Thermal Analysis of Semi-Circular Pin Fins for Application in Electronics Cooling

Saroj Yadav, Kumari Ambe Verma, Mukul Ray, Krishna Murari Pandey

Abstract*: Efficient thermal energy management of a system is always a prime requirement for many equipment and industries. The performance of almost all devices are affected by the thermal conditions of the system, which also include the surroundings. Generation of heat is an unavoidable phenomenon for any device that runs on external power sources. The generated heat in such systems must be dissipated to the surrounding. It requires efficient utilization of the surface area with minimum flow losses in the system. It was always desirable to have a heat sink in the all devices that occupy minimum space with maximum effectiveness. The present work is an effort to analyze conjugate heat transfer physics in a 3-D system of aluminum pin fins, with air as the working fluid. A finite element solver, COMSOL 4.3a has been used in simulating a staggered pin fin arrangements placed over a base plate. The solver is validated using the empirical data of previous literature. The thermal analysis has been performed on semicircular pin fins with uniform cross section. Consideration is given to staggered arrangement of semicircular fins with various relative distances between two sections of a circle with various inlet velocities. Heat transfer coefficient, Nusselt number, skin friction coefficient and pressure coefficient are four parameters that are taken into consideration for analyzing all the fin geometries in the current study. The proposed shapes are designed to increase the wetted surface area by keeping the fin material volume constant.

Keywords: Extended surfaces; Nusselt number; Heat transfer co-efficient; Finite element method; Conjugate heat transfer.

I. INTRODUCTION

Thermal analysis and its management is a key parameter during engineering processes designing. A system often experience heating during working condition. To avoid the heating effect of the system in various industries, heat control techniques can be favorable. Activities like in air travel, heat exchangers, microelectronic and electronic equipment, Internal Combustion engines, etc. are permanently in prerequisite of heat-dissipation techniques. In this aspect, various researchers have already been explored the stretched surfaces or called as fins. These are very useful for many cases. Heat transfer rate can be improved by increase number of fins, this is also increase the dimension and the weight of the system and also its cost. To reduce the operating cost of this complex heat transfer system, focus should absolutely be done on the size as well as weight of the fins with maximum heat transfer rate. Active mode of heat transfer is used to increase the diffusion rate for specific thermal system. Convection and radiation heat transfer mode are dominating for fin type heat transfer system over conduction mode. Consequently, Heat transfer of such system is entirely dependent on the temperature of the extended surface, convective heat transfer coefficient of the fluid and its wetted surface area also. As convective heat transfer totally concerned with thermos-physical properties of the fluid, connected surface, and fluid velocity. So it can concise that extension of surface area can promote heat transfer. However extended surface area supplement the cost of the system. This problem is still present and many researchers are doing for the same to get optimum solution. Addition of fins are also a challenging situation for conduction resistance. Furthermore, widespread heat transfer of the surface has not been professionally assured due to geometrical complexity of the system. Numerous work has been performed in the area of shape and profile of the fins to enhance the heat transfer (Gorla and Bakier 2011 and Tsai et al. 1999). Kanaskar et al. (2015) has presented a literature appraisal on the enhancement of heat transfer with the aid of perforated heat sinks. Tullius et al. (2012) has used short micro pin fins in order to optimize the geometry and thus magnify heat transfer rate through convection from heated surface. Authors used six different geometrical shaped fins – circular, square, triangle, ellipse, diamond, and hexagon in staggered arrangement and they concluded that triangle shape fins has the highest Nu and Elliptical and circular fins depict the smallest pressure drop. Liao et al. (2014) presents a numerical assessment of the flow and thermal characteristics of in-line pin-fins in a wedge duct using air, steam, and mist/steam as coolant. Later, authors concluded that mist/steam coolant produces a lesser friction coefficient and a higher aspect of thermal performance which in results improve the heat transfer rate. Sajedi et al. (2016) studied the influence of splitter plate application on the circular and square pin fin heat sinks in order to determine and thus improves the hydro-thermal behavior of a pin fin heat sink. Yadav et al. (2016) has done the investigation of micro-channel heat sink included plain rectangular micro-channel, and micro-channel with cylindrical pin fins and they found in terms of pressure drop and heat transfer enhancement, micro-channel with cylindrical pin fin is much superior than other cases. Lawson et al. (2011) experimentally investigates the effect of pin fin spacing including span wise and stream wise on array heat transfer and pressure drop. Further, authors ended up with a conclusion that heat transfer in an array of pin fin can be increased while diminishing pressure drop by increasing span wise and decreasing

