Experimental Study of Heat Transfer from Horizontal Rectangular Fins with Perforations under Natural Convection

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Abstract—**In this paper experimental study of heat transfer from horizontal rectangular fins with perforations under natural convection is presented. The parameters varied for the experiment are heater inputs (Q= 25-100 W), even fin spacing (S = 2-14 mm) 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 average Nusselt number (Nu_a), base Nusselt number (Nu_b), and Rayleigh number (Ra). The natural convection results are compared with results of plain Horizontal Rectangular Fin Array (HRFA) from previous researcher study. It is observed that there is a significant effect of variation in fin spacing on average and base heat transfer coefficients. For natural convection study the optimum fin spacing is found to be 10 mm. For perforated fins h_a is enhanced by about 15% as compared to plain horizontal fins.**

Keywords— Perforated Horizontal Rectangular Fins, Natural Convection, Heat Transfer

I. INTRODUCTION

Thermal management of electronic components is a current issue which is increasingly gaining importance in line with the advances in technology. The consistency of the electronics of a system is a major factor in the overall reliability of the system. The tininess of electronic systems has resulted in a remarkable increase in the amount of heat generated per unit volume. The high rates of heat generation result in high operating temperatures for electronic equipment, which compromises its safety and reliability. The failure rate of electronic equipment increases exponentially with temperature. Heat removal in an efficient way is necessary in order to maintain reliable operation of electronic devices. The main purpose of improving the thermal control of electronic systems is to increase cooling capacity. Extended surfaces or fins are in common use in the electronic industry and serve to enhance the thermal capability of natural convective cooling with air.

Starner K.E. et.al [1] experimentally investigated rectangular fin arrays for horizontal and vertical orientation under natural convection firstly. They had used four fin arrays sets positioned with base vertical, at 45° and horizontal to determine average heat transfer coefficients. **Harahap F et.al [2]** investigated the horizontal fin arrays to obtain relevant dimensionless parameters, the governing equations of continuity, momentum and energy were examined on the basis of similarity and a correlation was proposed. **Jones C.D. et.al. [3]** experimentally determined average heat transfer coefficients for horizontal arrays over a wide range of spacing. A simplified correlation for average heat transfer coefficient for all the arrays tested was suggested. The following important findings reported as the heat transfer rate is maximum for an optimum fin spacing exists at the given fin height, length and ΔT and the average heat transfer coefficient increases with spacing and asymptotically approaches the flat plate value for large spacing. **Mannan K.D. [4]** determined the effect of all related geometrical parameters of the fin array on its performance. The wide range of length: 127 mm < L < 508 mm. Height: 25.4 mm < H < 101.6 mm and spacing: 4.8 mm < S < 28.6 mm, with ΔT varying from 39° C to 156° C. Some important inferences of authors study are: the fin spacing is most important geometrical parameter and from a given height, length and temperature difference, an optimum spacing exists for which the heat transfer rate is maximum. The fin length is another important geometrical parameter. Short fin arrays perform better than long arrays, due to prevailing single chimney flow pattern up to L/H < 5. **Sane N.K. et.al [5]** solved the governing equations neglecting the velocity component normal to the fin flats in the case of single chimney flow problem and employing vorticity – stream function formulation. They concluded that beyond certain values of S/Hon lower side, the single chimney flow pattern ceases to exist due to choking effect on the entering side flow. They proved that the point of maximum heat transfer lies at the transition from single chimney flow pattern to sliding chimney and the corresponding S/H becomes the optimum spacing. **Dayan.et.al. [6]** studied a horizontal rectangular hot fin array under natural convection analytically, numerically and experimentally. Optimum analyses were conducted by author to determine the minimum fin height that provides the necessary cooling capability of a specified array base area. It is proved that the optimal fin spacing varies within a narrow range which depends primarily on the array length. **Sane S.S. et.al [7]**.investigated the horizontal rectangular fin array under natural convection by providing notch at the centre and suggested the selection of optimum notch dimensions, and spacing by analyzing variety of fin arrangements for performance improvement. It is observed that total heat flux as well as the heat transfer coefficient increase as the notch depth increases. This analysis reveals that the recommended single chimney flow pattern is maintained for the notched fin arrays. The performance of notched fin arrays is 30 to 50% superior than corresponding unnotched arrays, in terms of average heat transfer coefficient. **Suyawanshi S.D. et.al [8]** experimentally investigated normal and inverted notched fin arrays **(INFAs**) for variation in fin spacing, and %of area removed in form of inverted notch are the parameters. It is concluded that the average heat transfer coefficient for INFAs is nearly 30 to 40 % higher as compared to normal array. The value of ha increases with spacing giving an optimum value at about S=6mm. When single chimney flow pattern is present, in mid channel stagnant bottom portion becomes ineffective. Modified array is designed in inverted notched form and that has proved to be successful retaining single chimney together with removal of ineffective fin flat portion. **Pawar A.L. et.al [9]** experimentally investigated the perforated horizontal fin array under free convection. The authors concluded that the values of h_a are 15–20% higher for PFAs giving better performance, for smaller spacing, increment in ha is small due to the flow constriction effect . The spacing giving an optimum value at about S=6 mm. Single chimney flow pattern is retained in PFAAs also with a wider chimney zone. The modified array is designed in perforated form and that has proved to be successful retaining single chimney together with the removal of ineffective fin flat portion.

