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Performance of Elliptical Pin Fin Heat Exchanger with Three Elliptical Perforations

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Abstract

A computational investigation has been carried out to assess the heat transfer and pressure drop characteristics of elliptical pin fins arranged in a rectangular duct in a staggered manner. Two types of elliptical pin fins have been considered for this purpose, namely, solid elliptical and perforated elliptical pin fins. Three perforations of elliptical cross sections are considered in the elliptical pin fins. Pressure, temperature and velocity profiles at different locations within the computational domain are considered for different inlet velocities. The heat transfer and pressure drop characteristics along the computational domain are presented and the overall performance, which is defined here as the heat transfer per unit pressure drop, of the heat exchanger is also assessed. The results show that the perforated elliptical pin fins perform better than the solid elliptical pin fin both in terms of heat transfer and pressure drop characteristics.

Keywords: Perforated elliptical pin fins; Heat transfer; Pressure drop; Overall performance.

1. Introduction

Fins are used to enhance convective heat transfer in a wide range of engineering applications. Pin fin arrays are frequently used for cooling of electronic components. They are commonly used as extended surfaces in compact heat exchangers to augment heat transfer and turbulence [1]. Fins help in achieving a large total heat transfer surface area without the use of a large primary surface area. Due to an ever increasing heat production density of electronic components there is a need of compact and energy efficient pin fin heat exchanger. It is important to design a heat exchanger by considering both the pressure loss and heat transfer characteristics. Staggered pin fins are widely used in many thermal systems to enhance their capacity to transfer high heat at low pressure drop and thus to reduce the size of heat exchangers. Various techniques have been developed by many researchers during the last few decades to enhance convective heat

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transfer. Sahiti *et al.* [2] showed that the best fin geometry from the overall performance point of view is the straight elliptical pin fin under the fully developed laminar flow conditions. Dewan *et al.* [3] showed the effect of pin fin spacing and material on performance of fin heat exchanger. Bergelin *et al.* [4] showed that in range 200 < Re < 5000, the flow should be considered in transition regime, while it is laminar below 200.

Sparrow *et al.* [5] experimentally investigated the pressure drop characteristics of diamond-shaped pin fin arrays, which were used on the space shuttle. Using the k–ε turbulence model, Yue-Tzu *et al.* [6] performed numerical simulation of a heat sink with fins of non-uniform height with a confined impingement cooling to examine the effects of the fin shape of the heat sink on the thermal performance. They considered Re of 15,000 and 25,000 and 12 non-uniform fin height designs. Their results showed that the junction temperature can be reduced by increasing the fin height near the center of the heat sink. Sahin *et al.* [7] experimentally studied the heat transfer enhancement and corresponding pressure drop over a flat surface equipped with circular cross-section perforated pin fins in a rectangular channel. The rectangular channel had a cross section area of 100–250 mm^2. Taguchi experimental design method was used for optimum design parameters. Results showed that the use of circular cross-section pin fins may lead to heat transfer enhancement and the optimum result were found for a Reynolds number of 42,000, fin height of 50 mm and streamwise distance between fins of 51 mm. Elyyan *et al.* [8] performed a detailed computational investigation of heat transfer enhancement from dimpled fins using direct numerical and large eddy simulations over a wide range of Reynolds number covering laminar, transitional and turbulent flow regimes.

