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Effect of triangular perforation orientation on the heat transfer augmentation from a fin subjected to natural convection

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ABSTRACT

This study examined heat transfer enhancement from a horizontal rectangular fin embedded with triangular perforation under natural convection compared to equivalent solid fin. The parameters considered were geometrical dimensions and orientation of the perforations. The study considered the gain in fin area and extent of heat transfer enhancement due to perforations. It was found that the heat dissipation from the perforated fin for certain range of perforation dimension can result in improvement in heat transfer over the equivalent solid fin, and the heat transfer enhancement of the triangular perforation that its base is parallel to the fin side (Triangle 3) is the highest.

Key words: Perforated fin, Triangular Perforation, Finite Element, Heat dissipation, Heat transfer enhancement.

NOMENCLATURE

A: cross sectional area of the fin
A_c: cross sectional area of the perforation
A_e: cross sectional area of the finite element
b: triangular perforation dimension
g: acceleration due to gravity
h: heat transfer coefficient
k: thermal conductivity of fin material
L: fin length
L_c: characteristic length
N_c: total number of perforations
Nu: average Nusselt number
Nu_c: average Nusselt number of the inner perforation surface
N_x: number of perforations in x direction
N_x: number of perforations in y direction
OA: open area of the perforated surface
P_c: perimeter of perforation opening
Pr: Prandtl number.
Q: heat dissipation rate
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
S: perforation spacing
T: temperature
t: fin thickness
Tri. 1: Triangular number 1
Tri. 2: Triangular number 2
Tri. 3: Triangular number 3
W: fin width
x: longitudinal direction or coordinate
y: transverse (lateral) direction or coordinate

GREEK LETTERS

β: volumetric expansion coefficient
μ: dynamic viscosity
ν: kinematics viscosity

Subscripts and superscripts

f: non-perforated (solid) fin
fp: perforated fin
pc: perforation inner surface (within the perforation)
ps: remaining solid portion of the perforated fin
t: fin tip
x: in or along the x-direction
y: in or along the y-direction
u: upper surface of fin
l: lower surface of fin
max: maximum
n: nominal
∞: ambient
b: fin base
m: mean (average)

INTRODUCTION

The removal of excessive heat from system components is essential to avoid damaging effects of overheating. As a general rule, the total heat losses is by the convective heat transfer process [1, 2] which should be enhanced as possible. Therefore, the enhancement of heat transfer is an important subject of thermal engineering. Free convection phenomenon has been object of extensive research. The importance of this phenomenon is increasing day by day due to the enhanced concern in science and technology about buoyancy induced motions in the atmosphere [3]. The heat transfer from surfaces may in general be enhanced by increasing the heat transfer coefficients between the surfaces and there surroundings, by increasing the heat transfer area of the surfaces, or by both. In most cases, the area of heat transfer is increased by utilizing extended surfaces in the form of fins attached to walls and surfaces.

Fins as a heat transfer enhancement devices have been quite common. As the extended surface technology continues to grow, new design ideas emerge including fins made of anisotropic composites, porous media, and interrupted plates [4]. 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 taking into account, however, the ease of manufacturing of the fin shape [5].

This study aims mainly at examining the extent of heat transfer enhancement from a horizontal rectangular fin under natural convection as a result of introducing body modifications (perforation which leads to interruptions) to the fin. The modifications in this work are vertical equilateral triangular perforations made through the fin thickness. The study investigates the influence of triangular perforation orientation on heat transfer enhancement. The modified fin, called perforated fin, is compared to the corresponding solid (non-perforated) fin in terms of heat transfer dissipation 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.

Large number of studies has 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 heat transfer coefficient [6, 7]. One popular heat transfer augmentation technique involves the use of rough surfaces of different configurations. The surface roughness aims at promoting surface turbulence that is intended mainly to increase the heat transfer coefficient rather than the surface area [8]. It was reported that non-flat surfaces have free convection coefficients that are 50% to 100% more than those of flat surfaces [9]. Several other researchers 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 [10, 11]. Perforated plates (fins) represent an example of surface interruption [10] and are widely used in different heat exchanger, film cooling, and solar collector applications [9, 12]. An experiment study was performed by Elshafei [13] on natural convection heat dissipation from hollow/perforated circular pin fin sinks. He found that the heat dissipation from the hollow/perforated pin fins was better (bigger) than that of solid fins.

Despite the fact that correlations for the convection heat transfer coefficient within cavities and over the surfaces of non-perforated plates are readily available [6, 9], literature search indicated a lack of such relations for the perforated surfaces under natural convection. Consequently, perforated surface heat transfer coefficient were estimated based on the concepts of augmentation ratio [10] and open area ratio [12, 14].

