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Optimization of triangular fins with/without longitudinal perforate for thermal performance enhancement†

A. Shadlaghani, M. R. Tavakoli*, M. Farzaneh and M. R. Salimpour
Department of Mechanical Engineering, Isfahan University of Technology, Isfahan 84156-83111, Iran

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Abstract

This study aimed at determining a suitable pattern to allow for a better design of the fins used in heat sinks. Flow was considered laminar and steady, and the studied heat transfer mechanism was forced convection. Considering a fixed fin volume, the shape of fin cross section and its dimensions were optimized to maximize the heat transfer rate in a given physical condition. Numerical results showed that at a constant fin base area, heat transfer rate was higher in a fin with a triangular cross section compared to the fins with rectangular or trapezoidal cross sections. Investigation of optimum dimensional ratio in triangular fins showed that an increased height/thickness ratio enhanced the heat transfer rate. The effect of vertical position of the longitudinal perforations with different cross sections but similar volume ratios on the thermal performance of triangular fins was also examined. Results showed that perforation enhanced the thermal performance of the fins. Perforations with square and circular cross sections had almost identical thermal performances and dissipated more heat compared to those with triangular perforations.

Keywords: Force convection; Heat sink; Longitudinal perforate; Numerical optimization; Triangular fin

1. Introduction

Heat transfer enhancement in heat systems has always been an important step when designing them. One of the most effective methods in heat transfer enhancement is the use of extended surfaces, i.e. fins. The term “extended surface” is used when the conduction inside the body and the convection from boundaries occur simultaneously. The thermal conductivity of fins greatly affects the temperature distribution along the fin and, consequently, the heat transfer rate. The choice of fins for various applications depends on factors such as size, weight, increased pressure drop, and increased convection coefficient. Many studies have been conducted on the use of fins with various shapes to increase the heat transfer rate. Kim et al. [1] provided relationships for the thermal optimization of rectangular fins in the developed laminar flows. Their results were compared with the numerical and experimental methods for natural convection heat transfer. The analytical relationships showed that the optimum fin thickness depends only on the height and the conductivity of the fin, as well as the fluid heat transfer coefficient, and is independent of the Rayleigh number, fluid viscosity, and fin length. Dougan et al. [2] studied a heat sink with different fins which had the same surface area using the numerical convection and radiation heat transfer method and investigated the effects of geometric parameters on the optimal design of the fins. El Sayed [3] experimentally examined the optimum position of a set of rectangular fins. Johnson and Moshfegh [4] conducted different experiments in wind tunnels with different dimensions and studied the heat transfer rate of heat sinks with different geometries that were heated from their base by a constant heat flux. A relationship for the Nusselt number and pressure drop was presented in terms of dimensions of the heat sink and the wind tunnel. Yaghoubi and Velayati [5] studied the convection heat transfer in turbulent flow through a set of cubes mounted on a flat surface and provided relationships for average Nusselt number and the efficiency of the fins. The use of longitudinal and transverse perforates in fins has also been addressed in the field’s literature in order to reduce the weight and increase the heat transfer rate. Shaeri and Yaghoubi [6, 7] numerically simulated the heat transfer enhancement in rectangular fins in laminar and turbulent flows in heat sinks. The effect of longitudinal perforations (in the direction of flow) inside the fins was studied, and it was concluded that perforation increases the performance of the fins. In another study [8], the effect of different arrangements of transverse perforations with rectangular cross sections in rectangular fins was investigated. Ismail et al. [9]

*Corresponding author. Tel.: +98 313 391 9012, Fax.: +98 313 391 2628
E-mail address: mrtavak@cc.iut.ac.ir
† Recommended by Associate Editor Chang Yong Park

numerically studied the effect of cross section of perforations on the increase in the heat transfer rate in rectangular fins and found that perforations with circular cross section had the greatest effect in increasing the heat transfer and reducing the pressure drop. The effects of perforations with circular cross section on an arrangement of cylindrical fins mounted on a smooth surface on the Nusselt number and friction coefficient was experimentally studied by Sahin and Demir [10]. Huang [11] examined the overall heat transfer coefficient of a set of horizontal rectangular fins for air conditioning application using numerical methods. It was concluded that the perforations in the fins increase the overall heat transfer coefficient more than 2-folds. Constructal theory that was first introduced by Bejan [12] is a useful method for fin geometrical optimization. For instance, Si et al. [13] investigated the thermal performance of a heat sink with different multi-stage arrangements of perforations using the concepts of constructal theory. In addition, a large number of researches investigated the thermal performance of pin fins arrays with different arrangements with and without perforate [14, 15].