Experiments were carried by Uzol *et al.* [9] for investigation of the wall heat transfer enhancement and total pressure loss characteristics for two alternative elliptical pin fin arrays and the results were compared with the conventional circular pin fin arrays. They showed that elliptical pin fin array is good from the overall performance point of view. Moshfegh and Nyiredy [10] compared five different turbulence models for pin fin heat sinks, namely, the standard k-ε model, RNG k-ε model, the realizable k-ε model, the k-ω model, and the Reynolds stress transport model. Yang *et al.* [11] performed an experimental study of pin fin heat sinks having circular, elliptic and square cross-sections both for inline and staggered arrangements. For the staggered arrangement, the heat transfer coefficient increased with a rise of fin density for all the three configurations. The elliptic pin fin showed the lowest pressure drops. For the same surface area at a fixed pumping power, the elliptic pin fin possessed the smallest thermal resistance for the staggered arrangement. The flow over tube banks with more than 16 rows is considered to be fully developed [12]. Dewan *et al.* [13] compared the numerical results with the experimental data reported in the literature for circular pin fin heat exchanger and a good agreement was obtained using the RNG k-ε turbulence model with the standard wall functions.

Shaeri *et al.* [14] numerically studied the conjugate conduction-convection heat transfer from a three-dimensional array of rectangular perforated fins with square windows arrangement on the lateral surface of fins and fluid flow. They used Navier-Stokes equations and RNG k-ε model. Their results showed that the perforated fins have higher total heat transfer and considerable weight reduction compared to those with solid fins.

The aim of the present work is to computationally assess the performance of the elliptical pin heat exchanger by introducing three elliptical perforations in its cross-sections. A three-dimensional computational domain is used and the effect of turbulence in the computational domain has been considered by using the RNG k-ε model. The results of the present study can be applied to the design of different heat exchangers used in industry, especially in the design of heat exchangers for electronic components.

2. Geometry and flow description

The flow and heat transfer over pin fins are analyzed assuming that these are basically similar to that over tube banks. The fluid dynamics of flow around a single cylinder is quite complicated [15].

The computational domain considered in the present study is the same as that presented by Dewan *et al.* [13], a three-dimensional rectangular duct of size 155.31×3.6×23.0 mm^3. A staggered array of elliptical pin fins mounted on the heated bottom wall maintained at a constant temperature of 343 K is considered. A staggered arrangement with 16 rows of pins is considered in the present work. For the flow to be fully developed the inlet block length was taken as $5d_h$ (10.0 mm) to avoid the influence of backflow streams and the outlet block length was taken to be $15d_h$ (30.0 mm), here d_h denotes the hydraulic diameter. The unidirectional flow was assumed at the inlet with a constant atmospheric temperature. The computational domain used in the present work for elliptical pin fins as shown in Figure 1 has a hydraulic diameter (d_h) approximately equal to 2.0 mm. Two symmetry planes are passed through the middle of the fins of two consecutive rows. The rectangular box minus the solid fins is the fluid domain considered in the present work. The fluid was chosen to be air. The physical properties of air considered are ρ = 1.225 $kg.m^{-3}$, μ = 1.789x10^{-5} kg. $(ms)^{-1}$ and C_p = 1006.43 Jkg^{-1} K^{-1} and Pr = 0.71. Aluminum with thermal conductivity k = 202.4 W/m-k was chosen to be the fin material.

The computational domain for perforated elliptical pin fins was exactly the same as that of the solid elliptical pin fins with the only difference that the elliptical pin fins were replaced by the perforated elliptical pin fins. The height of the pin fins in both cases was kept the same (23.0 mm). To keep the total surface area of the two types of pin fin identical for comparison, the minor axis of the perforated elliptical pin fin was varied (Table 1). The number of perforations in the elliptical pin fins was taken as three. The perforations were located at the heights of 5.75 mm, 11.50 mm and 17.25 mm from the base. The perforation was an elliptical through hole aligned with the direction of the flow. The dimensions of the pin fins are given in Table 1. The fin spacing in the computational domain for both types of pin fin is shown in Figure 2.

