice of 22 mm diameter is provided in the suction pipe to measure the discharge of air over the fin. Five thermocouples are arranged over the fin surface at a distance of 23 mm, 46 mm, 69 mm, 92 mm and 102 mm respectively, to get the temperature distribution over the fin surface. Sixth thermocouple is provided to record the air temperature.

The experiments are carried out for various heat inputs (Q) and various Reynolds number (Re) as given in table 1.

TABLE I
HEAT INPUTS AND REYNOLD'S NUMBER CONSIDERED

S. No	Heat Input		
	Q = 45 W	Q = 56 W	Q = 66 W
1	Re_1=2529.7	Re_1=2403.4	Re_1=2391.9
2	Re_2=3394.7	Re_2=3413.76	Re_2=3411.9
3	Re_3=4153.6	Re_3=4191.95	Re_3=4187.8
4	Re_4=4817.17	Re_4=4882.4	Re_4=4868.3

III. HEAT TRANSFER COEFFICIENT

1. Free Convection

The air is made to flow over the heated fin surface due to the density difference. In this case, the Grashoff number plays a major role in deciding the type of flow through the duct.

Grashoff number is given as $Gr = \dfrac{g\beta D^3 \Delta T}{\gamma^2}$

The product of Gr and Pr decides the type of flow and the Nusselt number is calculated using the equation, $Nu = C(Gr.\Pr)^m$. The values of C and m are presented in table-II.

TABLE II
VALUES OF C & m

C	m	Gr*Pr
1.675	0.058	10^{-10} to 10^{-2}
1.02	0.148	10^{-2} to 10^{2}
0.81	0.188	10^{2} to 10^{4}
0.48	0.25	10^{4} to 10^{7}
0.125	0.333	10^{7} to 10^{12}

Then the heat transfer coefficient is calculated using

$$Nu = \frac{hd}{K_{air}}$$

(i) For the plain cylindrical fin, the cross section area remained constant. Hence the average heat transfer coefficient is obtained as $h = \dfrac{Nu \times K_{air}}{d}$

(ii) For the tapered cylindrical fin, the cross sectional area is varying along its length. Hence local heat transfer coefficient is calculated at the location of thermocouple tapping provided over the fin surface as

$$h_x = \frac{Nu \times K_{air}}{d_x}$$

The average heat transfer coefficient for the tapered fin is calculated as $(h_{avg})_{taper} = \dfrac{\sum_{x=1}^{n} h_x}{n}$

The obtained local heat transfer coefficient is presented graphically as shown in the fig .

2. Forced Convection

In this case, blower attached to the duct is switched on to pass air over the heated fin surface. In this case, Reynolds number decides the type of flow. By using the orifice, the discharge of air is calculated. Since the duct is non-circular, hydraulic diameter of the duct section is calculated before calculating the Reynolds number.

Hydraulic Diameter,

$$D_h = \frac{Area}{wetted \quad perimeter} = \frac{4ab}{2(a+b)} = 0.12m$$

Reynolds number, $\mathrm{Re} = \dfrac{\rho v D_h}{\mu}$

In all the cases, the flow is observed to be turbulent. The Nusselt number is calculated using the equation,

$$Nu = 0.193(Re)^{0.618}(Pr)^{0.333}$$

Then the heat transfer coefficient is calculated using

$$Nu = \frac{hd}{K_{air}}$$

The obtained local heat transfer coefficient is presented graphically as shown in the fig.

Prandtl number is obtained from the property tables of air.

IV. RESULTS AND DISCUSSION

The obtained results are presented graphically as shown below:

Fig 6.1 Reynold's number vs heat transfer coefficient for different heat inputs

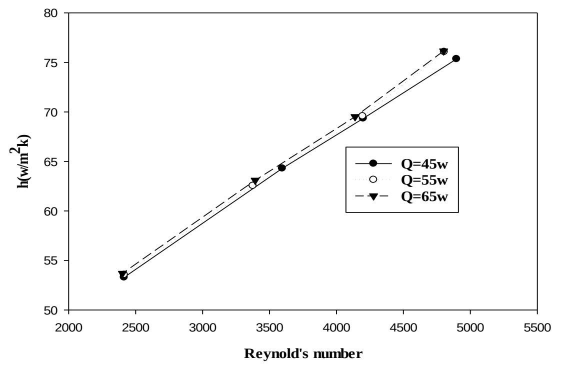


Fig 6.2 Length of the fin vs temperature distribution for Q=45w

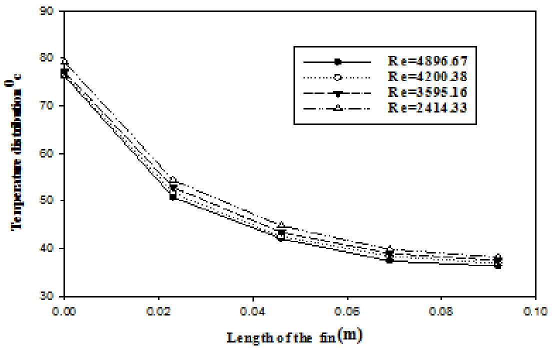


Fig 6.3 Length of the fin vs temperature distribution for Q=55w

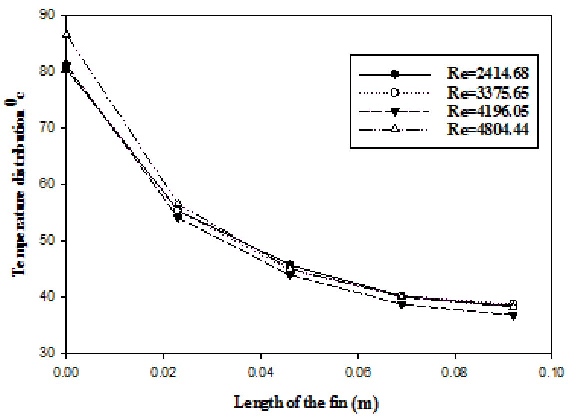


Fig 6.4 Length of the fin vs temperature distribution for Q=65w

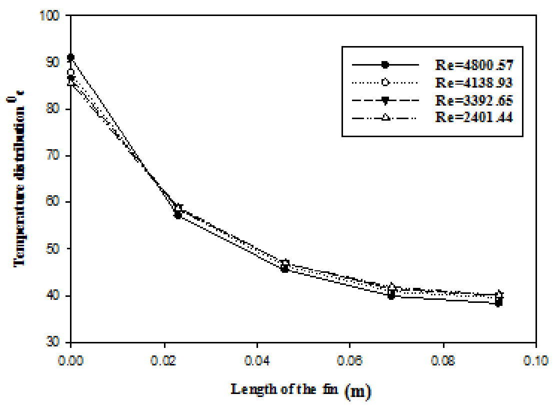


Fig 6.5 Reynold's number vs nusselt number for different heat inputs

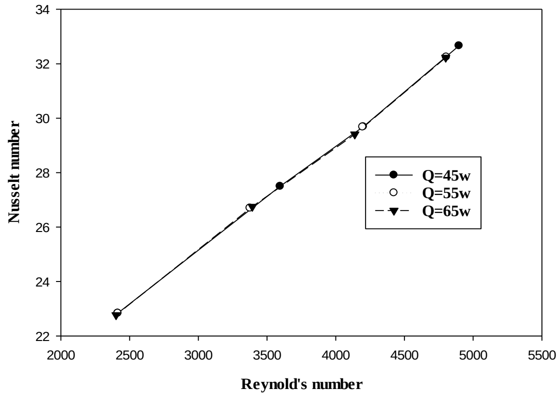


Fig 6.6 Length of the fin vs temperature distribution for different heat inputs

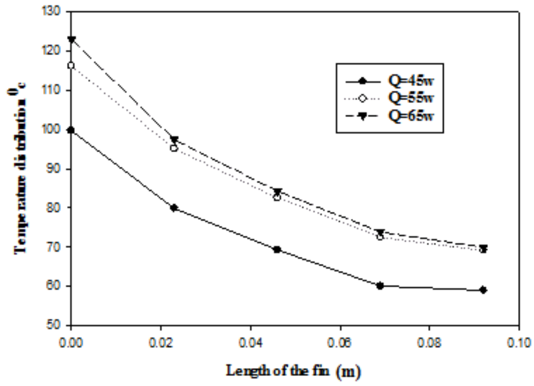
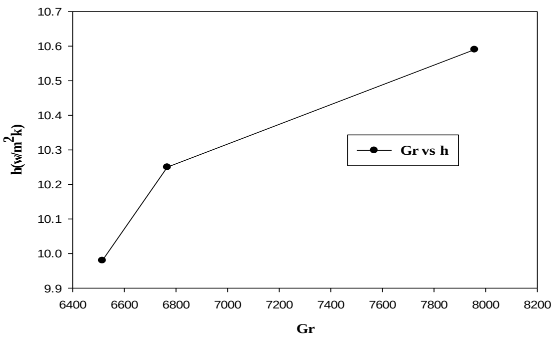