The overall objective of this study was to evaluate the potential of heat transfer enhancement when body perforations of triangular cross section are introduced to a horizontal rectangular plate (fin) under natural convection conditions. The specific objectives of the work may be summarized as follows:

1- Evaluate the influence of equilateral triangular perforation factors (dimensions, lateral spacing, thermal conductivity and orientation) on the enhancement of heat transfer dissipation rate of the fin.
2- Determine the values of parameters that would result in maximum heat transfer enhancement of the perforated fin compared to the solid counterpart.

HEAT TRANSFER ANALYSIS

The classical analysis of fins, which assumes one-dimensional heat conduction as the Biot number is small (less than 0.01) can be considered. The perforated fin with triangular perforations analyzed in this study is shown in figure 1, 2 and 3. The symmetry part considered for heat transfer analysis (shown hatched). For this part the transverse (lateral) Biot number in (z) direction (Bi_z) can be calculated by ($Bi_z = h_{pc}.t / 2k$) and the transverse Biot number in (y) direction (Bi_y) can be calculated by ($Bi_y=h_{ps}.(S_y+b/2)/k$) for Tri. 1 and Tri. 2. And it is ($Bi_y=h_{ps}.(2S_y+b.\mathrm{Sin}\,(60^o))/k$) for Tri. 3.the As the values of (Bi_z) and (Bi_y) less than 0.01, then the heat transfer in (z) and (y) directions can be assumed lumped and one dimensional solution can be considered. If the values of (Bi_z) and (Bi_y) are greater than 0.01 then the heat transfer solution must be two or three dimensions. In this study the parameters of the perforated fin are taken as they lead to values of (Bi_z) and (Bi_y) smaller than 0.01. The analysis and results reported are based on the following assumptions:

The analysis and results reported in this study are based on the following assumptions:

1. Steady, one-dimensional heat conduction; 2. Homogeneous and isotropic fin material with constant thermal conductivity; 3. No heat sources/ sinks in the fin body; 4. Uniform base and ambient temperatures; 5. Uniform heat transfer coefficient over the whole fin solid surface (perforated or solid); 6. Uniform heat transfer coefficient within the perforation.

Based on these assumptions, the energy equation of the fin along with the boundary conditions may be stated as below [15]

$$\frac{d}{dx}\left(k\frac{dT}{dx}\right)=0 \qquad (1)$$

The associated boundary conditions are
1- At the base surface (x = 0), $T = T_b$
2- At the perforated surface, at the perforation inner lining surface and At the fin tip surface.

$$k.A_e \frac{dT}{dx}/_x + h_{ps}.A_{ps}(T-T_\infty)+h_{pc}.A_{pc}(T-T_\infty)+h_t.A_t(T_t - T_\infty)=0 \qquad (2)$$

The fin with triangular perforations of three orientations are shown in Figures (1, 2, and 3). The surface area of this fin including the tip may be expressed as

$$\begin{aligned}A_{fp} &= A_{ps}+A_t+N_c.A_{pc}\\ &=(2.W.L-2.N_c.A_c)+(W.t)+(N_c.A_{pc})\\ &=A_f+N_c(A_{pc}-2A_c)\\ &=A_f+N_x.N_y(3\,b.t-b^2.Sin(\pi/3)) \qquad (3)\end{aligned}$$

In this study, the energy equation presented in equation (1) is solved numerically utilizing one dimensional finite-element technique using the variational approach [15]. The corresponding variational statement has the following form:

$$\begin{aligned}I_n &= \frac{1}{2}\iiint_V k\left(\frac{dT}{dx}\right)^2 dV+\frac{1}{2}\iint_{A_{ps}} h_{ps}(T-T_\infty)^2 dA_{ps}\\ &+\frac{1}{2}\iint_{A_{pc}} h_{pc}(T-T_\infty)^2 dA_{pc}+\iint_{A_t} h_t(T_t-T_\infty)T\,dA_t \qquad (4)\end{aligned}$$

It is established that everything else being the same, heat dissipation from a fin, solid or perforated, depends on fin surface area and heat transfer coefficient. For the solid fin, both aspects are established. The average value of h is that for a single horizontal plate in natural convection and may be given by

$$h = Nu.\ k_{air}\ /\ Lc \qquad (5)$$

The average Nusselt number, Nu, is given by [3]

$$Nu = (Nu_u + Nu_l)/2 \quad (6)$$

$$Nu_u = \left[\left(\frac{1.4}{\ln\left(1 + \frac{1.4}{0.43\, Ra_o^{0.25}}\right)} \right)^{10} + \left(0.14\, Ra_o^{0.333}\right)^{10} \right]^{0.1} \quad (7)$$

And

$$Nu_l = \frac{0.527\, Ra^{0.2}}{\left(1 + \left(\frac{1.9}{p_r}\right)^{0.9}\right)^{\frac{2}{9}}} \quad (8)$$

