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Experimental Analysis of Heated Horizontal Rectangular Fin Array (HRFA) Under Forced Dominating Mode of Mixed Convection (FCDMM)

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***Abstract* - Present study represents experimental analysis of heated horizontal rectangular fin array (HRFA) under forced dominating mode of mixed convection (FCDMM). In experimental study heat transfer characteristics are investigated for fin with L/H=5; the effect of various parameters such as fin spacing, heater input and flow velocity on average heat transfer coefficient (h_a),base heat transfer coefficient (h_b) studied. The dimensionless number analysis such as Nusselt number, Reynolds number (Re), Grashoff number (Gr), Richardson number (Ri) is also presented. Different heat transfer characteristics such as h_a, h_b are determined and compared with experimental and numerical results of plain horizontal rectangular fin arrays (HRFA). In comparison it is observed that for modified fin geometry average heat transfer coefficients (h_a) increased by 17%.**

***Index Terms*-heat transfer, fin array ,mixed convection.**

INTRODUCTION

Fin arrays on horizontal and vertical surfaces are used in variety of engineering applications to dissipate heat to the surroundings. Needs for buoyancy driven ventilation appear in a variety of engineering applications. However, with the fins the fluid flow rate is reduced. Hence, if not properly designed it is possible that no improvement is achieved in terms of overall heat transfer. Therefore, only if the fins are properly designed, they are very attractive for the applications since they offer an economical, trouble-free solution to the problem. A designer must minimize the thermal resistance between the source of heat dissipation and the thermal sink which is essential in controlling maximum operating temperatures and consequently the long term reliability and performance of electronic components. The relationship is given in an increase in either the heat transfer coefficient or the surface area for heat transfer results in a reduction in the film resistance. The heat dissipation from fins under natural convection condition depends on the geometry and orientation of finned surface. Different types of fins ware used to increase the heat transfer rate. Fins are used on plane surfaces or cylindrical surfaces. Fins may be of having different cross sections. Depending on cross section we may have rectangular, parabolic or triangular fins. The heat can be removed effectively if the fluid flow and the resulting flow pattern are capable of removing the heat efficiently. Situations where both pressure forces and buoyant forces interact. How much each form of convection contributes to the heat transfer is largely determined by the flow, temperature, geometry, and orientation. The ratio Gr/Re^2 known as Richardson number (Ri), which gives a qualitative indication of the influence of buoyancy on forced convection. Mixed convection is fundamentally significant heat transfer mechanism that is present in many industrial and technological applications. In cases where Ri is close to 0.1 or less than 0.1 a condition occurs where a forced convection dominates in mixed convection which is a case of Forced Convection Dominating Mode Of Mixed Convection. (FCDMM). In such conditions effects of pure forced convections are yet to obtain.

LITERATURE REVIEW

Sane et al [1] investigated the problem of horizontal fin array for the single chimney flow pattern; and down and up flow pattern. They solved the governing equations neglecting the velocity component normal to the fin flats in the case of single chimney.

Acharya et al [3] presented some experimental results on the mixed convection heat transfer with longitudinal fins in a horizontal parallel plate channel. The effect on N_u number was most pronounced for smaller fin heights. Therefore, extended surfaces were used to enhance mixed convection heat transfer.

Incropera et al. [4] mixed convection flow and heat transfer in the entry region of a horizontal rectangular duct was investigated to resolve some of the conflicting trends concerning laminar mixed convection in a horizontal rectangular duct. Shete [8] carried out experimental and numerical study on combined convection heat transfer from vertical rectangular fin arrays. Results are generated in the form of variation in h_a, h_b, Nu_a and Nu_b.

H. N. Deshpande, et al [9] carried out experimental study for perforated horizontal rectangular fin arrays for Ri number less than 0.1.It is found that increasing trend of h_a with increase in fin spacing is sharp up to spacing that about 6 and 8 mm after which h_a is gradually decreases for all velocities and all heater input.