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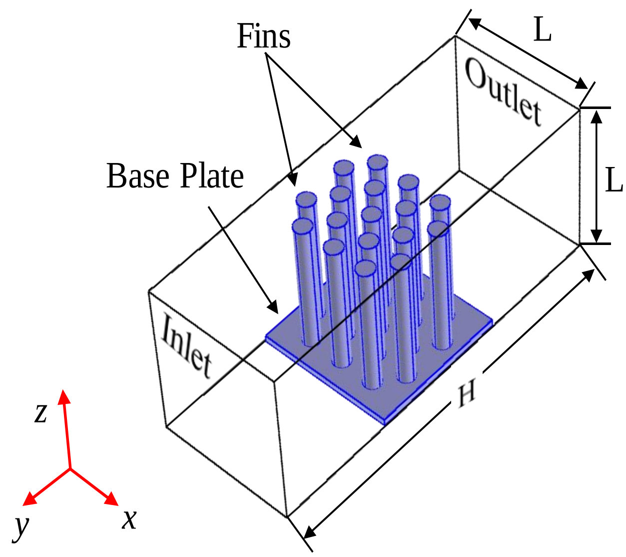
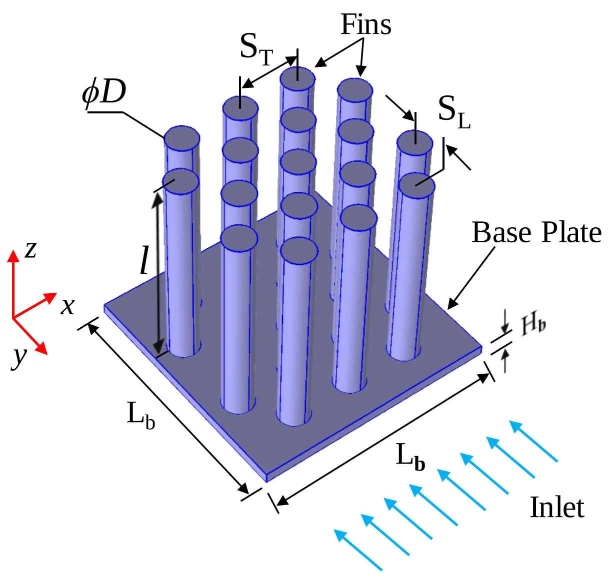
stream wise pin spacing. Park et al. (2015) designed a staggered pin-fin radial heat sink which has been optimized for cooling of light-emitting diode (LED) device. In this work, authors developed a numerical model to simulate various pin-fin array heat sinks. Further, the result has been verified experimentally. Maldonado and Roth (2012) simulates a three-dimensional two phase model for the snow-drift relocation around buildings utilizing deflected fins of various shapes and sizes and found that bent fins doesn't affect but it create shadows in relocated drifts. Pouryoussefi and Zhang (2015) has done number of experiments to examine the forced convective cooling performance of an air cooled parallel plate fin heat sink with and without insertion of cylindrical pin fins in between the plate fins. Authors found that the overall performance is higher for the case of plate fins heat sink with cylindrical fins. Abdoli et al. (2015) numerically investigated the thermal and flow analysis and presented the influence of six different micro pin-fin including hydrofoil shape micro pin-fin which shows the 30.4% reduction in pumping power and 3.2% growth in the ratio of convection over total heat load compared to conventional circular shape micro pin-fin design. Jandaud et al. (2016) has done the geometrical optimization of a three-dimensional CFD modelled heat sink using Variable Neighbourhood Search method in order to minimize the pressure drop and thermal resistance of the system. Arshad et al. (2018) has done the experimental study to observe the effect of fin geometry on flow-induced vibration response using parallel triangular finned tube array with P/Deff ratio as 1.62. The authors found that instability threshold is delayed with the increment in fin density. Jafari et al. (2018) has presented the numerical study on flow and heat transfer of a Nano fluid between a cold square enclosure and warm inner cylindrical pin-fins system filled with homogenous porous media using Lattice Boltzmann method and found increment in Rayleigh number and volume fraction of nanoparticles and further recommended it for heating the cold square enclosure. With an aim to enhance the heat transfer rate from a plate to fluid (air), a staggered arrangement of semi-circular pin fins are considered. The fins are taken into consideration by parting circular pin fins vertically into two halves. It is expected that the relative positioning of a semicircular pin fin with its adjacent opposite part will affect the heat transfer process. Therefore, in the current study, thermal analysis has been performed on semicircular pin fins with different longitudinal and transverse pitches. In the following, the formulations and the considered geometry is discussed in detail. The work has been performed numerically using FEM based commercial solver COMSOL 4.3a. The solver is validated using previous literature before performing the actual work.

II. FORMULATIONS AND GEOMETRY

In the present paper three dimensional model has selected to conduct numerical simulation. Bundle of aluminum pipe has taken in air circulation duct having $L \times H \times L$ dimension shown in figure 1 (a). Aluminum pipes or fins are having height *l* and diameter *D*. $L_b \times L_b \times H_b$ dimension of base plate has selected shown in figure 1 (b) in staggered manner. The experimental system is located inside the air-flow duct at a distance maintaining 1:3 ratio from the inlet. S_L and S_T, are the longitudinal and transverse pitch between two fins respectively. System has been taken for forced convection state. Validation has completely done with experimental done and explained in Yadav, S. and Pandey, K.M., (2018). Isothermal room temperature T_f and velocity $\vec{V}\left(=-\vec{u}_y\right)$ in the negative y – direction has been taken for incoming air inside the duct. Atmospheric pressure outlet (P_o) and at outflow condition $\left(\hat{n}\cdot k\nabla T=0\right)$ has chosen for numerical observation. The walls of the air duct is at no slip conditions $\left(\left|\vec{V}\right|=0\right)$ and at an isothermal room temperature T_f (Fig. 1a). The base plate of the fin system is supplied with a total uniform heat of Q_s.



(a)



(b)

Figure 1 (a) Numerical Simulation field and (b) Extended view of fins

In general, thermal behavior of any system can obtained easily by means of heat transfer modes and system characteristics. Thus, the energy equation is

$$\rho c_p\left(\frac{\partial T}{\partial t}+\vec{V}\cdot\nabla T\right)=\nabla\cdot\left(k\nabla T\right) \quad (1)$$

Where, $\nabla=\hat{i}\frac{\partial}{\partial x}+\hat{j}\frac{\partial}{\partial y}+\hat{k}\frac{\partial}{\partial z}$ in rectangular Cartesian co-ordinate and ρ, c_p, and k are the thermal-physical properties. Hence, the governing equations also comprise steady state continuity and momentum equations,

$$\nabla\cdot\left(\rho\vec{V}\right)=0 \quad (2)$$

$$\vec{V}\cdot\nabla\vec{V}=-\frac{1}{\rho}\nabla p+\nu\nabla^2\vec{V} \quad (3)$$

The governing equations (Eq. 1-3) has been discretized by means of finite element method (FEM). The discretization is achieved using COMSOL multi-physics, a commercially available FEM solver. The average Nusselt number is explained below

$$Nu=\frac{Q_s D}{k_f A_w\left(T_w-T_f\right)}=\frac{h_f D}{k_f} \quad (4)$$

where, A_w and T_w are the wetted surface area of the fin system and its average surface temperature, respectively. The pressure co-efficient and the coefficient of friction can be calculated using the following

$$C_f=\frac{2\tau_w}{\rho V^2} \quad (5)$$

$$C_p=\frac{2\Delta P}{\rho V^2} \quad (6)$$

where, τ_w and ΔP are local wall shear stress (N/m^2) and pressure difference (N/m^2) respectively.

