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EFFECT OF PERFORATION AREA ON TEMPERATURE DISTRIBUTION OF THE RECTANGULAR FINS UNDER NATURAL CONVECTION

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ABSTRACT

In many engineering applications extended surfaces known as fins, are used to enhance convective heat transfer. The problem of natural convection heat transfer for perforated fins was investigated in this work. An experimental study was conducted to investigate the natural convection heat transfer in a fin plate with circular perforations. The investigation is conducted to compare heat transfer rate of rectangular fins (15 fins) with a size of 100x270 mm embedded with circular vertical shaft. The patterns of perforations rectangular fins contain of 18 circular hole. The temperature distribution was examined for an array of rectangular fins. Experimental results show that the temperatures distribution along the non-perforated fin decreased from 49 to 42^{o}C but for the perforated fin with hole diameter of 20mm, temperature decreased from 67 to 47.4^{o}C with low power (200W). The temperature drop along the non-perforated fin decreased from 170 to 122^{o}C but the temperature drop for perforated fin with hole diameter 20mm decreased from 170 to 101.7^{o}C with high power (900W). Also, when the perforation diameter increased the decrement of the temperature between the base and tip of the fin was increases. Furthermore, when the perforation diameter increased, the heat transfer rate and the coefficient of heat transfer also increased.

Keywords: fin, natural convection, temperature distribution, perforation area.

INTRODUCTION

A heat sink is used to enhance heat transfer in many thermal engineering applications. With the increasing concerns on economic and energy saving consideration, many researchers attempt develop more efficient heat exchanger. Many types of extended surface are used in practice as heat transfer augmentation surface including offset, strip fins, louvered fins, perforated fins and wavy fins (Naik and Probert, 1987; Bayram and Alparslan, 2008). The heat transfer from any surface can be enhanced by increasing the surface area of heat transfer, increasing the coefficient of heat transfer between the surface and surroundings, or both. In general, an extended surfaces as fins are used to increase the surface area of heat transfer. Whenever a heat exchanger with extended surface is designed, certain equations for every type of fin are required. These equations have all been derived from the general heat conduction equation where certain simplifying assumption and boundary conditions are used in each case (Naik and Probert, 1987; Ameel *et al*., 2013).

Walunj at al., 2014, preformed review study on the analysis of natural convection heat transfer through rectangular fins. Various experimental studies were carried out to investigate effect of fin height, fin spacing, fin length and fin thickness over convective heat transfer. An experimental study was carried out by Fujii, (2007), and it was found that the thermal boundary layer was interrupted by fins. Al-Essa *et al*., (2009) were studied the enhancement of natural convection heat transfer from a rectangular fin embedded with equilateral triangular perforations. The results for perforated fin were compared with its equivalent solid one. It was found that the augmentation in heat dissipation and a reduction in weight with used perforations over that of the equivalent solid one.

Al-Doori (2011) carried out an experimental study to enhance the heat transfer of fins containing circular perforations. The study reported that the temperature drop along the non-perforated fin length was consistently lower than perforated fins. The heat dissipation rate enhancement from the perforated fin was significantly affected by the lateral spacing and size of perforation. Dorignac *et al*. (2005) performed an experimental study to determine convective heat transfer on a multi perforated plate. For a range of perforations spacing they developed an empirical relation for the wide range of a perforated flat plate. Sahin and Demir, (2008) experimentally studied overall heat transfer and the effect of various design parameters on heat transfer and friction factor for a heat exchanger with square cross sectional perforated pin fins in a rectangular channel. The aim of this study was to investigate and analyse the heat transfer performance of uniform rectangular fins used for cooling of electronic devices in natural convection environment with perforations. This study focuses on investigating the effect of increasing the perforations diameter on the distribution of temperature, the heat transfer rate and the coefficient of heat transfer. Meanwhile, show effect of increases the perforation area on the distribution of temperature and the coefficient of heat transfer.