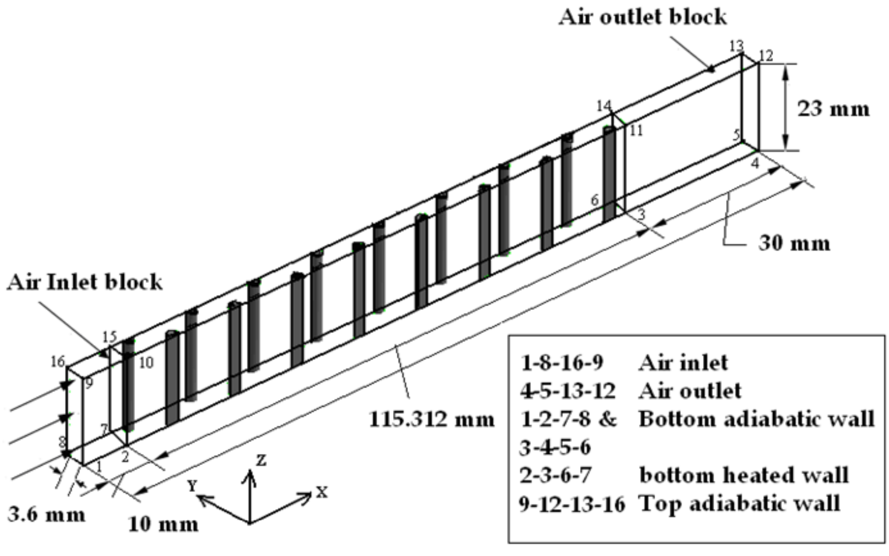

Figure 1. Computational domain for solid elliptical pin fins

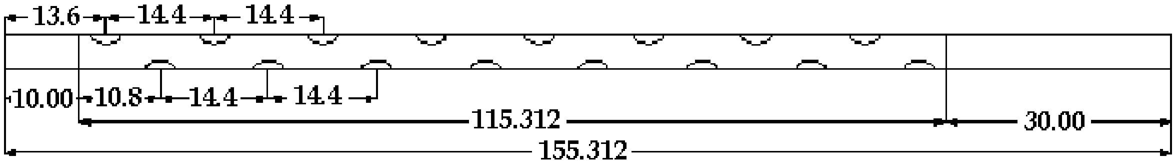

Figure 2. Pin fin spacing in the computational domain (all dimensions are in mm)

TABLE 1 DIMENSIONS OF PIN FINS CONSIDERED.

	Major axis (r_1) mm	**Minor axis (r_2) mm**	**Height (h) mm**
Solid elliptical pin fin	1.4	0.86	23.0
Perforated elliptical pin fin	1.4	0.955	23.0
Elliptical hole in perforated elliptical pin fin	0.7	0.43	------

3. Mathematical modeling and simulations

The flow was assumed to be steady in mean and incompressible and the governing time averaged continuity and momentum equations for three-dimensional turbulent flow were numerically solved. In addition the governing energy equation for obtaining the temperature field was also numerically solved. Thus the five primary variables of interest are three components of mean velocity, mean pressure and mean fluid temperature. The conjugate heat transfer from pin fin arrays implies the simultaneous solution of the above-mentioned equations and the energy equation in the solid. The RNG k-ε turbulence model was used to treat turbulence and the corresponding transport equations for turbulence kinetic energy (k) and its rate of dissipation (ε) were solved [3]. Turbulent heat fluxes in the thermal energy equation were modeled using the concept of eddy viscosity. The standard laws of the wall for velocity and temperature fields were used to treat turbulence in the vicinity of the solid wall. The same boundary conditions and assumptions as used by Dewan *et al*. [3] for circular pin fins were used.

The commercial CFD software FLUENT 6.3 was used to solve numerically the governing equations based on the finite volume method along with the boundary conditions. The second order upwind scheme was used to discretize the governing equations. The preprocessing tool GAMBIT was used for the creation of geometry and meshing. The segregated solver was employed to obtain the numerical solutions of the governing equations for the conservation of the mass, momentum, and energy and other scalar variables, such as, turbulence. The SIMPLE algorithm was used to relate velocity and pressure corrections to enforce mass conservation and to obtain the pressure field. Sutherland’s correlation was used for the molecular viscosity of air.

