ISSN 2395-1621

Experimental investigation of vertical stepped fin arrays under natural convection

#1Anurag B. Suke, #2Santosh G. Taji, #3N.K.Sane

1anuragsuke13@gmail.com
2santosh_taji@yahoo.com
3Narayan.sane@yahoo.com

#12G H Raisoni College of Engineering and Management, Pune, India
#3Walchand College of EngineeringVishrambag, Sangli, India
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ABSTRACT

a This paper represents the experimental Study of heat transfer from vertical stepped fin arrays under natural convection. The parameters varied for the experimentation are heater input (Q=50-200W), even fin spacing (S=6-22mm), different fin array geometry and its orientation by keeping fin length, fin height and width of array constant. The measurement of convective heat transfer is very critical and depends on estimation of average heat transfer coefficient. The experimentation results are presented in terms of various heat transfer parameters such as average heat transfer coefficient (h_a), base heat transfer coefficient (h_b). Dimensionless parameters such as average nusselt number (Nu_a), base nusselt number (Nu_b), and Rayleigh number (R_a). The natural convection results are compared with results of plain vertical rectangular fin array (VRFA). The results obtained are compared with earlier investigators result from standard referred papers of vertical fin arrays under natural convection. The correlation between average nusselt number (Nu_a) in terms of Rayleigh number (R_a) and S/H ratio found from vertical stepped fin array.

***Keywords*— a Fin Spacing,Natural Convection,Stepped Up,Stepped Down and Vertical rectangular fin arrays.**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Thermal processes have practical importance in the heat exchangers from the heat transfer point of view. For the application of heating and cooling widely used are heat exchangers. For designing the heat exchangers heat transfer, pressure losses, weight and price should be taken into account. Obtaining high heat transfer rates through various enhancement techniques can lead to substantial energy savings. In a heat exchanger heat transfer processes are mainly improved by turbulence promoters via providing suitable optimum flow condition having different geometrical features as well as orientations. The heat transfer process are closely related to the performance the effects of the sizes and orientations of these geometries. The high power density electronics requirements design of efficient cooling strategies is essential for reliable performance. Many of a failure mechanism in electronic

devices, such as inter-metallic growth, metal migration, and void formation, are closely related to thermal effects.

The operation of many engineering systems results in generation of heat. If the heat which is generated within a system is not dissipated rapidly to its surroundings the temperatures of the components may rise and the system may not function effectively and safely. The use of natural convection air-cooling extended surfaces provides a reliable, cheap and widely used method of cooling for dissipating unwanted heat. Besides, their design is simple, economic and without any acoustic noise. However the designs of these need to be optimized so the rate of convective heat dissipation through them are maximized. Without exceeding

a maximum temperature and by keeping the power input fixed, the convective heat transfer from an extended surface can be increased either by increasing heat transfer coefficient or the surface area or both of these quantities. Increasing the heat transfer area is preferred as the simplest method to enhance heat dissipation rate, because the use of better fluid to increase the heat transfer coefficient is not an economical and practical solution. The only the controllable variable to enhance the convective heat transfer rate from an extended surface is the geometry of the fins.

Fins are used to enhance convective heat transfer in a wide range of engineering applications, and offer a practical means for achieving a large total heat transfer surface area without the use of an excessive amount of primary surface area. Fins are commonly applied for heat management in electrical appliances such as computer power supplies or substation transformers. Other applications include Internal Combustion engine cooling, such as fins in a car radiator. It is important to predict the temperature distribution within the fin in order to choose the configuration that offers maximum effectiveness. Natural convection heat transfer is often increased by provision of rectangular fins on horizontal or vertical surfaces in many electronic applications, motors and transformers**.** The current trend in the electronic industry is miniaturization, making the overheating problem more acute due to the reduction in surface area available for heat dissipation.

II. LITERATURE

The problem of heat transfer from fin arrays has been studied extensively. Various geometries and orientations have been investigated.

