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AN EXPERIMENTAL INVESTIGATION OF TRAPOZOIDAL FINS ON A VERTICAL BASE IN FREE CONVECTION HEAT TRANSFER

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***Abstract**-In this paper an experimental investigation is carried out for trapezoidal fins with considering steady state free convection skim. The steam was generated in a container by applying heat and was supplied to a rectangular metallic box through an inlet pipe. Fins were attached at one side of the rectangular box. The plate along with fins was locked with the box. The plate could be easily removed and might set-up another plate in the box. By placing a certain plate in the box 100 °C steam at 100 °C was supplied in the box. The fin received heat from steam and transferred heat to atmosphere. As a result the steam considered due to release of latent heat of condensation. The purely mechanical experiment is carried out in different fin length in between 88 mm to 108 mm and fin height from 75mm to 95mm. Fin thickness was maintained at 9.5 mm. The effect of these dimensions on optimum fin height and length was also examined. The results indicated that the optimum fin heights are between 103 mm and 103.5 mm and fin lengths are between 86 mm and 87 mm in this work. Moreover, various comparisons on heat transfer quality are shown considering fin as finished, unfinished, finished and threaded.*

Keywords: Trapezoidal Fins, Free Convection Heat Transfer, Rectangular Box

1. INTRODUCTION

Performance improvement is the common feature of a system. To satisfy this especial needs many sophisticated design are utilized. The simplest and widely used method is air circulated natural convection. Natural convection air-cooling systems are reliable, and easy to maintain. Besides, their design is simple, economic and without any acoustic noise [1].

Newton's law of cooling states that, by keeping the power input fixed and without exceeding a maximum temperature, the convection heat transfer rate from a surface can be increased either by increasing the heat transfer coefficient h or the surface area A or both of these quantities in the equation $\dot{Q}_c = h \times A \times \Delta T$. The convection heat transfer coefficient can be increased by use of a better fluid but this is not an economical solution. Use of a liquid as a coolant, for example, requires a heat exchanger, a pump, piping and other instruments. However, air is inexpensive and often readily available. Since it is not practical and economical to increase the heat transfer coefficient but increasing and optimizing the heat transfer area is widely preferred as the simplest and preferable method to enhance heat transfer [2].

Fins are appendages intimately connected to the primary surface for the augmentation of heat transfer. The most frequent application is one in which an extended surface has been used specifically to enhance the heat transfer rate between a solid and an adjoining fluid. Such an extended surface has been termed a fin [3].

The selection of suitable fin depends on different parameters like geometrical shape, fin spacing, fin height, base thickness, kinds of materials and surface finish, etc. Karagiozis et al. [4] has reported about experiments with triangular fins in vertical and horizontal arrangement under natural convection heat transfer process. Several studies of free convection from rectangular fins were conducted previously [4–15].

In order to achieve the desired rate of heat dissipation, with the least amount of material, the optimal combination of geometry and orientation of the finned surface is required. Among the geometrical variations, trapezoidal fins are considered because of their simple construction, low cost and effective cooling capability. On the other hand rectangular fins are the most commonly encountered fin geometry, but it requires more materials for the same dimensions. In case of triangular fin the successive heating and cooling of fin increases the brittleness and fatigue of the tip.

In this study, the steady state natural convection heat transfer from a wide range of vertical trapezoidal fin configurations protruding from a vertical base is investigated experimentally. The main objective of this experimental work is to determine the effects of geometric parameters and base-to-ambient temperature difference on the heat transfer performance of the fin and obtain results which estimate the optimum fin dimension values for maximum convection heat transfer rates from the fin.

2. 1D MATHEMATICAL MODEL

Considering the one dimensional fin exposed to a surrounding fluid at a temperature T_a. The temperature of the base of the fin is T_w.