EXPERIMENTAL DETAILS

The experimental test facility was specially designed and fabricated for this purpose. Figure 1 shows the view of the experimental test rig. The experimental setup consists of heat sink provided by four heating

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elements with heating capacity of 670 W and data acquisition system. The heating elements generated the heat inside the heat sink. The heat sink chosen for experiments are aluminium cylinder shaft of 100 mm diameter and 270 mm length as shown in Figure-2. Four holes were drilled in the cylinder in which four heating elements were pressed. The power supplied by heating element was 2680 W. Fifteen aluminium straight fins were fitted radially. The fins were 100 mm long, 270 mm wide and 2 mm thick. These fins were divided into five groups and each group consists of three similar pate as shown in Table-1.

Table-1. Geometry of various types of fin array studied.

Case	Fin length (mm)	Fin height (mm)	Thickness (mm)	Number of perforation per fin	Perforated fins diameter (mm)
1	100	270	2	0	-
2	100	270	2	18	8
3	100	270	2	18	12
4	100	270	2	18	16
5	100	270	2	18	20

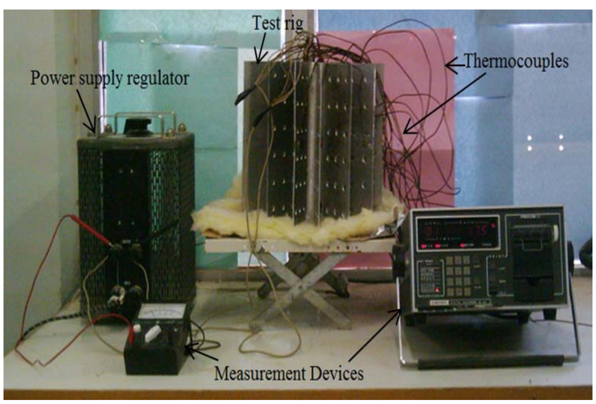

Figure-1. Experimental setup.

A variable transformer of type 50B with an input of 220 V and 50-60 Hz and output of 0-240 V, 20 A and 7.5 kVA was used to regulate the voltage supplied to the heating elements as shown in Figure-2. The temperatures were measured by twenty seven K type thermocouples at different locations. The heat supplied from the heat elements ranges during the experiments from 200W to 900W. To measure base temperature of the fins one thermocouple was fixed on the aluminum shaft. In addition, the air temperature was measured by another thermocouple. Twenty five of thermocouples were divided into five fins groups equally. Each thermocouple was fixed to the surface of the test fin at equal space (20 mm) locations along the fin length. The apparatus was allowed to run for approximatly 120 minutes, until the steady state was achieved. The recording of temperature was started after the steady state had been achived.

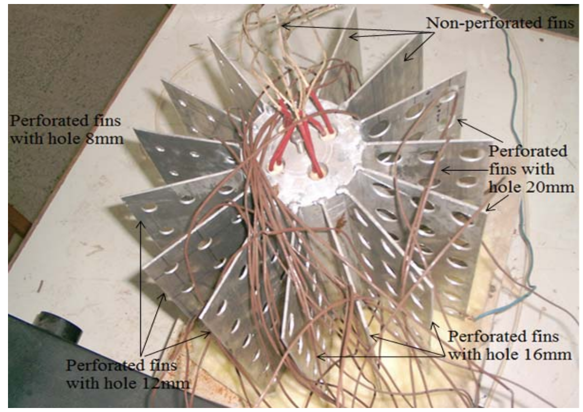

Figure-2. The design of heat sink.

MATHEMATICAL ANALYSIS OF PERFORATED FIN

The study considered the comparison of heat dissipation from the perforated fin and the solid one. From view point of view of thermal engineering, thermal systems must be designed and sized to generate, transmit, or dissipate the appropriate amount of heat with the required demand. The optimum operation condition of thermal system depends on the application including cooling and heating of certain component. In this study, the number of perforations was N_x in the *x*- direction (*L*), N_y in the *y*- direction (*W*) and the perforation cross sectional area (A_c) were assumed. For the uniform longitudinal rectangular perforated fin, the surface area can be expressed as follows:

$$A_{fp} = A_{ps} + A_t + N_c A_{pc} \tag{1}$$

$$A_{fp} = (2W.L - 2N_c.A_c) + (W.t) + (N_c.A_c) \tag{2}$$

$$A_{fp} = A_f + N_C(A_{pc} - 2A_c) \tag{3}$$

Equation (3) can be rearrange and written as equation (4):

$$A_{fp} = A_f + N_x \times N_y(A_{pc} - 2A_c) \tag{4}$$

The surface area ratio of the fin (*RAF*) defined by equation (5). It represent the ratio between the surface area of conventional fin (A_f) to surface area of perforated fin (A_{fp}).