Taji.S.G. et al. [11] experimentally and numerically investigated the horizontal rectangular fin array by varying fin spacing and keeping length, height same for different heater inputs and different flow velocities in assisting mode of mixed convection as possibility of performance enhancement.

EXPERIMENTATION

An experimental setup is designed and developed to carry out the experimental investigation on horizontal rectangular fin array.

The length and height of fin flats used is kept constant(L=200,H=40). The insulating siporex block was used to reduce the leakage of heat from bottom and sides of the fin array. However the side and bottom heat losses were measured and accounted for conduction losses.

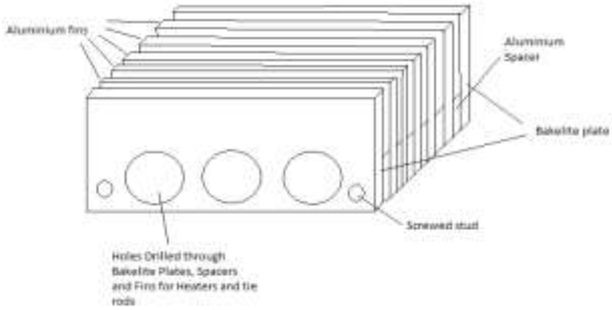


Fig 1- schematic diagrams of fin array assembled

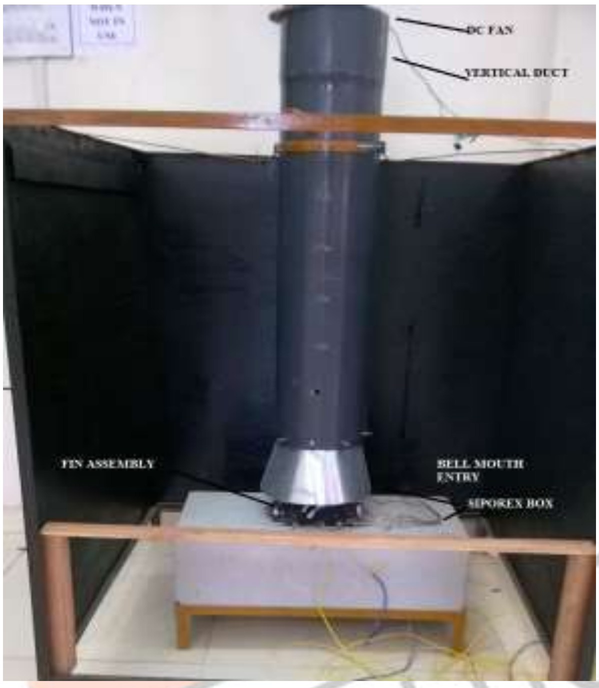


Fig 2- Experimental set up

The six arrays with fin spacing 2 to 12 mm are tested for plate fin configuration. A detail of fin array assembly is as given in Table 2.. Plate fins of height 70 mm, spacers of height 30 mm and plates of thickness 2 mm is used in the assembly (Fig.1). Fin arrays are formed by assembling plate fins and separate spacer pieces and tied together by tie bolts of 6mm diameter and of length 125mm .It is ensured to keep minimum air gap between the plate fins fin and spacers by proper tightening of the nuts. All fin flats are cut to the same size simultaneously. All surfaces are checked for flatness and are carefully cleaned and polished. Fin flats and spacers are clamped together while drilling holes for placing the cartridge heaters which are used for heating the sink. The desired air velocity through the test section is obtained by controlling the fan speed of DC fan. When measuring velocity the duct diameter should be measured. To determine average velocity. The rate of heat input for all three heaters are measured with voltmeter and ammeter. Product of voltage (V) & current (I) gave value of heater input. Temperatures at different locations are measured with calibrated Copper Constantan thermocouples. Eight thermocouples, mounted at appropriate locations for base and fin for realistic temperature measurement of the fin surface and one thermocouple for ambient temperature is used for HRFA. Six thermocouples are used for measurement of temperature of siporex and Bakelite at predefined locations which are used for heat loss calculation. Thermocouples T1 to T8 are placed on surface of the fin flat, T9 and T10 are located on the base of the fin channel. Thermocouples T13, T14, and T15, T16 are placed in the Siporex block for measuring conduction loss through side and bottom of the fin Data logger is used to indicate and store the data of temperature measurement every minute. This record helps in deciding attainment of steady state. Observations are recorded in the observation table when the steady state is reached. Readings are taken at least thrice for the same configuration and heater input to ensure the validity and repeatability of readings.