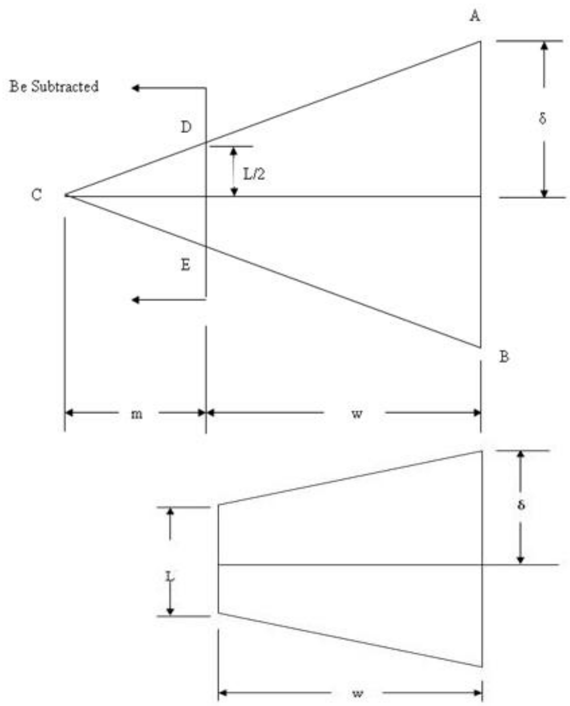

Fig. 1 Schematic diagram of trapezoidal fin

From Fig-1 we have, the temperature differential $(\frac{dT}{dx})$ for trapezoidal fin would be-

$$(\frac{dT}{dx})_{ABCD} = (\frac{dT}{dx})_{ABC} - (\frac{dT}{dx})_{DEC} \text{.........................(1)}$$

Now, the mathematical expression for triangular fin as follows:

The generalized differential equation for temperature distribution is

$$y_x \frac{d^2T}{dx^2} + \frac{dy_x}{dx}\frac{dT}{dx} - \frac{h}{k}T = 0$$

Considering the straight fin of triangular profile, since $x = 0$ is chosen at the fin tip and the maximum thickness at the base $x = w$ is 2δ then $yx = \delta(\frac{x}{w})$ and the general differential equation (1) reduces to

$$x\frac{d^2T}{dx^2} + \frac{dT}{dx} - N^2 wT = 0$$

Where N is a parameter, given by

$$N = \sqrt{\frac{h}{k\delta}}$$

This is a distinguish form of Bessel's differential equation and on Comparing it with the generalized Bessel's equation and solution. We determine its general solution as,

$$T = C_1 J_o(2iN\sqrt{wx}) + C_2 Y_o(2iN\sqrt{wx})$$

Where J_0 and Y_0 are zero order Bessel's functions of the first and second order respectively.

The integration constants C_1 and C_2 are to be determined such that T satisfies the required boundary conditions,

$$T = T_0 \text{ at } x = w, \frac{dT}{dx} = 0 \text{ at } x = 0$$

It is seen that $J_1(0) \,\&\, Y_1(0) = -\infty$

So applying the secondary conditions as,

$$\left(\frac{dT}{dx}\right)_0 = 0 = \left[-C_1 Ni\sqrt{\frac{w}{x}} J_1\left(2iN\sqrt{wx}\right)\right]_0 - \left[C_2 Ni\sqrt{\frac{w}{x}} Y_1\left(2iN\sqrt{wx}\right)\right]_0$$

We find that C2= 0 and the general solution must then be represented by

$$T = C_1 J_o(2iN\sqrt{wx})$$

The evaluation of C1 by application of the first boundary Condition leads to the particular solution for T which reads,

$$\frac{t - t_g}{t_0 - t_g} = \frac{J_o(2iN\sqrt{wx})}{J_o(2iNw)}$$

The rate of heat dissipation from the surfaces of the straight triangular fin is,

$$T = C_1 J_0(2iN\sqrt{wx})$$

$$T_0 = C_1 J_0(2iNw)$$

$$C_1 = \frac{T_0}{J_0(2iNw)}$$

$$T = \frac{T_0 J_0(2iN\sqrt{wx})}{J_0(2iNw)}$$

$$\frac{dT}{dx} = \frac{T_0}{J_0(2iNw)}\left[\frac{d}{dx} J_0(2iN\sqrt{wx})\right]$$

$$= \frac{T_0}{1 + N^2w^2 + \frac{N^4w^4}{12} + ---}\left[\frac{d}{dx}(1 + N^2w^2 + \frac{N^4w^4}{2^2} + ---)\right]$$

$$= \frac{T_0}{1 + N^2w^2 + \frac{N^4w^4}{12} + ---}\left[N^2w + \frac{N^4w^2x}{2} + ---)\right]$$

