

Experimental Investigation of Temperature Distribution along the Length of Uniform Area Fin for Forced and Free Convection

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Abstract. The overheating of an industrial component sometimes may leads to system failure. The convection heat transfer from a heated surface can be effectively enhanced by employing fins on that surface. This Paper emphasizes on the experimental investigation of temperature distribution along the length of pin shaped fin. The analysis is performed on a 100 mm long fin made up of brass with 19.6 mm diameter having thermal conductivity as 111 W/m.K. Temperature at different section of the fin along its length is evaluated experimentally and theoretically. The influence of convection mode viz natural & forced convection and variable heat input on the temperature distribution is evaluated. The result outcomes are then compared with the widely accepted analytical relations. A comparison of convective heat transfer coefficient for uniform and non-uniform area fin is also presented. The results by experimental and analytical method are found to be in good agreement for free convection phenomenon.

1. Introduction

Many of the industrial components generate some amount of heat when operated. There is a need of removal of this heat to surrounding atmosphere else this may elevate the temperature of the system which may lead to major heating problems also results in failure of that device. In order to the continuous and efficient working of any instrument, generated heat within the instrument must be dissipated to the surrounding. Various methods that are employed for cooling of heated instrument vary with its application and its cooling capacity. Fins are widely used to enhance the heat transfer rate in electronic circuits and vehicle engines. Fins can be related as a single or series of the extended surface provided on heated component's surface to intensify the heat dissipation from the surface to the surrounding cold fluid. These extended surfaces increase the total heat transfer area and thereby magnifies the heat dissipation rate. Depending on the application, different geometrical configurations of fins are available as uniform cross-section (rectangular and pin) fin and non-uniform cross-section (triangular, elliptical and trapezoidal) fin. From the earlier days, several kinds of research have been done in the area of heat transfer [1–3]. Y. Pratapa et al. [4] performed an experimental study to evaluate temperature distribution of a pin fin. The experimental results were validated with the findings of ANSYS using FEM technique. L. Chapman et al. [5] experimentally analyzed the performance of differently shaped fin as elliptical, straight and cross-cut fins under the atmosphere of low air flow rate. They reported the lowest value of overall thermal resistance for the straight fin. Yang et al. [6] reported an experimental heat transfer analysis for pin, circular, square and elliptical shaped fin. They evaluated the influence of fin density on the heat dissipation rate for both staggered



and in-line fin arrangement. Their results suggested that for staggered arrangement, the coefficient of heat transfer increases with fin density for all differently shaped pin fins. Lowest pressure drop was reported for the elliptical fin. Nagarani and Mayilsamy [7] reported an experimental study on the effectiveness and heat transfer rate for circular and elliptical fins under-variable operating conditions. Their results showed that elliptical fin is found to be more effective than the circular fin.

Sahiti et al. [8] conducted experiments to study the heat transfer and frictional losses in a rectangular channel along with different shapes of pin-fin; e.g., ellipse, circle, and square. They observed highest heat transfer rate for the elliptical pin-fin arrays. Gawai et al. [9] experimentally investigated the heat transfer characteristics and pressure loss for pin fin of different materials. The effect of friction factor and nusselt number on the heat transfer characteristics has been analysed. Many researches are going on the heat transfer analysis of pin fin [10-15] although very less information about the temperature distribution over the fin length and influence of operating conditions on the heat transfer rate was found. Therefore, in this work, an experimental and analytical investigation is carried out to evaluate temperature distribution across the fin length at variable operating conditions. The experiments are conducted for both natural and forced convection atmosphere for both uniform and non-uniform area fin. The experimental findings were then compared with the analytical findings.

2. Experimental Description

The experimental setup is represented schematically by Fig.1. The apparatus is equipped with a blower that provides air flow at different rates. A control valve is also provided to regulate the flow from the blower. The equipment consists of a rectangular channel in conjunction with a nozzle attached to the blower. A 100 mm long pin fin of 19.6 mm diameter made up of brass is fitted at the base to a rectangular channel. The base of the fin is welded with a brass flange of 10 mm thickness and 47 mm diameter. A controlled amount of heat is supplied to the flange by means of an electric heater, which can be varied with the help of a variance. Six thermocouples were installed over the length of the fin as shown in fig. 2. The brass fin is kept inside a rectangular duct. An ammeter and voltmeter are also provided to record the current and voltage readings at a particular heat input. To vary the heat input the voltage may be increased or decreased with the help of a regulator.