As it can be observed, the previous works mostly investigated the square fins or pin fins in presence of perforate. Due to the lack of investigations about triangular fins, this study mainly focused on this kind of fins in presence of perforate by investigating the geometrical and hydraulic parameters of fins and different perforates. In other word, this was the first time that the effect of perforates on the thermal performance of triangular fins was investigated. Also, thermal performance of fins with three different cross sections (i.e., triangular, trapezoidal and square) was studied in this paper. Once the best fin design was determined, effects of perforations at different locations, as well as different cross sections of a single perforation, on the thermal performance of the fin were also investigated. The ratio of the perforation volume to fin volume was defined as the volume ratio of φ and was considered 0.05 in all the studied geometries.

2. Governing equation

The governing equations were the equations of continuity, momentum and energy for incompressible steady laminar flow by neglecting body forces:

$$\frac{\partial u_i}{\partial x_i} = 0$$
$$u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \nu \frac{\partial^2 u_i}{\partial x_i^2} \tag{1}$$
$$u\frac{\partial T}{\partial x_j} = \alpha \frac{\partial^2 T}{\partial x_i^2}.$$

Since the heat transfer in extended surfaces, such as fins, is a combination of conduction and convection, the temperature distribution in fins was calculated using the Fourier's law and Newton's law of cooling.

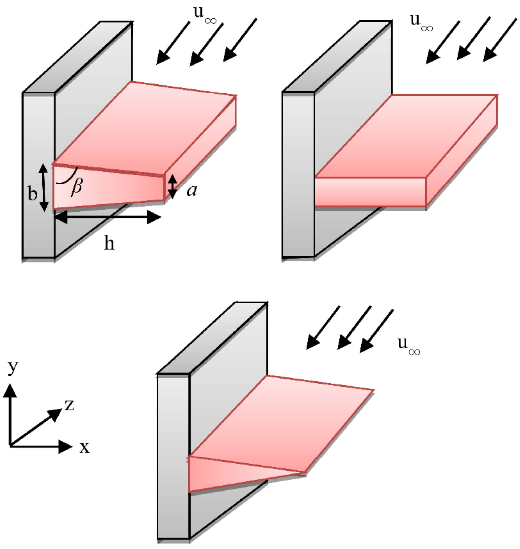

Fig. 1. Investigated fins with various cross section.

3. Problem description

In order to cool electronic equipment, an array of air-cooled finned channels is placed between the heat-generating electrical components. Heat sinks consist of a number of parallel fins of similar geometry. The fins are generally triangular, trapezoidal, and rectangular. Fig. 1 shows the studied fins with different cross sections. As can be seen in Fig. 1, at a given fin volume, the cross section changes as the angle β varies. The optimum angle was determined by examining the increase in the heat transfer rate achieved though changing the angle β. Clearly, if $\beta = 90$, the cross section of the fin becomes rectangular. The maximum change in angle β occurs when $a = 0$, which means a triangular cross section. The effect of fin dimensions (height and width) was examined though determining the optimum cross section. The optimum dimensions at a constant volume were then calculated.

The effect of the location and cross section shape of the longitudinal perforation was then investigated. Perforations in extended surfaces have always been one of the most important issues in this field since it reduces the heat sink weight in manufacturing process. Three perforations with square, circular and triangle cross section were modeled to bring insight into the optimum shape of perforations.