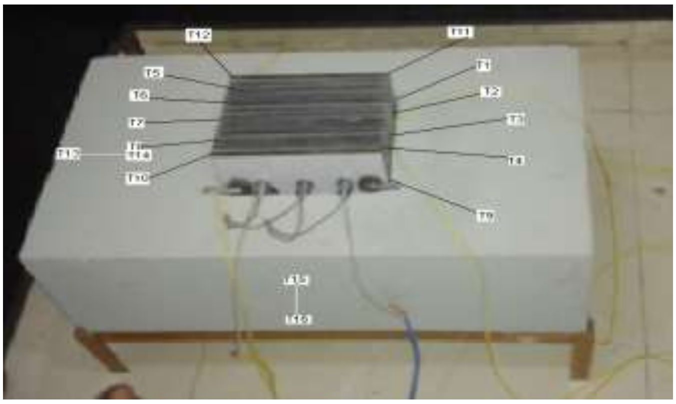


Fig 3.-Picture of mounting of thermocouples positions

Table 1 Details of parameters for experimentation

Set no	Fin Spacing [S] (mm)	No. of fins [N]	No. of channels	Air Velocity [V] (m/s)	Heater Input [q] (W)
1	2	25	24	0.85	25
2	4	17	16	0.95	50
3	6	13	12	1.05	75
4	8	11	10	1.15	
5	10	9	8	1.25	
6	12	8	7		

RESULTS & DISCUSSIONS

A. Effect of Fin Spacing On h_a

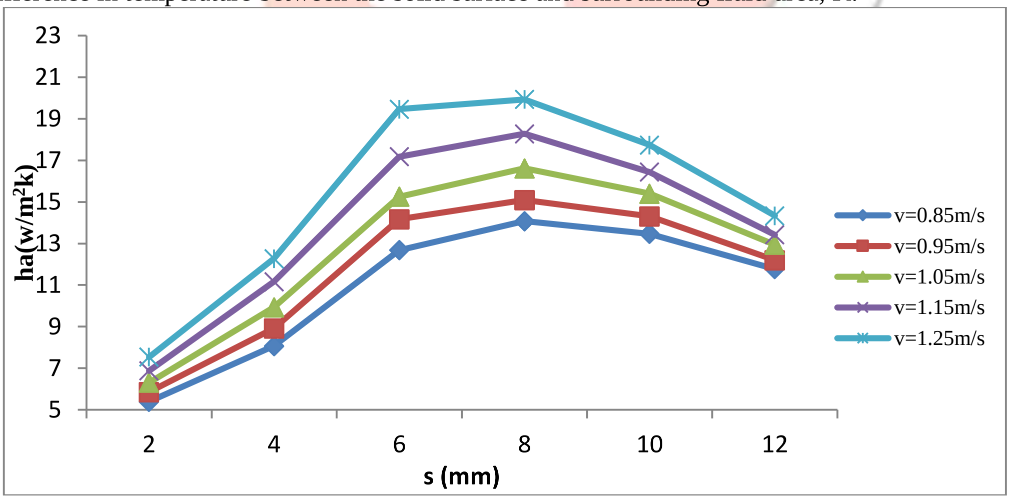
Heat transfer coefficient or film coefficient, in thermodynamics and in mechanics is the proportionality constant between the heat flux and the thermodynamic driving force for the flow of heat (i.e., the temperature difference, ΔT). h_a is average heat transfer coefficient which can be calculated by