The grid was generated using the commercial software GAMBIT 2.3.16. The unstructured meshes were used for all the geometries. The prismatic volume mesh was used for the solid elliptical pin fin geometries and tetrahedral volume meshes for the perforated elliptical pin fin geometries. A prismatic volume mesh was obtained from the triangular meshes with pave scheme on the surfaces and Hex/Wedge element with cooper scheme on the volume. The grid independence study was performed for two types of pin fins for the inlet velocities ranging from 1.5 m/s to 6.0 m/s. In all cases, the mesh was very fine in the critical regions. For the grid independence study (Table 2) three different types of mesh, namely, MT-1, MT-2 and MT-3, were selected according to the requirements of the geometries. A grid refinement beyond the mesh size MT-2 did not result in any improvement. Therefore MT-2 type mesh size was selected for all the subsequent computations. The computed results reported here were found to be independent of different values of residuals considered for different governing equations considered. The second-order upwind scheme was used to discretize the governing equations.

The validation of the present computational model was performed by comparing the present computations with the experimental data of Kays [16] for pin fin with circular cross-section. The average deviation between the present computations using the RNG k-ε model and the standard wall functions and experimental data [16] for Nu was observed to be less than 20% (Figure 3). The magnitude of deviation obtained in the present study is similar to that obtained in the literature for the validation of computational model for circular pin-fin heat sink. By considering that the behaviour of manufactured pin configuration under complex turbulent flow conditions is being modelled and computed, the present agreement can be considered to be satisfactory to perform comparisons of various pin fin cross-sections.

TABLE 2 DIFFERENT MESH CONSIDERED FOR GRID INDEPENDENCE STUDY.

Types of pin fins	Mesh types (MT)			Selected cells
	MT-1	MT-2	MT-3	
Solid elliptical	1059729	1366074	1478001	1366074
Perforated elliptical	863386	1193826	1835445	1193826

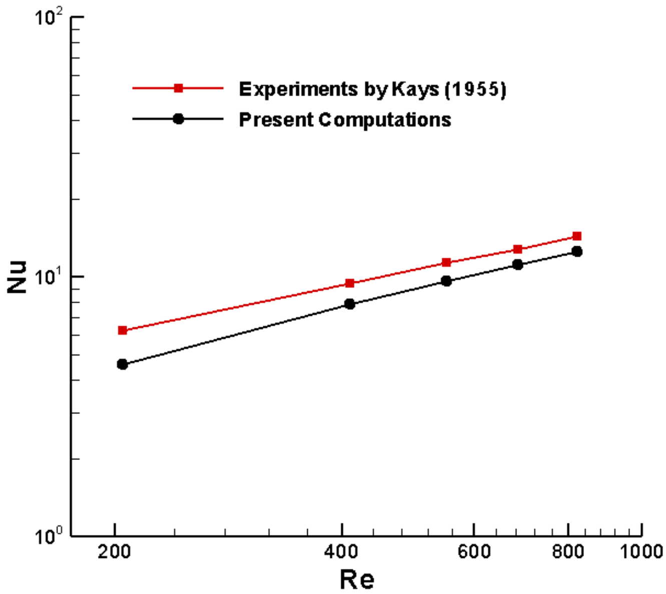

Figure 3. Validation of the numerical results of straight circular pin fins heat exchanger with experimental data by Kays [16]