Fig 6.7 Grashof number vs heat transfer coefficient



Tapered fin of tip dia =4 mm

Fig 6.8 Length of the fin vs heat transfer coefficient for different reynold's number for Q=65 w

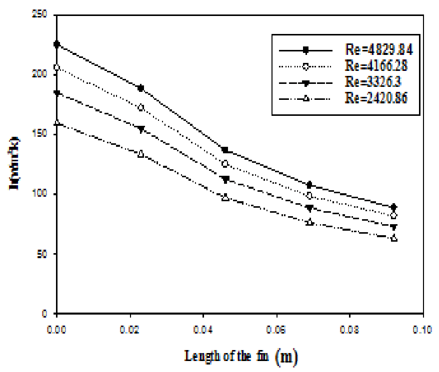


Fig 6.9 Length of the fin vs heat transfer coefficient for different reynold's number for Q=55 w

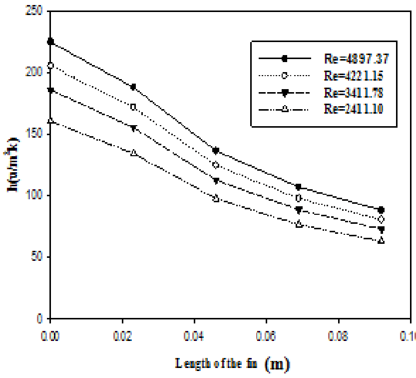
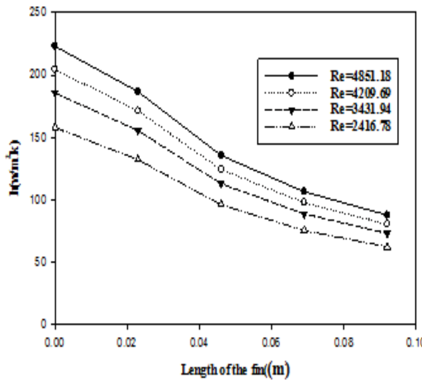


Fig 6.10 Length of the fin vs heat transfer coefficient for different reynold's number for Q=45 w



Tapered fin of tip dia= 6 mm

Fig 6.11 Length of the fin vs heat transfer coefficient for different reynold's number for Q=65 w

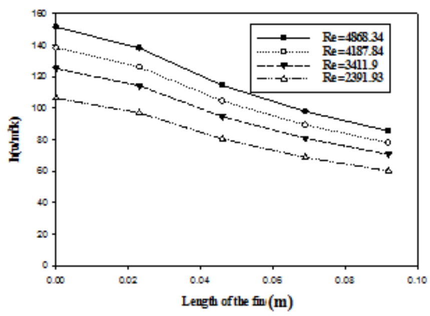


Fig 6.12 Length of the fin vs heat transfer coefficient for different reynold's number for Q=55 w

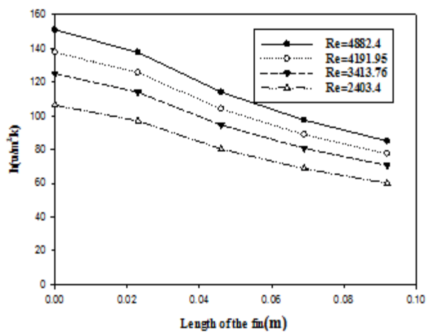
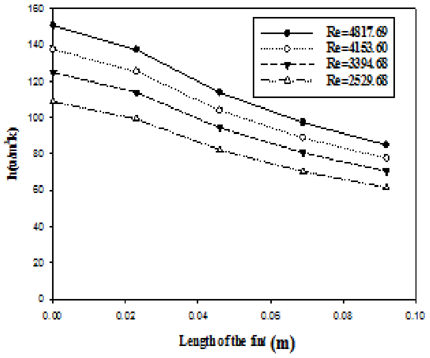


Fig 6.13 Length of the fin vs heat transfer coefficient for different reynold's number for Q=45 w



Tapered fin of tip dia =8 mm

Fig 6.14 Length of the fin vs heat transfer coefficient for different reynold's number for Q=65 w

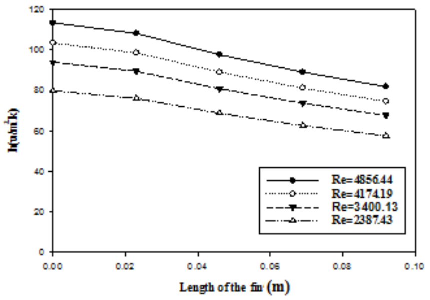


Fig 6.15 Length of the fin vs heat transfer coefficient for different reynold's number for Q=55 w

