Development of Energy Efficient Heat Sink by Augmentation of Natural Convection Heat Transfer Characteristics from Horizontal Rectangular Fin Array with Rectangular Notch at Middle

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Abstract – Heat transfer due to free convection of air from Notched, full rectangular fin array have been investigated experimentally. For study purpose short fin array has been selected which show single chimney flow pattern. Middle portion of fin array becomes ineffective due to low temperature difference between entering air & fin surface. So in present study, middle portion is removed by cutting rectangular notch and added where more fresh air come in contact with fin surface area. Results have been obtained over range of spacing from 12mm to 25mm and heat input from 25W to 100W. Length & height of rectangular fin array was kept constant. Comparison has been made between full, notched rectangular fin arrays. It is found that Notched rectangular fin array performed better as expected.

Keywords: Fin arrays, Average Heat transfer coefficient, Free convection, Spacing.

I INTRODUCTION

Starner and McManus, Harahan and McManus, Jones and Smith, Mannan have studied the general problem of free convection heat transfer from rectangular fin arrays on a horizontal surface experimentally and theoretically by Sane and Sukhatme. During their investigations, flow visualization studies have also been conducted and it has been found out that the single chimney flow pattern was preferred from the heat transfer stand point and was present in most of the lengthwise short arrays used in practice.

The present paper is consists of an experimental study on horizontal rectangular short fin arrays with notch, without notch at the center & dissipating heat by free convection. In case of a single chimney flow pattern, the chimney formation is due to cold air entering from the two ends of the channel flowing in the horizontal direction and developing a vertical velocity flow of air as it reaches the middle portion of fin channel resulting in the heated plume of air going in the upward direction Full & Notched fin arrays are investigated with different spacing & heat inputs.

Fig.1 Exploded view of fin array

Experimental setup is developed on the basis of simplicity and practicability. Fin arrays are assembled & manufactured using 2 mm thick commercially available aluminum sheet. Size of sheet is 120X40. It is observed meticulously that all the fin flats are cut to the same size simultaneously. All fins are glued to base plate with help of adhesive backing which sustain for high temperature. Holes were drilled for placing cartage heater in base plate.

II EXPERIMENTATION

The following procedure is used for the experimentation:

1. The fin arrays are assembled by gluing the required number of fin plates by using epoxy resign and positioning the thermocouples at the appropriate locations.
2. Cartridge heaters (02 numbers) are placed in their position, connected in parallel with power circuit.
3. Assembled array as above is placed in the slotted C4X insulating block.
4. Thermocouples are placed in the C4X block for measuring conduction loss. The assembled array with insulation is placed at center of an enclosure.
5. The decided heater input is given and kept constant by connecting to stabilizer, which is provided with

dimmerstat voltage.

6. The temperatures of base plate at different positions, C4X brick temperature and ambient temperature are recorded at the time intervals of 15 min. up to steady condition. (Generally it takes 2 to 3 hours to attain steady state condition).

Fig.2 Assembly of Rectangular fin array

Table.1 Parameters of Experimentation

Spacing in mm	Heater input in watt	Length of fin array in mm	Height of fin array in mm
12	25	120	40
14	50		
18	75		
25	100		

Readings were recorded on reading table when the steady state was reached. Readings were taken at least four times for four different configuration and heater input to ensure the validity and repeatability of readings. It is decided that variables for experimental work are spacing, heater input, and geometry. Spacing are 12mm, 14mm, 18mm and 25mm. Heater inputs are 25watt, 50watt, 75watt & 100 watt. The results were obtained from the observations.

Experimental Calculations

1. Conduction Loss =KA $\frac{dT}{dx}$

2. Radiation Loss = $\in \sigma A$ [Ts4- T∞4]

3. Heat Transfer Coefficients = $\frac{Q}{A\Delta T}$

4. Nusselt Number = $\frac{hL}{K}$

5. Grashof number = $\frac{g\beta(Ts - T\infty)Lc^3}{U^2}$

III RESULT & DISSCUSSION

Results have been obtained in terms of average heat transfer coefficient, base heat transfer coefficient, Average Nusselt number, Base Nusselt number, Grashof number.