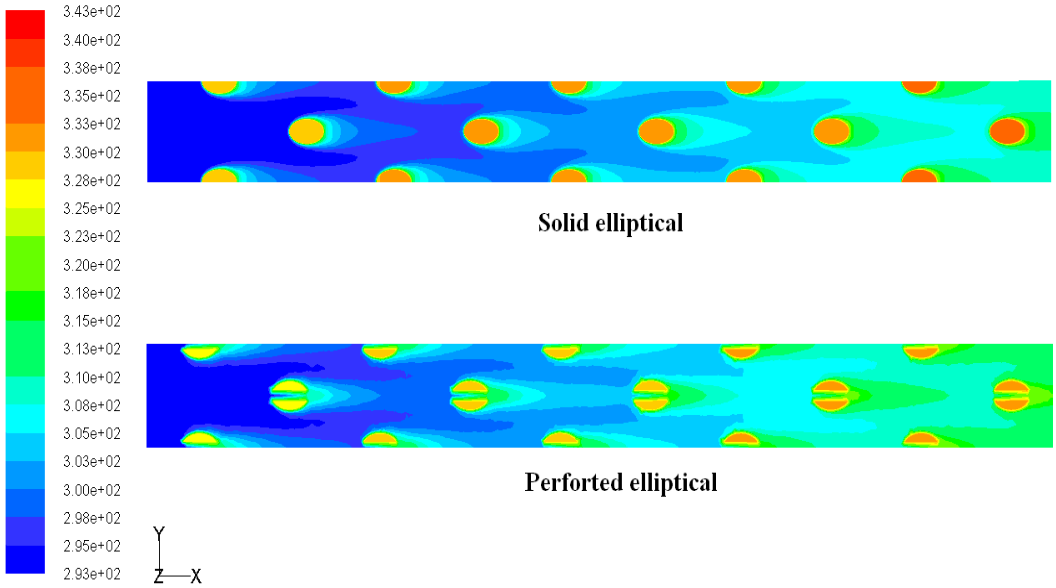

Figure 4. Temperature contours along the streamwise direction for inlet velocity of 3.0 m/s for two pin fins

4. Results and discussion

Temperature contours are shown in Figure 4 at inlet velocity of 3.0 m/s on the x-y planes along the streamwise direction. Contours are shown for first to fifth row of pin fins at a plane in the middle of the computational domain above the bottom heated plate. It is seen from Figure 4 that for both types of fins the fluid adjacent to the fins attains the maximum temperature. The fins take the heat from the bottom heated plate and the fluid takes away this heat from the fins. Thus the fluid temperature increases as it moves along the heat exchanger and the temperature difference between the fins and the surrounding fluid decreases along the length of the computational domain.

The static temperature plots are shown in Figure 5 for two types of pin fins at the inlet velocity of 3.0 m/s along a line passing through the middle of the computational domain, i.e., at x = 0 to 155.31 mm, y = 1.80 mm, z = 11.5 mm. It is observed that there are fluctuations in the temperature plots for solid and perforated pin fins (Figure 5). The fluid after coming in contact with the first row of fins, take heat away from the fins and gains thermal energy. The heated fluid upon its motion to the next row has to pass through the gap between the rows and pushes the cold fluid which was already there to the next row of fins. The colder fluid takes heat from the second row of fins which are at higher temperature than that at the first row. This process is repeated for the remaining fin rows. Hence there is fluctuation in the temperature along the domain. The static temperature increases along the computational domain and it is seen that the perforated elliptical fins attain the highest temperature at the exit compared to that of solid elliptical fins (Figure 5).

The global Nusselt number was calculated for different inlet velocities (from 1.5 m/s to 6.0 m/s) using the surface integral for the heated plate and the pin fins using the commercial CFD software FLUENT 6.3. It is seen from Figure 6 that the global Nusselt number of perforated elliptical pin fin is larger than that of the solid elliptical pin fins at the same Reynolds number. By changing the pin shape from solid elliptical to perforated elliptical, there is an increased interaction between fin and fluid, which increases the total heat transfer rate. The Nusselt number is seen to increase with an increase in Reynolds number.

