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Full Length Research Paper

Enhancement of natural convection heat transfer from a fin by rectangular perforations with aspect ratio of two

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In this study, the enhancement of natural convection heat transfer from a horizontal rectangular fin embedded with rectangular perforations of aspect ratio of two has been examined using finite element technique. The results for perforated fin have been compared with its equivalent solid one. A parametric study for geometrical dimensions and thermal properties of the fin and the perforations was carried out. The study investigated the gain in fin area and of heat transfer coefficients due to perforations. It was found that, for certain range of rectangular dimension and spaces between perforations, there is an augmentation in heat dissipation and a reduction in weight over that of the equivalent solid one. Also, the heat transfer enhancement of the perforated fin increases as the fin thickness and thermal conductivity increase.

Key words: Finite element, perforated fin, heat dissipation, heat transfer augmentation, natural convection, rectangular perforations.

INTRODUCTION

The removal of excessive heat from system components is essential to avoid damaging effects of burning or overheating. Therefore, the enhancement of heat transfer is an important subject of thermal engineering (Sahin and Demir, 2008a). Extended surfaces (fins) are frequently used in heat exchanging devices for the purpose of increasing the heat transfer between a primary surface and the surrounding fluid. Various types of heat exchanger fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered or pin fins, to a combination of different geometries, have been used (Sahin and Demir, 2008b). The study of improving heat transfer performance is referred to as heat transfer augmentation, enhancement or intensification. The heat transfer augmentation is very important subject in industrial heat exchangers and other thermal application. There are many techniques which are available for augmentation for single or two-phase heat transfer in natural or forced convection. Those techniques may be passive methods or active schemes (Kakac et al., 1981). The heat transfer improvement may in general be achieved by either of increasing the heat transfer coefficients, or the heat transfer surface area or by both (Sahin and Demir, 2008a, Sahin; Demir, 2008b). In most cases, the area of heat transfer is increased by utilizing extended surfaces in the form of fins attached to walls and surfaces (Bergles, 1981). Fins are normally used as heat transfer enhancement devices. As the extended surfaces (fins) technology continues to grow, new design ideas emerge including fins made of anisotropic composites, porous media, perforated and interrupted plates (Kakac et al., 1981; Bergles, 1981; Al-Essa and Al-Hussien, 2004; Mullisen and Loehrke, 1986). Due the high demand for lightweight, compact, and economical fins, the optimization of fin size is of great importance. Therefore, fins must be designed to achieve maximum heat removal with minimum material expenditure and easy manufacturing procedure (Prasad and Gupta, 1998; Kutscher, 1994; Al-Essa, 2000). Previous studies by Kakac et al. (1981), Bergles (1981), Al-Essa and Al-Hussien (2004), Prasad and Gupta (1998) and Al-Essa (2000) had introduced shape modifications by cutting some material from fins to make cavities, holes, slots, grooves, or channels through the fin body to increase the heat transfer area and/or coefficient. One popular heat transfer augmentation technique involves the use of rough surfaces of different configurations. The

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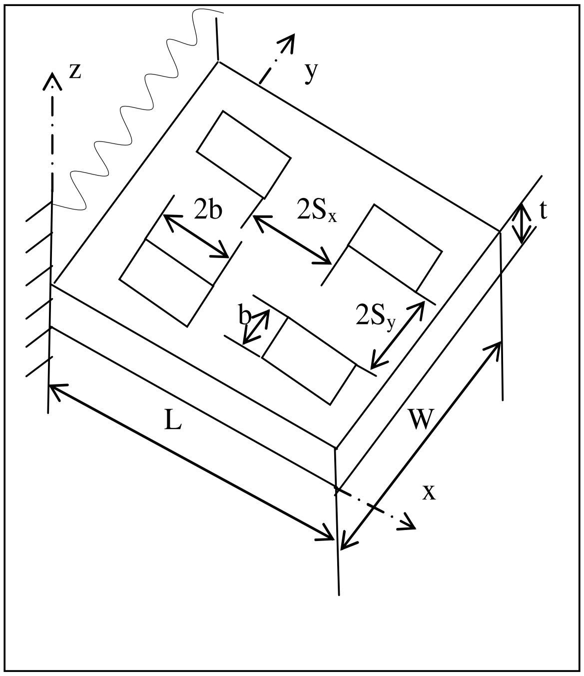