2.1 INVESTIGATION OF ELENBAAS [1]

One of the earliest works of relevance is due to Elenbaas, who studied the problem of heat dissipation by natural convection between isothermal parallel plates. They found that the heat dissipation, from vertical isothermal parallel plates was a function of the spacing, the height and temperature difference between the plates and the environment. A correlation is proposed in terms of *Nu* and *Gr* based on spacing *S* as the characteristicdimension.

$$Nu_s = \left(\frac{1}{24}\right)\left(\frac{S}{H}\right) Gr_s \Pr\left[1-\exp^{-35/(S/H)Gr_s \Pr}\right]^{3/4} \ldots(1)$$

Elenbaas equation for vertical parallel plates gives only a rough estimate of the value of the Nusselt numbers for vertical fin arrays. Many investigators have however, adopted it as a guideline and have reported their results in the form of an Elenbaas type of equation.

2.2 INVESTIGATION OF STARNER AND MCMANUS [2]

The very first investigation on both vertical and horizontal rectangular fin arrays is due to Starner and McManus. Natural convection heat transfer for four different fin arrays also with three different base plate was calculated.Flow pattern of these cases were observed by using new smoke filaments. Parameters which were varied in their study were the fin spacing as well as fin height.

2.3 INVESTIGATION OF WELLING AND WOOLDRIDGE [3]

Welling and Wooldridge used four fin spacing and three fin heights giving 12 configurations in all. For all configurations following dimensions were kept constant.

L = 203.2 mm, W_a= 66.3 mm, $2t$ = 2.286 mm, and t_b = 6.35 mm.

Variation in geometric and other parameters was done as H = 19.07 mm. 12.7 mm, 6.35 mm

S = 19.07 mm, 10.57 mm, 6.86 mm, 4.825 mm

L^+ = 0.0935. 0.0625, 0.0313,S^+= 0.935, 0.0517, 0.0338, 0.0237

They compared their values of Nusselt numbers with those of Elenbaas for vertical parallel plates and of Starner and McManus for vertical fin arrays. The Nusselt numbers of Welling and Wooldridge were based upon the hydraulic radius of the fin channel as the characteristic dimension. They have not given any general Elenbaas type equation.

2.4 INVESTIGATION OF SAIKHEDKAR AND SUKHATME [4]

Saikhedkar and Sukhatme solved the problem of natural convection heat transfer from vertical rectangular fin arrays theoretically. The governing equations were solved after neglecting the velocity component normal to the fin flats. A finite difference method was employed. The values of the dimensionless parameters were

$Pr = 0.7$, $500 \le Gr_s \le 10^5$, $0.1 \le S^+ \le 0.3$, and $0.1 \le L^+ \le 0.25$.

The computational results for Nu_s were plotted against Gr_s. It is seen that Nu_s values increases with Gr_s and S^+ for constant H^+ = 0.25. The computational results were correlated as follows:

$Nu_s = 0.079\, S^{+\,0.579} L^{+\,(-0.261)}\, Ra_s^{0.509}$,for $0 \le Ra_s^+ < 50$….. (2)

$Nu_s = 1.619(1+11.0/S^+)^{-0.436}(1 - 3.8L^+)^{0.058} Ra_s^{0.28}$,

For $250 < Ra_s^+ < 3 \times 10^4$……(3)

2.5INVESTIGATIONOF A.N.TIKEKAR AND N.K.SANE [5]

The comprehensive studies of natural convection heat transfer from the fin arrays having vertical fins on horizontal base surface and the arrays having vertical fins on vertical base surface. They studied more detailed theoretical and experimental analysis of the fin arrays. Analysis of vertical arrays having fins of different shapes. The fin shape studied include stepped (upward and downward), tapered (upward and downward), triangular (apex up and down), symmetrical triangular and rectangular notched. Each type of fin array is compared with its equivalent rectangular counterpart. From the theoretical analysis the stepped fin arrays were found to be favourable.

2.6 INVESTIGATION OF J.P. SHETE, N.K. SANE AND S.PAVITHRAN[6]

They presented numerical analysis of mixed convection from isothermal vertical rectangular fin array; at Richardson number of unity is presented. At this Richardson number, both natural and forced convection modes are comparable. Computations are done for assisting and opposing modes of mixed convection for a range of temperature differences. Other objective of their work is to find the optimum spacing zone. For given temperature differences, Nusselt numbers are small for natural convection and become large for mixed modes, at all spacings. For a range of temperatures, optimum spacing zone is found.

III.EXPERIMENTATION

Literature review revealed that there is no experimental work reported so far on vertical rectangular stepped fin array under natural convection. An experimental set-up is designed and developed to carry out the investigation on natural convection heat transfer over heated vertical rectangular stepped fin arrays. The main objective of the experiments is to determine the heat transfer characteristics experimentally and to obtain optimum spacing zone.

