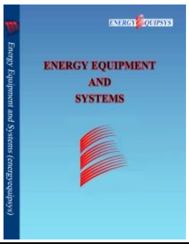
Energy Equip. Sys./ Vol. 5/ No.2/ June 2017/147-156

ENERGY EQUIPSYS

Energy Equipment and Systems

http://energyequipsys.ut.ac.ir
www.energyequipsys.com



Numerical simulation of laminar convection heat transfer from an array of circular perforated fins

Authors

Shekoufeh Mohammadi [a]
Afshin Ahmadi Nadooshan [a*]
Morteza Bayareh [a]

[a] Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran

Article history:

Received : 20 December 2016
Accepted : 21 January 2017

ABSTRACT

The present paper reports the laminar fluid flow and heat transfer of a heated array of circular-perforated and solid fins mounted over a flat surface using the finite-volume method. One to four circular cross-sectional perforations are made along the length of the fins. The SIMPLE algorithm is used for pressure-velocity coupling and the second order upwind technique is employed to discretize the governing equations. The simulations are done for a range of Reynolds number, 100 < Re < 350, corresponds to the flow velocity of 0.37-1.28 m/s. The average Nusselt number, the average friction factor coefficient, and the perforated-fin effectiveness are calculated. The results demonstrated that for any Reynolds number, the heat transfer decreases and the friction factor increases with the number of perforations. It was observed that the fin with circular perforation had a higher effectiveness in comparison with the rectangular-perforated fins.

Keywords: Laminar Flow, Perforated Fin, Nusselt Number, Reynolds Number.

1. Introduction

The enhancement of heat transfer rate is very important as many industrial applications require rapid heat removal from heated surfaces. Heat transfer techniques that are commonly used in different industrial sectors include passive or active methods, or a combination of these methods. Passive techniques do not need power, but active methods require external energy sources [1].

Heat exchangers, which are one of the most practical engineering devices, are progressively requiring increasing amounts of energy. Extended surfaces or fins are examples of the passive techniques that are used in heat exchangers to enhance the heat transfer rate. Rectangular fins are commonly used owing to their low cost and simplicity of manufacturing.

Fin optimization is necessary to reduce the cost of production by maintaining better performance. There are several suggestions to optimize these types of fins, which consider factors such as perforation, porosity, and groove [2]. Perforated fins have been used as one of the most useful optimization methods. The most significant challenge in designing perforated fins is in having to achieve high heat transfer and a reduction of fin weight, simultaneously.

Bilen et al. [3] investigated the geometrical effect of rectangular blocks on heat transfer from a surface by using a Taguchi approach. They found that the heat transfer increased with an increase in the Reynolds number as well as by changing the angle of the blocks. EI-Sayed et al. [4] performed an experimental investigation to study the heat transfer from an array of rectangular fins for different flow directions. They observed that the highest rate of heat transfer occurs in the case of parallel-flow orientation. Baskaya et al. [5] studied the heat transfer between the flow and a horizontal fin array, and they revealed that the

*Corresponding author: Afshin Ahmadi Nadooshan
Address: Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran
E-mail address: ahmadi@eng.sku.ac.ir

overall heat transfer increased by increasing the height of the fins and by decreasing by their lengths. Dijek et al. [6] numerically investigated the subsonic laminar three-dimensional flow around a block that was located on the floor of a periodic channel. They identified five flow features at different Reynolds numbers. In addition, they also calculated a correlation between the position of separation and the attachment point with the Reynolds number. Kundu et al. [7] performed an analytical study on the optimum design of a porous fin with different profiles. They demonstrated that the heat transfer for the exponential profile is slightly higher than the convex profile; the exponential profile's heat transfer is, however, much higher than the rectangular profile. In addition, they also showed that the rate of heat transfer increased with an increase in porosity. Kundu and Lee [8] examined an analysis to determine the minimum shape of porous fins in the presence of radiation and convection heat transfers. Unlike in the case of solid fins, they found that the radiation effect increases the performance of the porous fins. It has been demonstrated that the optimum shape of a porous fin depends on the thermophysical properties of the fluid and fin porosity. Velayati and Yaghoubi [9] investigated turbulent fluid flow around a series of rectangular bluff fins. They showed the impact of the blockage ratio and the Reynolds number on the average Nusselt number, and they calculated a correlation between the Nusselt and the Reynolds numbers for different blockage ratios.