Figure 1. The fin with (b by 2b) rectangular perforations.

surface roughness aims to promote surface turbulence that is intended mainly to increase the heat transfer coefficient rather than the surface area (Kakac et al., 1981; Bergles, 1981) For the free convection from machined or formed rough surfaces with air, it was found that heat transfer coefficient can be increased up to 100% (Kakac et al., 1981). It was reported that non-flat surfaces have free convection coefficients that are 50 to 100% more than those of flat surfaces (Bergles, 1981). Several other studies reported a similar trend for interrupted (perforated) fins attributing the improvement to the restarting of the thermal boundary layer after each interruption indicating that the increase in convection coefficient is even more than enough to offsets lost area, if any (Bergles, 1981). Perforated plates and fins represent an example of surface interruption (Kakac et al., 1981; Bergles, 1981, Al-Essa and Al-Hussien, 2004) and are widely used in different heat exchanger, film cooling, and solar collector applications. This study aims to examine the extent of heat transfer enhancement from a horizontal rectangular fin under natural convection as a result of introducing surface modifications (interruptions) to the fin. The modifications in this work are vertical rectangular (b by 2b) perforations made through the fin thickness. This study investigates the influence of perforations on heat transfer rate or heat dissipation rate of the perforated fin. The modified fin, as it is called perforated fin, is compared to the corresponding solid (non-perforated) fin in terms of heat transfer rate. The study eventually attempts to make the best use of the material and size of a given fin, which involves some sort of optimization.

ANALYSIS

The perforated fin with rectangular perforations under consideration is shown in Figure 1. The transverse Biot number in (z) direction (Bi_Z) can be calculated by ($Bi_Z = h_{ps}.t/2k$) and the transverse Biot number in (y) direction (Bi_y) can be calculated by ($Bi_y = h_{ps}(S_y + b/2)/k$). If the values of ($Bi_Z$) and ($Bi_y$) are less than .01 (Aziz and Lunadini, 1995; Razelos and Georgiou, 1992) then the heat transfer in (z) and (y) directions can be assumed lumped and one dimension solution can be considered. If the values of (Bi_Z) and (Bi_y) is greater than 0.01 then the heat transfer solution should be two or three dimensions. In this study the parameters of the perforated fin are restricted as they lead to values of (Bi_Z) and (Bi_y) smaller than 0.01. The calculations in this study are based on assuming steady state one-dimension heat conduction, uniform heat transfer coefficient over the whole fin solid surface (perforated or solid) and uniform heat transfer coefficient within the perforation, negliing radiation effects, no heat sources/sinks in the fin body with uniform base and ambient temperatures homogeneous and isotropic fin material with constant thermal conductivity and the side area of the fin is much smaller than that of its surface area. Based on the previous assumptions, the energy equation of the fin along with the boundary conditions may be stated as):

$$k\frac{d^2T}{dx^2} = 0$$ (Rao, 1989) (1)

The associated boundary condition at the fin base (x = 0) is

$$T_b = 0$$ (Al-Essa, 2006]) (2)

And at the perforated surfaces is:

$$k\frac{dT}{dx}l_x + h_{ps}(T - T_\infty) + h_{pc}(T - T_\infty) = 0$$ (Al-Essa, 2006) (3)

It is established that the heat dissipation from a fin for solid and perforated one depends on fin surface area and heat transfer coefficients. The average value of heat transfer coefficient of the solid fin surface (h_{sf}) in natural convection is given by

$$h_{sf} = Nu.k_{air}/L_c \quad \text{where} L_c = L.W/(2L + 2W)$$ (Raithby and Holands, 1984). (4)