$$h_a = \frac{Q_{conv}}{A_e \times \Delta T}$$

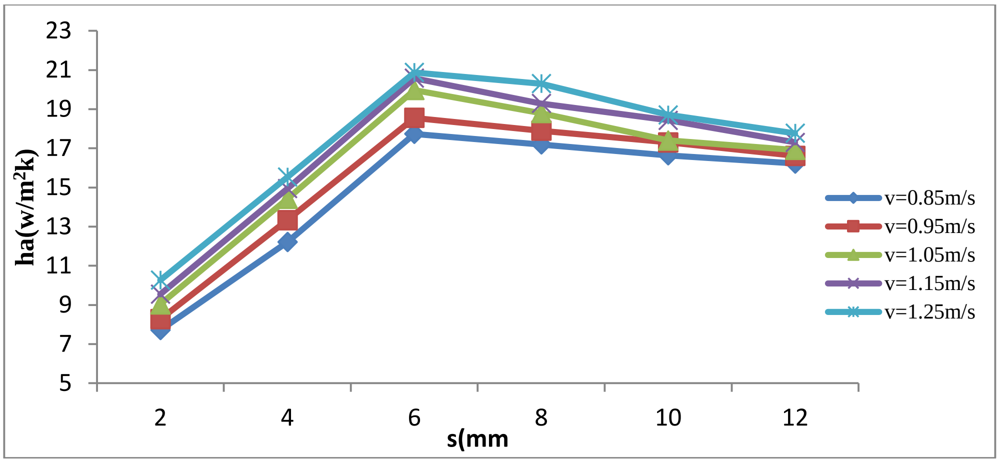
where Q_{conv} = rate of heat loss due to convection . W

A_e = Total exposed area m^2

ΔT= difference in temperature between the solid surface and surrounding fluid area, K.



a) Variation of h_a with fin spacing 'S' for 25w input



b) Variation of h_a with fin spacing 'S' for 50 input

Fig. 4 -Variation of h_a with fin spacing 'S'

Figure 4 Shows the effect of fin spacing on h_a with 25W heater input. As the fin spacing increases the h_a increases. For 25W heater input the highest value of h_a is around 20 W/m^2 K at the spacing of 8 mm. The increasing trend is steep up to spacing about 8 mm. after which there is a gradual decrease. Similarly for 50W heater input h_a increases with fin spacing S and attains maximum value 20.8 w/m^2 for spacing 6 mm

B. Effect of Fin Spacing On h_b

Figure 5 shows the effect of fin spacing on h_b.. for different heater input. h_b. is base heat transfer coefficient which can be calculated

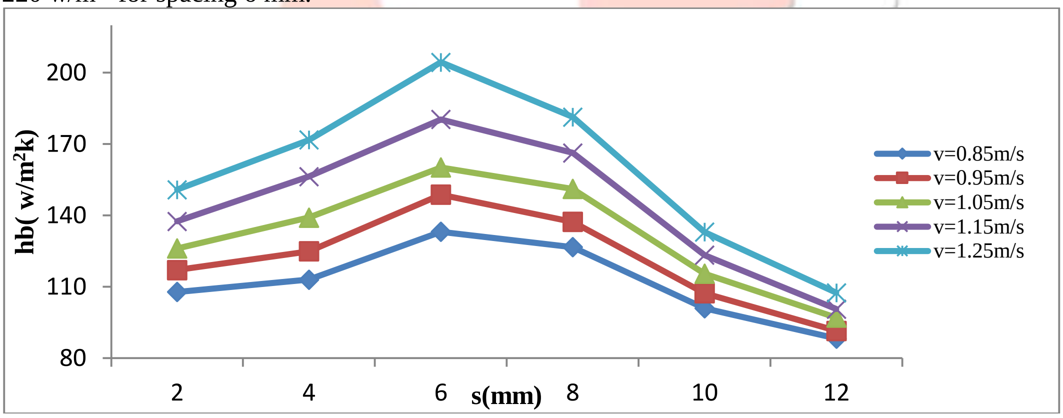
$$h_b = \frac{Q_{conv}}{A_b \times \Delta T}$$

