Experimental Study of Heat Transfer from Perforated Horizontal Rectangular Fins for Richardson Number Less than 0.1

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ABSTRACT

The heat transfer from perforated horizontal rectangular fins under natural as well as force convection dominating mode of mixed convection (Richardson Number < 0.1) is experimentally investigated. The fin length, fin height and width of fin array are maintained constant. The experiments are carried out with heater inputs 25, 50, 75 W, even fin spacing 2 to 14 mm, and air flow velocity 0.95 to 1.25 m/s. It is observed that the variation in fin spacing and flow velocity is having considerable effect on heat transfer from fins. Very small amount of fan energy input is required to raise the average heat transfer coefficient by about 40%. In this study it is observed that the maximum value of average heat transfer coefficient enhanced for same fin spacing and its also at lower fin spacing as compared to natural convection study. The present experimental results are correlated by using dimensionless parameters.

Keywords— Force convection Dominating Mixed Convection, Perforated Horizontal Rectangular Fins.

I. INTRODUCTION

The electronic equipment breakdown increases exponentially with higher operating temperature. The operating temperature of electronic system increases because of high rates of heat generation due to compact arrangement of components. This affects the reliable working of the electronic equipment. To absorb or dissipate thermal energy, so as to maintain the device temperature below, the maximum allowable temperature given by the device manufacturer, the heat sinks are used. The extended surface or fins are used as heat sinks eg. Horizontal rectangular plain fins with natural or force convection heat transfer mode. The researchers investigated the rectangular fins with various performance improvement geometries such as notch at centre of fins [1, 2], perforation on fin flat [3] in horizontal arrangement. The inactive zone which does not contribute much in heat dissipation is created at the central bottom portion of fin array channel. The notch or inverted notch at the central bottom portion of fin can be considered as heat transfer augmentation method.

The convection heat transfer coefficient is a strong function of the Reynolds number (Re) in forced convection and the Grashoff number (Gr) in natural convection. The parameter Gr/Re² is known as Richardson number (Ri), represents the importance of natural convection relative to forced convection. When Richardson number (Ri) is less than the 0.1 it is the case of force convection dominating mode of mixed convection. In this experimentation the velocity range (0.95 to 1.25 m/s) is selected in such a way that Ri is less than the 0.1 therefore it is a case of force convection dominating mode of mixed convection.

Sane S.S. et al. [1], Suyawanshi S.D. et al. [2] investigated that the performance of notched, inverted notched horizontal rectangular fin arrays under natural convection are about 30 to 45% superior to the corresponding without notched arrays, in terms of average heat transfer coefficient. Pawar A.L. et al. [3] examined that for the perforated horizontal fin array under natural convection the values of average heat transfer coefficient are about 30% to 45% superior to the corresponding without notched arrays, in terms of average heat transfer coefficient. Pawar A.L. et al. [3] examined that for the perforated horizontal fin array under natural convection the values of average heat transfer coefficient are about 30% to 45% superior to the corresponding without notched arrays, in terms of average heat transfer coefficient. Optimization of the geometry in stacks of parallel plates that generate heat and to find the optimal spacing for volumes with maximal heat transfer density to mixed convection was the research work of Ochende T. B. et al. [5]. As per Dogan M et al. [4] the mixed convection heat transfer in the natural convection dominated mode depends on the fin height, spacing and initially with increase in fin spacing the average convection heat transfer coefficient increases and takes maximum value after which to decrease. Shete J.P. [6] experimentally and numerically studied isothermal rectangular vertical fin...
arrays losing heat under mixed convection with the variation in parameters fin height, fin spacing, velocity of the air and heater input. Taji.S.G. et al. [7-8] 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. Study of flow patterns and validation of the same were also the key parameters.

II. EXPERIMENTATION

As per literature available until now the heat transfer from perforated horizontal rectangular fins under force convection dominating mode of mixed convection flow velocity is not investigated yet. Therefore the experimental investigations are carry out on horizontal rectangular fins with perforations in triangular pattern under natural as well as forced dominating mixed convection by keeping the fin length and height constant.