II. EXPERIMENTATION

For experimentation seven arrays with even fin spacing 2 to 14 mm are tested. Horizontal Rectangular fins plate of length 205 mm, height 70 mm, spacers of height 30 mm and thickness 2 mm with perforations in triangular form at the fin centre are used in the assembly as shown in Fig 1. Fin arrays of 100mm are formed by assembling fins and separate spacer pieces by tie bolts. The minimum air gap between the fins and spacers is ensured by proper tightening of the nuts. The fin array sides are insulated with Bakelite and base is housed in siporex insulating block (1200 mm × 240 mm × 220 mm) in order to reduce heat loss from bottom and sides. Fin array configuration and mounting is shown in Fig 2. For temperature measurement seventeen copper constantan thermocouples are mounted at proper locations. The direct metal to metal contact between thermocouples and fin surface is ensured. Thermocouples are cut from same spool of wire and the error in the temperature is adjusted. Eight thermocouples are used to measure fin surface temperature, two are used to measure base fin temperatures, four thermocouples are also provided in siporex block temperature measurement and two thermocouples are Bakelite temperature measurement.

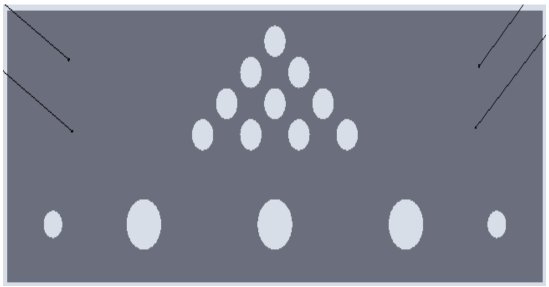

Fig.1. Details of Fin Flat used for Experimental Analysis

Fig.2. Photograph of Fin Array Used for Experimental Analysis

Fig.3. Photograph of Digital Wattmeter and Digital Temperature Indicator

Digital Wattmeter is used to measure heater input of three cartridge heaters. The digital wattmeter is calibrated and direct value in Watt is displayed by it.

2.1 Heat Loss Calculations

The measurement of convective heat transfer is very vital and depends on estimation of average heat transfer coefficient. The evaluation of average heat transfer is accurate if the losses calculated are perfect. The heat transfer from array is mainly by convection in addition to various other losses. The assembly is placed in siporex block to reduce loss of heat by conduction. An attempt was made to measure these losses mainly conduction loss from bottom and sides of horizontal rectangular fin array. The end fins have bakelite strips to reduce end loss. The radiation losses from array and bakelite end plates are also measure. The estimation of conduction loss from bottom and side, radiation loss form bakelite ends and fin array. As mentioned in above points the set up has provisions for measurements of temperature at bottom siporex and side siporex with respect to fin array for conduction loss estimation. For calculation of radiation losses temperatures of end bakellite plate and fin surfaces are noted. The conduction and radiation losses are calculated as above method.

III. RESULTS AND DISCUSSION

The experimental investigation carried out for horizontal rectangular fin array with perforations in triangular form under natural convection. The objective of present chapter is to investigate possible heat transfer characteristics. The results obtained from the observations for both experiments are presented here in the form of graphs. Results obtained are compared with previous investigators. The fin spacing, heater input and air flow velocity are the key parameters of an experimental investigation.