An experimental work was performed by Molki and Hashemi-Esfahanian [10] to determine the heat transfer of a perforated baffle blockage that was attached to the wall. It was found that the perforation improves the thermal performance for the Reynolds number ranged between 5000 and 30000. Sara et al. [11] investigated the geometric parameters on the enhancement of the heat transfer of perforated rectangular blocks and reported that the enhancement increased by increasing the degree of perforations. They also showed that the thermal performance of the blocks with inclined perforations was higher than those with straight perforations. Heat transfer in a heat exchanger with square cross-sectional perforated fins was investigated experimentally by Sahin and Demir [12].

They found that the rate of heat transfer is affected by the Reynolds number, the spaces between the fins, and fin height. Shaeri and Yaghoubi [13] examined a three-dimensional study of laminar flow around an array of perforated fins. It was revealed that the position and the shape of the generated vortexes behind the fins depend on the geometry of the perforations. They showed that the enhancement of heat transfer is considerable, especially at higher Reynolds numbers. They developed a correlation between the perforated-fin effectiveness, the Reynolds number, and the porosity.

The two-dimensional experimental investigation was presented by Igarashi and Mayumi [14]. They considered the fluid flow and the heat transfer around a rectangular fin for a different angle of attack and calculated a correlation between the Nusselt and the Reynolds numbers. Karabacak and Yakar [15] considered the effect of the perforations on the heat transfers from the perforated-finned heat exchangers. They showed that the heat transfer of the perforated-finned cases was 12% and 6% higher than that of the imperforated positions at the Reynolds numbers above and below the critical value, respectively. Huang et al. [16] studied the enhancement of natural convection heat transfer for different kinds of perforated-fin arrays. It is revealed that the fin-based perforation increases the heat transfer of large-fin arrays. In addition, they showed that the heat transfer performance of fins increases with the total perforation length.

2. Governing Equations and the Numerical Method

The three-dimensional laminar fluid flow around an array of perforated fins is presented. Since the flow is steady state incompressible, the equations of continuity, momentum, and energy are, respectively, written as follows:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

$$u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \nu \frac{\partial^2 u_i}{\partial x_i^2} \tag{2}$$

$$u_j \frac{\partial T}{\partial x_j} = \alpha \frac{\partial^2 T}{\partial x_i^2} \tag{3}$$

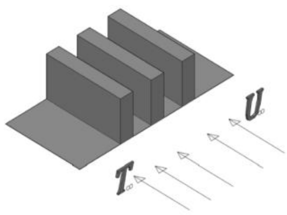
In addition, Fourier's heat conduction equation is solved for the solid phase. The Navier–Stokes equations are solved by a control volume approach. The SIMPLE algorithm is used for velocity-pressure coupling. The second order upwind scheme is used to discretize the momentum and the energy equations.

3. Problem Set up and Boundary Conditions

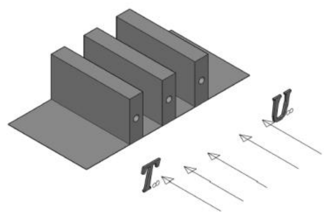
The solid and the perforated-fin arrays simulated in present study are illustrated in Fig. 1. Air with inlet velocity U_∞ blows across the fin array, which is assumed to be made of aluminum with a thermal conductivity of 202.4 W/mK. The Reynolds number ranges between 100 and 350, and corresponds to a flow velocity of 0.37–1.28 m/s. The length, thickness, and height of the fins are 24, 4, and 12 mm, respectively. The circular perforations have a diameter equal to 1.84 mm. The porosity, φ, is defined as the ratio of void volume to the total volume of the fin. According to the diameter of circular perforations, a porosity of 0.0555–0.111 corresponds with 1–4 circular holes. Since the Richardson number (Ri = Gr/Re^2) indicates the free and forced convection heat transfer, the Richardson number is calculated based on the fin height. A very low value of the Richardson number ($Ri < 0.1$) confirmed that forced convection is the dominant form of heat transfer in this study.