4. Numerical analysis

Since the Richardson number of the air flowing through the heat sink was much less than one ($Gr/\mathrm{Re}^2 \ll 1$), forced convection heat transfer mechanism was considered here. The Reynolds number was calculated based on the thickness of the fin base $(\mathrm{Re} = \frac{VD}{\nu})$. The studied aluminum fins had a conductivity of 202 w/mk. The dimensions of the solution domain were selected so that it does not affect the results and the flow

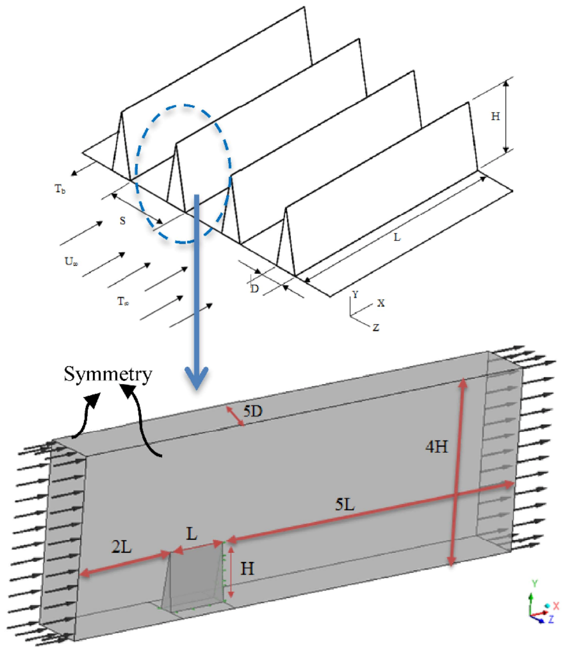

Fig. 2. The cubic solution domain and its dimensions.

patterns, such as vortices [16]. Given the symmetry and the similarity of the fins (Fig. 2), a single fin was investigated, and the results can be generalized to the whole heat sink by definition of the appropriate boundary conditions. A cubic domain (Fig. 2) was considered around the fin after examining different dimensions for solution domain. The hydrodynamic and thermal boundary conditions were considered as follows:

- Inlet: laminar flow with a prescribed velocity and a constant temperature of 25°C.
- Outlet: Zero static pressure.
- Walls: no-slip wall condition.
- Lower wall of the solution domain: the wall was divided into three sections. The middle section was subjected to a constant temperature of 70°C, while the two other sections were adiabatic. The no-slip hydrodynamic boundary condition was considered for all three sections.
- Upper wall of the solution domain: free slip wall condition.
- Sidewalls of the solution domain: symmetry boundary condition was selected since only one fin was modeled.

Navier-Stokes and energy equations were discretized through the finite volume method using Ansys CFX commercial package. The convection term of the equations was solved using the high resolution scheme. In order to achieve the acceptable results, the convergence criteria were set at 10^{-5} for steady momentum and energy equations.

4.1 Computational grid

Two important dimensionless parameters of Nusselt number and friction drag coefficient were used to evaluate the performance of the fins. They are defined as follows:

Table 1. Typical grid study for rectangular fin.

NO.	Element number	$\overline{Nu}$	$\overline{C_f}$
1	798881	2.26	0.044
2	975811	2.27	0.045
3	1223284	2.23	0.043
4	1461299	2.21	0.043

$$\bar{c}_f = \frac{\bar{\tau}_w}{0.5\rho u_\infty^2} \tag{2}$$

$$Nu = \frac{\bar{h}b}{k_f}. \tag{3}$$

Nusselt number was calculated based on the fin base thickness. Average convection coefficient was calculated based on the volumetric temperature of the free flow over the fin as follows:

$$\bar{h} = \frac{q_f}{A_f(T_b - T_\infty)}. \tag{4}$$

One of the most important steps in numerical simulations is to investigate the sensitivity of the results to the number and size of the generated elements. Four different grids were examined to determine the independence of the numerical solution from the solution domain mesh. To check the quality of the grid, it was refined until changes in the two parameters of friction coefficient and Nusselt number were negligible. Table 1 represents the changes in these dimensionless numbers for a rectangular fin ($\beta = 90°$) in a velocity of 0.4 m/s.

As can be seen in Table 1, the maximum variations in the friction coefficient and Nusselt number reached less than 1% as the grid was refined. The same procedure was performed in other simulations. The third grid was able to yield results with lower computational cost compared to the fourth grid, while it was acceptably accuracy; therefore, it was selected for further simulations.