…………………………………………………….. (2)

From equation (2), calculating $(\frac{dT}{dx})$ for triangle ABC and $(\frac{dT}{dx})$ for triangle DEC, the temperature gradient $(\frac{dT}{dx})$ can be calculated from equation (1) for trapezoidal fin ABED.

For $x = 0 \rightarrow 108mm, T_0 = 100^0 C, N = 0.691$

We calculated the value of temperature gradient from the above expressions.

$$\left.\frac{dT}{dX}\right|_{ABC} = 34.17$$

$$\left.\frac{dT}{dX}\right|_{DEC} = 15.32$$

Now from equation (1),

$$\left.\frac{dT}{dX}\right|_{x=w=108mm} = 18.85$$

3. EXPERIMENTAL SET-UP AND PROCEDURE

The set-up consists of rectangular box in vertical position holding the fins. Box size was $(18'' \times 18'' \times 6'')$. Fins were attached to $\frac{1}{16} in$ MS plate of $(18'' \times 18'')$ size shown in (Fig. 2). A burner is used to heat the steam generating unit to generate steam and that is supplied to the rectangular box.

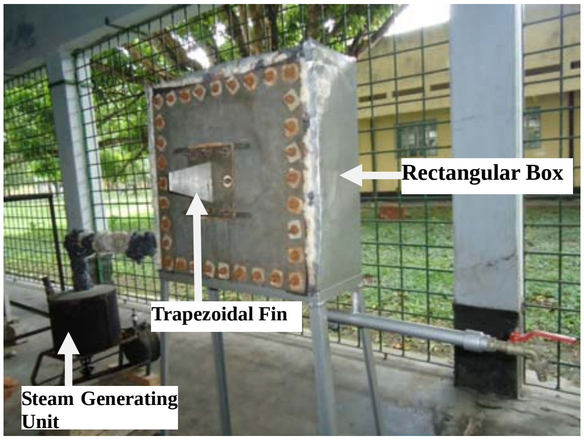

Fig. 2 Photograph of experimental setup

Temperature remains constant during latent heat of condensation. Thus using this principle the latent heat of condensation principle was applied for maintaining constant temperature at the plate. The steam was generated in a container by applying heat and was supplied to a rectangular metallic box through an inlet pipe. The temperature of inlet steam was measured by a thermometer, placing at inlet pipe. Fins are attached at one side of the rectangular box. The plate along with fins was locked with the box by nut and bolts and was sealed by using gasket. By placing a certain plate in the box 100 °C steam at 100 °C was supplied in the box. The fin received heat from steam and transferred heat to atmosphere. As a result the steam considered due to release of latent heat of condensation. The condensed water was delivered by outlet pipe and collected in a beaker. The total 12 fins are observed. All results are taken after heating the steam generating unit for 2 hours. That is done for constant heat supply.

During an experiment, the temperatures before and after the fins under investigation are measured by thermo-couples made of Copper-Constantan.

4. RESULTS AND DISCUSSION

The experimental data obtained from more than 12 different fin configurations are experimented. These results are utilized to reveal the effects of geometric parameters, fin height and fin length, and the effects of base-to-ambient temperature difference on the steady-state heat transfer rates from finned surfaces. The experimental data are presented in different figures to examine the effect of each parameter separately.

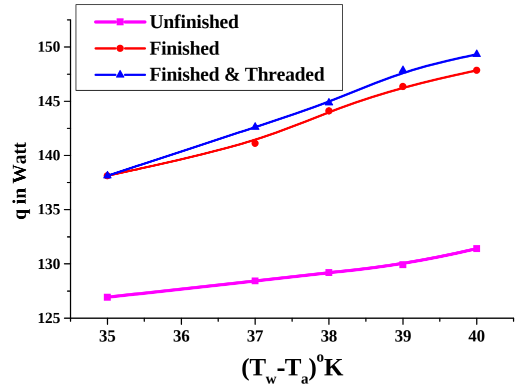

Fig. 3 Variation of Convection Heat transfer rate with temperature rise at different fin condition.