where Q $_{conv}$ = rate of heat loss due to convection . W

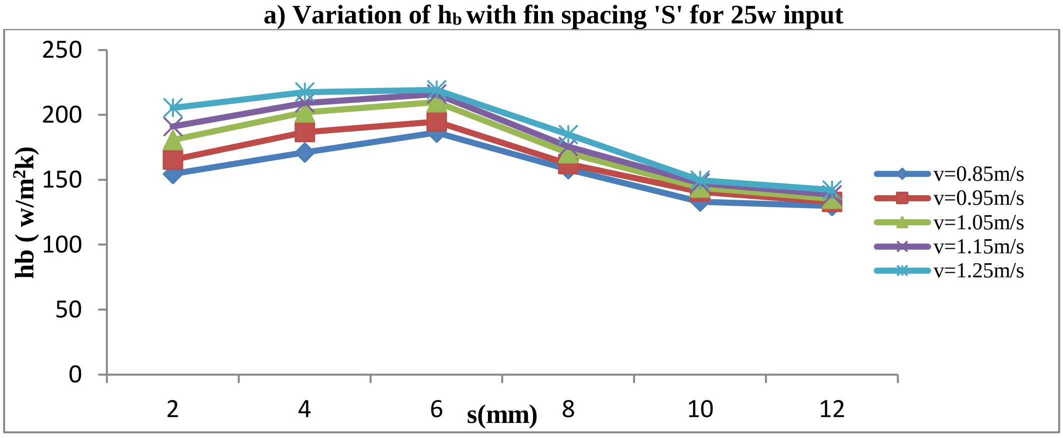
A_b = base area m^2

ΔT= difference in temperature between the solid surface and surrounding fluid area, K

Figure 5 Shows the effect of fin spacing on h_b with 25W heater input. As the fin spacing increases the h_b increases. For 25W heater input the highest value of h_b is around 200 W/m^2 K at the spacing of 6 mm. The increasing trend is steep up to spacing about 6 mm. after which there is a sharp decrease. Similarly for 50W heater input h_b increases with fin spacing S and attains maximum value 220 w/m^2 for spacing 6 mm.



a) Variation of h_b with fin spacing 'S' for 25w input



b) Variation of h_b with fin spacing 'S' for 50w input

Fig. 5 -Variation of h_b with fin spacing 'S'

C. Effect of Fin Spacing On Nu_a

Nusselt number (Nu) is a dimensionless parameter indicative of the ratio of heat transfer by convection to conduction across a fluid layer. It is a coefficient indicating the rate at which geometry transfer heat. Average Nusselt number (Nu_a) is the mean of Local Nusselt numbers over the entire heat transfer surface. Therefore, its value depends on the geometry or surface of heat transfer. The type of the flow regime or region is not a constraint for the use of average Nusselt number as heat transfer coefficient. It can be used to estimate the heat transfer at the entrance region as well as in a fully developed region. Fully developed Nusselt number is only for the fully developed region. It is an average Nusselt number estimated when the flow is fully developed. The Nusselt number is a constant throughout the region, but its value depends on the surface thermal condition