The average Nusselt number, Nu, can be given by [14]:

$$Nu = (Nu_u + Nu_l)/2 \qquad (5)$$

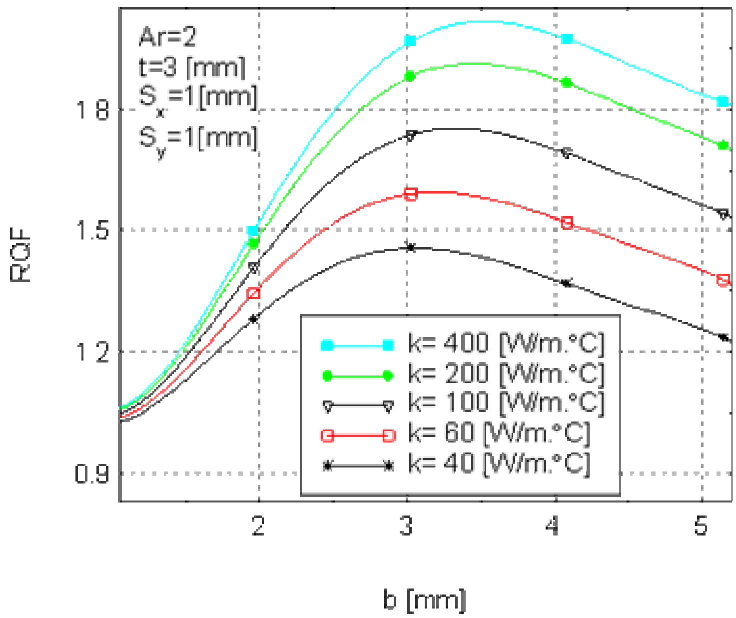

Figure 2. Ratio of heat dissipation rate (RQF) vs. perforation dimension with variable fin thermal conductivity.

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$$Nu_u = [(1.4/\ln(1+1.4/(0.43Ra^{0.25})))^{10} + (0.14Ra^{0.333})^{10}]^{0.1} \quad (6)$$

$$Nu_l = 0.527Ra^{0.2} / (1 + (1.9/Pr)^{0.9})^{2/9}$$

For the fin considered in this study, both perforated and non-perforated, the fin tip is a vertical surface for which the Nusselt number is given by

$$Nu_t = 0.5\left(\left(\frac{2.8}{\ln\left(1+\frac{2.8}{0.515Ra^{.25}}\right)}\right)^6 + \left(.103Ra^{0.333}\right)^6\right)^{(1/6)} \quad (8)$$

$$h_t = Nu_t \,.\, k_{air} / L_c$$ (Raithby and Holands, 1984).
where $L_c = L.t/(2L + 2t)$

(9)

For the perforated fin there are three distinct heat transfer coefficients that exist. The first one is the heat transfer coefficient of the remaining solid portion of the perforated fin surface (h_{ps}) which can be calculated by the following expression:

$$h_{ps} = (1 + 0.75\,ROA).h_{ss}$$ (Al-Essa and Al-Hussien, 2004) (10)

$$ROA = OA / OA_{max}$$

The second one is the heat transfer coefficient within the perforation (h_{pc}). I (6) Nu_c) is given as:

$$Nu_c = \left[\left(\frac{Ra_c}{18.7}\right)^{-1.5} + (0.62Ra_c^{0.25})^{-1.5}\right]^{1/-1.5}$$ (Raithby and Holands, 1984) (11)

The third one is the heat transfer coefficient at fin tip (h_t) which its Nusselt number (Nu_t) is given by equation (8). To investigate the perforated fin performance, the ratio of heat dissipation rate of the perforated fin to that of the solid one (RQF) is introduced and given by:

$$RQF = Q_{pf} / Q_{sf} \quad (12)$$

Where the heat dissipation rate of the perforated fin (Q_{pf}) is computed numerically according to the finite element heat transfer solution described in Al-Essa (2006) which used the variational approach method in matrix notation to formulate the algebraic equations of the problem. The formulation of these equations can be found in details in Rao (1989). The heat dissipation rate of the solid fin (Q_{sf}) is computed according to the exact solution described in Incropera and Dewitt (2007). The material weight of the perforated fin is compared with that of the non-perforated one by weight ratio RWF which is given by:

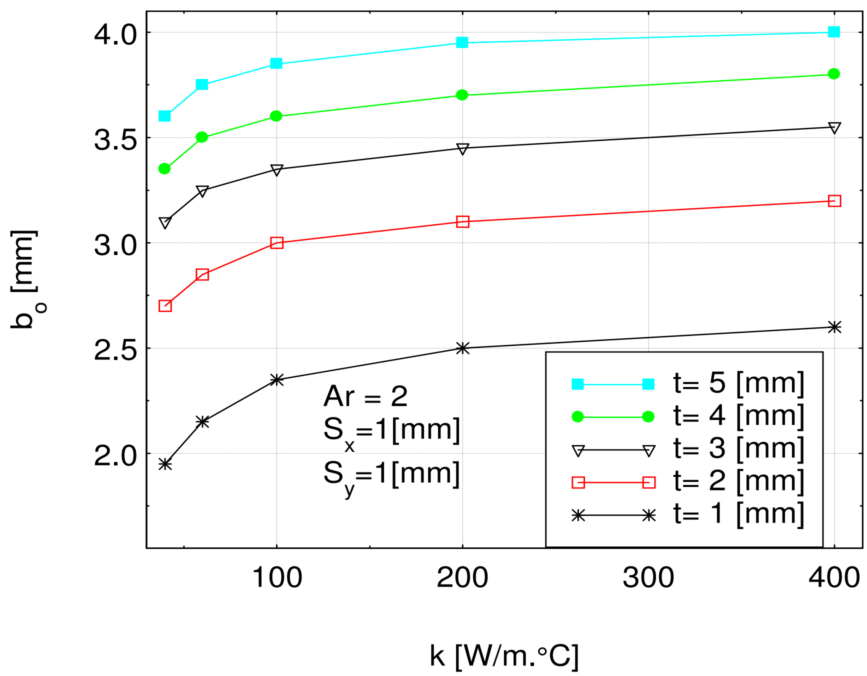

Figure 3. Values of the optimum perforation dimension (b_o) as a function of the fin thermal conductivity with variable fin thickness.

$$
\begin{aligned}
RWF &= W_{pf} / W_{sf} \\
&= 1 - (N_x . N_y . 2 b^2 . t) \;/\; (L.W.t)
\end{aligned} \tag{13}
$$

RESULTS AND DISCUSSION

The perforated fin results are compared with solid counterpart to evaluate the improvement in heat transfer caused by introducing the perforations. It is assumed that both fins have the same dimensions (the fin length is L= 50 mm and its width is W = 200 mm), same thermal conductivity, same base temperature $(T_b = 100^o C)$ and same ambient temperature $(T_\infty = 20^o C)$. A parametric study was carried out to investigate the effect of the perforation dimension, longitudinal spacing and lateral spacing on the perforated fin performance.

Ratio of heat dissipation rate (RQF)

The ratio of heat dissipation rate from the perforated fin to that of the corresponding solid one (RQF) was studied in terms of perforation dimension (b) for different values of fin thickness and thermal conductivity. The results are shown in Figure 2. It is shown that thicker fins produced larger heat transfer enhancement at any value of (b). The variation of (RQF) with (b) at different values of (t) showed a consistent trend of increase in (RQF) with increasing (b) to a maximum value followed by a decrease. This trend may be explained by net effects of changing in fin heat transfer surface area and heat transfer coefficients due to perforations. Furthermore, it is clearly seen that (RQF) has a strong relation with that of the perforation dimension (b). The perforation dimension at which (RQF) has its maximum value is referred to as the optimum perforation dimension (b_o). Figure 2 indicates that the optimum dimension (b_o) is a function of the fin thickness and its thermal conductivity. The approximate values of (b_o) can be obtained from the graphs and their curves like that shown in Figure 2.

To obtain the approximate values of the optimum perforation dimensions (b_o), other fin thicknesses of (1, 2, 4 and 5 mm) and thermal conductivities of (40, 60, 100, 300, 500 W/m.^{o}C) are considered. The obtained values of (b_o) are plotted in Figure 3 from which it is obvious that (b_o) has an obvious relation of the fin thickness while it is slightly varies with its thermal conductivity.