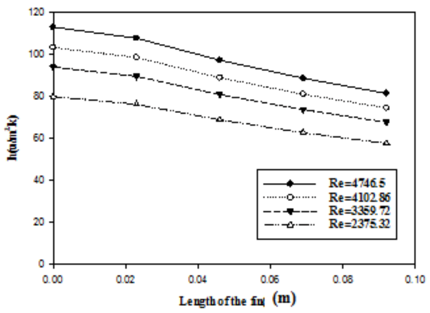
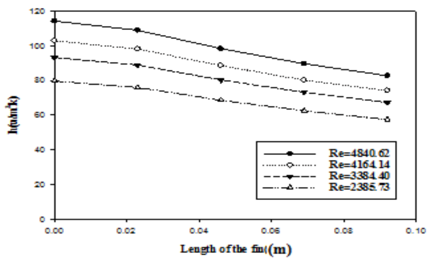


Fig 6.16 Length of the fin vs heat transfer coefficient for different reynold's number for Q=45 w



Tapered fin of tip dia=4mm

Fig 6.17 Length of the fin vs temperature distribution at different reynold's numbers for Q=65w

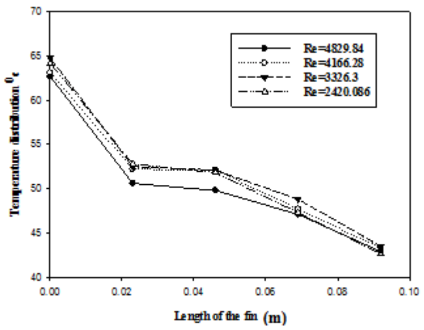


Fig 6.18 Length of the fin vs temperature distribution at different reynold's numbers for Q=55w

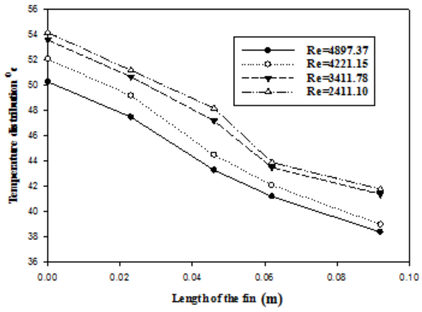
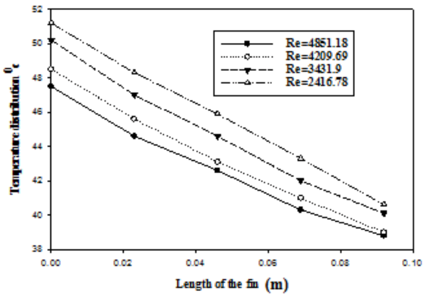


Fig 6.19 Length of the fin vs temperature distribution at different reynold's numbers for Q=45w



Tapered fin of tip dia=6mm

Fig 6.20 Length of the fin vs temperature distribution at different reynold's numbers for Q=45w

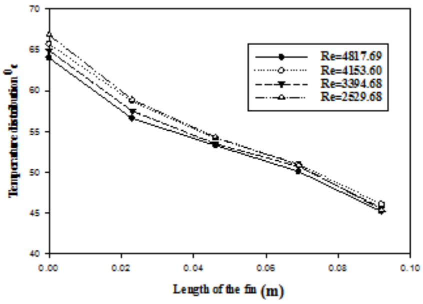


Fig 6.21 Length of the fin vs temperature distribution at different reynold's numbers for Q=55w

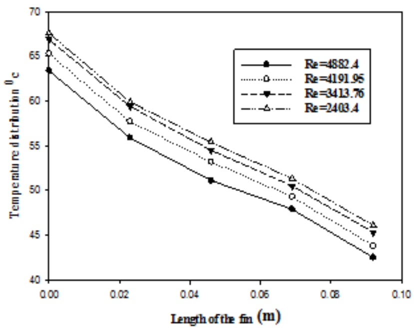
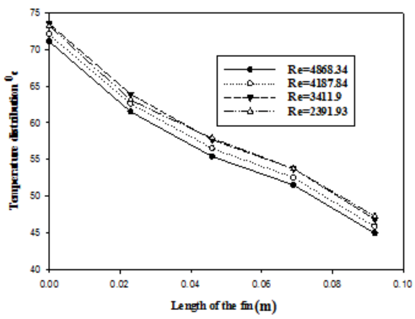


Fig 6.22 Length of the fin vs temperature distribution at different reynold's numbers for Q=65w