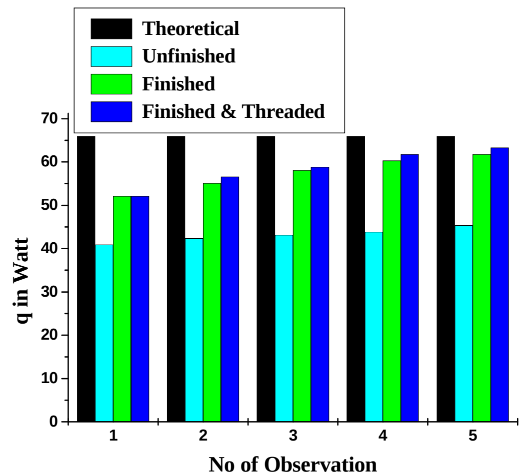

Fig. 4 Comparison of actual Convection Heat Transfer with theoretical value of different fin condition

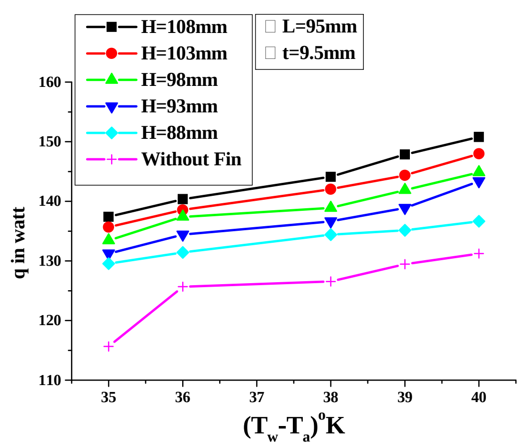

Fig. 5 Variation of Convection Heat Transfer rate with Fin Height

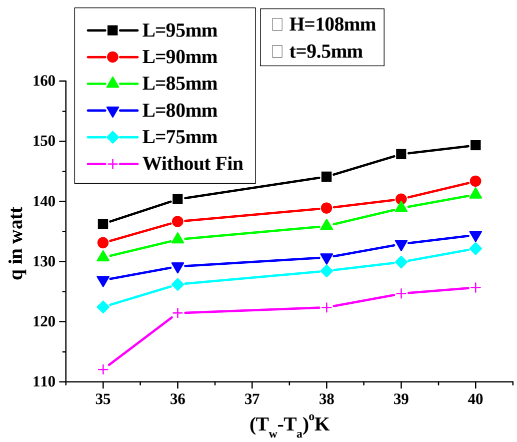

Fig. 6 Variation of Convection Heat Transfer rate with Fin Length

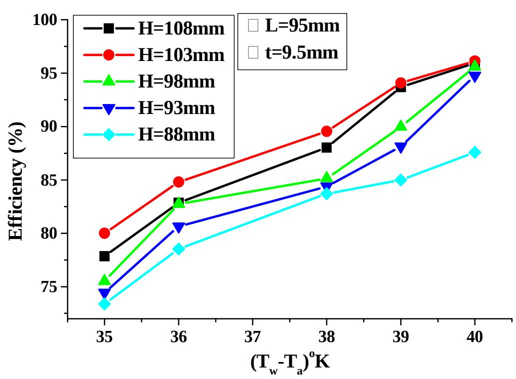

Fig. 7 Variation of efficiency with Fin Height

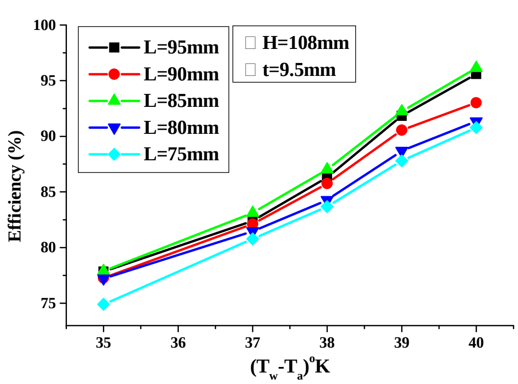

Fig. 8 Variation of efficiency with Fin Length

Figures 3 and 4 depict heat transfer rate in various types of fin such as unfinished, finished and threaded. The quality of heat transfer rate is hugely increased for finished & threaded surface with respect to unfinished surface.