We will prepare different vertical fin array assembly by changing different fin spacing.

Table No.-1:- Different fin array assembly details

Sr. No.	Spacer (mm)	No. of Spacer	Fin thickness (mm) (t)	No. of fin	Total VFA assembly length(mm) (w)
1	10	8	2	9	98
2	22	4	2	5	98
3	12	7	2	8	100
4	8	10	2	11	102
5	6	12	2	13	98

Fig. 3.1 Photograph of Fin Array Used for Experimental analysis

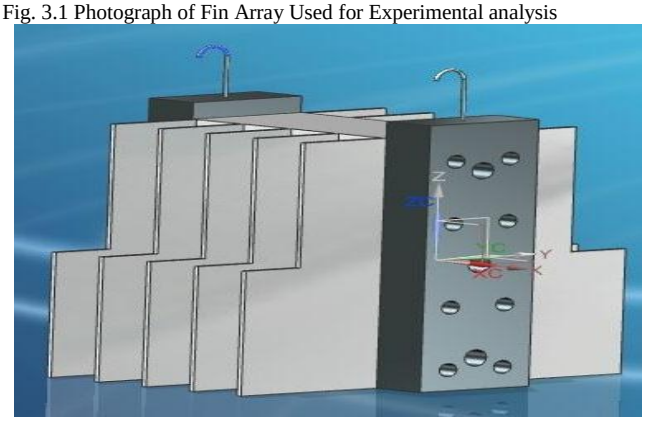


Fig 3.2.Vertical Stepped Fin Array Assembly

These fin array assembly will hang in enclose (1m ×1m x1m). Attaching thermocouples at different places of fin. By varying, power supply to cartage heater we will take readings after achieving stady state observations under natural mode of convection. The fin spacing and heater input are the parameters of an experimental investigation. Also, we will try to observe flow visualization by smoke detection technique. Finally we will compare the different assembly readings and we will try to find optimum numbers of fin, fin spacing, and
fin geometry.

IV. RESULT & DISCUSSION

The experimental work has been carried out under natural convection over heated vertical stepped fin array. The results are plotted in terms of various heat transfer parameters like average heat transfer coefficient(h_a), base heat transfer coefficient(h_b), average nusselt number (Nu_a), base nusselt number(Nu_b). Results obtained are compared with previous investigators for natural convection. As spacing increases from S=6-22mm, the value of h_a are higher for large spacings. On increasing fin spacing, initially, average heat transfer coefficient increases and then reaches a maximum value of 8.37 W/m^2K(Q=200W) after which it starts to decline. When the fin spacing is smaller 6mm, the resistance against the flow is formed due to the intersection of boundary layers developed on fin surfaces and as a result, the rate of heat transfer from fin array decreases. At lower spacing ha values are lower as expected, whereas at wider spacing ha values on higher side because of the free flow.

4.1 Validation of experimental work:

In order to assess the present experimental result, a comparison has been made with the previous literature as shown in fig 4.1. The present experimental data is superimposed on the same plot. It is observed that the present data is confirming the trends obtained by previous investigators.Fig 4.1 shows the effect of Ra_s^+Vs. Nu_s is based on fin spacing & results of vertical fin arrays are compared with standard literature available. It is found that results of vertical rectangular fin are in between stepped up & stepped down vertical fin for same surface area. Stepped down fin shows better performance than all other investigators result. Stepped up shows worst performance than vertical stepped down fin array and rectangular fin array.