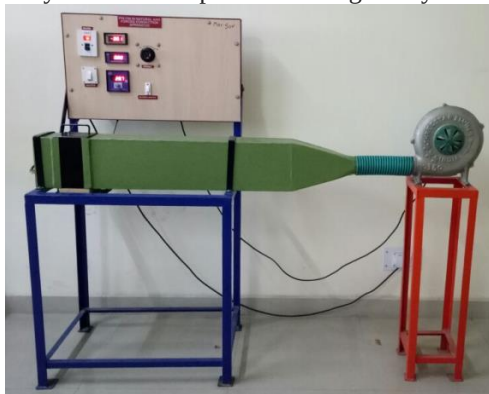


Figure 1. Experimental Set-up of Pin Fin

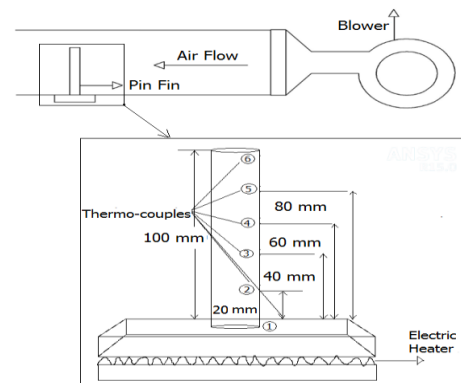


Figure 2. Schematic of Experimental Set-up

3. Analytical Solution Approach

The analytical relations and solutions for uniform area fin are available in several literatures [16-18]. In this study, the fin of finite length with insulated tip was considered. The analytical relations were made by considering some of the assumptions as listed below:

- 1-D conduction in the y-direction.
- The fin material is having constant thermal conductivity.

- No internal heat source is placed inside the fin.
- Radiation heat transfer is neglected

3.1 Free Convection

Heat transfer mechanism that does not need any external agent to generate fluid motion, the flow movement is solely by density difference due to temperature gradient is termed as free convection. Two dimensionless numbers that dominate the free convection heat transfer are Prandtl and Grashof number, can be written as:

$$\text{Prandtl number, } P_r = \frac{\mu C_p}{k} \quad (1)$$

$$\text{Grashof number, } G_r = \frac{\rho^2 D_c^3 \beta g \Delta T}{\mu^2} \quad (2)$$

For the case of flow over vertical cylinder, Nusselt number is represented as:

$$N_u = 0.59(G_r P_r)^{\frac{1}{4}}; \text{ When } 10^4 < G_r P_r < 10^9 \text{ [Laminar Flow]} \\ = 0.59 \left(\frac{\rho^2 D_c^3 \beta g \Delta T C_p}{\mu k} \right)^{\frac{1}{4}} \quad (3)$$

$$N_u = 0.10 (G_r P_r)^{\frac{1}{3}}; \text{ When } 10^9 < G_r P_r < 10^{12} \text{ [Turbulent Flow]} \quad (4)$$

Also,

$$N_u = \frac{h D_c}{k} \quad (5)$$

3.2 Forced Convection

In convection mode of heat transfer if the movement of surrounding fluid is provided by some external agents like fans, blower, pumps etc. then it is termed as forced convection. The fluid flow may be of the laminar or turbulent type. The motion of fluid amplify the heat transfer, flow with higher velocity will increase the heat dissipation rate. Forced convection is dominated by two dimensionless numbers named as Prandtl and Reynolds numbers, can be expressed as:

$$\text{Prandtl Number, } P_r = \frac{\mu C_p}{k} \quad (6)$$

$$\text{Reynolds Number, } R_e = \frac{D_c \rho V}{\mu} \quad (7)$$

The case of flow over vertical cylinder is considered to obtain temperature distribution over the fin length.

The expression for Nusselt number can be written as:

$$N_u = 0.3 + \frac{0.62 R_e^{0.5} P_r^{\frac{1}{3}}}{[1 + (\frac{0.4}{P_r})^{\frac{1}{4}}]^{0.25}} \left[1 + \left(\frac{R_e}{28200} \right)^{\frac{5}{8}} \right]^{0.8}; \text{ for turbulent flow:} \quad (8)$$

$$N_u = 0.023 (R_e)^{0.8} (P_r)^{\frac{1}{3}} \\ = 0.023 \left(\frac{D_c \rho V}{\mu} \right)^{0.8} \left(\frac{\mu C_p}{k} \right)^{\frac{1}{3}}; \text{ for laminar flow} \quad (9)$$

Temperature distribution for finite length fin with insulated tip is expressed as;

$$\frac{t - t_{atm}}{t_0 - t_{atm}} = \frac{\cosh m(L-x)}{\cosh mL} \quad (10)$$

Where m is fin parameter which is given by;

$$m = \sqrt{\frac{hp}{kA}} \quad (11)$$

The position of fin in the rectangular channel can be observed from fig. 3.