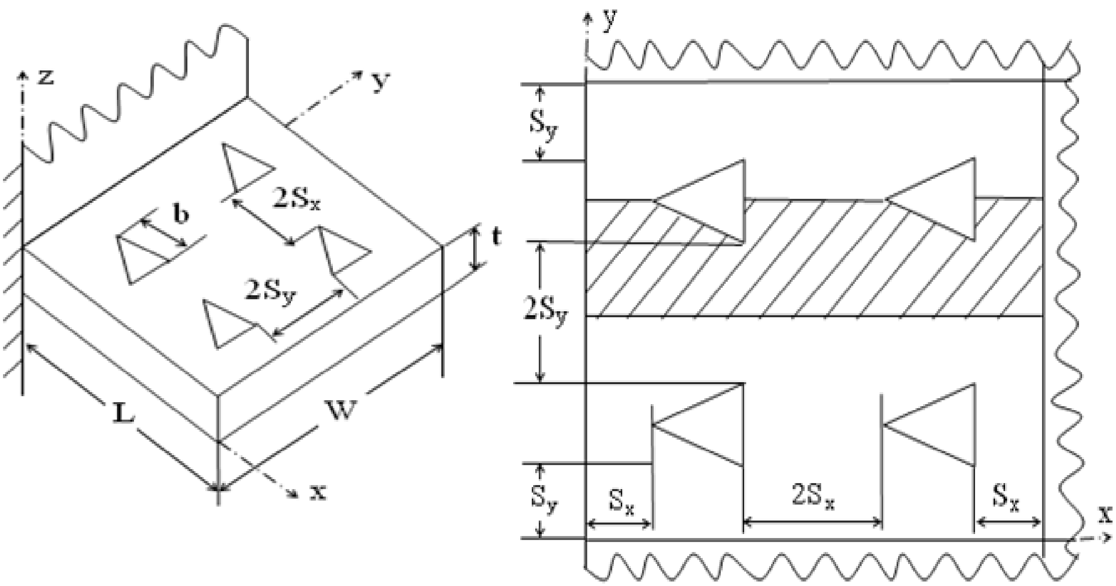

Fig. 1:- Fin with equilateral triangular perforation and the symmetrical part (shown hatched) of the perforated fin considered for analysis (first orientation, Tri. 1)

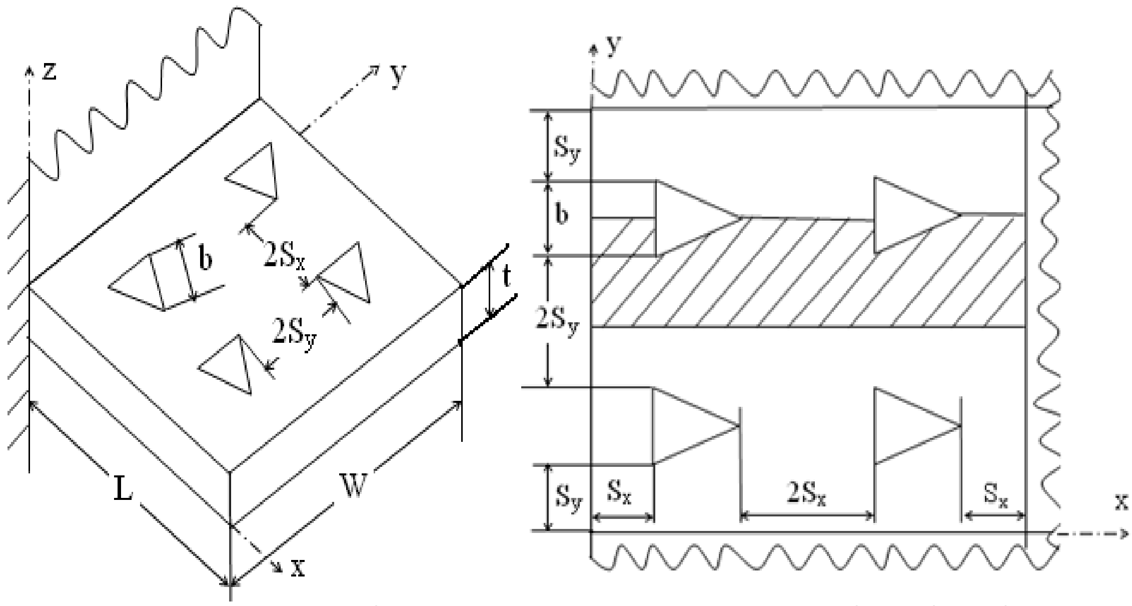

Fig. 2:- Fin with equilateral triangular perforation and the symmetrical part (shown hatched) of the perforated fin considered for analysis (second orientation, Tri. 2)

