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Enhancement of the heat transfer through perforated fin by natural convection

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In the present paper we have investigated the enhancement of the heat transfer using fins with circular perforation. The experiment was conducted for different voltages to obtain variation of temperature in the entire setup fabricated for this study. The area ratio factor, weight ratio factor and volume ratio factor are calculated for present setup. It has been observed that the temperature drop along the perforated fin length is consistently higher than that for the equivalent non perforated fin.

Keywords: Heat transfer, Perforation, Fins, Convection, Area ratio factor, Weight ratio factor, Volume ratio factor

1 Introduction

With advancement of technology, there is a need of better heat transfer equipments. Methods of enhancing heat transfer can be classified into two groups – active method and passive method. There is a requirement of external power source in active method while the passive method does not require any external power source[1].

Heat transfer under natural convection from a rectangular horizontal fin having square perforations is numerically studied[2]. The parameters under consideration in this study are the geometrical dimensions of perforations and their orientation. A comparison is made between the heat loss from the fin with perforations of different orientations. It is observed that there is a greater enhancement of heat loss from a fin with perforations square and parallel to the base in comparison to that of inclined square perforations.

Enhancement of heat transfer and the corresponding pressure drop in a rectangular channel having pin fins with square cross-sectional perforations have been studied[3].

The effectiveness of convective heat transfer for air flow of low speed though thin and isothermal plates with perforations having crosswind on the upstream face have been studied[4]. The aim of this study is to gather information that will help designers to optimize the spacing and hole size of the perforations.

Combined conduction-convection heat transfer and fluid flow over a 3-D array of rectangular fin with square perforations on lateral surface of fin have been studied numerically[5].

Heat dissipation under natural convection from a rectangular horizontal fin with square perforations of two orientations has been numerically studied[6]. The parameters under consideration in this study are the geometrical dimensions of perforations and the orientation of perforation. A comparison is made between heat dissipation from the fin with various orientations of the perforations. It is observed that there is a greater enhancement of heat loss from a fin with perforations square and parallel to the base in comparison to that of inclined square perforations.

The use of infrared thermography technique is done to study the thermal performance of plate fin heat sinks under confined impinging jet conditions. The parameters included in this study are: Reynolds number (Re), the height(H/L) and the width(W/L) of the fins and the impingement distance(Y/D) and how these parameters affect the thermal performance of plate-fin heat sinks have been discussed[6].

Heat dissipation under natural convection from a rectangular horizontal fin with square perforations of two orientations has been numerically studied. The parameters under consideration in this study are the geometrical dimensions of perforations and the orientation of perforation. A comparison is made between heat dissipation from the fin with various orientations of the perforations. It is observed that there is a greater enhancement of heat loss from a fin with perforations square and parallel to the base in comparison to that of inclined square perforations[7].

A theoretical study of the effect of fin spacing, fin height and length, temperature difference between fin and surroundings under natural convection heat transfer from an array of horizontal fins was done.

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Finite volume based computational fluid dynamics (CFD) code was used to solve 3-D elliptic governing equations[8].

Heat dissipation under natural convection from a rectangular horizontal fin with triangular perforations (their bases parallel and towards the fin tip) is studied. Then comparison of heat dissipation is done with an equivalent solid one. The parameters under consideration are the thermal properties of the fin and the geometrical dimensions of the fin and the perforations. The enhancement in heat dissipation and the reduction in fin weight due to perforations are considered. The results show an increase in heat dissipation from the fins with triangular perforations over a certain range of perforation dimensions and the space between the perforations in comparison to a solid fin[9].

A new type of plate-pin fin heat sink in constructed based on plate fin heat sinks, which is made of PFHS and some columnar pins grouped between plate fins[10]. A comparison of these two types of heat sinks is made based on experiments and numerical simulations.

2 Experimental Method

The experimental setup includes a heat sink supplied with heating elements and data acquisition system. The heat is generated within the heat sink by means of one heating element power of 670 W. All the experimental data are recorded by the data acquisition system. The heat sink chosen for experiments are aluminum cylinder of 60 mm diameter and 200 mm length. One hole was drilled in the cylinder in which one heating element was pressed. The power supplied by heating element was 670 W. Five aluminum straight fins were fitted radially. The fins are 100 mm long, 200 mm wide and 2 mm thick. There is one non-perforated fin and four perforated fins. These fins were divided into four groups as shown in Table1.

A variable transformer of type 2P1 with input 240 V and 50 Hz and output 0-240 V, 20 A and 7.5 kVA were used to regulate the voltage supplied to the heating elements. The experimental data were measured by a hand held, battery operated digital temperature sensor. Temperature was recorded on the surface of the test fins.