The results obtained for different fin spacing (S = 2, 4, 6, 8, 10, 12 and 14mm), for heater input (Q = 25, 50, 75, 100 W) under natural convection. The heater inputs are selected so as to achieve actual working range temperature of the application for which the array can be used.

The heat is supplied to fins by cartridge heater; majority of heat transfer is by convection (Q_{conv}). Heat losses are also considered as per section 2.1 to determine Q_{conv} hence heat transfer coefficient more precisely. The heat loss through base and side of the siporex block by conduction is noted as Q_{cond} and heat loss for fin array and bakelite ends by radiation is noted as Q_{rad}. The 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 average Nusselt number (Nu_a), base Nusselt number (Nu_b), Rayleigh number (Ra) For determination dimensionless parameters fluid properties such as thermal conductivity, kinematic viscosity of air and volumetric thermal expansion coefficient are evaluated at mean film temperature T_{mf}.

3.1 Effect of Fin Spacing (S) on Average Heat Transfer Coefficient (h_a)

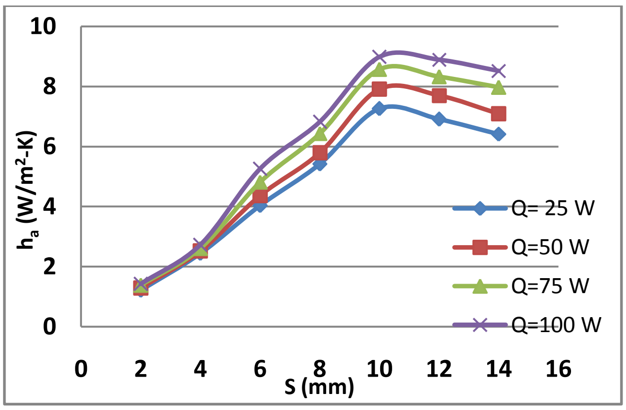

Fig.4. Effect of Fin Spacing (S) on Average Heat Transfer Coefficient (h_a)

Average heat transfer coefficient (h_a) is important parameter in the convection heat transfer study. It can be determined as dividing convection heat transfer (Q_{conv}) by product of exposed fin area and temperature difference between average fin surfaces, ambient. The exposed area deceases with increase in fin spacing.

Fig 4 shows the effect of fin spacing 'S' variation on average convective heat transfer h_a with various heater input. As the fin spacing increases, the average heat transfer coefficient (h_a) increases for the fin array up to maximum value at S = 10 mm then and then (h_a) decreases with the increase in fin spacing (S = 12 and 14 mm). At starting, h_a values are very small for S = 2 and 4 mm 1.43 to 2.72 W/m^{2}K at 100W. The highest value of h_a is 8.98 W/m^{2}K at 100W and 10 mm fin spacing. The increasing trend of h_a is steep up to spacing that about 10 mm after which h_a is gradually decreases to 8.52 W/m^{2}K for S=14 mm and 100 W. This trend is same for all heater inputs.

For smaller fin spacing flow between fin spacing get blocked hence less fin surface area is effective in convective heat transfer hence small value of h_a. Whereas, for larger fin spacing, the fluid through the fin channel is flowing more freely without fin array interference so value of h_a increases. But this rise in h_a with increase in fin spacing is up to particular spacing beyond that air moves out without contacting fin surface so temperature increases hence h_a decreases. Thus, there is a significant effect of variation in fin spacing on average heat transfer coefficient.

3.2 Effect of Fin Spacing (S) on Base Heat Transfer Coefficient (h_b)

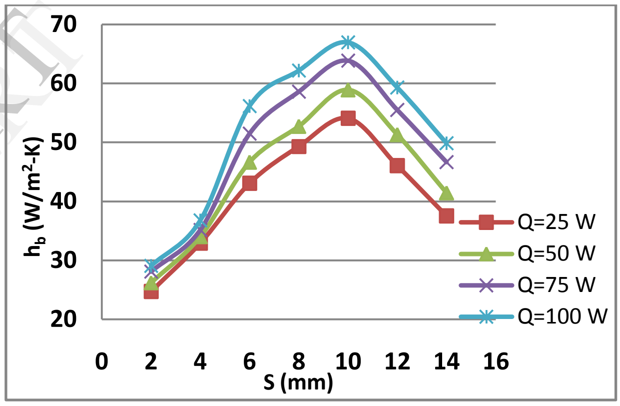

Fig.5. Effect of Fin Spacing (S) on Base Heat Transfer Coefficient (h_b)

Base heat transfer coefficient (h_b) is also the significant parameter in study of the convection heat transfer. It can be determined as dividing convection heat transfer Q_{conv} by product of base area for heat transfer and temperature difference between average fin surfaces, ambient. The base area for heat transfer remains constant with increase in fin spacing.