$$RAF = \frac{A_{fp}}{A_f} \tag{5}$$

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

Fin weight reduction ratio (*RWF*) can be expressed as Equation. (7), it is come from the ratio of the weight perforated fin to the weight of solid fin:

$$RWF = \frac{W_{fp}}{W_f} = \frac{(W_f - N_x \times N_y \times A_c \times t \times \rho)}{W_f} \tag{7}$$

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$$RWF = 1 - \frac{(N_x \times N_y \times A_c \times t \times \rho)}{L \times W \times t \times \rho} = 1 - \frac{N_x \times N_y \times A_c}{L \times W} \quad (8)$$

The fin surface area of heat transfer can be calculated by equation (9):

$$A_{fp} = A_f - 2N_c.A_c + N_c.A_p \quad (9)$$

$$A_{fp} = A_f + N_c(A_p - 2A_c) = A_f + \pi N_C.b\left(t - \frac{b}{2}\right) \quad (10)$$

The *RAF* can be expressed as equation (11):

$$RAF = 1 + \frac{\pi \times b \times N_x \times N_y\left(t - \frac{b}{2}\right)}{(2W \times L + W \times t)} \quad (11)$$

Equation (12) was used to evaluate the heat transfer coefficient by natural convection for array of parallel flat plates (Mousa, 2000).

$$Nu = \frac{h \times B}{k} = \frac{Ra}{24}\left(1 - e^{\frac{-35}{Ra}}\right)^{0.75} \quad (12)$$

where B is the average space between adjacent fins.

$$Ra = \frac{\rho^2 \times g \times \beta \times C_p \times B^4 \times \Delta T}{\mu \times k \times L} \quad (13)$$

Numerous researchers described that the coefficient of surface heat transfer for the perforated fin as a function of open area ratio (*ROA*). The open area ratio of the perforated surface was defined as equation (14) (Sable *et al*., 2010; Al-Doori, 2011):

$$ROA = \frac{OA}{OA_{max}} \quad (14)$$

where OA is the actual open area $= A_c.N_c$

$$OA = A_c.N_x.N_y \quad (15)$$

OA_{max} is the maximum possible perforations open area, which is defined as equation (16)

$$OA_{max} = A_c.N_{c,max} = A_c.N_{x,max}.N_{y,max} \quad (16)$$

where $N_{x,max}$ and $N_{y,max}$ are the maximum possible number of the perforations along the fin. In this study these numbers were assumed to be constant.

Equation (17) was used to estimate the perforated surface heat transfer coefficient ratio (R_h).

$$R_h = 1 + 0.75\frac{OA}{OA_{max}} \quad (17)$$

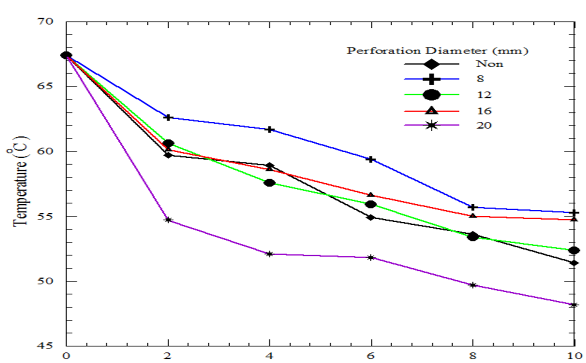
Equation (18) expresses the film heat transfer coefficient of the perforated surface (h_{ps}).

$$h_{ps} = (1 + 0.75\frac{OA}{OA_{max}})h \quad (18)$$

RESULTS AND DISCUSIONS

In this research the heat transfer by natural convection was examined at steady state to show the effect of the geometrical parameters of impressions. Figure-3 shows the temperature distribution along *x*-direction of the perforated fins and non-perforated fin. As shown in Figure-3, it is obvious that the temperatures along all fins decreased with the *x*-direction. It was also found that when the perforation diameter increased the temperature drop between the fin base and tip increased too. This was because the increasing in the perforated diameter led to decrease in thermal resistance of the perforated fin. The best heat dissipation appears in the fin plate with perforated diameter of 20mm. Meanwhile, when the heat supplied increased the temperature drop also increased.