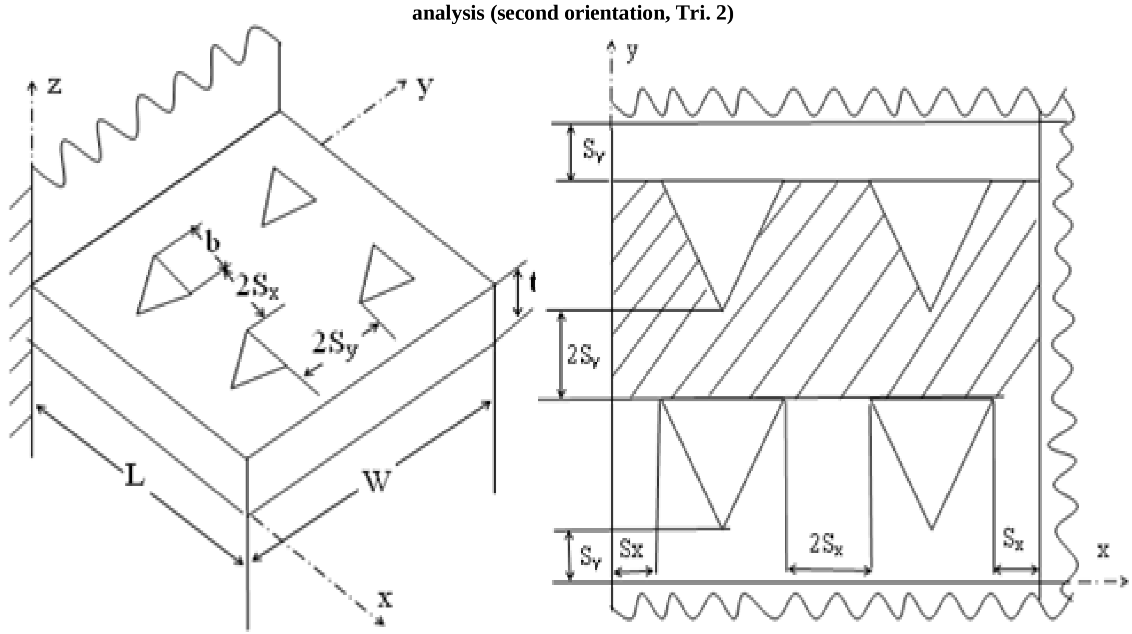

Fig. 3:- Fin with equilateral triangular perforation and the symmetrical part (shown hatched) of the perforated fin considered for analysis (third orientation, Tri. 3)

As for the perforated fin, the aspect of surface area was discussed in the previous section. However, three distinct heat transfer coefficients exist as discussed below.

1. Heat transfer coefficient of the solid portion of the perforated surfaces, h_{ps}

Literature search revealed that there is lack of correlations for the perforated surface under natural convection. Consequently, it was decided in this study to adapt approximate estimate for h_{ps}. Studies under convection conditions reported that h_{ps} was a function of the open area ratio and augmentation ratio [9]. The open area ratio, ROA, for a perforated surface is defined as

$$ROA = OA/OA_{max} \tag{9}$$

Equation (9) shows that ROA ranges between zero (for the non-perforated) and one (for maximum perforation effect). Based on averaging literature estimates of increases in heat transfer coefficient due to perforations (50 to 100%, taking mean value 75% for this study) and assuming a linear relationship between ROA and heat transfer coefficient, the following expression is introduced to estimate h_{ps} in terms of ROA and h [16, 17]:

$$hps=(1+0.75ROA)h \quad (10)$$

2. Heat transfer coefficient within the perforation, h_{pc}

Correlations for heat transfer coefficient in natural convection for triangular perforations are available. With uniform wall temperature, the Nusselt number, Nu_c, is as follows [6]:

$$Nuc=\left[\left(\frac{Ra_c}{14.3}\right)^{-1.5}+\left(0.62\,Ra_c^{\;0.25}\right)^{-1.5}\right]^{-\frac{1}{1.5}} \quad (11)$$

3. Heat transfer coefficient at fin tip, h_t

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

$$h_t = Nu_t \,.k_{air} \,/\, t \quad (12)$$

Nusselt number, Nu_t, is given by [6]

$$Nu_t = 0.5\left(\left(\frac{2.8}{\ln\left(1+\frac{2.8}{0.515\,Ra^{.25}}\right)}\right)^{6}+\left(0.103\,Ra^{0.333}\right)^{6}\right)^{\frac{1}{6}} \quad (13)$$

4- Ratio of heat dissipation rate (RQF)