Fig 5 shows the effect of variation in fin spacing 'S' on base convective heat transfer (h_b) with various heater input. As the fin spacing increases, the base heat transfer coefficient h_b increases for the fin array up to maximum value at S = 10 mm then and then h_b decreases sharply with the increase in fin spacing (S = 12 and 14 mm). The h_b values for S = 2 and 4 mm varies from 29.05 to 36.74 W/m^{2}K for 100W. The highest value of h_b is 66.92 W/m^{2}K at 100W and 10 mm fin spacing. The increasing trend is steep up to spacing that about 10 mm after which there is sharply decreases to 49.85 W/m^{2}K for S=14 mm and 100 W. This trend is same for all heater inputs.

For all fin spacing the base heat transfer area remains constant hence variation in h_b is depends only on fin surface temperature. For smaller fin spacing due to choking of flow, convective heat transfer is less hence small value of h_b. Whereas, for larger fin spacing, the fluid flow through the fin channel is flowing more freely without fin array interference therefore fin temperature start reducing therefore value of h_b increases. But this rise in h_b with increase in fin spacing is up to particular spacing beyond that air moves out without contacting fin surface so temperature increases hence sharp decrease in h_b. Thus, there is a significant effect of variation in fin spacing on base heat transfer coefficient.

3.3 Effect of (S/H) on Average Nusselt Number (Nu_a)

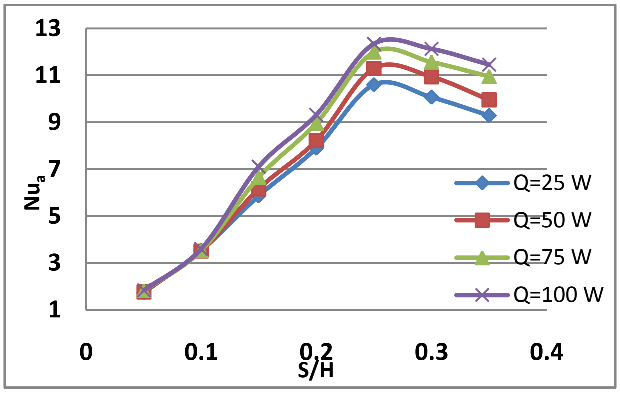

Fig.6. Effect of (S/H) on Average Nusselt Number (Nu_a)

Fig 6 shows variation of dimensionless parameter average Nusselt number (Nu_a) with(S/H).Un$_d$etermined from average heat transfer coefficient (h_a), fin height (H) and thermal conductivity (k) of air which depend on temperature. Trends obtained for Nu_a are same as that of h_a because the fin height is constant and thermal conductivity not vary drastically.

The effect of variation of ‘S/H’ on average Nusselt number (Nu_a) for various heater inputs is shown in Fig 6. As the ‘S/H’ increases, the average Nusselt number (Nu_a) increases for the fin array up to maximum value at S/H = 0.25 and then (Nu_a) decreases with the increase in fin spacing S/H = 0.3 and 0.35. At beginning, Nu_a values are very small for S/H = 0.05 and 0.1 (1.83 to 3.59 for 100W). The highest value of Nu_a is 12.35 at 100W and S/H = 0.25. The increasing trend is steep up to spacing that about 10 mm after which there is gradual decreases to 11.45 for S/H=0.35 and 100 W. This trend is same for all heater inputs.

3.4 Effect of (S/H) on Base Nusselt Number (Nu_b)

Variation of dimensionless parameter base Nusselt number (Nu_b) with (S/H) is as shown in Fig 7. The base heat transfer coefficient (h_b), fin height (H) and thermal conductivity (k) of air are the parameter from which Nu_b is determined. Nature of graph for Nu_b is same as that of h_b because the fin height is constant and thermal conductivity not varies drastically.