The weight reduction ratio (*RWF*) for the perforated fin can be calculated by equation (8). Figure-4 shows the relation between the weight reduction ratios (*RWF*) and perforation diameter for the perforated fin. It can be noted that the increase of the perforation diameter led to decrease the *RWF* for the perforated fin. Figure-5 shows the effect of the perforation diameter (*b*) on the fin surface area ratio (*RAF*). It is clear that the *RAF* is always smaller than unity. Furthermore, the *RAF* decreased with the increase of the perforation diameter. It indicates that *RAF* is a weak function of the fin length and width. This is because the effect of the fin tip area which is smaller as compared to the fin surface area and can be neglected. The heat transfer coefficient and fin area are the main significant parameters on the heat dissipation rate from the perforated fins. All film heat transfer coefficients in this study were assumed to be uniformed and equaled. As shown in Figure-6, the heat transfer coefficient ratio *Rh* was calculated and plotted with effect the perforation diameter. It can be seen that, the heat transfer coefficient ratio *Rh* was always more than 1 and increased up to the upper limit 1.208 as the perforation diameter increased to 20mm. However, the *Rh* decreased to the lower limit of 1when non-perforated fin plate was used.



(a)200W

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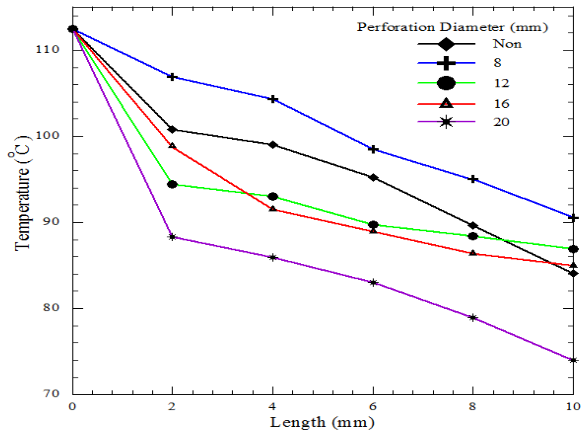
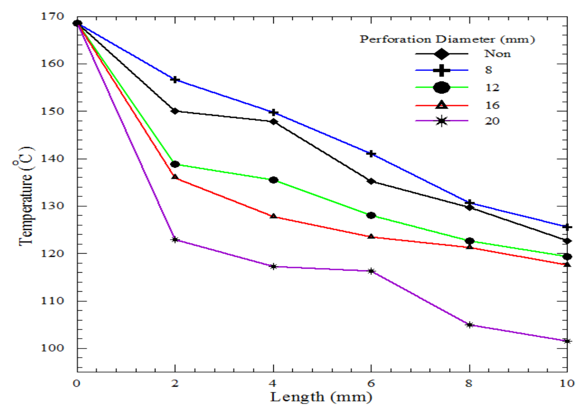


(b)500W



(c)900W

Figure-3. Temperature distribution along *x*-direction of the non-perforated and perforated fins.

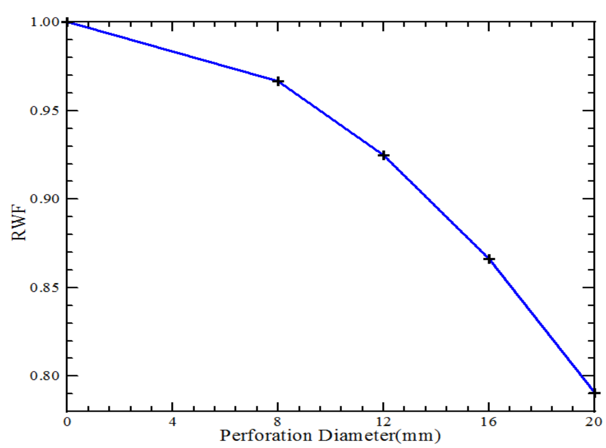


Figure-4. Effect of the perforation diameters on the weight reduction ratio (*RWF*) of the perforated fins.