3 Mathematical Analysis

Here, the number of perforations $N_x$in the *x*-direction (L) and N_y in the *y*- direction (W) are assumed. The perforation cross sectional area (A_c) is assumed and then the dimension of any perforation is calculated. The surface area of the uniform longitudinal rectangular perforated fin can be expressed as follows[1]:

$$A_{fp} = A_{ps} + A_t + N_c A_{pc} \quad \dots (1)$$

$$A_{fp} = (2W * L - 2N_c * A_c) + (W_c * A_{pc}) \quad \dots (2)$$

$$A_{fp} = A_f + N_c (A_{pc} - 2A_c) \quad \dots (3)$$

Where,

A_{fp} – surface area of perforated fin
A_{ps} – surface area of perforated surface
A_t – surface area of fin thickness
N_c– number of perforations on fin
A_{pc} – surface area of perimeter of perforation
A_c – perforation cross-sectional area
A_f – surface area of conventional fin

Equation (3) can be written as:

$$A_{fp} = A_f + N_x * N_y (A_{pc} - 2A_c) \quad \dots(4)$$

Now fin surface area ratio (RAF) is introduced to compare the heat transfer surface area of perforated fin (A_{fp}) to that of conventional one (A_f), and is given as:

$$RAF = \frac{A_{fp}}{A_f} \quad \dots(5)$$

$$RAF = 1 + \frac{N_x \times N_y (A_{pc} - 2A_C)}{A_f} \quad \dots(6)$$

A comparison of material volume of perforated fin and non-perforated fin is made by volume reduction ratio (RVF) as expressed by Eq. (7):

$$RVF = \frac{V_{fp}}{V_f} = \frac{(L \times W \times t - N_x \times N_y \times A_c \times t)}{L \times W \times t} \quad \dots(7)$$

$$RVF = 1 - \frac{N_x \times N_y \times A_c}{L \times W} \quad \dots (8)$$

Similarly, a perforated fin is lower in weight than a non-perforated fin. This aspect can be expressed by the fin weight reduction ratio (RWF) given by Eq. (10):

Table 1 — Number of perforations on 5 different fins

Fin	Diameter of perforation on fin (mm)	Number of perforations per fin
1	-	0
2	10	24
3	10	32
4	10	40
5	10	48

$$RWF = 1 - \frac{(N_x \times N_y \times A_c \times t \times \rho)}{L \times W \times t \times \rho} = 1 - \frac{N_x \times N_y \times A_c}{L \times W} \quad \dots (9)$$

$$RWF = \frac{W_{fp}}{W_f} = \frac{(W_f - N_x \times N_y \times A_c \times t \times \rho)}{W_f} \quad \dots(10)$$

Circular perforations are cut out from the fin body and studied. The number of perforations in longitudinal direction N_x, in the transverse direction N_y and the perforation diameter is b. The directional perforation spacing is S_x and S_y:

$$L = N_x * b + (N_x + 1)S_y \quad \dots(11)$$

$$S_x = \frac{L - N_x b}{N_x + 1} \quad \dots(12)$$

$$W + N_x b + (N_y + 1)S_y \quad \dots(13)$$

$$S_y = \frac{W - N_y b}{N_y + 1} \quad \dots(14)$$

The heat transfer surface area of the fin can be expressed as Eq. (15):

$$A_{fp} = A_f - 2N_c A_c + N_c A_p \quad \dots(15)$$

$$A_{fb} = A_f + N_c(A_p - 2A_c) = A_f + \pi N_c b\left(t - \frac{b}{2}\right) \quad \dots(16)$$

The RAF and RVF can be expressed as Eq. (17) and Eq. (18), respectively:

$$RAF = 1 + \frac{\pi * b * N_x * N_y \left(t - \frac{b}{2}\right)}{2W * L + W * t} \quad \dots(17)$$

$$RVF = 1 - \frac{N_x * N_y * \frac{\pi}{4} b^2}{L * W} \quad \dots(18)$$

An experimental correlation to estimate the convection heat transfer coefficient of array of vertical parallel flat plate is given by Eq. (19):

$$Nu = \frac{h * B}{k} = \frac{Ra}{24}\left(1 - e^{\frac{-35}{Ra}}\right)^{0.75} \quad \dots(19)$$

Where, B is the average space between adjacent fins.

$$Ra = \frac{\rho^2 * g * \beta * C_p * B^4 * \Delta T}{\mu * k * L} \quad \dots(20)$$