The seven fin arrays with even spacing from $s = 2$ to $14$ mm are selected for the experimentation (details as per Table 1). The horizontal rectangular fins with $L/H = 5$ ($L = 200$ mm, $H = 40$ mm, thickness $t = 2$ mm) are used in the assembly. In order maintained equal fin spacing plates of height = $30$ mm used as spacers.

As the long horizontal fins are used ($L/H = 5$) consequently air might not reach at the central section of the fins. This acts as ineffective central surface of fins which is removed by creating the triangular pattern ten perforations with $8$ mm diameter as shown in Fig.1. The flow pattern and heat transfer characteristics investigated as per previous researcher Taji S.G. et.al [8] is used to finalize these geometrical parameters of perforations.

![Dimensions of Horizontal Rectangular Fin with perforations in triangular form](image1.png)

![Dimensions of spacers used in Fin array Assembly](image2.png)

The fin arrays are produced by assembling fins and spacer by tie bolts, nuts and its ends are insulated with Bakelite plates. In order to reduce heat loss from bottom and sides the fin array base is housed in siporex insulating block.

In order to achieve forced convection dominating mode of mixed convection under assisting mode over the fin arrays the experimental set-up mainly consists of a stand mounted vertical duct with flow straighteners and DC fan. To obtain the suction effect at the test section DC fan is mounted at the top of vertical duct which generates the desired air flow velocity.

- The calibrated digital ammeter and voltmeter with accuracy $\pm 0.01$ mA and $\pm 0.1$ V respectively is used to measure power consumed by DC fan. The fan speed is a vital parameter therefore for steady voltage, the voltage stabilizer is used. In order to get $Ri < 0.1$ it is necessary to change air velocity from $0.95$ to $1.25$ m/s. The portable hot wire anemometer with an accuracy of $\pm 0.015$ m/s is used to measure the flow velocity. The heater input of three cartridge heaters provided for heating of fins is measured by using digital wattmeter with least count of $0.1$ W, accuracy $\pm 0.5$%.

**TABLE 1**
Details of Fin Array Configuration Tested For Forced Dominating Mixed Convection

<table>
<thead>
<tr>
<th>Set No.</th>
<th>Fin Spacing [S] (mm)</th>
<th>No. of fins</th>
<th>Air Velocity [V] (m/s)</th>
<th>Heater Input [q] (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>25</td>
<td>0.95</td>
<td>25</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>17</td>
<td>1.05</td>
<td>50</td>
</tr>
<tr>
<td>3</td>
<td>6</td>
<td>13</td>
<td>1.15</td>
<td>75</td>
</tr>
<tr>
<td>4</td>
<td>8</td>
<td>11</td>
<td>1.25</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>10</td>
<td>9</td>
<td></td>
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<tr>
<td>6</td>
<td>12</td>
<td>8</td>
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<tr>
<td>7</td>
<td>14</td>
<td>7</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Seventeen copper constantan thermocouples are mounted at appropriate locations for temperature measurement. In between thermocouples, fin surface is maintained. The eight thermocouples are used for fin surface temperature measurement, four thermocouples are provided in siporex block temperature measurement. The base fin, Bakelite plate temperatures measurement is carried out by using four thermocouples. The surrounding temperature is measured is one thermocouple.

The majority of heat transfer is by convection. An attempt is made to measure losses mainly conduction loss from bottom, sides of horizontal rectangular fin array and the radiation losses from fin array, bakelite end plates. The heat transfer coefficient is more precise as heat losses are considered.

III. RESULTS AND DISCUSSIONS

The results obtained for different fin spacing (even s = 2 to 14 mm) under natural convection as well as force dominating mixed convection heater input (Q = 25, 50, 75W). The air flow velocity is taken as 0.95 to 1.25 m/s.