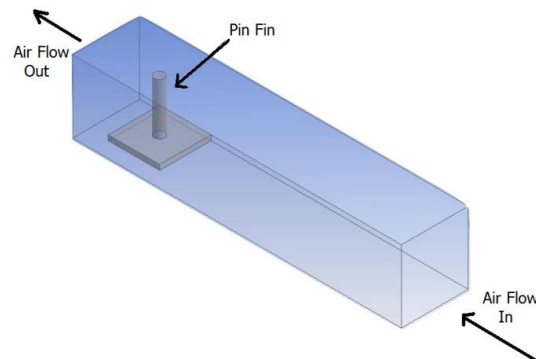


Figure 3. Position of pin fin

4. Results and Discussion

4.1 Influence of variable heat supply on temperature distribution

Case A: Free Convection

In this case, the investigation of temperature distribution over the fin length is performed under the natural circulation of air due to density difference. The temperature distribution at the various section of the fin was experimentally observed at variable heat input supplied by the heater. The variation of heat input is obtained by changing the voltage and current. The experimental results are compared with the analytical solutions for the case of free convection heat transfer over the vertical cylinder by using Eq. 1-5. The experimental temperature distribution findings are found to be in better agreement with the analytical results to those regions which are closer to the fin base (up to 20 mm region). Whereas, the temperature distribution away from the fin base (20 mm to 100 mm region) were significantly different for the experimental and analytical approach. Fig 4-7, shows the temperature distribution across the fin length by both experimental and theoretical approach at different heat input.

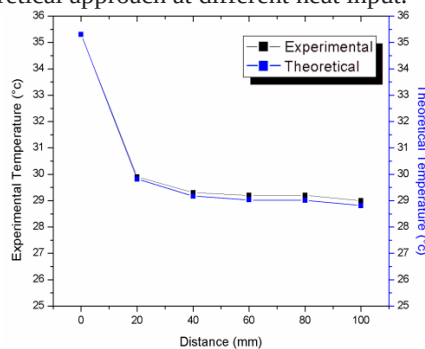


Figure 4. Temperature distribution for 5 watt Heat Input

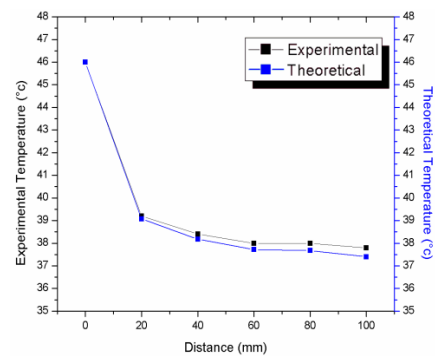


Figure 5. Temperature distribution for 7.5 watt Heat Input

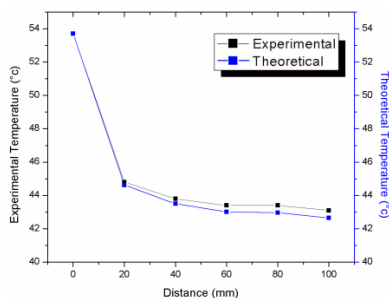


Figure 6. Temperature distribution for 10 watt Heat Input

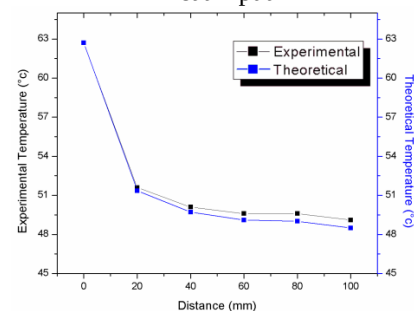


Figure 7. Temperature distribution for 12.5 watt Heat Input

Case B: Forced Convection

In this case, the air is forced to flow over the fin by employing a blower. The temperature profile at the different section of fin length is observed at variable heat input. Experimental results are compared with the findings obtained by employing widely accepted relations as given by Eq. 6-11. The variation of temperature distribution over the fin length by experimental and analytical approach at different heat input is represented by figure 8-11. It can be depicted that the results by analytical relation predict comparatively larger temperature value up-to half of the fin length and smaller value in the next half length. At 20 mm length section from the fin base, the temperature prediction by analytic relation was 12.72% higher than that by experimental finding whereas, it predicts 9.07% lesser magnitude of temperature than the experimental result at 80 mm section from the base for 5-watt heat input. The dominance of forced convection heat transfer increases as the distance of length section from the fin base increases while conduction mode is more responsible for heat dissipation near the fin base region. The dominance of different modes of heat transfer leads to the variation in experimental and analytical thermal behaviour.