Several experiments concluded that the surface heat transfer coefficient of perforated surface is function of open area ratio (ROA) of the perforated surface, which is defined by Eq. (21):

$$ROA = \frac{OA}{OA_{max}} \quad \dots(21)$$

Where, OA is the actual open area=$A_c.N_c$

$$OA = A_c.N_x.N_y \quad \dots(22)$$

OA_{max} is the maximum possible perforation open area, which is defined as Eq. (23):

$$OA_{max} = A_c.N_{c,max} = A_c.N_{x,max}.N_{y,max} \quad \dots(23)$$

Where, $N_{x,max}$ and $N_{y,max}$ are the maximum possible number of the perforations along the fin. These numbers related with the perforation spacing equal zero. The perforated surface heat transfer coefficient ratio (R_h) can be expressed as Eq. (24):

$$R_h = 1 + 0.75 \frac{OA}{OA_{max}} \quad \dots(24)$$

The film heat transfer coefficient of the perforated surface (h_{ps}) is expressed as Eq. (25):

$$h_{ps} = \left(1 + 0.75 \frac{OA}{OA_{max}}\right) h \quad \dots(25)$$

4 Results and Discussions

Variations of temperature with fin length and number of perforations at given voltage as obtained is shown in Table 2-6 Different parameters for

Table 2 — Temperature values across the fin length for varying no of perforations on fin for heat input at 220V

220v		Length (mm)					
		0	20	40	60	80	100
No of perforation	0	260.3	169.7	162.4	157.6	148.3	138.6
	24	260.3	168.8	158.9	149.8	141.3	132.9
	32	260.3	168.5	154.5	145.5	137.3	127
	40	260.3	165.1	153.8	144.4	136.7	126.5
	48	260.3	163.2	151.4	142.3	135.2	124.3

Table 3 — Temperature values across the fin length for varying no of perforations on fin for heat input at 200V

200v		Length (mm)					
		0	20	30	60	80	100
No of perforation	0	221	165.3	155.7	140.4	130.6	125.9
	24	221	163.2	146.8	131.1	126.2	119.3
	32	221	162.2	144.5	127.1	119.7	110.3
	40	221	160.2	143.6	125.8	118.6	109.7
	48	221	161.4	131.9	121.8	117.5	107.4

Table 4 — Temperature values across the fin length for varying no of perforations on fin for heat input at 180V

180v		Length (mm)					
		0	20	40	60	80	100
No of perforation	0	219.5	142.3	133.5	129.4	125.1	122.6
	24	219.5	136.5	128.3	125.6	120.8	114.9
	32	219.5	132.4	128	123.9	117.2	112.8
	40	219.5	134.8	125.8	121.7	114.9	110.4
	48	219.5	134.1	123	117.5	113.9	108.9

calculating RAF, RWF and R_h are as given in Table 7, 8, 9, & variation of RAF, R_h and RWF with no of perforations is given in Table 10

This study investigated perforation shape geometry which indicates that the increase or decrease in the surface area of the perforated fin with respect to that of the non-perforated one depends on the following parameters; the fin thickness, the total number of perforation Nc, and the perforation diameter b. However, *Afp*is greater or smaller than Af depends on the fin thickness and perforation diameter. The calculation show that the heat transfer surface area of the perforated fin is a function of the fin dimensions and the perforation shape geometry. The temperature distribution of the perforated fins and the non-perforated along x-direction is plotted in Figs 1-5. As shown in figure, it is obvious that the temperatures

Table 5 — Temperature values across the fin length for varying no of perforations on fin for heat input at 160V

160v		Length (mm)					
		0	20	40	60	80	100
No of Perforation	0	206.9	145.4	136.7	125.8	117.9	110.5
	24	206.9	138.8	125.5	114.5	107.5	104.1
	32	206.9	138.6	124.3	112.5	103.8	102.4
	40	206.9	137.2	122.9	110.2	101.8	98.3
	48	206.9	138.8	119.8	109.3	100.5	95.8

Table 6 — Temperature values across the fin length for varying no of perforations on fin for heat input at 140V

140v		Length (mm)					
		0	20	40	60	80	100
No of perforation	0	187.9	141.9	135.4	127.4	116	109.4
	24	187.9	139.6	126.8	114.5	108.2	105.2
	32	187.9	138.4	124	112.6	109.5	102.5
	40	187.9	136.6	123.2	109	101.5	99.6
	48	187.9	134.5	120.9	107.9	99.8	96.3

Table 7 — Parameters for calculating RAF on various fins

	RAF						
	Π	b	Nx	Ny	t	W	L
1	3.141592654	10	0	0	2	100	200
2	3.141592654	10	4	6	2	100	200
3	3.141592654	10	4	8	2	100	200
4	3.141592654	10	4	10	2	100	200
5	3.141592654	10	4	12	2	100	200