Fluid flow over a cylindrical body is often use in numerous engineering application. Number of correlation are present related to convective heat transfer coefficient in Khan *et al.* (2005). Khan *et al.* (2005) explained correlation for cylindrical pin fin heat sink. Hence approaching fluid velocity of $U\left(=\left|\vec{V}\right|\right)$, the Reynolds number for this flow is defined as

$$\mathrm{Re}_D=\frac{\nu U_{\max}}{D} \quad (5)$$

where ν is the kinematic viscosity of the flowing fluid and $U_{\max}$ is the maximum fluid velocity between the fins and explained below

$$U_{\max}=\max\left(\frac{S_T}{S_T-D}U,\frac{S_T}{2\left(S_D-D\right)}U\right) \quad (6)$$

where $S_D=\sqrt{S_L^2+\left(S_T/2\right)^2}$ is the diagonal pitch (Present Case).

The average Nusselt number for the fin is follow

$$Nu=\frac{hD}{k_f}=C_1\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3} \quad (7)$$

where, $C_1=\frac{0.61S_T^{0.091}S_L^{0.053}}{1-2e^{-1.09S_L}}$ [20].

The calculated values of Nu and h obtained from the Computational Fluid Dynamics (CFD) solution and the empirical relationship will prove the rightness of the numerical modelling. Additional, considering semi-circular cross-sectional shapes of pin fins with different transverse and longitudinal pitches between two semi-circular pin fins. The performances are checked and compared with the cylindrical pin fin. The considered shape is organized in staggered pattern and shown in Fig. 2. The results are discussed in the following section.

Coefficient of skin friction (C_f) and pressure (C_p) are two important parameters that affects the flow dynamics of the system. Skin friction coefficient is the ratio of shear stress to that of the dynamic pressure of the system. On the other hand, pressure coefficient is the ratio of pressure drop across the flow domain to the dynamic pressure of the system. A higher value of C_f and C_p imply higher losses along the length of the flow. In this study, both C_f and C_p are evaluated for cylindrical pin fins and compared with the newly proposed configurations of semicircular fin systems arranged in staggered fashion.

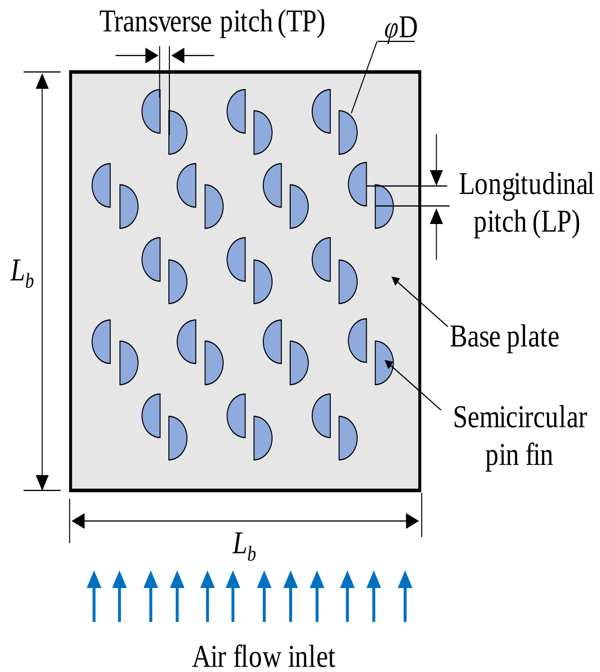


Figure 2 Fig. 2. Schematic diagram of semicircular pin fins on xy-plane.

III. RESULTS AND DISCUSSION

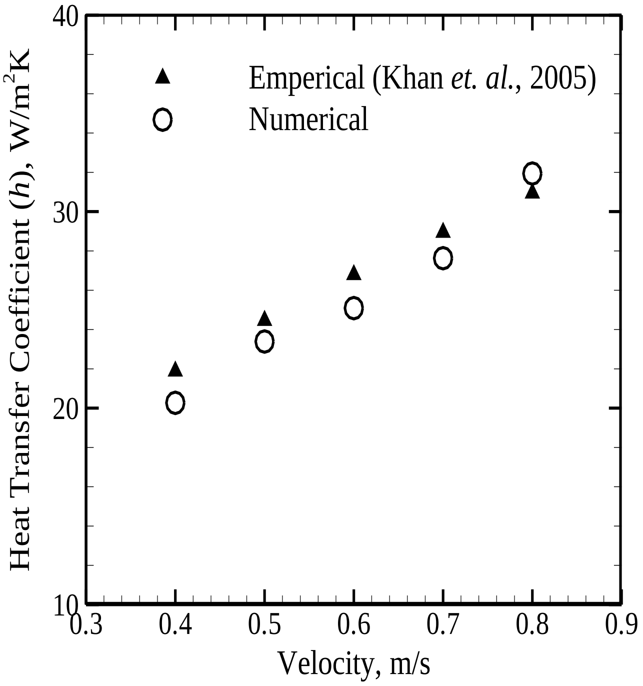
In the succeeding, the paper validate the formulations, the numerical model and the boundary conditions using the results acquired from the empirical relations given by Khan *et al.* (2005). The values of C_f and C_p are initially evaluated for the cylindrical pin fin system in staggered arrangement. Following it, the thermal and flow performance of the newly proposed system with different longitudinal and transverse pitches are evaluated.