The ratio of heat dissipation rate RQF from the perforated fin to that of the corresponding solid fin can be expressed as:

$$RQF = Q_{fp} / Q_f \quad (14)$$

where Q_{fp}: is the heat dissipation rate of the perforated fin which was computed using the finite element technique, and Q_f is the heat dissipation rate of the solid fin which was computed using the following analytical equation [8]

$$Q_f = k.A.m(T_b - T_\infty)\,\frac{Sinh(m.L) + (h_t/(m.k))\,Cosh(m.L)}{Cosh(m.L) + (h_t/(m.k))\,Sinh(m.L)} \quad (15)$$

where m is defined as

$$m=\sqrt{\frac{h\,.\,P_f}{k.A}} \quad (16)$$

RESULTS AND DISCUSSION

RQF, is studied in terms of perforation parameter (b) with S_y = 1 mm, S_x = 1 mm, L = 50 mm, W=200 mm, ambient temperature (T_∞ = 20 ^{o} C) and T_b = 100 ^{o}C for different values of fin thickness and thermal conductivity. The results are shown in Figures (4) and (5), which show that thicker fins produced larger heat transfer enhancement at any b. The variation of RQF with b at various t showed a consistent trend of increasing to a maximum value followed by a decrease. This trend may be explained by the net effects of changing area and heat transfer coefficients due to perforations. The increase of RQF in the first part of the curves is due to the fact that A_{fp} , h_{ps} and h_{pc} are increased with (b) which lead to increase the value of RQF. While, in second part the A_{fp} tends to decrease with (b) but h_{ps} and h_{pc} are increased, the net result is a decrease in RQF value. The maximum value of RQF is occurs at some value of perforation dimension symbolized as (b_o).

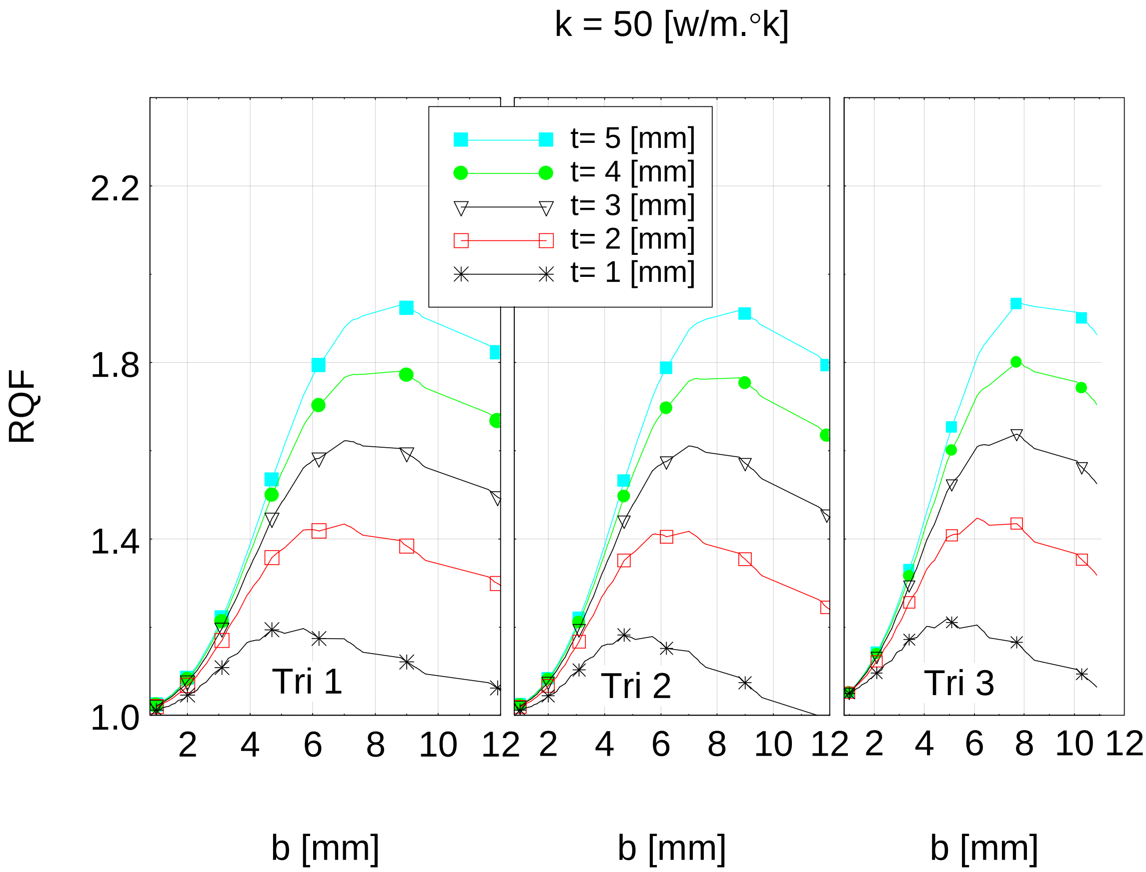

Fig. 4:- The perforated fin heat dissipation ratio vs the perforation dimension b with variable fin thickness t for fin thermal conductivity of k = 50 W/ m.°C