Figure 2 shows the computational domain. The simulations can be performed for one fin instead of the fin array; this is owing to the flow symmetry. Uniform flow is assumed for the plane *abcd* as the inlet surface, such that $u_\infty = u_{in}$, $T_{in} = T_\infty$, and the two other velocity components in the *y*- and the *z*-directions are equal to zero. The surface named *ijkl* is the output and the gradient of all the variables in the *x*-direction is zero. The planes *abji* and *dckl* have symmetry boundary conditions. The wall surface of *efgh* has a constant temperature of 70˚C. In addition, the wall surfaces of *fgli* and *adeh* are adiabatic. The ambient air temperature is 25 ˚C. In this study, the effect of radiation heat transfer can be ignored because the thermal radiative heat transfer is less than 8% of the total heat transfer.

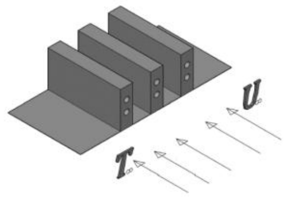
To avoid boundary effects, an appropriate computational domain is employed in a manner in which the domain extends 30D



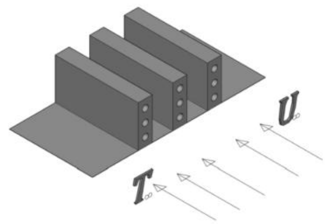
a) Array of solid fins



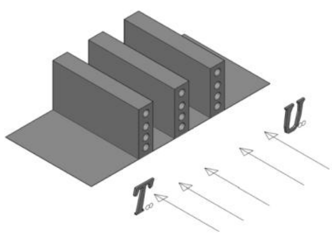
b) Array of fins with 1 perforation,φ= 0.0555



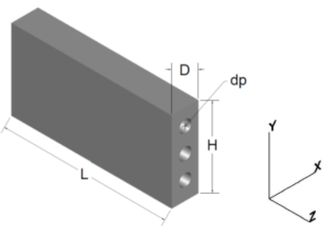
c) Array of fins with 2 perforations, φ = 0.111



d) Array of fins with 3 perforations, φ = 0.1665



e) Array of fins with 4 perforations, φ = 0.222



f) perforated fin geometry

Fig.1. The solid and perforated-fin arrays studied in the present work

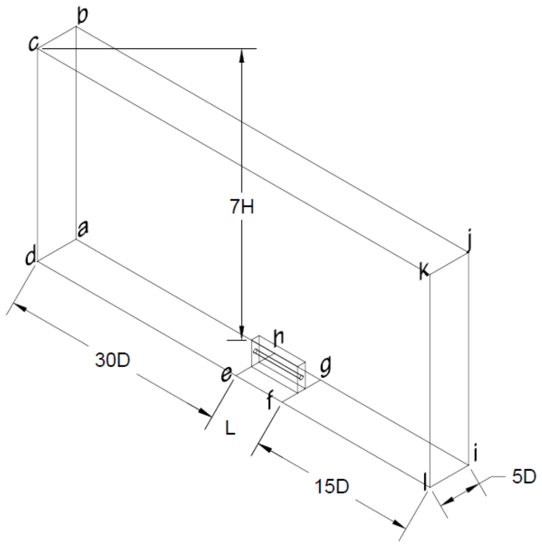


Fig.2. Computational domain for solid and perforated fins

downstream, $15D$ upstream, $7H$ in the y-direction, and $5D$ in the z-direction. It is found that the domain with $40D$ downstream, $20D$ upstream, $12H$ in the y-direction, and $10D$ in the z-direction leads to 0.3% change in the average Nusselt number.

4. Results and Discussion

The three-dimensional laminar fluid flow around an array of rectangular fins with circular perforations is investigated. The porosity effect on the heat transfer rate, the pressure drop, and the flow field around the perforated fin are presented. In the next two sections, the results of the grid-independent study and validation are discussed.

4.1. Grid-independent study

It is necessary to show that the results are not affected by the number of elements. It should be noted that to capture the recirculation region and the attachment point, the fin and the perforated surfaces should have the finest grid resolution.