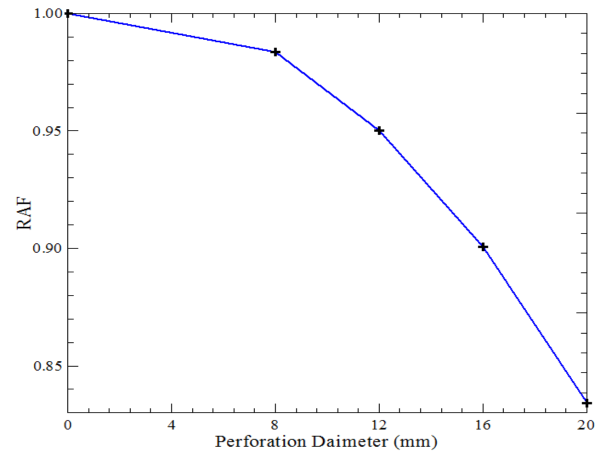


Figure-5. Effect of the perforation diameters on the fin surface area ratio (*RAF*).

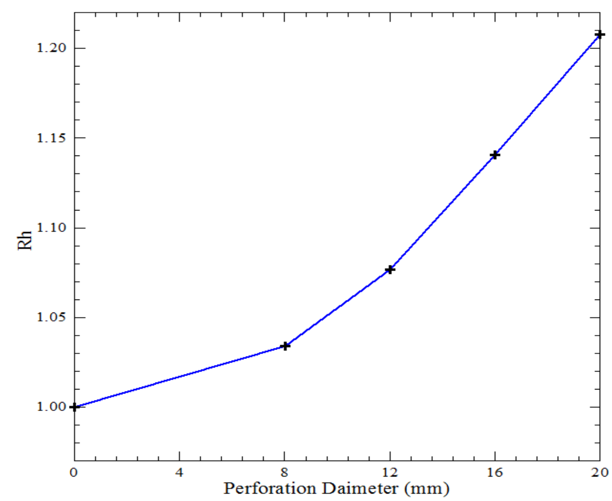


Figure-6. Effect of the perforation diameters on the perforated surface heat transfer coefficient ratio (*Rh*).

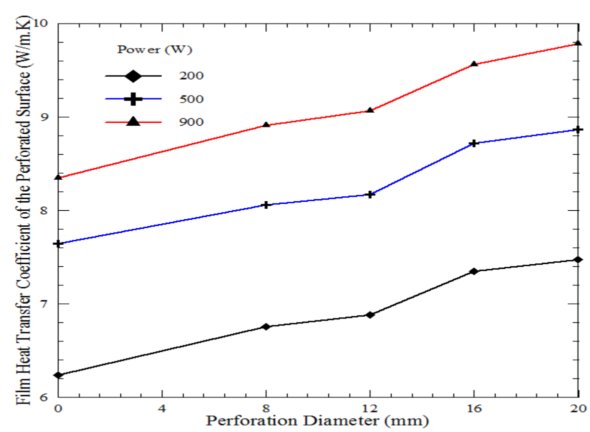


Figure-7. Effect of the perforation diameters on the film heat transfer coefficient of the perforated surface (h_{ps}).

Figure-7 presents the effect of the perforation diameters on the film heat transfer coefficient of the perforated surface (h_{ps}) for different heat supply. It can be seen that the increase of the perforation diameter size led to the increase of film heat transfer coefficient. Meanwhile, the film heat transfer coefficient of the perforated surface (h_{ps}) increased with the increase of the heat supply.

CONCLUSIONS

Along the perforated fin length the temperature drop is higher than that for non-perforated fin. The rate of temperature drop along the perforated fin decrease with decrease of the perforation dimension. The perforation dimension and lateral spacing have the significant effect on the gain in the heat dissipation rate of the perforated fin. For the perforated fin that contained a larger perforation diameter, the surface heat transfer coefficient ratio was higher than that contained small perforation diameter. The film heat transfer coefficient of the perforated surface (h_{ps}) increased with the increase of perforation diameter and power supply. The increases of perforation diameter and power supply have significant effect on the heat transfer rate and the coefficient of heat transfer.

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