Average nusselt no can be calculated by

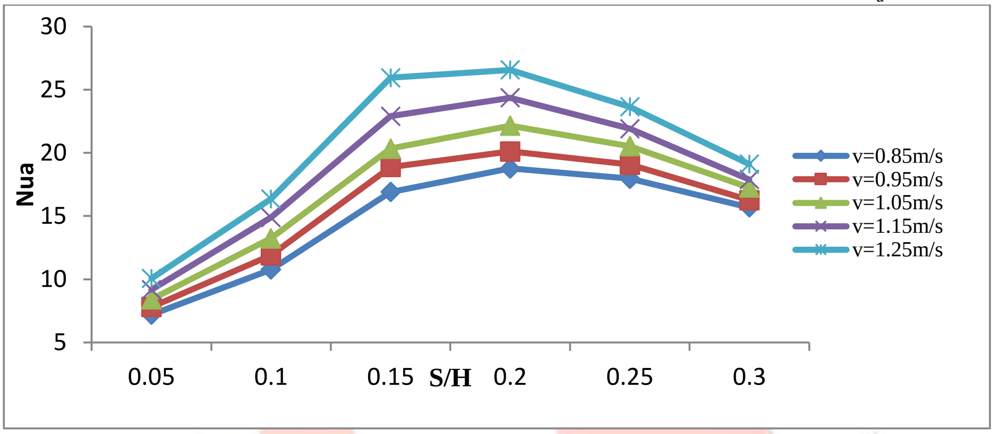
$$Nu_a = \frac{h_a \times H}{k}$$

where h_a = average heat transfer coefficient W/m^2

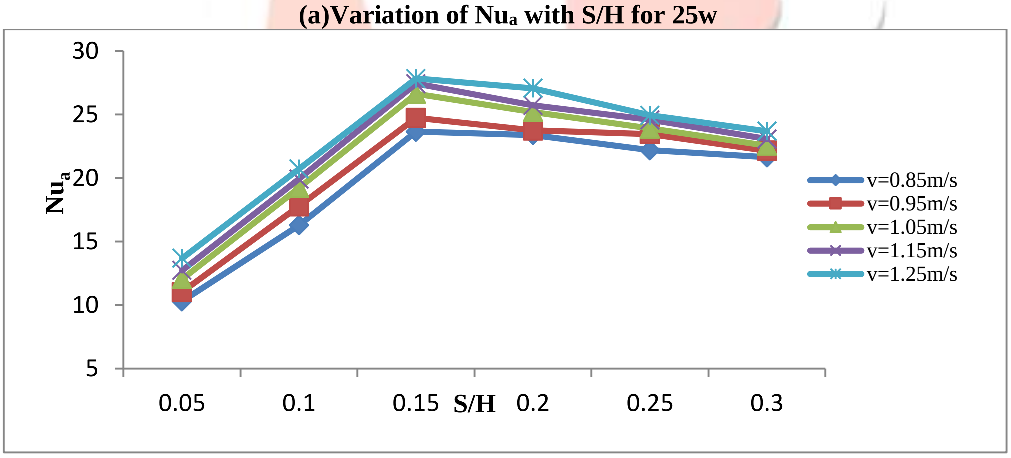
H = characteristic length m

k = Thermal conductivity at mean film temperature W/m K

It is observed from Fig 6 shows that with increase in the S/H ratio, Nu_a increases up to maximum value at S/H (S= 6 mm and H = 40 mm) = 0.15 for heater inputs 25W and 50W then Nu_a decreases with increase in S/H up to 0.3. The increasing trend of Nu_a is sharp up to S/H that about 0.15 and 0.2 for all velocities and heater inputs after which Nu_a is gradually decreased. It is also cleared from the graph that for same S/H with increase in flow velocity from 0.95 to 1.25 m/s the Nu_a also increases.