The effect of different numbers of grids on the average Nusselt number and the average friction coefficient is presented in Table 1. The results show that for an independency of issue from the number of grid points, a grid resolution with 70×80×210 points in the x-, y- and z-directions, respectively, are acceptable.

The corresponding grid configuration for a typical perforated fin is illustrated in Fig. 3.

Table 1. Grid independency of the results for perforated fin, Re_D= 250

Grid	$\overline{Nu}$	$\overline{C}_f$
64×56×168	2.9	0.261
70×80×210	2.877	0.254
88×98×262	2.89	0.252

Fig. 3. Grid configuration for a fin with one and two perforations

4.2. Validation

To validate the present simulations, three-dimensional numerical studies conducted by Shaeri and Yaghoubi [13] on heat transfer and fluid flow around an array of rectangular-perforated fins are used. In this respect, two fins, with one and two rectangular perforations, at the Reynolds numbers in the range $100 < Re < 350$ are considered. These results are displayed in Fig. 4(a) and show very good agreement for the range of the considered Reynolds numbers.

In addition, the results of the numerical scheme are compared with the experimental results of Igarashi and Mayumi [14] who studied the heat transfer around a rectangular cylinder, in which aspect ratio and the blockage ratio were equal to 5 and 0.066, respectively. The average Nusselt number for different Reynolds numbers are shown in Fig. 4(b). This figure shows that our results are in very good agreement with the experimental results.

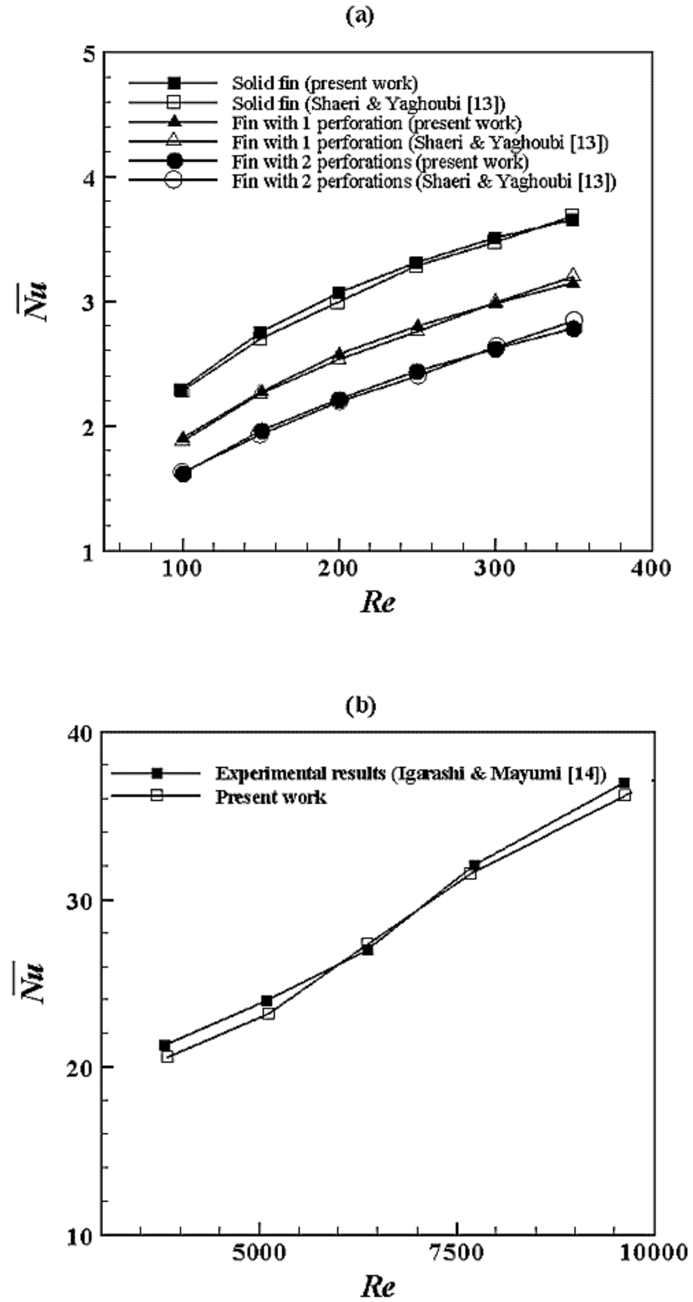


Fig. 4. Comparison of the average Nusselt number versus the Reynolds number, a) heat transfer and fluid flow around an array of rectangular-perforated fins, and b) heat transfer and fluid flow around a rectangular cylinder