The effect of lateral spacing (S_y) at RQF was studied in terms of (b_o), S_x = 1 mm, L = 50 mm and T_b = 100 °C for different values of fin thickness. The results are shown in Figures (6) and (7). The variation of RQF with S_y at various t showed a consistent trend of increasing to a maximum value followed by a decrease. This trend may be explained by the net effects of changing area and thermal resistance of the perforated fin due to lateral spacing (S_y). The increase of RQF in the first part of the curves is due to the fact that the effect of decrease of thermal resistance is larger than that of the decrease of area. While, in second part the effect of decrease of area is larger than that of the decrease of thermal resistance. The maximum value of RQF is occurs at some value of lateral perforation spacing symbolized as (S_{yo}).

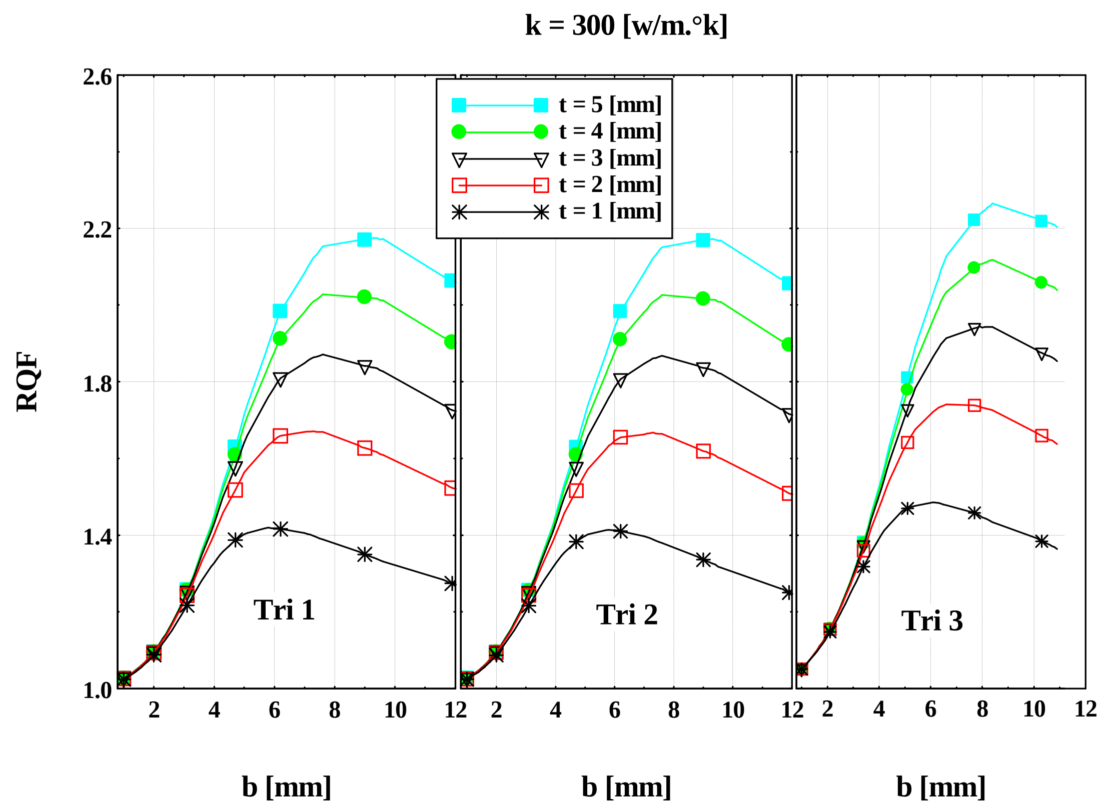

Fig. 5: The perforated fin heat dissipation ratio vs the perforation dimension b with variable fin thickness t for fin thermal conductivity of k = 300 W/ m.° C