Longitudinal spacing (S_x)

To investigate the effect of the perforation longitudinal

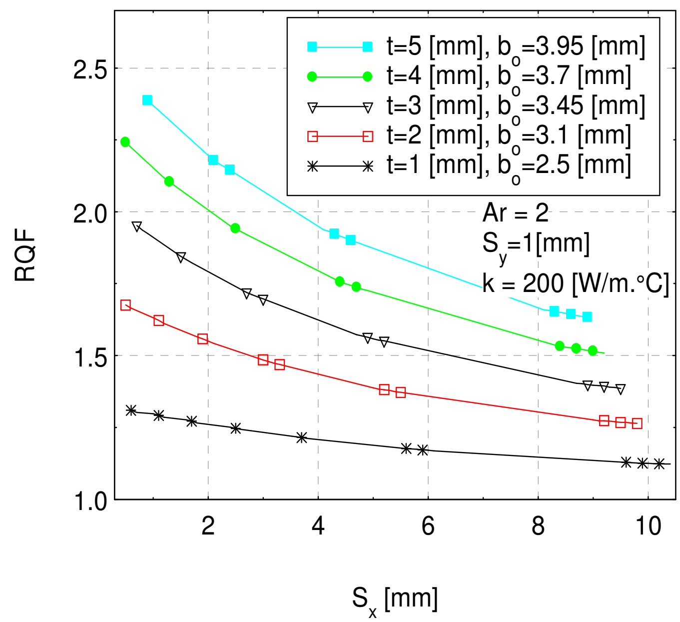

Figure 4. Ratio of heat dissipation rate (RQF) vs. longitudinal spacing with variable fin thickness.

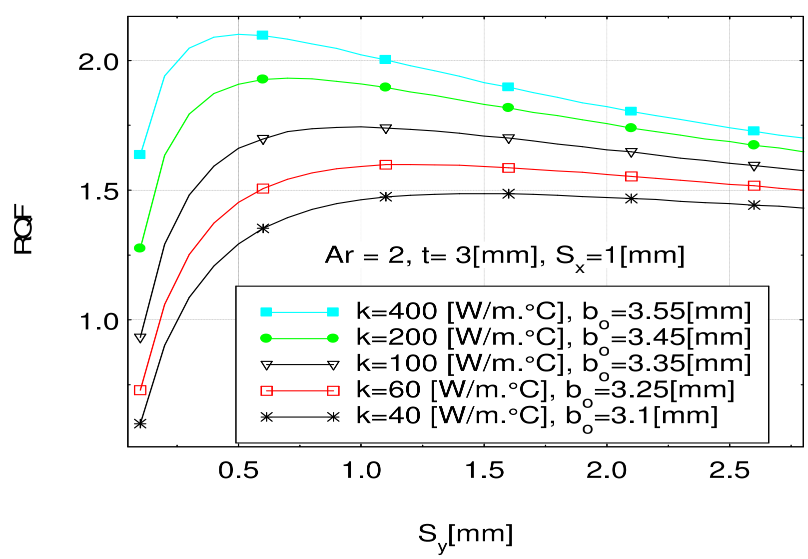

Figure 5. Ratio of heat dissipation rate vs. lateral spacing with variable fin thermal conductivity.

spacing (S_x) on the perforated fin performance, the heat dissipation ratio (RQF) is plotted as a function of the spacing (S_x) as shown in Figure 4. This Figure indicates that at any fin thickness, the (RQF) decreases with increasing of spacing (S_x). This behavior is due to the fact that increasing of (S_x) means less number of perforations, this means loss of the element that causes heat transfer enhancement. So according to (S_x) spacing it is preferable to minimize it as possible.