4.3. Fluid Flow

Since the flow field for the present work is three-dimensional, the computational domain is divided into two cross sections of the fin in the direction of y, where $y = 0.5H$ and $y = 0.9H$. The fluid streamlines around the solid and the perforated fins at the two sections are illustrated in Fig. 5. The Reynolds number is equal to 250. It is observed that the flow field around both cases is symmetrical and similar to the case of rectangular-perforated fins [13]. Two similar vortexes are generated at the rear portion of the fin. It is notable that the rear wake size decreases with distance from the fin base. This is a result of the lower fluid viscosity effect near the fin tip. The average friction factor coefficient is calculated as follows:

$$\bar{C_f} = \frac{\overline{\tau_w}}{1/2\rho u^2} \tag{4}$$

where $\overline{\tau_w}$ is the average wall shear stress over the fin side surface. Variation of the average friction factor coefficient as a function of the Reynolds number for different kinds of perforated fins is illustrated in Fig. 6.

In all these cases, include the solid fin, the average friction factor coefficient is decreased by the Reynolds number. It is observed that at any Reynolds number, the friction factor decreases with the porosity degree; therefore, the solid fin has the largest value of $\bar{C_f}$ and the fin with four perforations has the lowest value.

The effect of porosity is dominant at lower Reynolds numbers. In addition, the case of the fin with three circular perforations is compared with the fin with three rectangular perforations, where $\varphi = 0.0555$. For any Reynolds number, $\bar{C_f}$ is larger for the rectangular-perforated fin than the circular-perforated fin.

4.4. Heat Transfer

To study the perforation effect on the rate of heat transfer, the Nusselt number is defined as follows:

$$\overline{Nu} = \frac{\bar{h}d}{\kappa} \tag{5}$$

,where $\bar{h}$ is the average convection heat transfer. Figure 7 shows the average Nusselt number versus the Reynolds number. It is observed that $\overline{Nu}$ increases with the Reynolds number. For any Reynolds number, the average Nusselt number decreases as the amount of perforation increases.

This is due to a reduction in flow velocity for the case of perforated fins. In addition, one can see that the amount of average Nusselt number for the circular perforated fin is higher than that for the rectangular one. This is because of the difference in the heat entrance region in perforation holes. Equation 6 expresses the entrance length in a tube. Using this equation, the entrance length for a tube with circular and rectangular cross section is equal $x_{h,c}$=7.4, $x_{h,r}$=4.4, respectively.

$$\frac{x}{D} = 0.05\, Re_D Pr \tag{6}$$

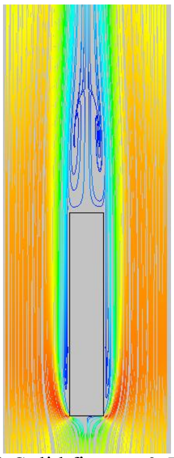
It shows that the length of fully developed zone in the circular perforation is higher than that in the rectangular one. Therefore, the flow in the circular perforated fin leads to higher heat transfer coefficient in comparison with the rectangular perforated fin. To evaluate the thermal performance of a perforated fin, perforated fin effectiveness is defined as the Eq.(7):

$$\varepsilon_{pf} = \frac{Q_{pf} - Q_{sf}}{Q_{sf}} * 100 \tag{7}$$

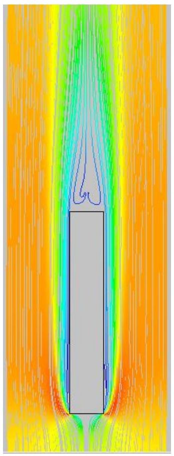
Figure 8 displays perforated fin effectiveness as a function of Reynolds number for all cases considered. Effectiveness increases by the increase of the amount of perforations and Reynolds number. As the amount of perforation increases, the fluid flow increases and it results in more heat transfer. The impact factor of the circular perforated fin is higher than the rectangular one due to higher heat transfer coefficient for the circular perforated fin.