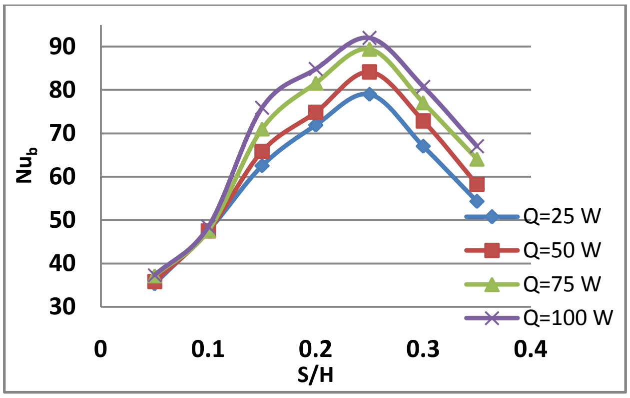

Fig.7. Effect of (S/H) on Base Nusselt Number (Nu_b)

Fig 7 shows the effect of variation of ‘S/H’ on average Nusselt number (Nu_b) with various heater input. It is observed that as the ‘S/H’ increases, the base Nusselt number (Nu_b) increases for the fin array up to maximum value at S/H = 0.25 and then (Nu_b) decreases with the increase in fin spacing (S/H = 0.3 and 0.35). At start, values of Nu_b are very small for change S/H = 0.05 and 0.1 (37.35 to 48.49) for 100W. The highest value of Nu_b is 92 at 100W and S/H = 0.25. The increasing trend is steep up to spacing that about 10 mm after which Nu_b drastically decreased to 67 for S/H=0.35 and 100 W. This trend is same for all heater inputs.

3.5 Effect of (ΔT) on Average Heat Transfer Coefficient (h_a)

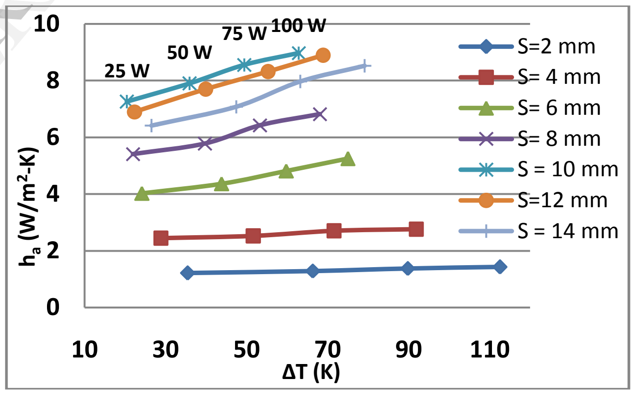

Fig.8. Variation of h_a with ΔT

Fig 8 explains effect of variation of temperature difference (ΔT) on the average heat transfer coefficient (h_a).The (ΔT) is the difference between average fin surface temperature (T_s) and surrounding fluid temperature (T_∞). For a particular fin spacing p$_a$ncreases with increase in ΔT and heater input. For lower fin spacing S= 2 and 4 mm increase in ha is small with increase in ΔT and heater input.

For 100 W and S=2 to 10 mm ΔT decreases from 112.5 K to 62.85 K and h_a increases from 1.43 W/m^2K to 8.98 W/m^2K. For 100 W and S= 12 and 14 mm ΔT increases gradually from 68.85 K to 79.15 K and hence h_a decrease from 8.8 W/m^2K to 8.5 W/m^2K. The similar effect is for all fin spacing and remaining wattages.

The value of h_a for same heater input increases because ΔT and fin exposed area are decreases with increase in fin spacing. This effect is only up to S= 10 mm and then onwards even though fin spacing increases S= 12 and 14 mm the value of h_a decreases because of increase in ΔT.

3.6 Effect of (ΔT) on Base Heat Transfer Coefficient (h_b)

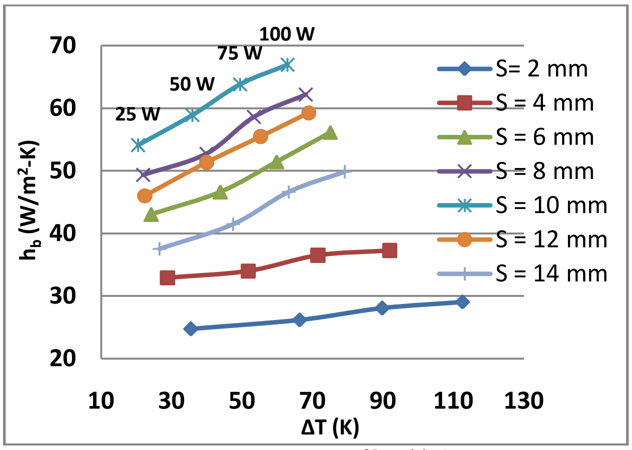

Fig.9. Variation of h_b with ΔT

Figure 9 shows effect of variation of temperature difference (ΔT) on the base heat transfer coefficient (h_b).The (ΔT) is the difference between average fin surface temperature (T_s) and surrounding fluid temperature (T_∞). For a specific fin spacing h_b increases with increase in ΔT and heater input.