(a)Variation of Nu_a with S/H for 25w

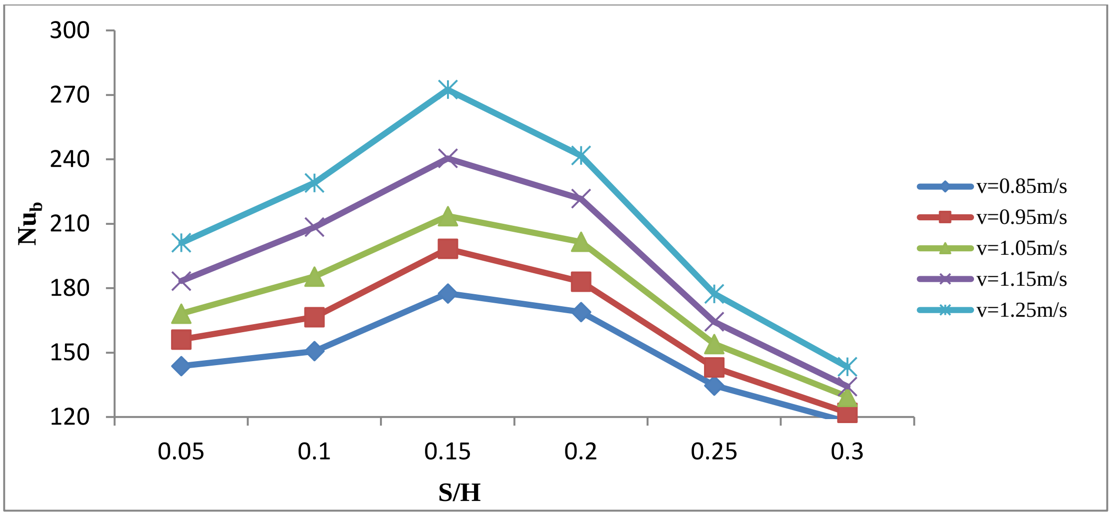


(b) Variation of Nu_a with S/H for 50w

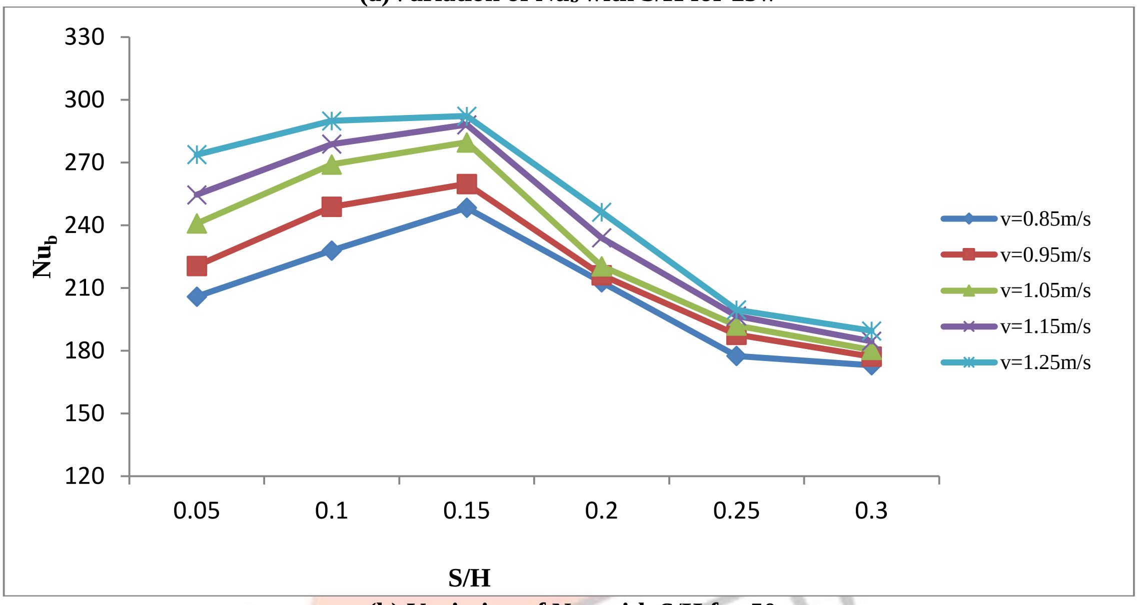
Fig 6-Variation of Nu_a with S/H

D. Effect of Fin Spacing On Nu_b

Figure 7 shows variation of base Nusselt number with fin spacing. It is clear that as the value of Nu_b increases as fin spacing increases. It reaches to its maximum value and again decreases. The reason for decrement in Nu_b may be due to the chocking of fluid flow at smaller spacing. Optimum fin spacing is decided by the highest value of base Nusselt number. It is observed that the optimum fin spacing for the two arrays is in a band of 6 to 8 mm. The Nu_b (Fig) increases up to maximum value of S/H = 0.15 and then with S/H up to 0.30 Nu_b decreases drastically. The increasing trend of Nu_b is sharp up to S/H that about 0.15 and 0.2 after which Nu_b is gradually decreases for all velocities and heater inputs. It is also cleared from the graph that for same S/H as flow velocity increases from 0.95 to 1.25 m/s Nu_b also increases.