A. *Validation of the Solver*

The 17 numbers of aluminum cylindrical pin fins $\left(k = 237\,W/m.K, \rho = 2700\,kg/m^3 \,\&\, c_p = 900\,J/kg.K\right)$ have been selected and ordered in staggered manner as shown in Fig. 1(b). These fins are placed over base plate having dimension of 80 mm × 80 mm × 10 mm, Length, width and height respectively. Dimeter and length of the fins are 9.8 mm and 40 mm respectively. Longitudinal and transverse distance between two subsequent fins are $S_L = 15\,mm$ and $S_T = 20\,mm$, Size of the duct are 90 mm × 60 mm × 200 mm. Uniform heat (Q_s) is provided to the bottom of the plate of 2W. Nusselt number is a constraint that portrays the strength of the convection heat transfer rate compared to the conductive heat transfer rate of the fluid. The performance constraints (Nu and h_f) in the considered system are calculated computationally (Eq. 4) and using the empirical relations (Eq. 5-7). With tetrahedral grid to discretize the computational domain, the grid dependency tests has been performed for different sizes of the grid. Using a grid having 15,93,317 numbers of elements, the governing equations were solved for a steady-state condition. Table 1 shows evaluation between empirical and numerical data of Nu and h for many inlet velocity and fin temperature A comparative plot of the same is also presented in Fig. 3a and 3b. From the results it has been noticed that with increase in the inlet velocity of the air, the convective heat transfer coefficient increases. The increased value of *h* is due to decreased value of average surface temperature of the fin system due to higher convective heat transfer rate. With stable characteristics length, increase in *h* also increases the value of average Nu. The equated results of average Nu and *h* are found within the acceptable accuracy. The deviation in the numerical result from the empirical one is due to the ignored thermal radiation phenomenon in the mathematical modelling. Moreover, lower is the velocity of flow, more will be the error between the empirical and the numerical results. Because, at lower flow velocity, the effect of buoyancy force appears and dominates the effect of inertia force.

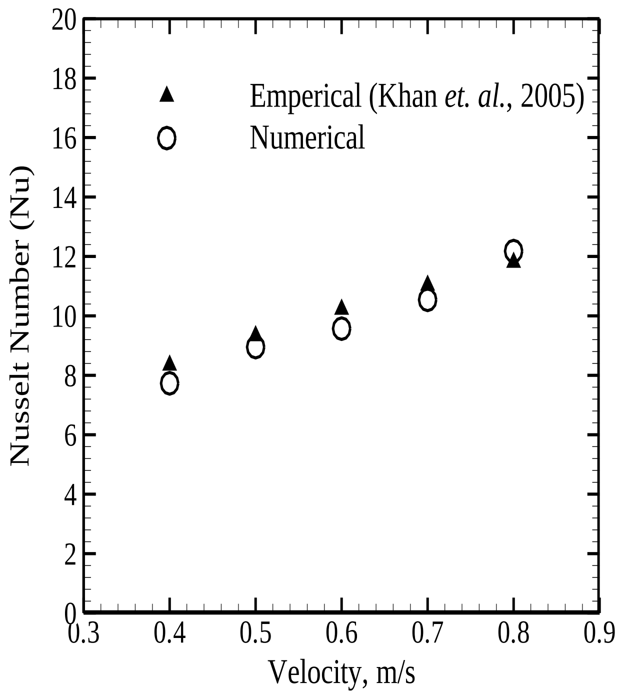
Table 1. Evaluation between empirical and numerical data of Nu and *h* for many inlet velocity and fin temperature

Velocity	T_fin	Empirical		Numerical	
		h	Nu	*h*	Nu
0.4	296.38	21.88	8.34	20.27	7.73
0.5	295.95	24.46	9.33	23.39	8.92
0.6	295.76	26.79	10.22	25.09	9.57
0.7	295.52	28.94	11.03	27.63	10.54
0.8	295.2	30.94	11.8	31.94	12.18



(a)

Having validated the numerical solver, the fins of semi-circular shape are considered for study. The fins are modelled in such a way that the volume of the system is always maintained constant. Further, the thermal analyses are performed by varying the longitudinal and the transverse pitch between a pair of semicircular pin fin. The considered values of pitches are D/2, D/4, D/6 and D/8.

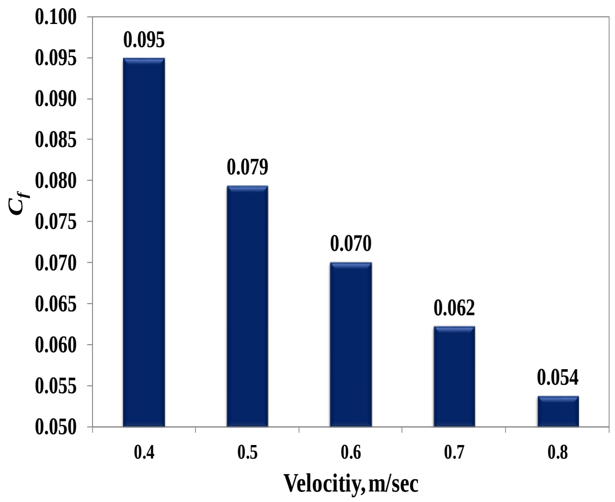


(b)

Figure 3 Comparison of empirical and numerical values of (a) h and (b) Nu for a cylindrical pin fin system.

B. *Cp and Cf of Cylindrical Pin Fin Geometry*

In any system with fluid flow, it is very important to analyze the flow characteristics. Understanding the flow losses in such systems helps in improving the system from flow dynamics point of view. In this context, coefficient of friction (C_f) and coefficient of pressure (C_p) are two important parameters. In the following the variation of C_f and C_p are plotted in Fig. 4 corresponding to increasing value of velocity. It can be noted that as the velocity of flow increases, the shear stress experienced by the solid surface increases, due to increase in local Reynolds' number. Moreover, the value of dynamic pressure also found to increase with increase in the inlet velocity of the fluid. However, the rate of increment of shear stress is lesser than the rate of increment of dynamic pressure with respect to incoming flow velocity. Hence, the result is the reduction in Cf with increase in velocity. Similar, justification can be put forward for the Cp too. As the incoming velocity increases, the dynamic pressure and the pressure drop increases. However, due to lesser rate of increment of pressure drop compared to dynamic pressure, the net result is reduction in the value of Cp.