Lateral spacing (S_y)

Figure 5 shows that the effect of the perforation lateral

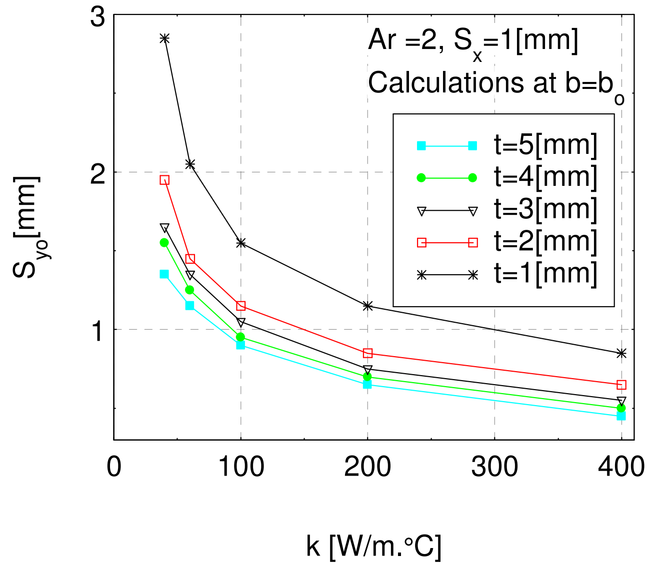

Figure 6. Optimum lateral spacing vs. thermal conductivity with variable fin thickness.

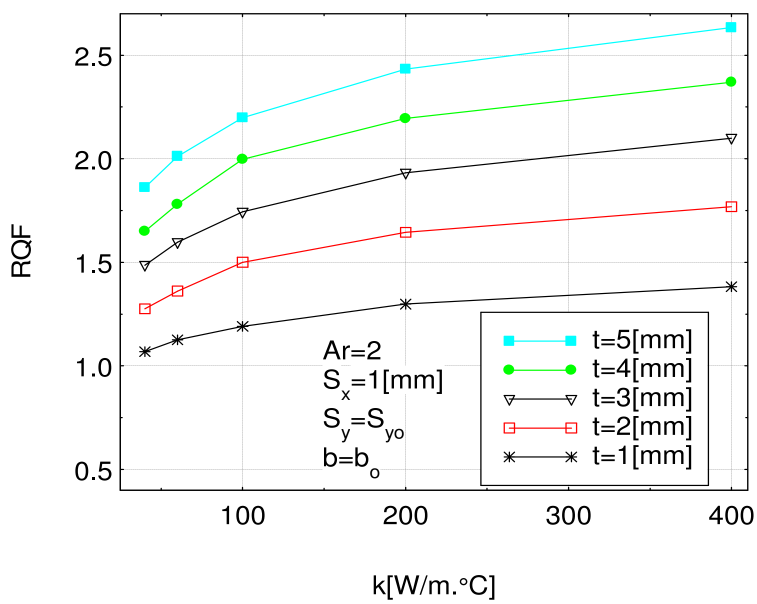

Figure 7. Ratio of heat dissipation rate vs. thermal conductivity at (b_o and S_{yo}).

spacing (S_y) on the perforated fin performance for various values of the fin thickness and its thermal conductivity. RQF increases for low values of (S_y) and then tends to decline thereafter. The conflicting effects of fin thermal resistance and surface area are responsible for this trend. Also, it is noted that (RQF) heavily dependson the lateral spacing. It is indicated in Figure 5 that (RQF) values which closely related to the optimum perforation dimension (b_o) are strong function of the lateral spacing. In this region, the ratio (RQF) attains maximum value and then decreases as the lateral spacing increases. This means that there is an optimum associated value of the lateral spacing which isabbreviated as (S_{yo}). The values of (S_{yo}) strongly depends on the fin thickness and its thermal conductivity.