In designing the heat exchanger to achieve better performance, it is important to ensure minimum weight. The parameter M is defined as the ratio of the heat transfer rate per unit volume of a perforated fin to the heat transfer rate per unit volume of a solid fin (Eq. 8). If M > 1, we can conclude that the perforated fin has a higher thermal performance in comparison to the solid fin.

$$M = \frac{Q_{pf}}{V_p} / \frac{Q_{sf}}{V_s} \tag{8}$$

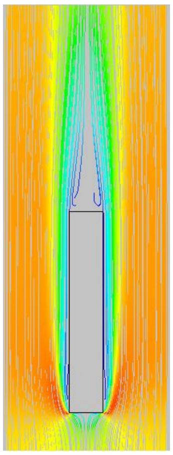


(a) Solid fin, $y = 0.5H$

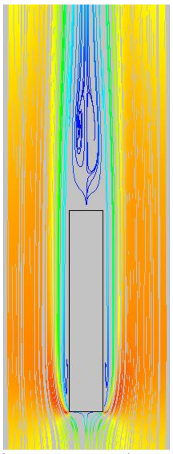


(b) Solid fin, $y = 0.9H$

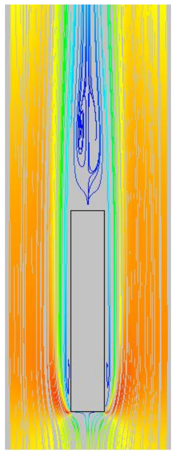
(c) Fin with 2 perforations, $y = 0.5H$



(d) Fin with 2 perforations, $y = 0.9H$



(e) Fin with 4 perforations, $y = 0.5H$



(f) Fin with 4 perforations, $y = 0.9H$

Fig. 5. Fluid path lines at various sections of the different fins, $\boldsymbol{Re_D} = 250$

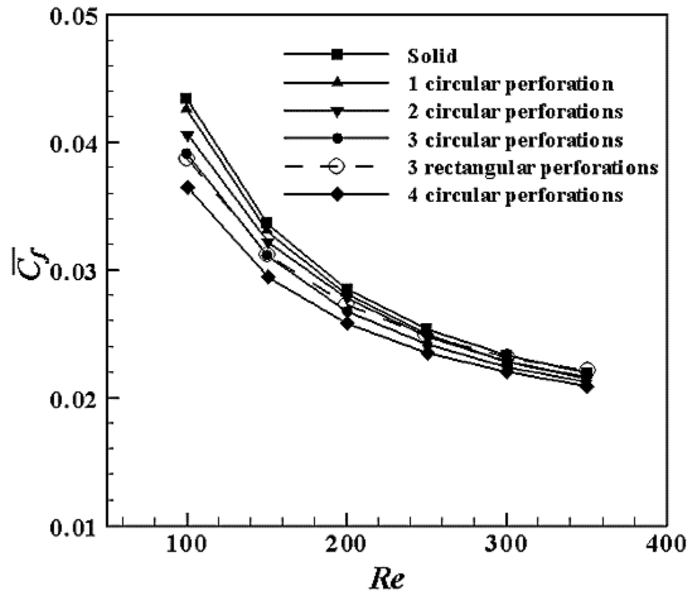


Fig. 6. Variation of the average friction factor coefficient with the Reynolds number over the faces of the various types of fins