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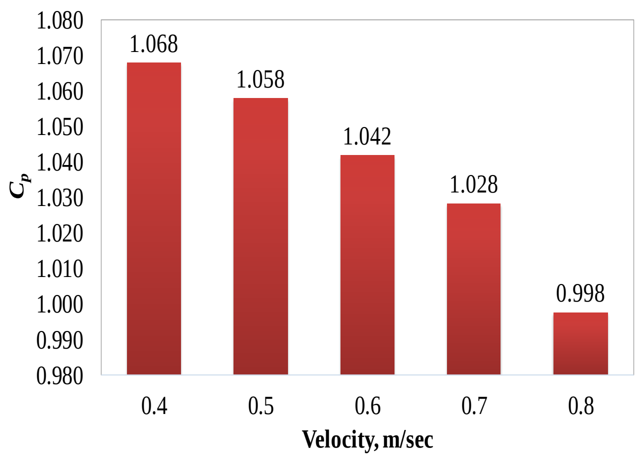


(b)

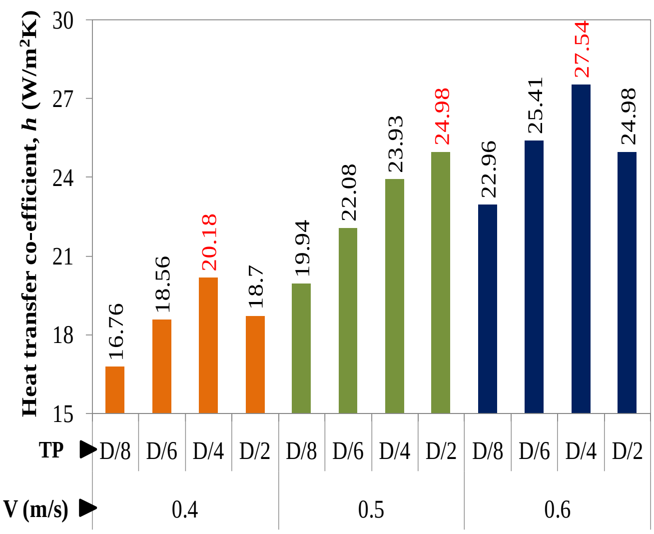
Figure 4 Variation of Cf and Cp with velocity for cylindrical pin fin system with staggered arrangement.

C. *Effect of Transverse Pitch of the Semicircular Pin Fins on Heat Transfer*

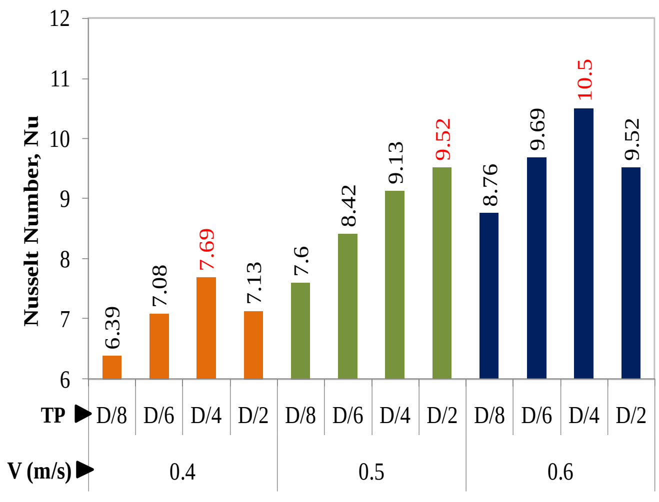
The center to center distance in a semicircular fin in the x-direction is the transverse pitch considered in the current study. Initially, the longitudinal distance between a pair of fin is kept constant at zero. The variation in the transverse pitch is done with values D/2, D/4, D/6 and D/8 at various velocities viz., 0.4, 0.5 and 0.6 m/s. The initial observation of the variation of h shows an increment in its value, with increase in velocity at a particular transverse pitch. At a flow velocity of 0.4 and 0.6 m/s, it has been found that initially when the transverse pitch increases the value of h also increases up to a case of D/4. Then the value of heat transfer coefficient decreases up to D/2. However, the scenario is not the same in case of inlet velocity V=0.5 m/s. For this case, the maximum value of heat transfer co-efficient is observed for a TP of D/2. At velocity 0.4 and 0.6 m/s, the average surface temperature of the fin has been found minimum for the case with TP of D/4 (Fig. 6a). The scenario is changed in case of velocity 0.5 m/s. At this value of inlet velocity, the local maximum velocity inside the rectangular duct with the fin system is observed to be higher in case of TP= D/2, i.e., Vmax= 1.537 m/s (Fig. 6b). The situation can be well visualized by velocity contours of all the considered cases. Figure 7 shows the velocity magnitude of semicircular staggered pin fin geometry. A 2-D zx- plane is considered at 10 mm above the base plate. As air flows over the fins from left to right (Fig. 7), the fluid velocity increases near the walls (top and bottom boundaries). The higher is the maximum velocity in the domain, higher will be the amount of heat carried away by the fluid. Hence, higher value of h and Nu.

D. *Effect of Longitudinal Pitch of the Semicircular Pin Fins on Heat Transfer*

Under this heading, the effect of the longitudinal pitch between a pair of semicircular fin has been discussed. Figure 8a and 8b shows the variation of h and Nu for different values of longitudinal pitch. The study has been performed by varying the longitudinal pitch from 0.0 – D/2 and fixing the transverse pitch at a particular value (viz., D/8, D/6, D/4 and D/2).