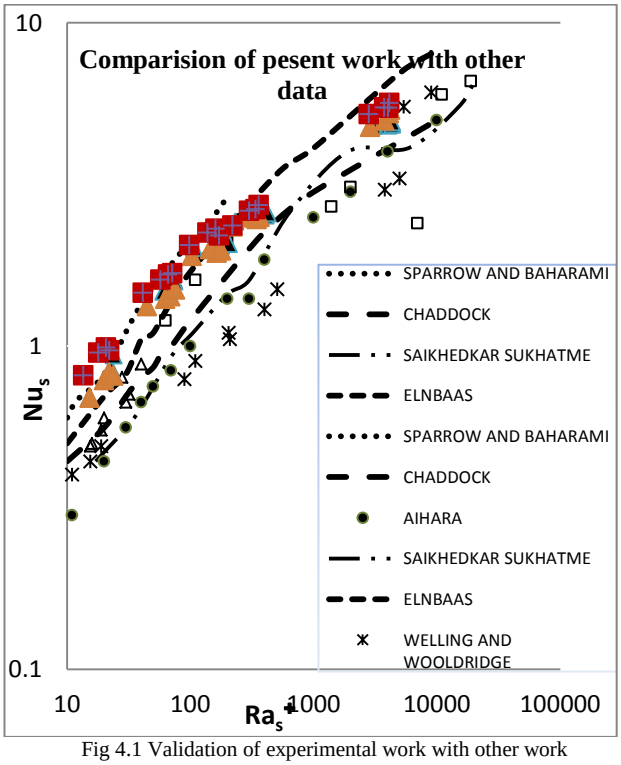


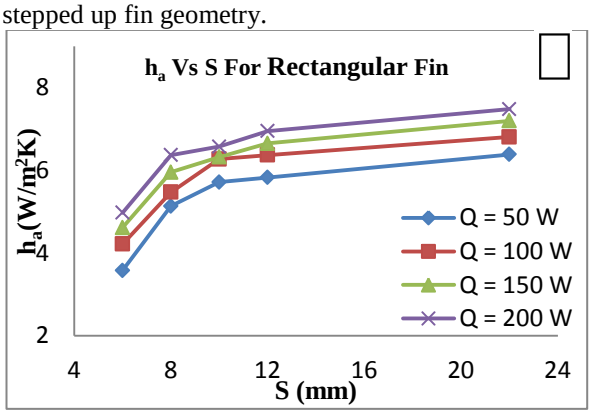
Fig 4.1 Validation of experimental work with other work

4.2 Effect of fin spacing on h_a

Fig 4.2 (a, b, c) shows effect of spacing 'S' on average heat transfer coefficient 'h_a' with different heat input for three different geometry. As the fin spacing increases, the h_a increases for fin array. In the beginning, h_a values are very small. The highest value of h_a is 8.37 W/m^{2}K (for stepped down fins) at S=22mm. h_a is increases from 3.31 to 8.37 W/m^{2}K, when fin spacing changes from 6mm to 22mm. Thus there is significant effect of fin spacing on average heat transfer coefficient. For Stepped Down fin geometry we can achieve maximum h_a as compared to other geometry. It is evident from the graphs that the average heat transfer coefficient increases with the heater input.

4.3 Effect of fin spacing on h_b

Fig 4.3 (a, b, c) shows the effect of fin spacing 'S' on base heat transfer coefficient 'h_b' with heat input for three different geometry. As the fin spacing increases, the h_b increases up to some value and again decreases. At S=8mm h_b is maximum (82.35W/m^{2}K) for stepped down fin array at 200W heat input. It is minimum at S=6mm (49.02W/m^{2}K) for stepped up fin array at 50W heat input. For value of higher input h_b is high and vice a versa. $h_b$is maximum in case of stepped down fin geometry and minimum for stepped up fin geometry.



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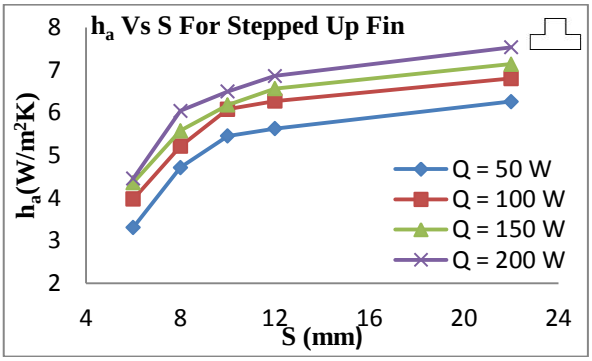