III A Effect of Fin Spacing on Heat Transfer Coefficient

In the convection heat transfer study important parameters are average heat transfer coefficient ($h_a$) and base heat transfer coefficient ($h_b$)

The effect of variation fin spacing ($s$) and flow velocity ($v$) on $h_a$ and $h_b$ for specific heater input ($Q = 50, 75$ W) is shown in Fig 3, 4 respectively. The results of forced dominating convection are also compared with results of natural convection.

It is observed from Fig 3 (a), (b) that for a certain velocity as the fin spacing increases, the $h_a$ increases up to maximum value at fin spacing of 6 mm, then decreases with increase in fin spacing up to 14 mm. 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. In case of natural convection ($V = 0$ m/s) maximum values of $h_a$ are of order of 6.8-8.5 W/m²K for $Q = 25$ to 75 W respectively at fin spacing of 10 mm and for forced dominating mixed convection maximum values of $h_a$ are of order of 21.8 – 25.8 W/m²K for $Q = 25$ to 75 W respectively at $v = 1.25$ m/s and $s = 6$ mm.

As the fin spacing ($S$) increases from Fig 4 (a), (b), the base heat transfer coefficient ($h_b$) increases up to maximum value at $s = 6$ mm and then $h_b$ decreases with increase in fin spacing up to $s = 14$ mm. For all velocities and heater inputs the increasing trend of $h_b$ is sharp up to spacing 6 mm after which $h_b$ is sharply decreases. It is also cleared from the graph that $h_b$ also increases for same fin spacing as flow velocity increases from 0.95 to 1.25 m/s. In case of natural convection ($v = 0$ m/s) maximum values of $h_b$ are of order of 53.6-63.2 W/m²K for $Q = 25$ to 75 W respectively at $S = 10$ mm and for forced dominating mixed convection maximum values of $h_b$ are in order of 247.3 – 266.4 W/m²K for $Q = 25$ to 75 W respectively at $V = 1.25$ m/s and $S = 6$ mm. Thus for forced dominating mixed convection maximum value of $h_b$ increased about...
76% and is also at lower fin spacing as compared to natural convection.

![Figure 4 a, b: Effect of fin spacing (S) on base heat transfer coefficient (h_b) for Q= 50, 75 W](image)

**III B Effect of (∆T) on Heat Transfer Coefficient**

The effect of deviation of temperature difference (∆T) on the h_a and h_b is described in Fig.5. The difference between average fin surface temperature (T_s) and surrounding fluid temperature (T_∞) defined is as the ∆T. By varying heater inputs, fin spacing and velocity of air the effect of ∆T on heat transfer coefficient is given. To study the variation of h_a and h_b by changing ∆T for various heater inputs and fin spacing considered for velocity V= 1.25 m/s.

With increase in ∆T and heater input for same fin spacing the h_a increases. From Fig.5 (a) for v = 1.25 m/s, 75 W and s =2 to 6 mm ∆T decreases from 16.0 to 14.03 K and h_a increases from 11.31 to 24.21 W/m²K. In case of heater input Q = 75 W and s = 8 to 14 mm ∆T increases gradually from 15.57 to 27.09 K and hence h_a decreases from 25.1 to 22 W/m²K. If heater input is same for fin spacing s = 6 mm onwards as the fin spacing increases the value h_a decreases because of increase in ∆T.

![Figure 5 a, b: Effect of (∆T) on Heat Transfer Coefficient for v=1.25 m/s](image)