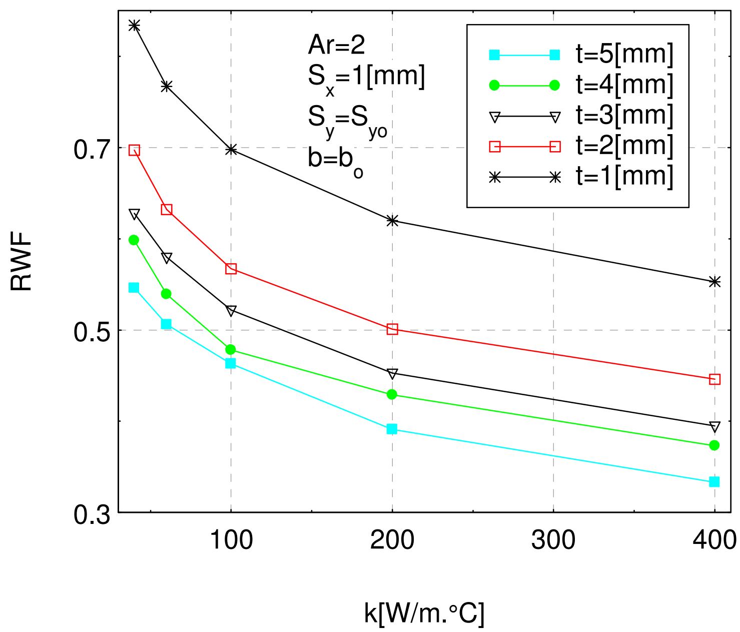

Figure 8. Ratio of perforated fin (RWF) vs. thermal conductivity at (b_o and S_{yo}).

The values of (S_{yo}) are plotted in Figure 6. This Figure shows that (S_{yo}) decreases as fin thermal conductivity and its thickness increase.

To summarize the advantages of using the perforated fin at the optimum values of perforation geometry, the ratios of heat dissipation (RQF) and of weight (RWF) as a function of the fin thickness with variable fin thermal conductivity at the optimum perforation dimension (b_o) and optimum lateral spacing (S_{yo}) are plotted in Figures 7 and 8. These Figures show that the using of perforations in finsleads to enhance heat dissipation and decrease the fin weight. The enhancement increases and the weight decreases as the fin thickness and its thermal conductivity increases. This means that using of perforated fin leads to enhance heat dissipation rates and at the same time decreasing the expenditure of the fin material.

Conclusion

This study showed that for certain values of rectangular perforation dimension, the perforated fin enhances heat transfer. The magnitude of enhancement is proportional to the fin thickness and its thermal conductivity. Also, the extent of heat dissipation rate enhancement for perforated fins is a function of the fin dimensions, the perforation geometry and the fin thermophysical properties. Furthermore, the gain in heat dissipation rate for the perforated fin is a strong relation of both, the perforation dimension and the lateral spacing. This function attains a maximum value at given perforation dimension and spacing, which are called the optimum perforation dimension (b_o), and the optimum lateral spacing (S_{yo}) respectively. Finally, not only the perforation of fins enhances heat dissipation but also decreases the weight of the fin.

NOMENCLATURE

A, cross sectional area [m^2]; **Ar,** aspect ratio; **Bi,** Biot number **b,** rectangular perforation dimension[m]; **h,** heat transfer coefficient [W/m^2.^{o}C]; **k,** thermal conductivity [W/m.^{o}C]; **L,** fin length [m]; l, unit vector; L_c, characteristic length[m]; **N,** Number of perforations; **Nu,** average Nusselt number; Nu_c, average Nusselt number of the inner perforation surface; **OA,** open area of the perforated surface; OA_{max}, maximum open area of the perforated surface; **Ra,** Rayleigh number; Ra_c, Rayleigh number of the perforation inner lining surface; **ROA,** ratio of open area; **RQF,** Ratio of heat dissipation rate of perforated fin to that of non-perforated fin; **RWF,** ratio of the perforated fin weight to that of the non-perforated fin; **S,**

perforation spacing [m]; **T,** temperature [^{o}C]; **t,** fin thickness [m]; **W,** fin width [m].

Subscripts and superscripts: b, fin base; **l,** lower surface of fin; **max,** maximum; **pc**, perforation inner surface (within the perforation); **pf,** perforated fin; **ps,** remaining solid portion of the perforated fin; **sf,** solid (non-perforated) fin; **ss,** solid surface; **t,** fin tip; **u,** upper surface of fin; **x,** longitudinal direction or coordinate; **y,** transverse (lateral) direction with the fin width or coordinate*;* **z:** transverse (lateral) direction with the fin thickness or coordinate. ∞ **:** ambient.

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