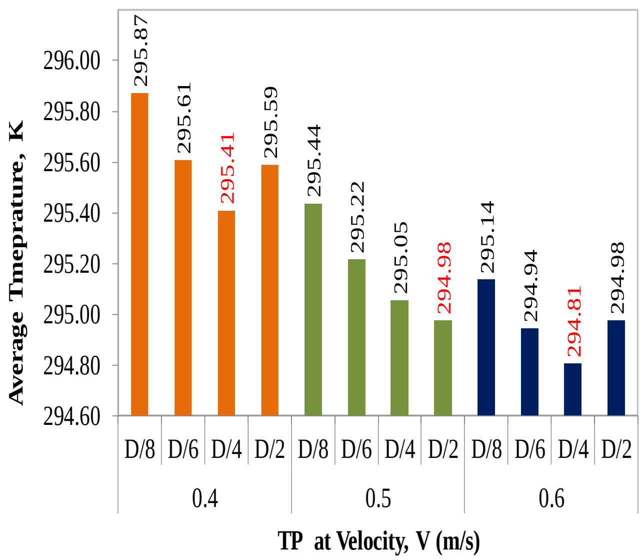
(a)



(b)

Figure 5 Variation of (a) h and (b) Nu for various transverse pitches at different velocities

All these studies are performed at a particular value of inlet velocity of 0.5 m/s. It has been found that the highest values of h and Nu are observed for a TP of D/2 at zero LP. Comparing all cases of LP at different TPs, it is found that the average surface temperature of the fin system is also found minimum for TP of D/2 at zero LP. Lower value of fin temperature at fixed amount of supplied heat flux imply that higher is the amount of heat carried away by the fluid.



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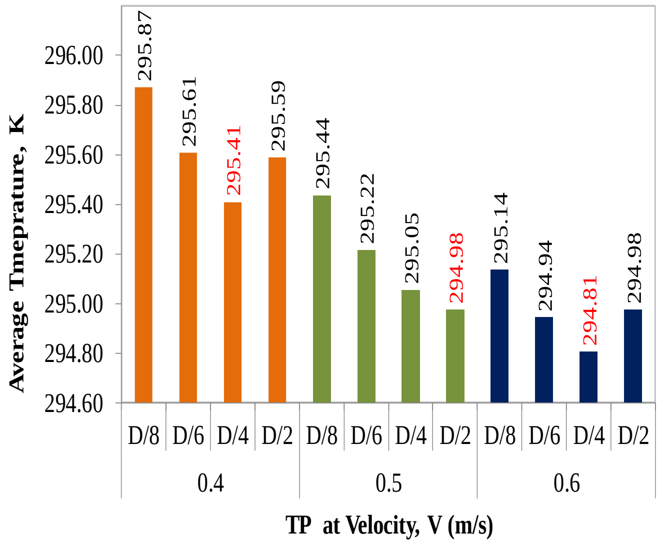


(b)

Figure 6 Variation of (a) average surface temperature and (b) maximum velocity of fin system at various transverse pitches at different velocities

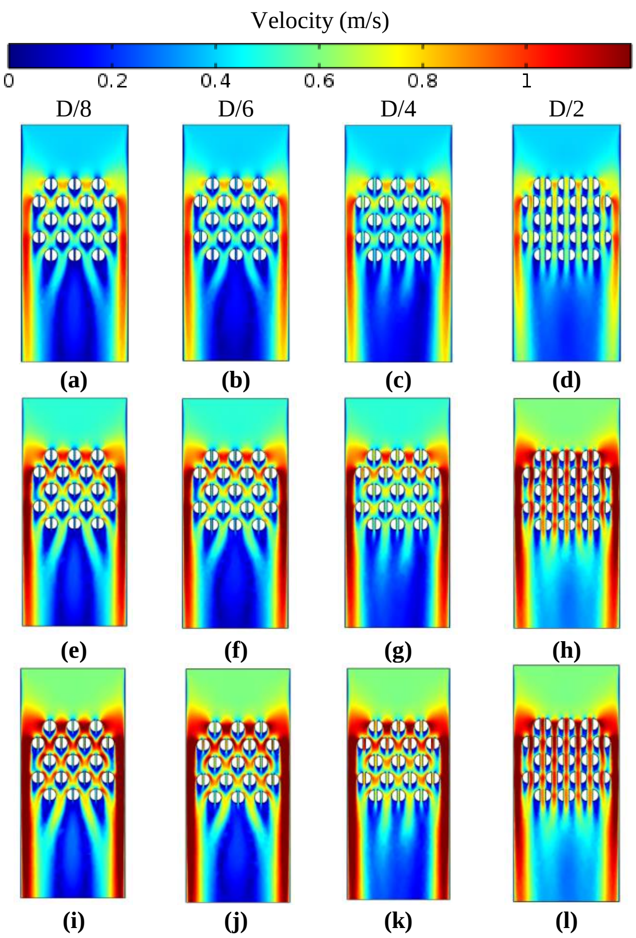
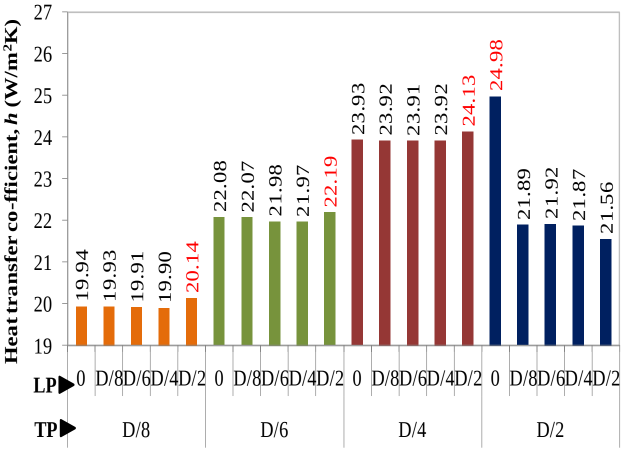
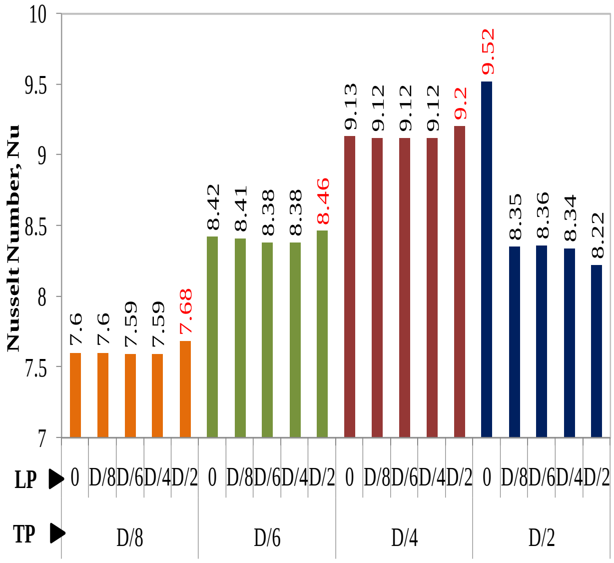