**III C Effect of (S/H) on Nusselt Numbers**

The variation of dimensionless parameter average Nusselt number (Nua) and base Nusselt number (Nub) with ratio of fin spacing to fin height (S/H) are shown in Fig 10 and 11 respectively. Nu_a and Nu_b are determined from h_a, h_b, fin height (H) and thermal conductivity (k) of air which depend on temperature. The obtained results are represented in the graphs. Each graph shows effect of variation of S/H and flow velocity (V) on Nu_a for specific heater input (Q = 25, 100 W).
It is observed from Fig 6 that with increase in the S/H ratio, \( N_{ua} \) increases up to maximum value at \( s/H = 0.15 \) then \( N_{ua} \) decreases with increase in S/H up to 0.35. The increasing trend of \( N_{ua} \) is sharp up to S/H that about 0.15 and 0.2 for all velocities and heater inputs after which \( N_{ua} \) 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 \( N_{ua} \) also increases. For natural convection (\( v = 0 \) m/s) maximum values of \( N_{ua} \) are of order 10.6 -12 for Q= 25 to 75W respectively at S/H=0.25 and in case of forced dominating mixed convection maximum values of \( N_{ua} \) are in range 31-34.4 for Q= 25 to 75W respectively at \( V = 1.45 \) m/s and S/H = 0.15.

The \( N_{ub} \) (Fig 7) increases up to maximum value of S/H = 0.15 and then with increase in S/H up to 0.35 \( N_{ub} \) decreases drastically. The increasing trend of \( N_{ub} \) is sharp up to S/H that about 0.15 and 0.2 after which \( N_{ub} \) 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 \( N_{ub} \) also increases. For natural convection (\( v = 0 \) m/s) maximum values of \( N_{ub} \) are 79-89.4 for Q = 25 to 75 W respectively at S/H = 0.25 and for forced dominating mixed convection maximum values of \( N_{ub} \) are in order 327-350 for Q = 25 to 75 W respectively at \( V = 1.25 \) m/s and S/H = 0.15.

Therefore it is cleared that for forced dominating mixed convection maximum values of \( N_{ua} \), \( N_{ub} \) improved.

IV. PROPOSED CORRELATION FOR \( Ri < 1 \)

The dimensionless parameter Reynolds number (Re) is representation of flow velocity in force dominating mode of mixed convection (\( Ri < 1 \)) heat transfer. The buoyancy force is also a component in heat transfer which is very less. In the correlation buoyancy forces in terms of Grashoff number (Gr) is considered. The fin geometry of the surface as well as fin spacing (s/H) which is significant heat transfer governing parameter is also considered in
correlation. Therefore at last an effort is made to correlate the experimental results obtained in the present force convection dominating mode of mixed convection study. Such a correlation is quite helpful from the point of view of the designers. The Nu_a is correlated with the other relevant governing parameters Re, Gr_H and S/H. The correlation is obtained by a least square fit applied to a set of data based on the experimental results:

\[ Nu_a = 0.34 \times Re^{0.45} \times Gr_H^{0.118} \times (S/H)^{0.23} \]

The correlation is valid for heater input Q = 25 to 75 W; S/H = 0.1 to 0.35, Re_H = 1400 to 2400, Gr_H = 4 \times 10^4 to 25 \times 10^4, Pr = 0.7 to 0.72, Ri < 0.1 and air as working fluid. The average error between Nu_a experimental and Nu_a correlation is 0.633%.

V. CONCLUSIONS

1. The average heat transfer coefficient (h_a) is augmented by about 40% with very small fan energy input 0.8 to 1.5 W. Therefore very small amount of energy input is required to raise heat transfer from fin array.
2. The 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.
3. It is observed that for forced dominating mode of mixed convection maximum value of h_a increased about 67% and maximum value of h_a is also at lower fin spacing as compared to natural convection.
4. The increasing trend of h_b with increase in fin spacing is sharp up to spacing that about 6 mm after which h_b is sharply decreases for all velocities and all heater inputs.
5. In forced dominating mixed convection maximum value of h_b increased about 76% and maximum value of h_b is also at lower fin spacing as compared to natural convection.
6. The optimum spacing region is 4-8 mm which is wider under force convection dominating mode of mixed convection as compared to natural convection.
7. The highest value of Nu_a and Nu_b are 34.4 and 350 respectively at 75 W and S/H = 0.15.
8. The Nu_a is correlated with the other relevant governing parameters Re, Gr_H and S/H. The results obtained by the correlation are in good contract with actual experimental results.

REFERENCES

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