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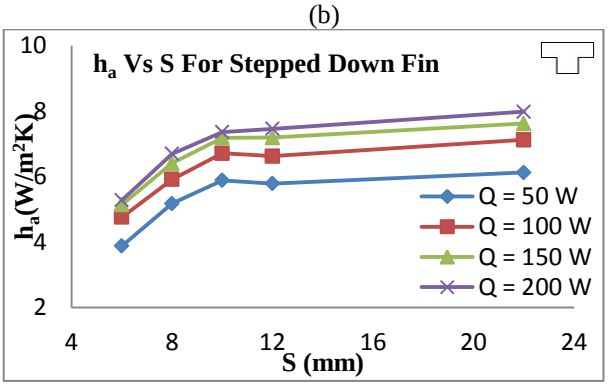
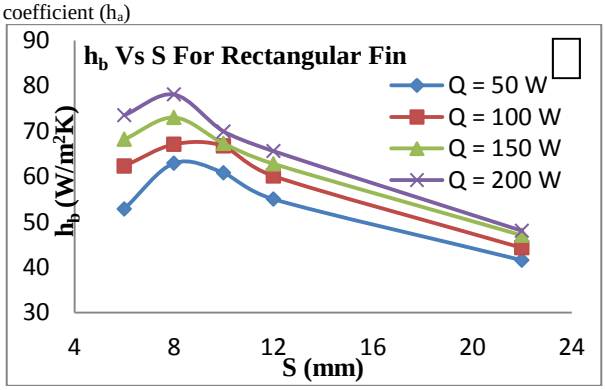
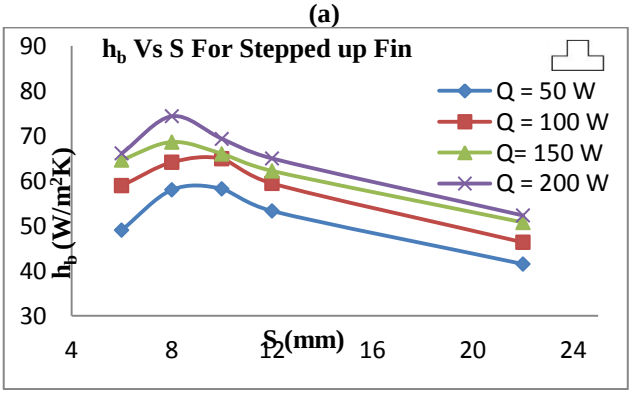


(c)

Fig 4.2 (a, b, c) Effect of fin spacing (S) on average heat transfer coefficient (h_a)



(a)



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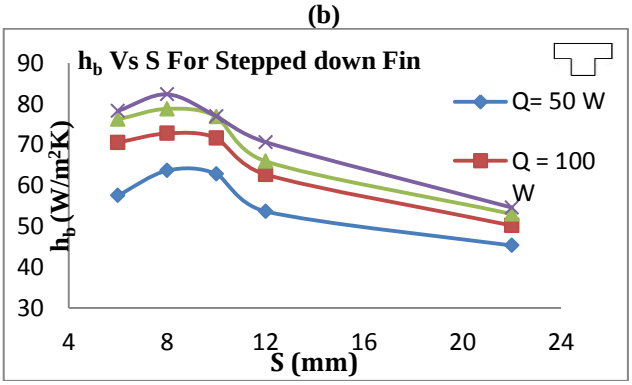


(c)

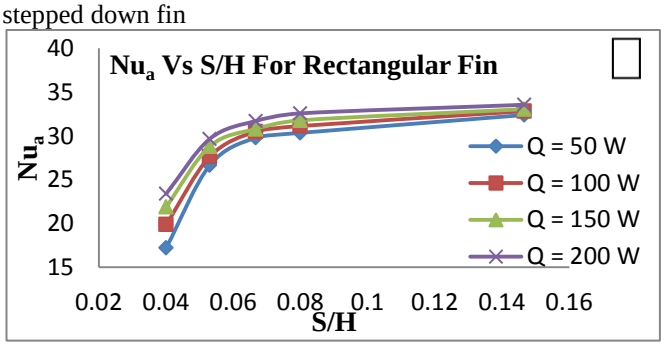
Fig.4.3 (a, b, c) Effect of fin spacing (S) on base heat transfer coefficient (h_b)

4.4 Effect of S/H on Nu_a

Fig 4.4 (a, b, c) shows the effect of Ratio of spacing to height 'S/H' on average Nusselt Number 'Nu_a' with heat input for three different geometry. As the fin spacing increases, the Nu_a increases for fin array. this is due to reason that with increase in spacing, the fluid can flow more freely through the fin channel. In the beginning, Nu_a values are very small for S/H= 0.04 (Nu_a=17.30). The highest value of Nu_a is 38.52 (for stepped down fins) at S/H=0.147. Nu_a is increased from 17.30 to 38.52, when fin spacing changes from 6mm to 22mm. Thus there is significant effect of fin spacing on average Nuselt Number. For Stepped Down fin geometry we can achieve maximum Nu_a as compared to other geometry. Increase of Nu_a with increase in fin spacing is due to decrease in effective fin surface area.