Table 8 — Parameters to calculate RWF on various fins

	RWF				
	Nx	Ny	Ac	L	w
1	0	0	78.54	200	100
2	4	6	78.54	200	100
3	4	8	78.54	200	100
4	4	10	78.54	200	100
5	4	12	78.54	200	100

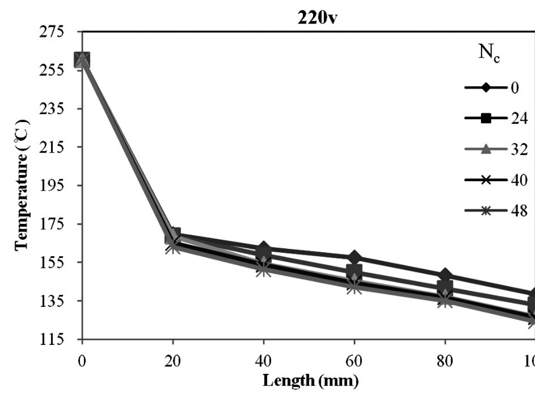

Fig. 1 — Distribution of temperature on fins when heated with 220V heat source.

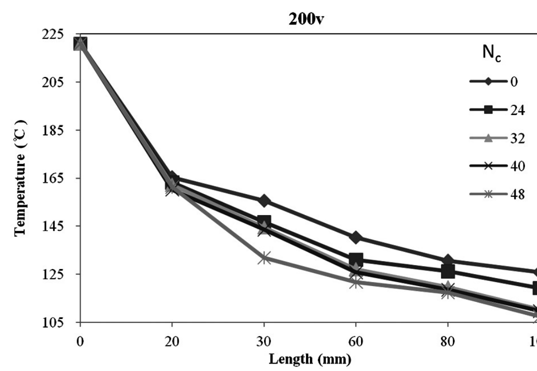

Fig. 2 — Distribution of temperature on fins when heated with 200V heat source.

Table 9 — Parameters to calculate R_h for various fins

	R_h						
	Ac	Nx	Ny	Nxmax	Nymax	OA	OAmax
1	78.54	0	0	4	12	0	3769.92
2	78.54	4	6	4	12	1884.96	3769.92
3	78.54	4	8	4	12	2513.28	3769.92
4	78.54	4	10	4	12	3141.6	3769.92
5	78.54	4	12	4	12	3769.92	3769.92

Table 10 — Variation of RAF, RWF, and R_h on various fins

	No of perforation	RAF	R_h	RWF
1	0	1	1	1
2	24	0.943733	1.375	0.905752
3	32	0.924977	1.5	0.874336
4	40	0.906221	1.625	0.84292
5	48	0.887465	1.75	0.811504

along the non-perforated fin are higher than those of the perforated one in most cases. It is also indicated the temperature drop between the fin base and tip increases as the number of perforation are increased. This is because thermal resistance of the perforated fin decreases as the perforation diameter isincreased.

It indicated that RAF is weak function of the fin length and width. This is because the effect of the fin tip area which is smaller surface compared to that of the fin surface area and can be neglected. The temperature distribution along the fin has important effect on the fin performance. Higherfin temperatures exist as the fin thermal resistance is decreased. Fig. 6 shows the relation between RAFand number of perforations. This figure shows thatRAF is smaller than unity. Heat dissipation rate of the perforated fin depends on the heat transfer coefficient and fin area. In this study, all the film heat transfer coefficient are assumed to be unity and increasing up to the upper limit 1.25 as the number of perforation increased, however, decreasing down to the lower limit of 1.1. The calculation of R_h was plotted against the number of perforation in Fig. 7. The ratio RWF is plotted as function of the number of perforation in Fig. 8. The figure shows that the weight reduction ratio of the

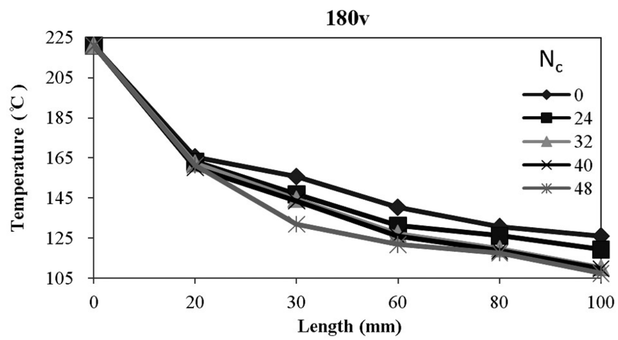

Fig. 3 — Distribution of temperature on fins when heated with 180V heat source.