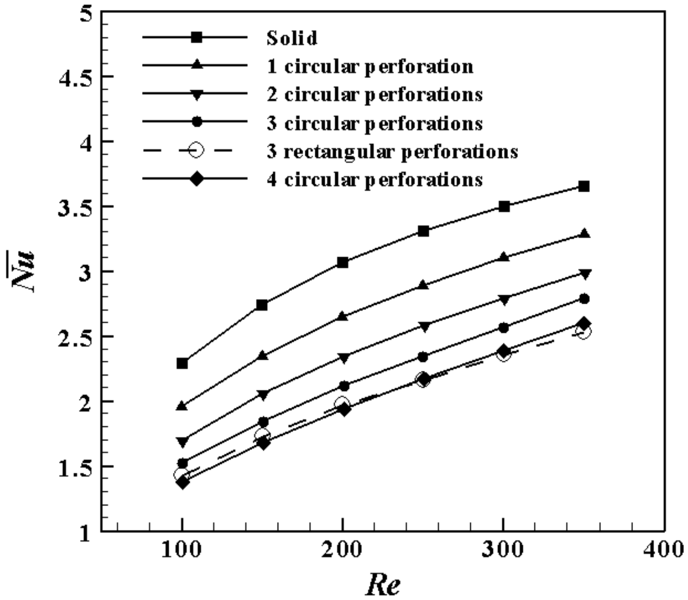


Fig. 7. Variation of the average Nusselt number with the Reynolds number over the faces of the various types of fins

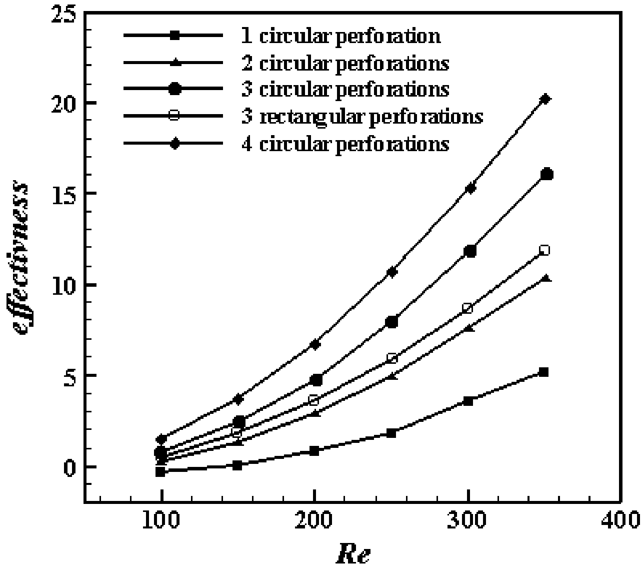


Fig. 8. Perforated-fin effectiveness as a function of the Reynolds number

Figure 9 shows the variation of the rate of heat transfer per unit volume versus the Reynolds number. As the perforation number increases, the heat transfer per unit volume of the fin increases.

This is a result of the reduction of fin size. As the Reynolds number increases, the fluid flow velocity increases, and this results in an increase in heat transfer. Figure 9 demonstrated that the ratio of heat transfer in a circular-perforated fin is better than the heat transfer in a rectangular-perforated fin, especially at larger Reynolds numbers.

5. Conclusion

Three-dimensional laminar flow around an array of circular-perforated fins was studied numerically and the following results were obtained:

1- The average friction factor coefficient decreases with the Reynolds number for solid and perforated fins. It is observed that at any Reynolds number, the friction factor decreases with the number of perforations.
2- The average Nusselt number increases with the Reynolds number. For any Reynolds number, the Nusselt number decreases with the number of perforations. The highest amount of

Nusselt number corresponds to the solid fin.

3- The effectiveness increases with an increase in the amount of perforations and the Reynolds number. As the amount of perforation increases, the fluid flow increases, and this results in more heat transfer.

4- As the perforation number increases, the heat transfer per unit volume of the fin increases. This is a result of the reduction in fin size. As the Reynolds number increases, the fluid flow velocity increases, and this results in an increase in heat transfer.

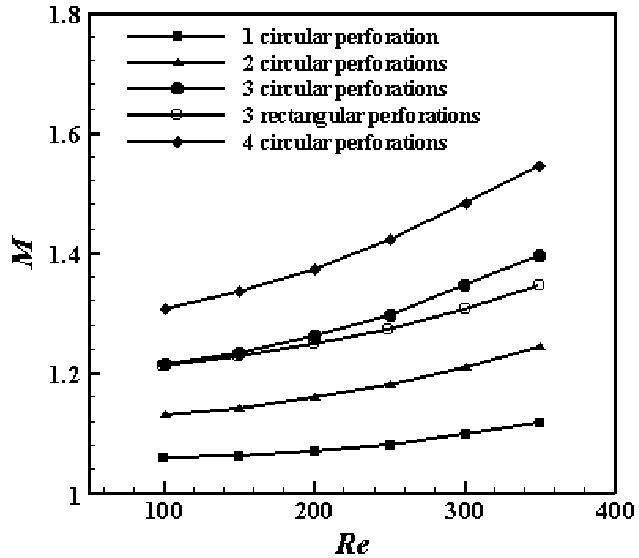


Fig. 9. Heat transfer of unit volume versus the Reynolds number for perforated fins.