As observed in Figures 5 through 6, the heat transfer rate from fin depends on fin height, fin length and base-to-ambient temperature difference. It is seen that the convective heat transfer rates from the fin increases with fin height and base-to-ambient temperature difference. The effect of extending the fin length from 75 mm to 95mm results in higher steady-state convective heat dissipation from the fin. However, the curves demonstrating the behaviors of fin heights show similar trends for similarly thickness and different fin length. The heat transfer rates measured from five fin heights are close to each other at low base-to-ambient temperature differences whereas at high base-to-ambient temperature differences, the heat transfer rates tend to slightly diverge with the variation in fin height.

On the other hand, considering varying fin length and fin height the effect of efficiency is increased more rapidly as shown in Figures 7and 8.

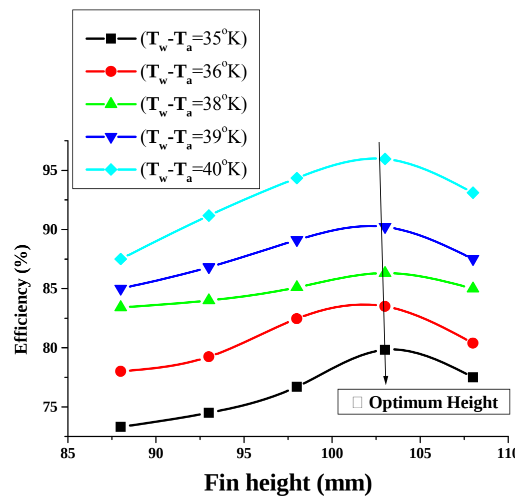

Fig. 9 Variation of efficiency with Base-to-Ambient Temperature Difference

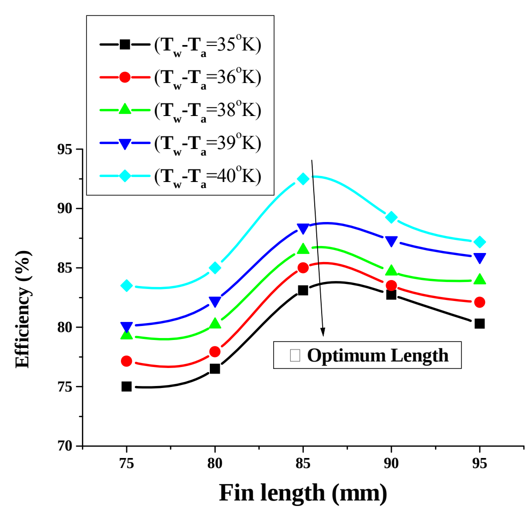

Fig. 10 Variation of efficiency with Base-to-Ambient Temperature Difference

From examination of Figures 9 and 10, it can be deduced that the efficiency from an fin increases with both fin height and length, and after reaching its maximum point, it starts decreasing at a given fin height, fin length and base-to-ambient temperature difference. Similarly it has been said that heat transfer rate is also in the same fashion as efficiency is. The corresponding fin height and length value of the maximum convection heat transfer rate point is called optimum fin height and length.

5. CONCLUSIONS

In this study, the steady-state natural convection heat transfer from vertical trapezoidal fins protruding from a vertical base was investigated experimentally. The effects of geometric parameters, fin height, fin length and base-to-ambient temperature difference on the heat transfer performance of fin were discussed. A relation for the optimum fin length and height value that maximizes the heat transfer rate was obtained.