4.2 Verification

In order to verify the present numerical method, simulation was firstly performed for a solid triangular fin array experimented in Ref. [17]. The set-up consisted of rectangular duct (120*70 mm^2) in vertical position holding the nine aluminum fin arrays, which were longitudinally installed in the upper part of the duct (200 mm from the top). The base plate had the dimension of 110 *mm* in width and 100 *mm* in height, which was exposed to constant heat flux from the bottom. Range of velocities was considered between 0.1 to 0.78 m/s. Nusselt numbers calculated in present study shown in Fig. 3 were compared with those reported by Ref. [17]. Note that, scale of axes is logarithmic as it used in the experimental work. Comparisons demonstrate the accuracy of numerical simulations in

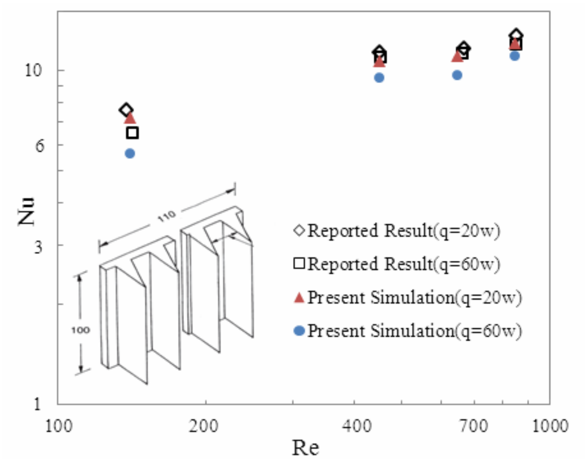

Fig. 3. Verification of results by experimental report [17].

prediction of thermal properties of heat sinks. Therefore, numerical approach including solution domain, grids, and boundary conditions were considered as mentioned earlier, for the rest of simulations.

5. Results and discussion

The simulation results of perforated and non-perforated fins in different conditions are presented in this section.

5.1 Non-perforated fins

This section presents the simulation results of rectangular, trapezoidal and triangular fins at a constant volume of 1440 mm^3. The base areas of all fins were the same and equal to 52×4 mm^2. For the fin volume to be constant, the fin height was changed as the angle β varied. Variations in the friction drag coefficient and Nusselt number for different angles of β at a velocity of 0.4 m/s are presented in Table 2.

As can be seen in Table 2, a decreased angle β (conversion of rectangular area to triangular area) increased the heat transfer rate due to the increased heat transfer surface. Increased triangular fin height resulted in an increase in the heat transfer surface and, consequently, the heat transfer rate. Since it was determined that the triangular fins had a better performance than rectangular or trapezoidal fins, the geometric parameters of its cross section were optimized. Considering a fixed cross-sectional area, different ratios (H/b ≥ 1) were examined to approximate the optimal dimension of the cross section. Figs. 4 and 5 show the effect of the ratio *H/b* on the heat transfer rate and friction coefficient at different Reynolds numbers.

As can be seen from these figures, an increased height to thickness ratio increased the heat transfer surface and, consequently, friction coefficient and heat transfer rate. The right and optimal choice of geometric parameters of cross section increased the convection heat transfer by approximately 50%, while the friction coefficient increased only about 12%. Therefore, it can be concluded that in triangular fins, given the design limitations, the greater the height of the fin, the higher the heat transfer rate.

In Fact, an increased fin height caused a lower volume of the device operate in the maximum temperature and improved the thermal performance of the fin, which increased the heat transfer from the fins.

Table 2. Variations in heat transfer rate, friction drag coefficient, and Nusselt number for different angles β.