Tapered fin of tip dia=8mm

Fig 6.23 Length of the fin vs temperature distribution at reynold's number for Q=45w

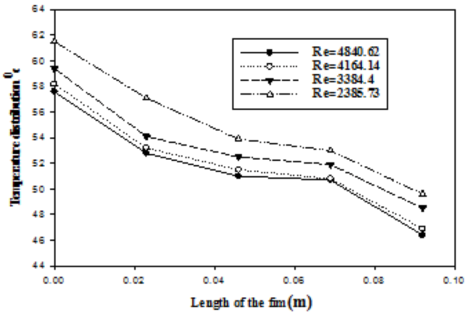


Fig 6.24 Length of the fin vs temperature distribution at different reynold's numbers for Q=55w

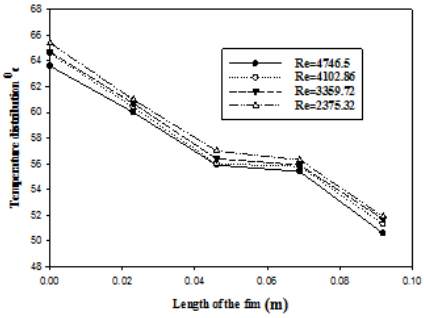
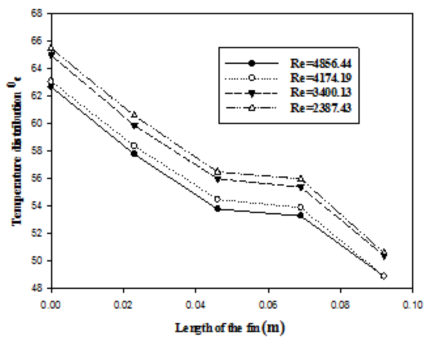


Fig 6.25 Length of the fin vs temperature distribution at different reynold's numbers for Q=65w



- From the results, it has been observed that the local heat transfer coefficient is decreasing from the big end to the small end of the fin and the value is observed to be very low at the small end corresponding to the diameter 4 mm. This is due to decrease in effective heat transfer area for convection.
- Further reduction in the fin tip diameter decreases the heat transfer coefficient and it converges to the insulated tip boundary condition.
- Along the length of the fin, the temperature is observed to be decreasing and these values are much lower corresponding to Reynolds number = 4817. Also increase in Reynolds number implies decrease in viscous forces, and the effect is increase in mass flow rate of air, resulting in higher heat dissipation rate to surroundings. Hence the temperature distribution over the fin surface for Re=4817 is less than corresponding to other values of Reynolds number.

VI. CONCLUSION

In this work we studied the heat transfer through a pin fin by considering plain and tapered fins of various tip diameters made of mild steel. Experiments were conducted on the pin fins for different heat inputs in free convection and along with different Reynolds numbers in forced convection. It has been observed that

- Local heat transfer coefficient decreases along the length of the tapered fins towards the tip of the tip.
- Heat transfer coefficient as well as rate of heat transfer is high in forced convection than compared with free convection.

REFERENCES

[1] Paisarn Naphon, Anusorn Sookkasem (2007), Investigation of heat transfer characteristics of tapered cylinder pin fin heat sinks, from Elsevier journals.

[2] Sampath SS, SawanShetty, ChithiraiPonSelvan M, *Estimation of Heat Dissipation from Plate with Multiple Tapered and Rectangular Fins.* European Journal of Advances in Engineering and Technology, 2015, 2(5): 123-128

[3] B. Kundu, P.K.Das, *Performance analysis and optimization of straight taper fins with variable heat transfer coefficient,* International Journal of Heat and Mass Transfer 45 (2002) 4739–4751.

[4] Balaram Kundu, *An Analytical study of the effect of dehumidification of air on the performance and optimization of straight tapered fins,* International common journal of Heat Mass Transfer Vol. 29. No. 2, pp. 269-218, 2002.

[5] Jitamitra Swain, Kumar Gaurav, Dheerendra Singh, Prakash Kumar Sen, Shailendra Kumar Bohidar, *Comparative Study on Heat Transfer in Straight Triangular Fin and Porous Pin Fin under Natural Convection*, International Journal of Innovation and Scientific Research ISSN 2351-8014 Vol. 11 No. 2 Nov. 2014, pp. 611-619.

[6] *Fundamentals of Engineering Heat and Mass Transfer*, 4th Edition by R.C.Sachadeva.

[7] A Heat Transfer Textbook Third Edition by John H. Lienhard IV and John H. Lienhard V.

[8] Heat and Mass Transfer Data Book by C.P. Kothandaraman and S.Subramanyan.