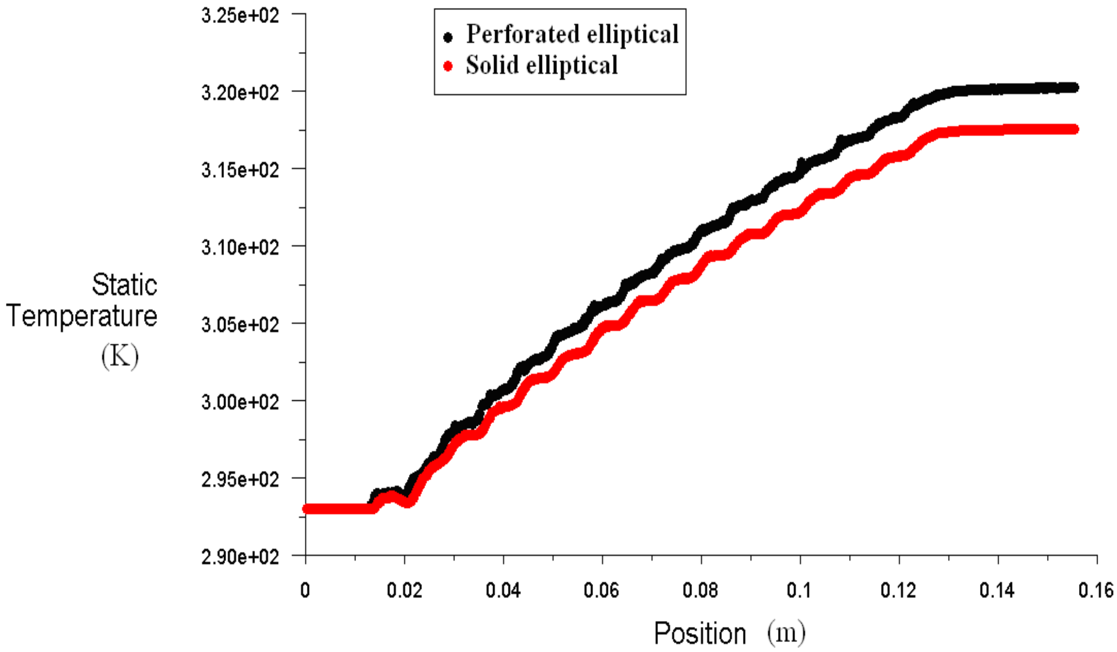

Figure 5. Comparison of static temperature for two pin fins at inlet velocity of 3.0 m/s

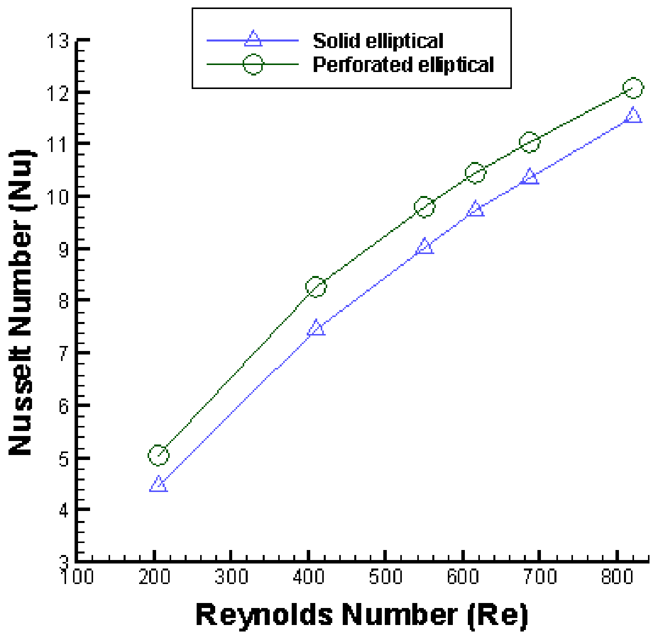

Figure 6. Variation of Nusselt number with Re for two types of pin fins.