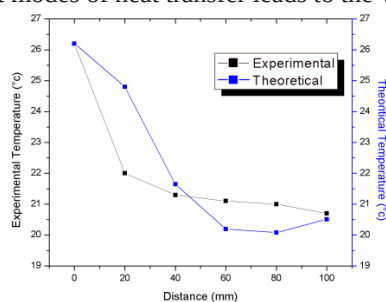


Figure 8. Temperature distribution for 5 watt Heat Input

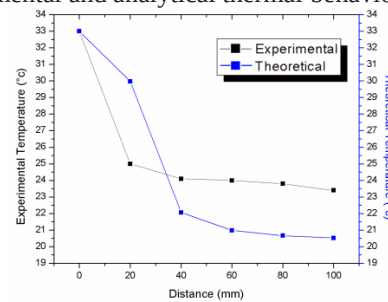


Figure 10. Temperature distribution for 10 watt Heat Input

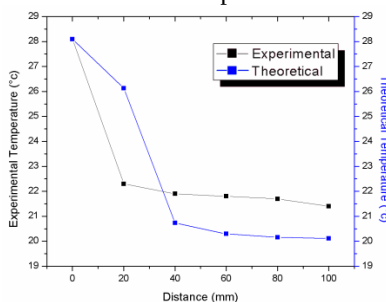


Figure 9. Temperature distribution for 7.5 watt Heat Input

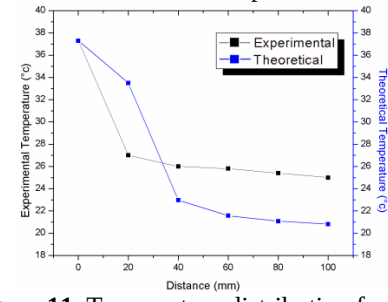


Figure 11. Temperature distribution for 12.5 watt Heat Input

4.2 Effect of geometrical variation on temperature distribution & convective heat transfer (h):

A comparison of convective heat transfer coefficient (h) and temperature distribution for uniform area fin (pin fin) and non-uniform area fin (trapezoidal fin) is performed. Both the fins were made up of brass. Fig. 12 represents the temperature effect of variable heat input on the convective heat transfer coefficient. The results depict a large difference in the value of 'h' for uniform and non-uniform area fin. The magnitude of 'h' for pin fin is found to be 2.2 times higher than trapezoidal fin at the same heat input of 5 Watt.

Table 1. Temperature distribution for Uniform and Non-Uniform cross-section fin

Distance from the base (x, mm)	Temperature for pin fin (°C)	Temperature for trapezoidal fin (°C)
0	62.7	62.7
20	51.6	57.3
40	50.1	52.7
60	49.6	51.4
80	49.3	49.6
100	49.1	49.2

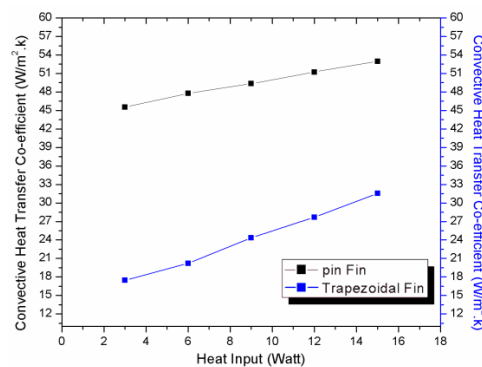


Figure 12. Convective heat transfer coefficient for uniform and non-uniform area fin

5. Conclusion

In this work, an experimental investigation has been performed to study the temperature distribution over the length of a 100 mm long pin shaped fin. Temperature profile at the various section of fin length was studied under variable heat input for both natural and forced convection atmosphere. The experimental findings were then compared with the analytical results. Investigations were also made to evaluate the effect of geometrical aspects on heat transfer characteristics of fin. The following conclusion can be drawn:

1. The results by widely accepted analytical relations were in better agreement with the experimental findings for natural convection.
2. The heat transfer rate for forced convection was 12% higher than that for natural convection at the same operating conditions.
3. Convection mode of heat transfer was found to be more dominating than conduction at the region away from the fin base.
4. The convective heat transfer coefficient (h) for uniform area fin was much higher than non-uniform area fin. At the same operating condition, the convective heat transfer coefficient for pin fin was 54% higher than the trapezoidal fin.
5. Uniform area fin shows better heat transfer characteristics than non-uniform area fin.

Nomenclature

C_p	specific heat of the fluid
μ	viscosity of fluid
k	thermal conductivity of the material of the fin
D_c	characteristic length of fin
P	Density of Fluid
ν	kinematic viscosity of fluid
h	coefficient of convective heat transfer
d	Fin diameter
V	Air Velocity
T_0	Base Temperature
T_{atm}	Atmospheric Temperature
k	Coefficient of thermal conductivity
h	Convective heat transfer coefficient
p	Perimeter of fin
A	Fin Area

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