Experimental results showed that the larger fin height results in higher convection heat transfer rates from the fins. As the temperature differences between fins and ambient decrease, the effect of fin height becomes insignificant. On the other hand, the variation in fin height influences the rate of convection heat transfer slightly for smaller fin length with respect to larger fin length.

The effect of fin length on convection heat transfer performance of fin arrays was also observed. As a result of experimental data, the change of fin length from 75mm to 95 mm causes an increase in convection heat transfer rates for each fin configuration.

It is concluded from the tested 12 fin configurations that the optimum fin lengths are between 86 mm and 87 mm. The optimum fin length and fin height results are presented in Table 1. As seen from Table 1, optimum fin dimension depends on fin height, fin length and base-to-ambient temperature difference. However, neither of these parameters changes the value of the optimum fin dimension for fin thickness 9.5 mm.

Table 1: Optimum fin height and length

	H_{opt} (mm)	L_{opt} (mm)
ΔT (^oK)	t= 9.5 mm	t= 9.5 mm
35	103	86
36	103.1	86.2
38	103.3	86.5
39	103.4	86.7
40	103.5	87

7. REFERENCES

[1] Kakac ¸ S, et al (1994) Cooling of electronic systems. NATO ASI Series, Kluwer Academic Publishers, Dordrecht.

[2] Kraus AD (1965) Cooling electronic equipment, Prentice-Hall, New Jersey.

[3] Kai-Long Hsiao, Conjugate Heat Transfer for Free Convection along a Vertical Plate Fin. Journal of Thermal Science Vol.19, No.4 (2010), pp. 337−345

[4] Karagiozis A; Raithby GD; Hollands KGT (1994) Natural convection heat transfer from arrays of isothermal triangular fins in air. ASME J Heat Transfer 116: pp. 105-110

[5] Starner KE; McManus HN (1963) .An experimental investigation of free convection heat transfer from rectangular fin arrays' Heat Transfer 273–278

[6] Welling JR; Wooldridge CB (1965). Free convection heat transfer coefficients from vertical fins. J Heat Transfer 439–444

[7] Harahap F; McManus HN (1967) Natural convection heat transfer from horizontal rectangular fin arrays. J Heat Transfer 32–38

[8] Jones CD; Smith LF (1970) Optimum arrangement of rectangular fins on horizontal surfaces for free convection heat transfer. J Heat Transfer 6–10

[9] Fitzroy ND (1971) Optimum spacing of fins cooled by free convection. J Heat Transfer 462–463

[10] Sparrow EM; Bahrami PA (1980) Experiments on natural convection heat transfer on the fins of a finned horizontal tube. Int J Heat Mass Transfer 1555–1560

[11] Sparrow EM; Niethammer JE (1981) Effect of vertical separation distance and cylinder-to-cylinder temperature imbalance on natural convection for a pair of horizontal cylinders. Trans ASME 638–643

[12] Leung CW, et al (1985) Heat exchanger design: thermal performances of rectangular fins protruding from vertical or horizontal rectangular bases. Appl. Energy 123–140

[13] Kadle DS; Sparrow EM (1986) Numerical and experimental study of turbulent heat transfer and fluid flow in longitudinal fin arrays. J Heat Transfer 16–23

[14] Leung CW; Probert SD (1989) Thermal effectiveness of short-protrusion rectangular, heat-exchanger fins. Appl Energy 1–8

[15] Sobhan CB, et al (1990) Experimental studies on steady free convection heat transfer from fins and fin arrays. Warme Stou -bertragung 345–352

8. NOMENCLATURE

Symbol	Meaning	Unit
T	Temperature	^0K
A	Area	m^2
D	Diameter	m
g	Acceleration of gravity	m/s^2
h	Heat transfer co-efficient	W/m^2- ^0K
m	Mass	kg
Q	Total heat	J
q	Heat transfer rate	W
R	Thermal resistance	$^0K/W$
t	Thickness	m
ΔT	Temperature difference	^0K
l	Length	m
L	Latent heat of condensation	kJ/kg
υ	Kinetic viscosity	m^2/s