References

[1] Dewan A., Mahanta P., Sumithr Raju K., Kumar S., Review of Passive Heat Transfer Augmentation Techniques, Journal of Power and Energy (2004) 218: 509-527.

[2] Kiwan S., Al-Nimr M. A., Using Porous Fins for Heat Transfer Enhancement, Journal of Heat Transfer (2001) 5: 123-790.

[3] Bilen K., Yapici S., Celik C. A., Taguchi Approach for Investigation of Heat Transfer from a Surface Equipped with Rectangular Blocks, Energy Conversion and Management (2001) 42: 951-961.

[4] EI-Sayed S. A., Mohamed Sh. M., Abouda A. L., Experimental Study of Heat Transfer and Fluid Flow in Longitudinal Rectangular-fin Array Located in Different Orientations in Fluid Flow, Experimental Thermal and Fluid Science (2004) 29: 113-128.

[5] Baskaya S., Sivrioglu M., Ozek M., Parametric Study of Natural Convection Heat Transfer from Horizontal Rectangular Fin Arrays, International Journal of Thermal Science (2000) 39: 797-805.

[6] Dijk A.V., Lange H. C., Compressible Laminar Flow Around a Wall–Mounted Cubic Obstacle, Journal of Computational Fluids (2007) 36: 949-960.

[7] Kundu B., Bhanja D., Lee K.S. A Model on the Basis of Analytics for Computing Maximum Heat Transfer in Porous Fins, International Journal of Heat and Mass Transfer (2012) 55: 7611-7622.

[8] Kundu B., Lee, K.S., Exact Analysis from Minimum Shape of Porous Fins under Convection and Radiation Heat Exchange with Surrounding, International Journal of Heat and Mass Transfer (2015) 81: 439-448.

[9] Velayati E., Yaghoubi M.,Numerical Study of Convective Heat Transfer from an Array of Parallel Bluff Plates, International Journal of Heat and Fluid Flow (2005) 26: 80-91.

[10] Molki M., Hashemi-Esfahanian A., Turbulent Convective Mass Transfer Downstream of a Perforated Baffle Blockage, International Journal of Heat and Fluid Flow (1992) 13: 116-123.

[11]Sara O.N., Pekdemir T., Yapici S., Yilmaz M., Heat-Transfer Enhancement in a Channel Flow with Perforated Rectangular Blocks, International Journal Heat and Fluid Flow (2001) 22: 509-518.

[12]Sahin, B., Demir A., Thermal Performance Analysis and Optimum Design Parameters of Heat Exchanger Having Perforated Pin Fins, Energy Conversion and Management (2008) 49: 1684-1695.

[13]Shaeri M.R., Yaghoubi M., Numerical Analysis of Turbulent Convection Heat Transfer from an Array of Perforated Fins, International Journal of Heat and Fluid Flow (2009) 30: 218-228.

[14] Igarashi T., Mayumi Y.,Fluid Flow and Heat Transfer Around a Rectangular Cylinder with Small Inclined Angle (the case of a width/height ratio of a section of 5), International Journal of Heat and Fluid Flow (2001) 22: 279-286.

[15]Karabacak R., Yakar G .,Forced Convection Heat Transfer and Pressure Drop for a Horizontal Cylinder with Vertically Attached Imperforate and Perforated Circular Fins, Energy Conversion and Management (2011) 52: 2785-2793.

[16] Huang G. J., Wong S. C., Lin, C. P., Enhancement of Natural Convection Heat Transfer from Horizontal Rectangular Fin Arrays with Perforations in Fin Base, International Journal of Thermal Science (2014) 84: 164-174.