Fig. 7. Contour plots of velocity magnitude for various transverse pitch at inlet velocity of (a-d) 0.4; (e-h) 0.5 and (i-l) 0.6m/s

This leads to higher convection rate of heat transfer and the convective heat transfer coefficient. With fixed characteristics length of the system, the variation of Nu show the same pattern as that of h. In case of TP varying from D/8 to D/4, for all LP, increasing the TP between a pair of alternate pin fins leads to reduction in the distance between oppositely facing neighboring fins. That leads to more stagnation of the fluid near the fins adding to the convective resistance in the system. Therefore, in case of TP varying from D/8 to D/4, with increasing LP, the average fin temperature reduces, and h and Nu increases (Fig. 8c). Upon comparison of a cylindrical pin fin with the semicircular fin at same inlet velocity of 0.5 m/s, the case with TP as D/2 and zero LP yields 6.8% enhancement in the heat transfer rate. This enhancement is due to the combined effect of increased wetted surface area and thermal interaction between the solid and the fluid. The highest possible thermal interaction between the solid surface and the fluid is obtained in the current geometry and the configurations by optimizing the interspatial distances between the fins.



(a)



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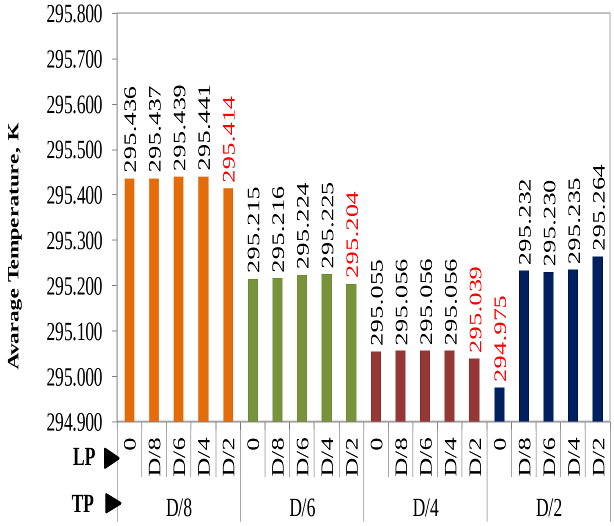
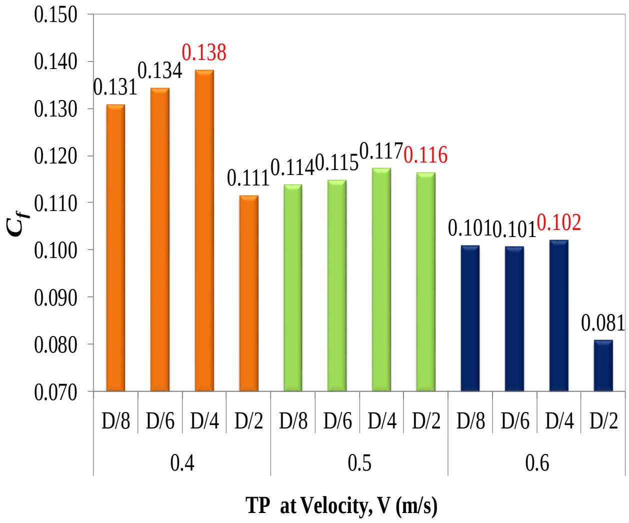


(c)

Figure 8 Variation of (a) h (b) Nu and (c) average skin surface temperature for various longitudinal pitches at 0.5 m/s

E. *Effect of Transverse and Longitudinal Pitches of the Semicircular Pin Fins on Fluid Flow*

Splitting of a pin fin of circular cross section into two semicircular fins help in increasing the wetted surface area of the fins. From heat transfer point of view, this modification has been found advantageous in case of fins with zero LP and D/2 TP. An increase in the wetted surface area adds to losses in the fluid flow. It can be understood from Fig. 9 and 10. Figure 9a and 9b show the variation of surface averaged skin friction and pressure coefficients for various cases of zero LP at various TPs at different velocity. The values in the bar plot are highlighted in red for the cases with maximum heat transfer rate. At higher value of velocity, the value of Cf and Cp reduces. At a particular value of inlet velocity, TP =D/2 yields the minimum skin friction coefficient (Fig. 9a). However, such a general inference cannot be made for Cp.