(a)Variation of Nu_b with S/H for 25w



(b) Variation of Nu_b with S/H for 50w

Fig 7 -Variation of Nu_b with S/H

E Validation for Forced Dominating Mode of Mixed Convection (FCDMM)

For a given fluid, it is observed that the parameter Gr/Re2 represents the importance of natural convection relative to forced convection. This is not surprising since the convection heat transfer coefficient is a strong function of the Reynolds number Re in forced convection and the Grashof number Gr in natural convection. A plot of the non dimensionalised heat transfer coefficient for combined natural and forced convection on a vertical plate is given in Fig. 8 for different fin spacing corresponding plots are obtained for 50 W heater input.[10]

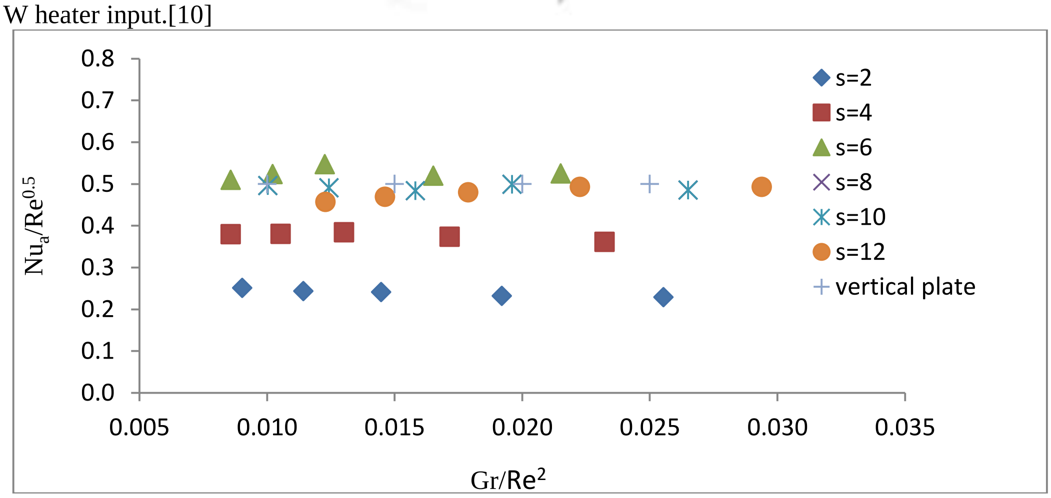


Fig 8- Validation with vertical plate under mixed convection.

CONCLUSION

The important findings of the experimentation are as follows:

1. Forced dominating mode of mixed convection can be validated by governing parameters like Gr/Re2,Nu/$Re^{0.5}$
2. For 50W heater input ha increases with fin spacing S and attains maximum value 20.8 w/m^2 for spacing 6 mm .for heater input 75 W h_a increases with fin spacing attains maximum value 22.97 w/$m^2$for S=8mm.The increasing trend of h_a is sharp up to spacing that about 6 and 8 mm after which h_a is gradually decreases for all velocities and all heater input. It can be summarized that for forced dominating mode of mixed convection optimum spacing is reduced to 6-8 mm as compared to natural convection.
3. As the fin spacing increases the h_b increases. For 25W heater input the highest value of h_b is around 200 W/m^2 K at the spacing of 6 mm. The increasing trend is steep up to spacing about 6 mm. after which there is a sharp decrease for all heater inputs.
4. With increase in the S/H ratio, Nu_a increases up to maximum value at S/H (S= 6 mm and H = 40 mm) = 0.15 for heater inputs 25W and 50W then Nu_a decreases with increase in S/H up to 0.3. The increasing trend of Nu_a is sharp up to S/H that about 0.15 and 0.2 for all velocities and heater inputs after which Nu_a is gradually decreased.
5. The average heat transfer coefficient has increased significantly at cost of very small energy input to the DC fan under forced dominating mode of mixed convection.

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