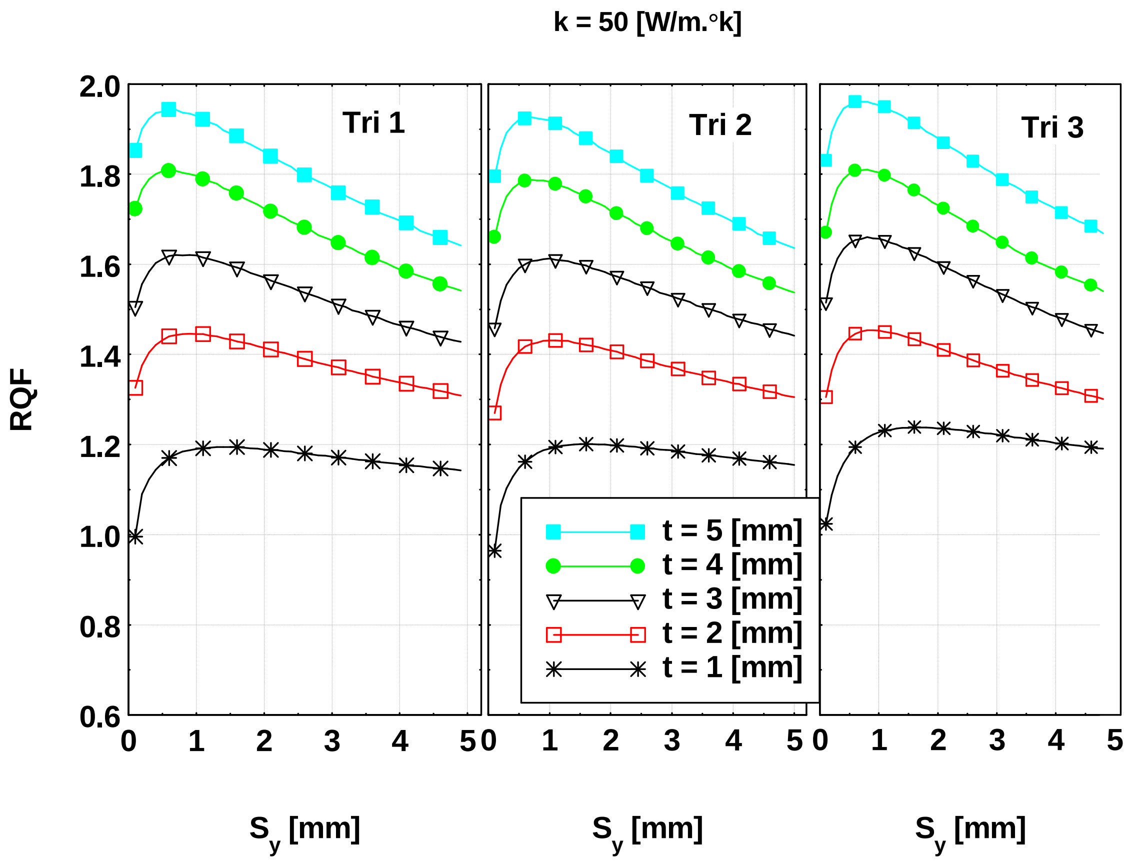

Fig. 6:- The perforated fin heat dissipation ratio vs the perforation lateral spacing with variable fin thickness and its related b_o for fin thermal conductivity of k = 50 W/ m.°C