For 100 W and S=2 to 10 mm ΔT decreases from 112.5 K to 62.85 K and h_a increases steeply from 29.05 W/m^{2}K to 66.95 W/m^{2}K. For 100 W and S= 12 and 14 mm ΔT increases gradually from 68.85 K to 79.15 K and hence h_b decrease sharply from 59.26 W/m^{2}K to 49.85 W/m^{2}K. The similar effect is for all fin spacing and remaining wattages.

The value of h_b increases with in fin spacing as ΔT decreases as base fin area is constant. This effect is only up to S= 10 mm and then onwards even though fin spacing increases S= 12 and 14 mm the value of h_b decreases drastically because of increase in ΔT.

3.7 Comparison of Heat Transfer from Plain and Perforated Fins

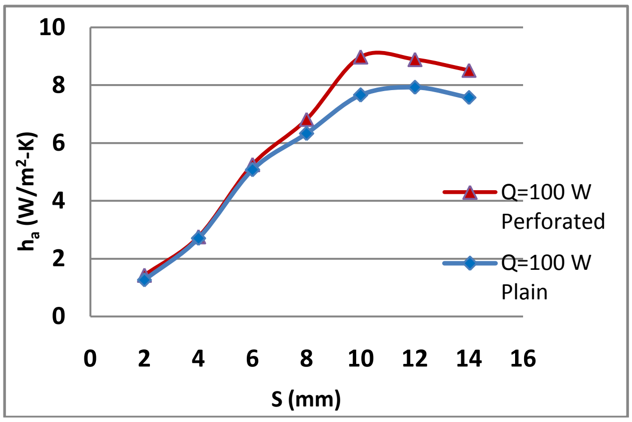

Fig.10. Comparison of present h_a with Plain HRFA

Figure 10 shows comparison between heat transfer from perforated and plain horizontal rectangular fins. For plain horizontal rectangular fins previous researcher Taji's **[10]** data is used. For this comparison, average heat transfer coefficient (h_a) from present data and plain horizontal rectangular fins under natural convection is used. The trend obtained for present study is same as that of plain fins. For comparative point of view only 100 W readings are used. Trends obtained for remaining wattages are also same shown in Figure 10. The graph also shows that maximum value of h_a for perforated fins is at S = 10mm and for plain fins at S = 12mm. It is observed that for perforated fins h_a is enhanced by about 15%. This shows that perforations crated at the centre of fins helps to improve heat transfer. From the comparison, it is cleared that the results for present study are satisfactory.

3.8 Correlation for Natural Convection

The experimental data obtained for natural convection is used for correlating the results in dimensionless form.

In natural convection heat transfer is due to the buoyancy force. It is governed by the dimensionless Grashof number, which represents the ratio of the buoyancy force to the viscous force acting on the fluid. Nu_a is dimensionless convective heat transfer coefficient. Natural convection heat transfer from a fin surface depends also on the fin geometry of the surface as well as fin spacing. It also depends on the variation of temperature on the surface and the thermo physical properties of the fluid involved. From Nu vs S/H plot, as the 'S/H' increases, the average Nusselt number (Nu_a) increases for the fin array up to maximum value at S/H = 0.25 and then (Nu_a) decreases with the increase in fin spacing S/H = 0.3 and 0.35. That means the S/H is also important parameter governing heat transfer from horizontal fins under natural convection.

Therefore at last an attempt is made to correlate the experimental results obtained in the present natural convection study. Such a correlation is quite helpful from the point of view of the designers. The average Nusselt number Nu_a is correlated with the other relevant governing parameters of the natural convection i.e. Ra_H and S/H.

Following correlation is obtained by a least square fit applied to a set of data based on the experimental results.

$$\mathbf{Nu_a = 16.29 \times Ra_H^{\,0.072} \times (S/H)^{1.01}} \qquad (1)$$

The correlation is valid for heater input Q = 25 to 100 W; S/H = 0.05 to 0.35, Pr = 0.7 and air as working fluid. The average error and RMS (Root mean Square) deviation is calculated as 3.11%, 3.88 respectively. The results obtained by the correlation are in good agreement with actual experimental results.