4.5 Effect of S/H on Nu_b

Fig.4.5 (a, b, c) shows the effect of ratio of spacing to fin height 'S/H' on base Nusselt Number 'Nu_b' with different heat in put for three different fin geometry. Nu_b is maximum at S/H=0.053 i.e. at S=8mm (Nub=387.22) for stepped down fin. From 0.04 to 0.053 Nu_b increases suddenly and from 0.053 to 0.15 Nu_b continuously decreases. It reaches to its maximum value and again decreases. The reason for less value of Nu_b may be due to the chocking of fluid flow at smaller spacing i.e. S=6 mm and at larger spacing due to improper contact of air with fin surface. Optimum fin spacing is decided by the highest value of Nusselt no. i.e. Nu_b is 387.22 at S=8 mm for stepped down fin



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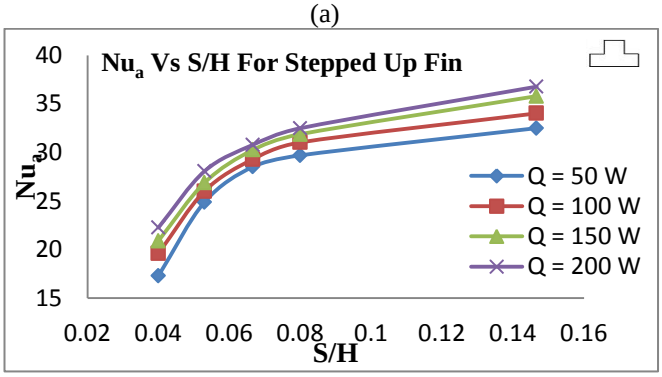

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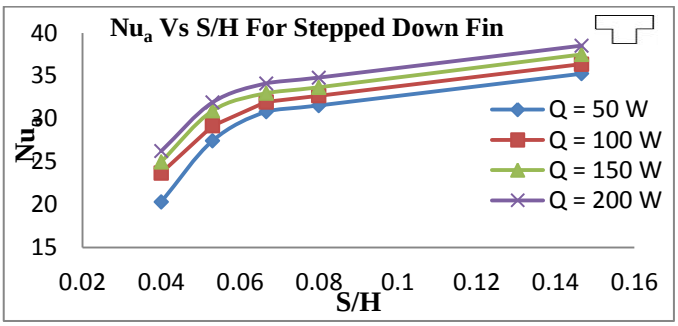

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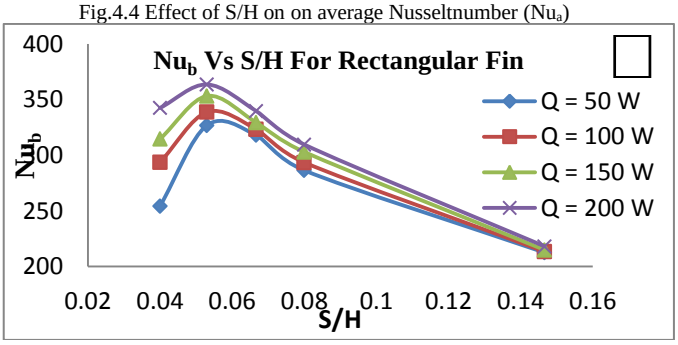
Fig.4.4 Effect of S/H on on average Nusseltnumber (Nu_a)



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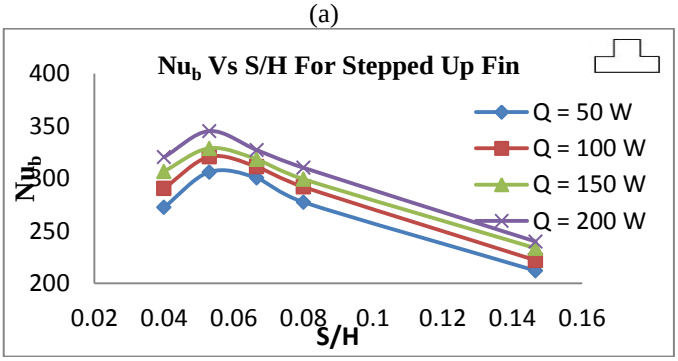