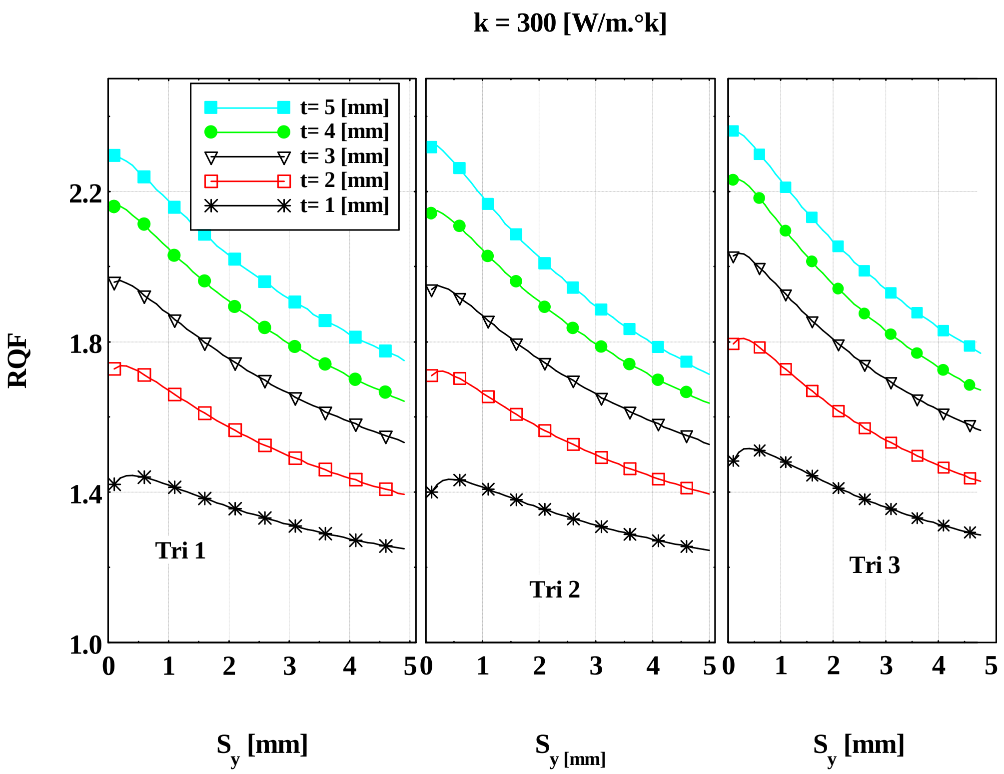

Fig. 7: The perforated fin heat dissipation ratio vs the perforation lateral spacing with variable fin thickness and its related b_o for fin thermal conductivity of k=300 W/m.°C

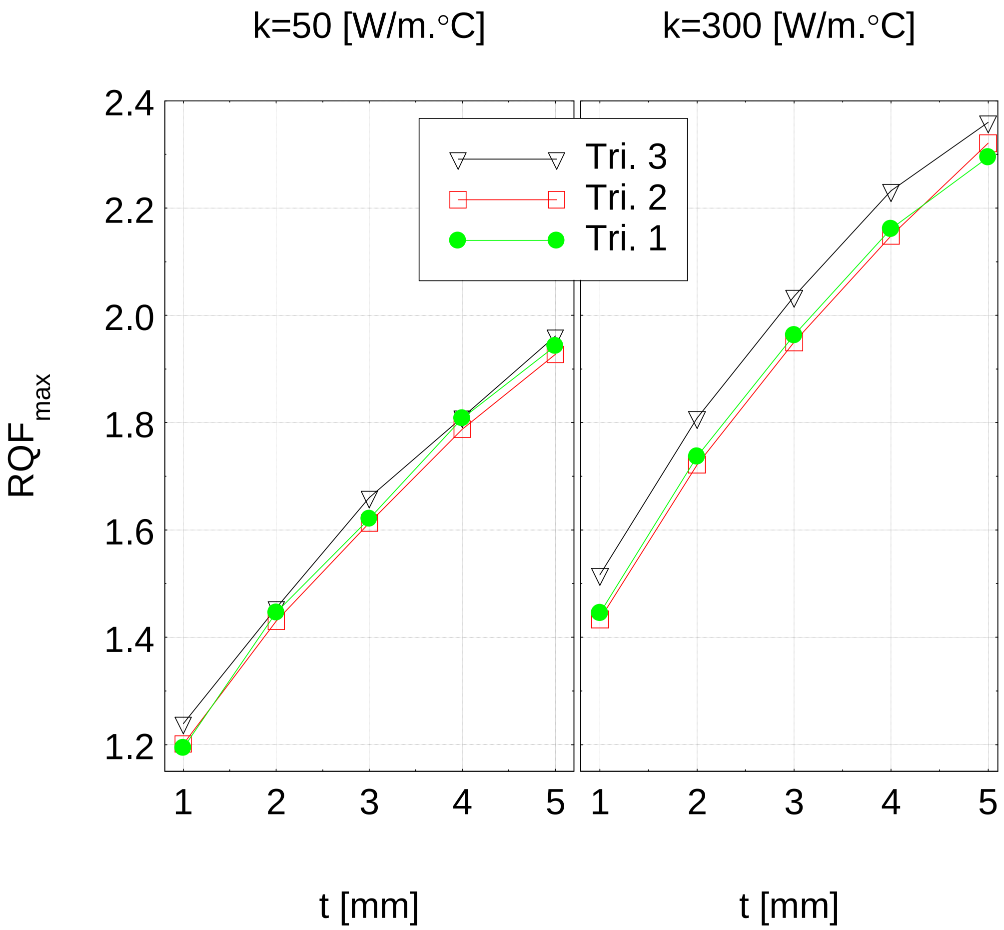

Fig. 8:- Maximum heat dissipation ratios of the perforated fin vs fin thickness with its related b_o, S_{yo}, t, b_o and S_{yo}

To compare the performance of the three studied triangular perforations, the maximum RQF at b_o and S_{yo} was plotted as a function of fin thickness in figure (8). This figure shows that performance of the third triangular perforation is better than the other two cases.

CONCLUSION

1- Introducing triangular perforations to fin body increases surface area and heat dissipation for a given range of the perforation dimension.
2- For certain values of perforation dimension, the perforated fin can enhance heat transfer. The magnitude of enhancement proportional to the fin thickness.
3- The gain in heat dissipation rate for the perforated fin is a strong function of both, the perforation dimension and lateral spacing. This function attains a maximum value at given perforation dimension and spacing, which is may called the optimum perforation dimension b_o, and optimum lateral spacing S_{yo}, respectively.
4- The third triangular perforations is better than the other two cases.

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