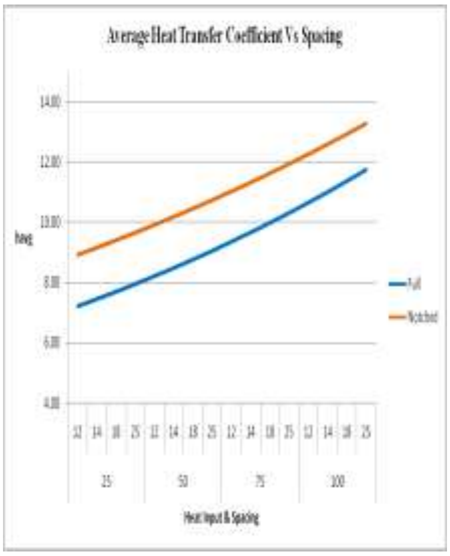

Fig.4 Graph of Average heat transfer coefficients Vs spacing

Fig. 4 show the effect of fin spacing on h_a with heater input as the parameter. As the fin spacing increases h_a increases for full fin array, as expected. The highest value of h_a is 13.95 W/m^2 K at the spacing of 25 mm. The increasing trend is steep up from spacing about 18 mm. Before which there is a gradual rise. The trend of increase in h_a and hence in the Nusselt number with fin spacing is observed in case of the notched array also with increase in h_a values at every point. The notched configurations yield higher values, thus indicating superiority over full fin arrays.

Also fig.4 shows the relative performance of fin array with notch and that of without notch. It is evident from the graph that h_a increases with the heater input, maintaining the superiority of notched array. It is clear that for the given heater input h_a of notched array is 10 to 30% higher than corresponding full fin array. Average heat transfer coefficient of Notched fin array is 32% higher than full fin array for 12mm spacing.. Average heat transfer coefficient of Notched fin array is 14% higher than full fin array for 14mm spacing. By doing data analysis, Percentage increase in average heat transfer coefficient of notched fin array in comparison with full fin array is increased as the spacing increases. It is shown that 12mm spacing is more effective when comparison have been made between Notched & Full fin array.

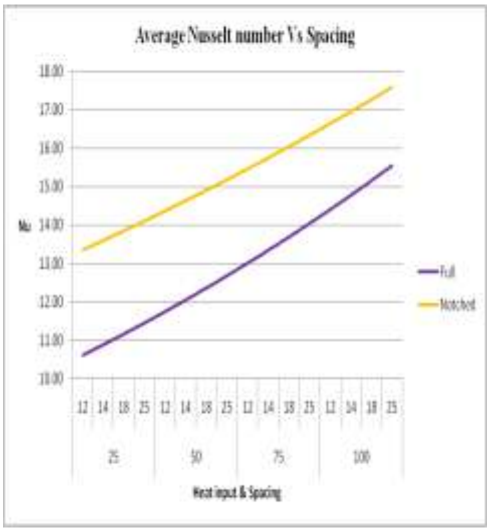

Fig.5 Graph of Average Nusselt number Vs Spacing

It is clear from the fig.5 that as spacing increases the average Nusselt number Nu_a increases for the notched fin array. The increasing trend is gradual up to a spacing of 14 to 18 mm. After that the rise is sudden. The notched configurations yield higher values, thus indicating superiority of notched fin array over Notched & full fin array. The highest Nu_a is about 18 W/m^2 K for the notched fin array at heater input of 100 W. In general it is observed that the Nu_a increases with increase in fin spacing, this is due to reason that with increase in spacing, the fluid can flow more freely through the fin channel. This may be attributed to the phenomenon of lateral boundary layer interference at lower fin spacing. Nu_a dimensionless number increases from 11 to 18 with increase in heat input from 25 to 100 W for notched fin array which is higher than that of full fin array. Best fin spacing is above 18 to 25 mm

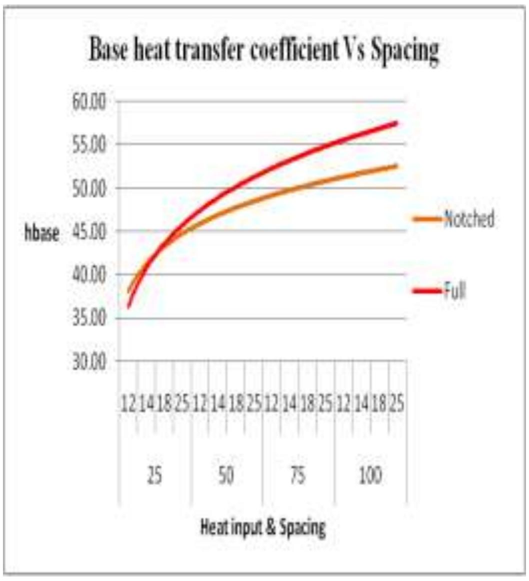

Fig.6 Graph of Base heat transfer coefficients Vs Spacing

Fig. 6 show the effect of fin spacing on h_b with heater input as the parameter. From the figure it is clear that the values of h_b decreases as fin spacing increases. It starts to its maximum value at fin spacing about 12 mm and again decreases gradually. This trend can be attributed to restriction of entry of air in the channel at smaller fin spacing. The trend of increase in base heat transfer coefficient with the maxima at a fin spacing of 14 mm is observed in case of the full fin array. It is therefore concluded that performance of full fin array is bettering terms of base heat transfer coefficient. At the spacing of 18mm, h_b is nearly 61 W/m^2 K for the full fin array and is of the order of 55 W/m^2 K for the notched fin array. This is due to decrease in heat transfer area.