β	$\overline{Nu}$	$\overline{C_f}$	Q(w)
90	2.23	0.043	0.53
87	2.5	0.047	0.73
85.5	2.67	0.055	0.86
84	2.75	0.061	1.01

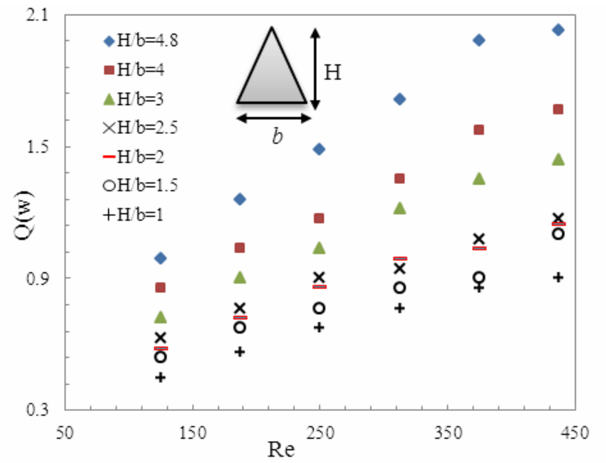

Fig. 4. Variations in thermal potential of the fin by the different dimensions of cross section at different Reynolds numbers.

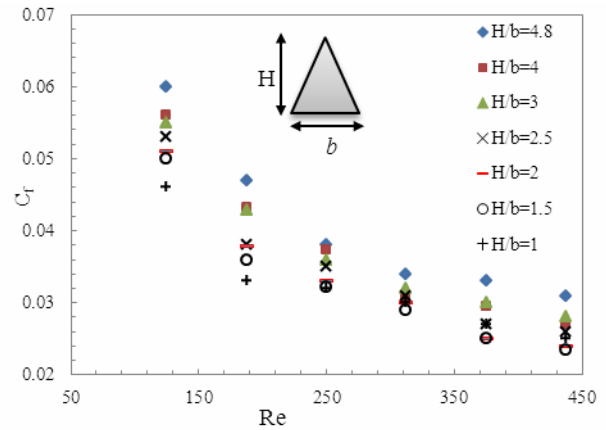

Fig. 5. Variations in friction coefficient by the different dimensions of cross section at different Reynolds numbers.

5.2 Perforated fins

This section discusses the effects of longitudinal perforations with circular, square, and triangular cross sections with similar volume ratio of $\varphi = 0.05$ in a triangular fin. The cross-

Table 3. Variations in Nusselt number by different heights of the center of mass of the perforation.

H_c/H	Q(w)		
	Triangular	Circular	Square
0.1	1.252	1.273	1.272
0.2	1.231	1.278	1.275
0.3	1.265	1.295	1.286
0.4	1.228	1.282	1.277
0.5	1.241	1.278	1.252

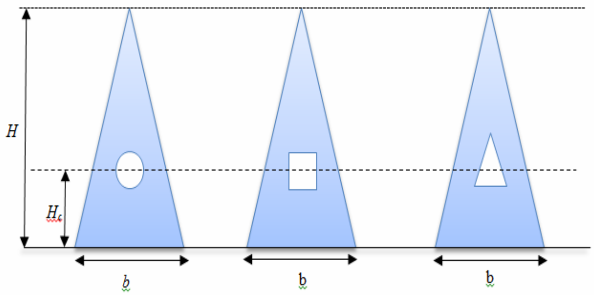

Fig. 6. Schema of triangular fin along with the different perforations.

sectional area of perforations were similar and equal to 209 mm^2, while only the lateral surfaces of the perforations were different. Perforations were examined in a triangular fin with the ratio of $H/b = 3$. Fig. 6 shows the arrangement of perforations inside the triangular fins. The values of heat rate for a velocity of 1 m/s versus the ratio of height of the center of mass of the perforation to the total fin height are presented in Table 3. Results presented in Table 3 show that the heat rate is larger in perforated fins compared to the non-perforated fins (shown in Fig. 4). The main reason for this is that perforations increase the heat transfer rate by increasing the solid-air contact area, and consequently reducing the fin temperature. In addition, as will be noted later, the efficiency of the perforated fins was significantly increased compared to the non-perforated fins.

The maximum heat transfer rate occurred at $H_C / H = 0.3$ (Table 3) because the conduction and convection resistances corresponding to the material thickness (distance between base plate to the perforate location) and fluid velocity had a different behavior along the fin height. In other words, as the perforation height increases, the convection resistance decreases due to the increased velocity of the fluid passing the perforation. On the other hand, the conduction resistance increases due to the increased thickness and reduced cross-section perpendicular to the conduction heat transfer. It is clear that heat transfer is maximized when the sum of the two resistances is minimized, which occurs in a specific height ratio.