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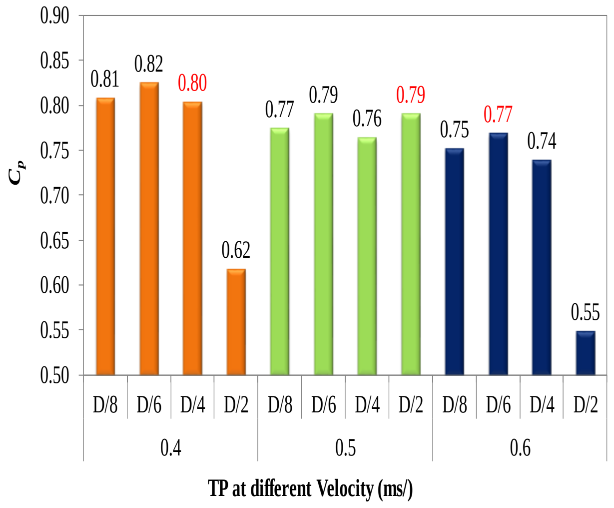
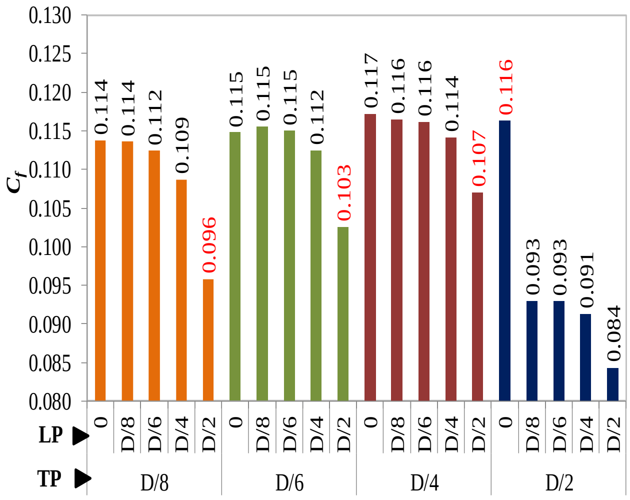


(b)

Figure 9 Variation of (a) Cf and (b) Cp for various TP at various inlet velocity

Figure 10a and 10b show the variation of surface averaged skin friction and pressure coefficients for various cases of LP at various TPs at 0.5 m/s inlet velocity. For TP varying from D/8 to D/4, the value of Cf is observed to be minimum for LP=D/2. Whereas, in these configurations, the Cp is found maximum. For TP=D/2, the minimum Cf and Cp is found in LP= D/2 and D/8, respectively. Increase in the wetted surface area in the present configuration of semicircular pin fin system increases the value of Cf, compared to pin fins of circular cross-sections. The maximum increment in the value of Cf compared to cylindrical pin fin system is approximately 48%. Whereas, the modification helps in reducing the pressure drop in the system. The minimum reduction in pressure drop compared to cylindrical pin fin is found as 9%, for TP=D/6 and LP=D/2.



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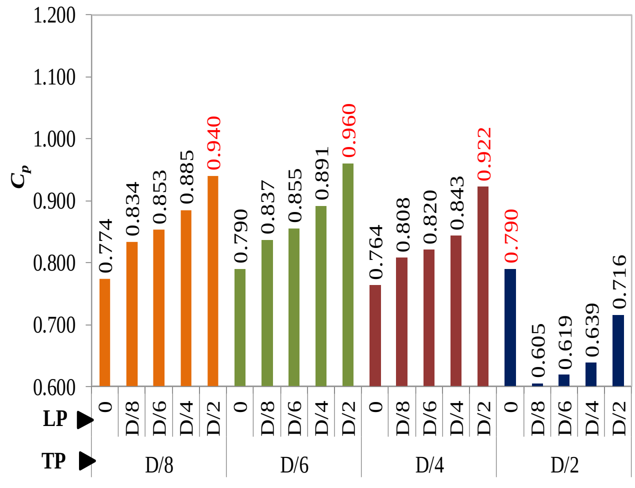

(b)

Figure 10 Variation of (a) Cf and (b) Cp for various TP and LP at 0.5 m/s inlet velocity

IV. CONCLUSION

In the present study, a system of pin fin geometry is analyzed numerically using conjugate heat transfer physics. The numerical solver is validated using the empirical relationships provided on cylindrical pin fins by Khan et. al. [20]. It has been observed that the heat transfer rate increases with increase in the flow velocity, whereas, Cp and Cf reduces. The semi-circular pin fins are configured by keeping volume of the system constant and splitting a cylindrical pin fin into two. This helps in increasing the wetted surface area of the system. In the present configuration of pin fins, increment in the surface area does not always ensure increment in the heat transfer rate. However, it affects the pressure drop and frictional flow losses in the system. Increase in the flow velocity, in a particular LP-TP configuration enhances the convective heat transfer rate, which is same for the cylindrical pin fin. The observation on Cp and Cf are also similar as that of cylindrical pin fin. In most of the cases of semi-circular pin configurations, the systems at 0.5 m/s inlet velocity yields lower h, Nu, Cp and higher Cf, when compared with cylindrical fins. The semicircular pin fins with TP=D/2 and zero LP results in increment of h, Nu and Cf by 6.8%, 6.72% and 46.8%, respectively, when compared with cylindrical pin fin at same inlet velocity of 0.5 m/s. For the same configuration, the value of Cp is observed to reduce by 33.92%. Out of all the considered cases, the lowest value of h (=19.9 W/m2K) and Nu (=7.59) is observed for case LP=D/4 and TP=D/8 at inlet velocity 0.5 m/s. At same inlet velocity, the maximum value of Cp (= 0.96) and Cf (=0.117) are found for the cases with (LP=D/2, TP=D/6) and (LP=0, TP=D/4), respectively. The minimum value of Cp (= 0.605) and Cf (=0.084) for the same configuration of semicircular pin fin is observed for (LP=0, TP=D/2) and (LP=D/2, TP=D/2), respectively.

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