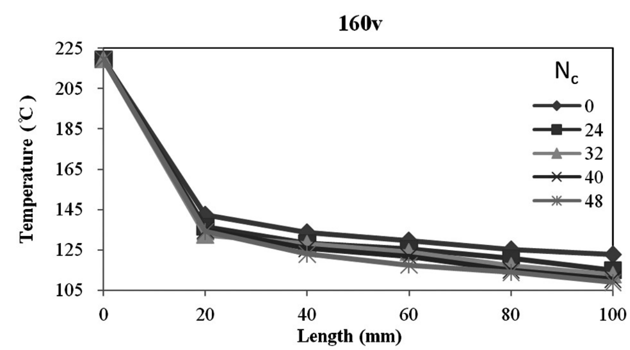

Fig. 4 — Distribution of temperature on fins when heated with 160V heat source.

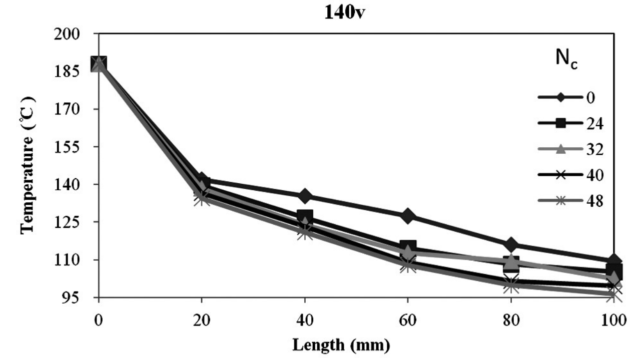

Fig. 5 — Distribution of temperature on fins when heated with 140V heat source.

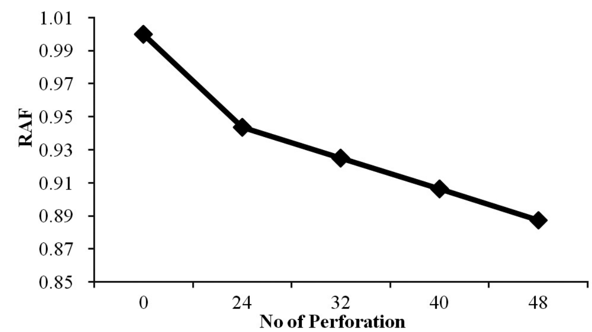

Fig. 6 — Variation of RAF with number of perforation on fins.

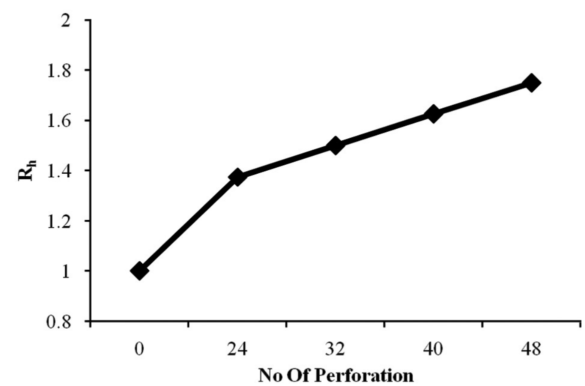

Fig. 7 — Variation of R_h with number of perforation on fins.

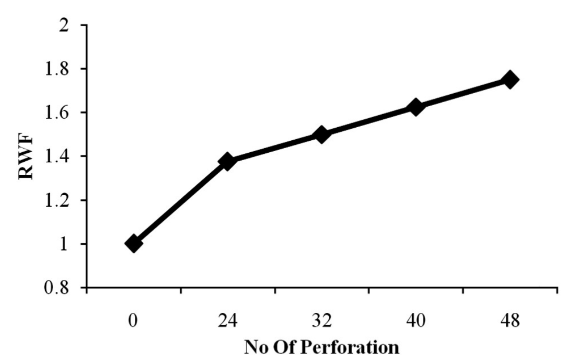

Fig. 8 — Variation of RWF with number of perforation on fins.

perforated fin continues to decrease as number of perforation isincreased.

5 Conclusions

i. The temperature drop along the perforated fin length is consistently higher than that for the equivalent non-perforated fin.
ii. It contains a larger number of perforations higher than the perforated fin that contained a small number of perforations.
iii. The gain in heat dissipation rate for the perforated fin is a strong function of the perforation dimension and lateral spacing.
iv. Decreasing the perforation dimension reduces the rate of temperature drop along the perforated fin.
v. Heat transfer coefficient for perforated fin that contained a larger number of perforations is higher than the perforated fin that contained a small number of perforations.

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