Based on the results of this study, the following correlations are developed to predict the heat transfer rate in terms of perforation location

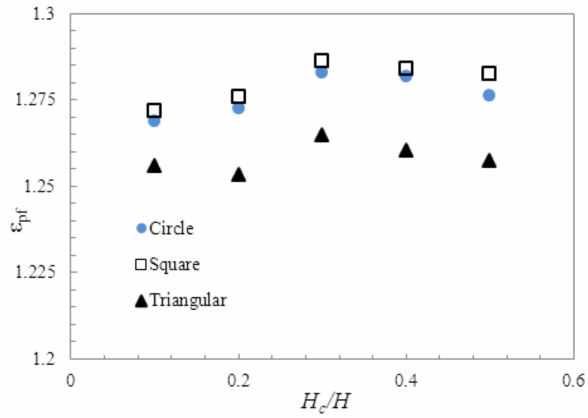

Fig. 7. Variations of ε_{pf} in different perforation shapes versus the location of the center of mass of the perforation.

$$\begin{aligned}
Q\big|_{Tri} &= 33.75(H_C/H)^4 - 40.75(H_C/H)^3 + \\
&16.613(H_C/H)^2 - 2.5875(H_C/H) + 1.403 \\
Q\big|_{Cir} &= -2(H_C/H)^3 + 1.2571(H_C/H)^2 - \\
&0.1843(H_C/H) + 1.2794 \\
Q\big|_{Squ} &= 102.92(H_C/H)^4 - 123.92(H_C/H)^3 + \\
&51.371(H_C/H)^2 - 8.4908(H_C/H) + 1.701 .
\end{aligned} \tag{5}$$

Perforations are intended to reduce the mass of the fin and increase heat transfer rate though increasing the heat transfer surface. The efficiency of perforated fins is an important parameter in determining its performance compared to the non-perforated fins. It is defined as

$$\varepsilon_{pf} = \frac{Q_{pf} - Q_{sf}}{Q_{sf}} \tag{6}$$

where Q_{sf} is the heat transfer from the non-perforated fin, Q_{pf} is the heat transfer from the perforated fin, and ε_{pf} is the efficiency of the perforated fin. Fig. 6 shows ε_{pf} of the triangular fin with different perforation shapes and different distances from the base surface.

As can be seen in Fig. 7, at a given perforation volume, the perforation with a square cross section was more efficient than those with circular or triangular cross sections with a negligible difference between square and circular perforates. The difference between the heat transfer rates in different perforations is due to the larger contact surface of the square perforations with the fluid flowing inside compared with other perforations at a constant volume.

Another important factor used for thermal evaluation of the perforated fins compared to non-perforated fins is the comparison of dissipated heat per unit fin volume. Accordingly, the parameter η is defined as Eq. (7):

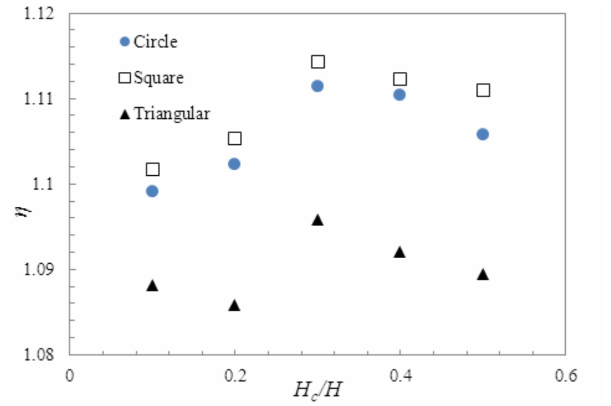

Fig. 8. Variations in η for the three perforations studied in different heights.

$$\eta = \frac{\frac{Q_{pf}}{\forall_{pf}}}{\frac{Q_{sf}}{\forall_{sf}}} = \frac{Q_{pf}\forall_{sf}}{Q_{sf}\forall_{pf}} \tag{7}$$

where $\forall_{sf}$ is the non-perforated fin volume, $\forall_{pf}$ is the perforated fin volume, and η is the ratio of dissipated heat per unit fin volume in the non-perforated fin to the perforated fin.