IV. CONCLUSIONS

Based on above experiential study following conclusions can be drawn

1. As the fin spacing increases, the average heat transfer coefficient (h_a) and base heat transfer coefficient (h_b) increases for the fin array up to maximum value at S = 10 mm and then h_a, h_b decreases with the increase in fin spacing (S = 12 and 14 mm). The highest values of h_a, h_b are 8.98 W/m^{2}K, 66.92 W/m^{2}K respectively at 100W and 10 mm fin spacing. There is a significant effect of variation in fin spacing on average and base heat transfer coefficients.

2. The optimum fin spacing is found to be 10 mm under the natural convection study.
3. Trends obtained for Nu_a and Nu_b Vs S/H are same as that of h_a and h_b Vs S respectively as expected. The highest value of Nu_a and Nu_b are 12.35, 92 respectively at 100W and S/H = 0.25.
4. It is observed that for perforated fins h_a is enhanced by about 15% as compared to plain horizontal fins. This shows that perforations crated at the centre of fins helps to improve heat transfer. From this comparison, it is cleared that the results for present study are satisfactory.
5. From comparison between present work under natural convection and previous investigator data available from literature it is observed that the results obtained from present data are satisfactory and clearly shows enhancement as compared with data from plain fins.
6. The average Nusselt number Nu_a is correlated with the other relevant governing parameters of the natural convection i.e. Gr_H and S/H. The results obtained by the correlation are in good agreement with actual experimental results.

REFERENCES

[1]. **Starner K.E. and McManus H.N.**, 'An experimental Investigation of free convection heat transfer from rectangular fin arrays', Journal of Heat Transfer, ASME, 85, pp.273-278 (1963).

[2]. **Harahap F. and McManus H.N., '**Natural convection heat transfer from horizontal rectangular fin arrays', Journal of Heat Transfer, ASME, 89, pp.32–38, (1967).

[3]. **Jones C.D. and Smith L.F.**, 'Optimum arrangement of rectangular fins on horizontal surfaces', J of Heat Transfer, ASME, 92, pp.6-10, (1970).

[4]. **Mannan K.D.**, 'An experimental investigation of rectangular fins on horizontal surfaces', Ph.D Thesis, Ohio State University, (1970).

[5]. **Sane N.K. and Sukhatme S.P.,** 'Natural convection heat transfer from horizontal rectangular fin arrays', Int Proc. of 5th Int Heat Transfer Conference, Tokyo, Japan, Vol. 3, NC3.7, pp.114–118 (1974).

[6]. **Dayan A., Kushnir R., Mittelman G. and Ullmann A.**, 'Laminar free convection underneath a downward facing hot fin array', Int J of Heat and Mass Transfer, 47, pp.2849–2860, (2004).

[7]. **Sane S.S., Paishwad G.V. and Sane N.K** 'Natural convention heat transfer enhancement in horizontal rectangular fin array with inverted notch', 18th National & 7th ISHMT-ASME Heat and mass Transfer Conference at IIT, Guwahati, 312- 317, Jan 04-06, 2006

[8]. **Suryawanshi S.D. and Sane N.K.,** 'Natural convection heat transfer from horizontal rectangular inverted notch fin arrays', ASME J. of Heat Transfer, 131, 082501, pp. 1-6 (2009).

[9]. **Pawar A.L., Taji. S.G., and Sane N.K** "An experimental investigation on perforated horizontal rectangular fin array under natural convection" 1st National Conference on innovations in mechanical engineering at SIT Lonavala (April 2012)

[10]. **Taji. S.G., Parishwad G.V., and Sane N.K** "Enhanced performance of horizontal rectangular fin array heat sink using assisting mode of mixed convection" International Journal of Heat and Mass Transfer 72C (2014), pp. 250-259

[11]. **Nag P.**K. 'Heat and Mass Transfer', Tata McGraw Hill Education Private Limited, 2nd Edition, (2007).

[12]. **Holman J.P. and Souvik B.,** 'Heat Transfer', Tata McGraw Hill Education Private Limited, 10th edition. New Delhi, pp. 348-350, (2011)

[13]. **Yunus A.C. and Afshin J.G.**, 'Heat and Mass Transfer', Tata McGraw Hill Education Private Limited,
4th Edition, (2011)