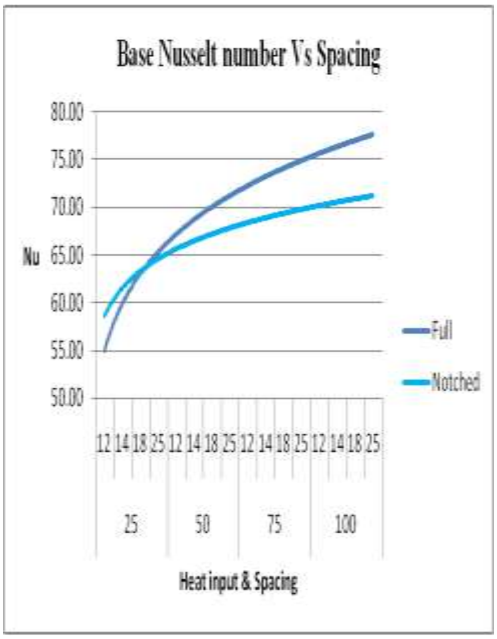

Fig.7 Graph of Base Nusselt number Vs Spacing

Fig. 7 shows variation of base Nusselt number with fin spacing for notched fin array & Full fin array. It is clear that as the value of Nu_b decreases as fin spacing increases. It reaches to its maximum value and again decreases. The reason for decrement in Nu_b may be due to less surface area at higher spacing. So that full configurations yield slightly higher values, thus indicating superiority over notched fin array. It is seen that base Nusselt number is decreasing up to 18 mm spacing & after that it is increasing for notched fin array. Therefore, Best fin spacing is above 18 mm. Nu_b dimensionless number increases from 63 to 68 with increase in heat input from 25 to 100 W for Notched fin array which is higher than that of full fin array.

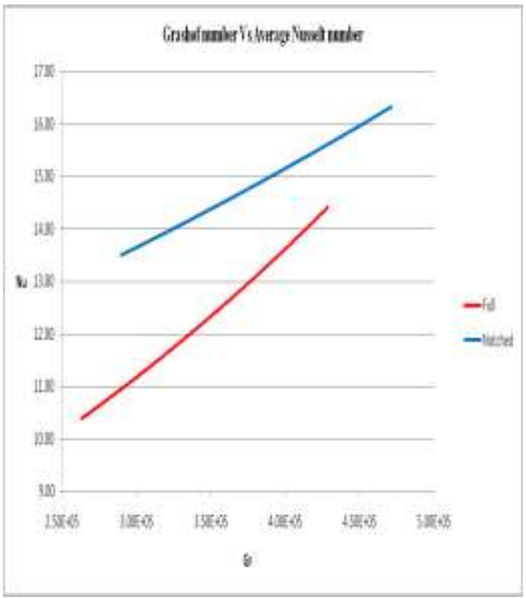

Fig.8 Graph of Grashof number Vs Nusselt number

Fig.8 represents Graph of average Nusselt number with Grashof number for notched fin & full fin array. It can be investigated that with the increases in Grashof number, average Nusselt number increases for a given spacing. The increase in average Nusselt number for notched fin array is more than the equivalent full rectangular fin array.

VI CONCLUSIONS

The problem of free convection heat transfer from horizontal rectangular fin array has been the subject of experimental as well as theoretical studies.

The important findings of the experimentation are as follows:

- Single chimney flow pattern reported to be preferred by earlier investigators is retained in the notched fin arrays as well by performing simple smoke test.
- Study shows that notched horizontal rectangular fin array is more effective than that full fin array.
- Rise in h_a for Notched fin arrays exhibit 10-30% higher than corresponding full fin array configuration.
- Average Nusselt number for notched fin arrays is 10-30% higher than corresponding full fin array.
- h_b & Base Nusselt number is continuously decreasing with increase in spacing for notched & full fin array.
- Grashof number for notched fin array is 8-15% higher than corresponding full fin array.
- Results show that Grashof number is less than 10^9. Therefore, free convection heat transfer with laminar flow of air is confirmed.

Nomenclature

A Cross Sectional Area of C4X bricks

$\frac{dt}{dx}$ Temperature Gradient along bricks

$\in$ Emissivity of Brick

σ Stefan Boltzmann's constant

g Acceleration due to gravity

β Coefficient of volume expansion

T_s Average Temperature of fin surface

T_∞ Temperature of Air

ʋ Kinematic viscosity of air

K Thermal Conductivity of C4X bricks

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