Fig. 8 shows the variations in η for three perforations studied in different heights from the base surface. Fig. 8 better illustrates the differences in the heat transfer from fins for different perforations. The transferred heat per unit volume of the fin with square perforation was greater than the two other perforations. Here, too, greater heat transfer surface in square perforation increased the heat transferred per unit fin volume. Figs. 6 and 7 indicate that the thermal performance of the perforated fin was maximized at the distance of $H_C / H = 0.3$.

Another important parameter in the evaluation of fins' performance is the ratio of thermal potential of the perforated fin to that of the non-perforated fin, which is defined by Eq. (8):

$$\frac{h_{pf}A_{pf}}{h_{sf}A_{sf}} = \frac{\frac{Q_{pf}}{T_s - T_\infty}}{\frac{Q_{sf}}{T_s - T_\infty}} . \tag{8}$$

Fig. 9 shows the variations in thermal potential for the two cases of perforated and non-perforated fins. The square and circular perforations yielded almost similar results, where the maximum thermal potential occurred around the height ratio of 0.3.

Thermal enhancement factor is another relevant factor used to simultaneously investigate the friction effects and heat transfer coefficient of fins. It is defined by Eq. (9):

$$TEF = \frac{Nu_{pf}}{Nu_{sf}}\left(\frac{c_{f(pf)}}{c_{f(sf)}}\right)^{-1/3} . \tag{9}$$

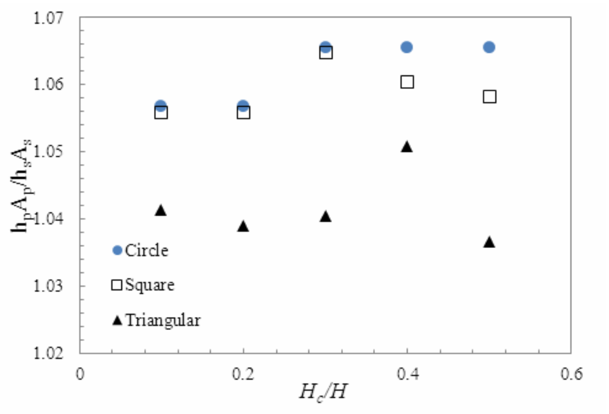

Fig. 9. Variations in thermal potential for the two cases of perforated and non-perforated fins for different perforation shapes.

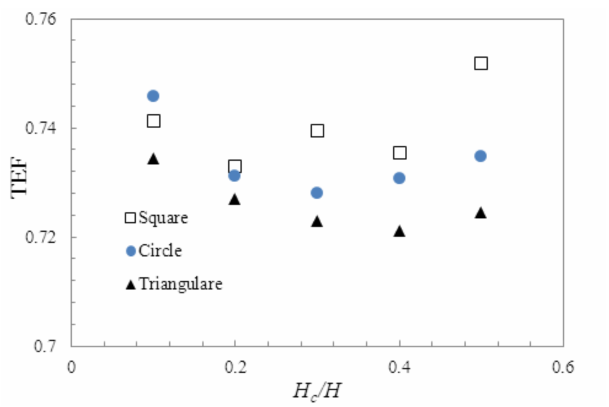

Fig. 10. Variations in TEF by the perforation height from the base surface.

Fig. 10 shows the diagram of thermal enhancement factor, where the fin with square perforation had a relatively better performance compared to other fins.

Perforation increases the flow pressure drop. An optimum design aims to maximize the heat transfer rate while minimizing the pressure drop. The parameter ζ was used to evaluate the effect of pressure drop on heat transfer rate in perforated fins compared to the non-perforated fins.