Heat transfer and pressure drop play an important role in the performance of pin fin heat exchanger. For a pin fin to be efficient it is necessary to have high heat transfer rate as well as small pressure drop. Therefore for arriving at good pin fin geometry we also need to focus on pressure variation in a pin fin heat exchanger. The total heat transfer and pressure drop at different Reynolds numbers were calculated. The total pressure drop is related to the input power required to drive the fan of a heat exchanger. It is calculated by the difference of the inlet pressure to the outlet pressure of the computational domain. At all Reynolds number the performance, which is defined here as the heat transfer per unit pressure drop, of the perforated elliptical pin fin is good and this behaviour can also be observed from Figure 7.

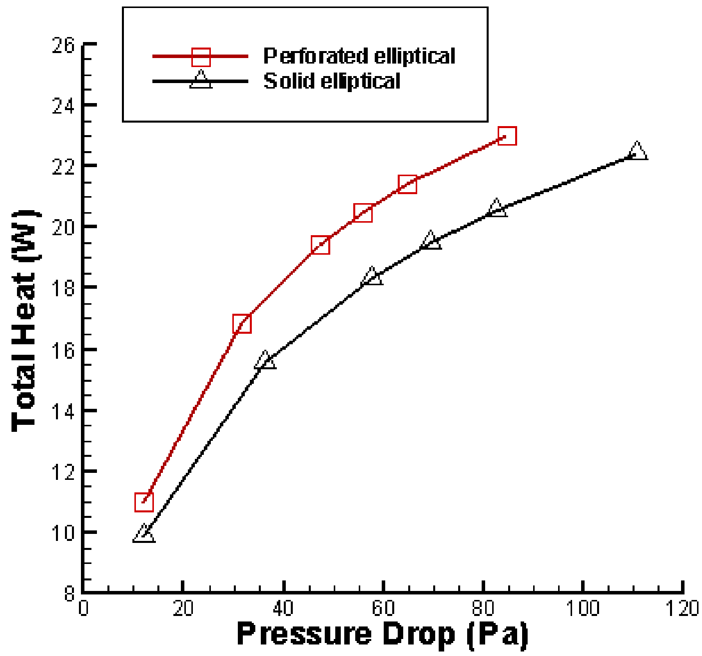

Figure 7. Pin fin performance plot

5. Conclusions

A performance comparison of solid and perforated elliptical pin fin heat exchangers has been presented in this paper. The conclusions drawn from the present study may be summarized as: (a) The results show that the elliptical pin fin heat exchanger with three elliptical perforations performs better than the corresponding unperforated (solid) elliptic pin fin heat exchanger; (b) By changing the solid elliptical pin fin into the perforated elliptical pin fin, the pressure drop decreases by an average of 12% and heat transfer increases by 5.6% and (c) The overall performance of elliptical pin fin heat exchanger increases by 23% by introducing perforations and this performance may be further improved by increasing the numbers of perforations in the elliptical pin fin.

It should be emphasized that the computation using the RANS equations based turbulence models, as employed in the present paper, is limited in predicting the behavior of complex flows such as those encountered in the present study. Therefore, more advanced techniques such as direct numerical simulation or large eddy simulation, though computationally more expensive, may provide more accurate predictions.

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Notations

D_h	Hydraulic diameter, m
H	Fin height, m
k	Thermal conductivity, W/m K
	Turbulence kinetic energy, m^2/s^2
L	Length, m
Nu	Nusselt number, --
Δp	Pressure Drop, Pa
Q	Heat transfer rate, W
Re	Reynolds number = $(UD_h)/\nu$, --
T	Temperature, K
U	Inlet air velocity, m/s
u, v, w	Velocity components along x, y, z directions, respectively, m/s
x, y, z	Streamwise, spanwise and vertical directions, respectively

Greek letters

ρ	Density, $kg.m^{-3}$
ε	Dissipation rate, m^2/s^3
δ_{ij}	Kronecker delta, --
τ	Shear stress, Pa
μ	Dynamic viscosity, kg $(ms)^{-1}$

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