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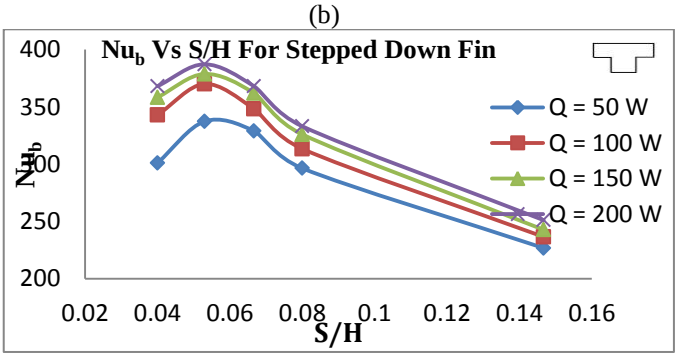

(c)

Fig.4.5 Effect of S/H on on average Nusseltnumber (Nu_b)

4.6 Uncertainity Analysis

All quantities measured during experiment are subjected tocertain uncertainties due to error in measurements. Themethod suggested by Kline and McClintock [1953]has beenadopted to carry out the error analysis. Errors in temperaturedifference are ±0.50C. This translates to an error in the heat transfer coefficient of ±4%.The uncertainty is calculated forall values and found out to be in the range of ±5%.

4.7 Flow Visualization

In the present research, the flow visualization study is conducted by simple smoke technique using dhoop stick.

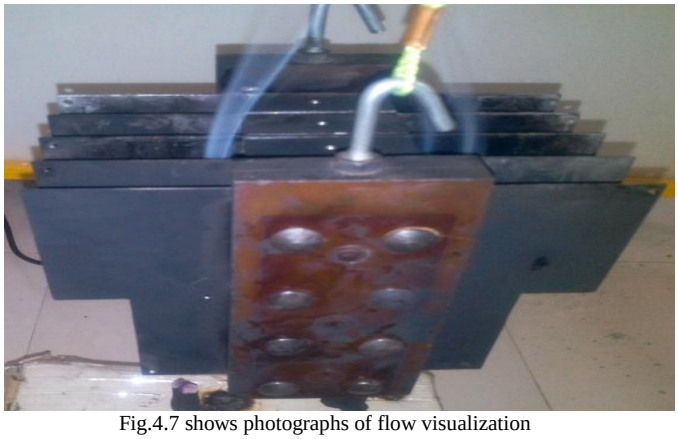

Fig.4.7 shows photographs of flow visualization

Figure 4.7 shows photographs of flow visualization for S=6-22mm for heater input of 200 W. It is clear from photographs that in case of higher spacing (s), air reaches to the centre of fin and carries maximum heat from vertical stepped down fins. However in case of lower spacing, air does not reach towards the centre of fin channel but leaves from the end of the stepped up fin.

V.CONCLUSION

The following conclusions may be drawn from the present investigation.

Experimental investigation of vertical stepped fin arrays is carried out under natural convection condition. Experimental results under natural convection are in good agreement with the previous experimental data.

1) Maximum heat transfer coefficient exist at S= 8mm in natural convection mode.
2) Figure 4.1 shows experimental results also a good match with the previous experimental work reported on natural convection.
3) From experimental results it is observed that the heat transfer coefficient h_a is very small in case of spacing 6mm (3.31 W/m^2K) whereas h_a in case of 22mm (8.37W/m^2K). The percentage increase of h_a are 39.54% from 6mm to 22mm.
4) From experimental results it is observed that maximum value of Nu_a is 38.52 for 22mm spacing and maximum value of Nu_b is 387.22 for 8mm spacing. Nu_a is increased by 44.91% from 6 to 22mm spacing whereas Nu_b is increased by 54.72% from 6 to 8mm spacing,this is due to reason that with increase in fin spacing, the fluid can flow more freely through the fin channel.
5) Base heat transfer coefficient values increases with optimum spacing and again decreases.
6) Experimental study has been carried out and compared with literature. The results obtained are matched well and showed similar trend and satisfactory agreement for heat transfer under natural convection. The optimum spacing is 8 mm under natural convection also agrees well with literature values of 8-10mm.

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