It is defined by Eq. (10):

$$\zeta = \frac{\frac{Q_{pf}}{\Delta p_{pf}}}{\frac{Q_{sf}}{\Delta p_{sf}}} = \frac{Q_{pf}\Delta P_{sf}}{Q_{sf}\Delta P_{pf}} \tag{10}$$

where ΔP_{sf} is the non-perforated pressure drop, ΔP_{pf} is the perforated fin pressure drop. Fig. 11 shows the variations in ζ versus the distance ratio of perforates. Pressure drop was higher in the square and triangular perforations compared to the circular one due to the presence of sharp corners and vortex flow in these areas and the resulting flow deceleration in the perforation. At the end, temperature distribution on the

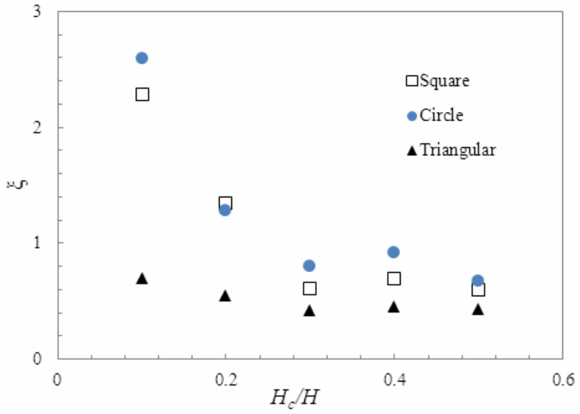

Fig. 11. Variations in the heat transfer rate versus pressure drop for perforated fins.

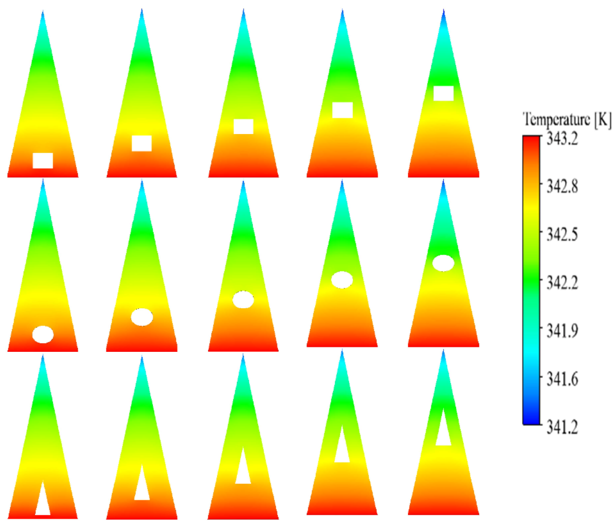

Fig. 12. Temperature distribution on the front face of fins with different perforates.

front surface of fins was illustrated in Fig. 12 for flow velocity of 1.4 m/s. Contours show that the temperature is reduced more around the perforates.

6. Conclusion

This paper comprehensively discussed dimensional optimization of the fins used in heat sinks through calculating the heat transfer rate considering a constant fin volume. Rectangular, trapezoidal, and triangular fins were first examined. Results showed that triangular fins had a better thermal performance due to their increased heat transfer surface. Afterward, the dimensions of the triangular fin cross section were investigated. Simulations showed that an increased ratio of fin height to fin thickness maximized the heat transfer rate. Another method used to enhance the convection heat transfer is the longitudinal perforation in fins. An important issue while using perforation in fin design is that although they increase the heat transfer rate by increasing the fluid-fin contact surface, they tend to decrease the heat transfer rate by decelerating the fluid due to the increased pressure drop in the perforation. Therefore, determination of the optimal perforation location and the optimal perforation cross section is crucial. Result showed that at a fixed volume, rectangular perforation increased the heat transfer rate more than the rectangular and circular perforations. The optimal location of the perforations was nearly in ratio of 0.3 from the fin base. In addition to the thermal parameters of perforated fins, the pressure drop and friction caused by the perforations were also shown to be very influential in the total performance of the perforated fins.

Nomenclature

Gr	: Grashof number
Q	: Heat transfer rate from fin surface
h_c	: Perforation height
α	: Fluid thermal diffusivity
μ	: Fluid kinematic viscosity
ρ	: Density

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Arash Shadlaghani received his M.Sc. in Mechanical Engineering from Shahrood University of Technology and Isfahan University of Technology, Iran, in 2010 and 2013, respectively. His major research areas include the use of experimental and computational fluid dynamic in thermofluids phenomena and their optimization.

Mohammad Reza Tavakoli is an assistant professor in Department of Mechanical Engineering. He received his Ph.D. in Mechanical Engineering from University of Maryland (College Park), USA in 2010. His research interests include multi phase flow, experimental and numerical heat transfer and porous media.