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Effects of A Pyramidal Pin Fins on CPU Heat Sink Performances

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ABSTRACT

The present investigation is carried out to analyze the forced convection heat transfer and fluid flow characteristics in a pin fin heat sink (PFHS). A new pins design has been proposed, which consists of a pyramid pin fins form characterized by a parameter named the ratio of pyramid (ROP), varies from 0 to 1 with a step of 0.2. A three other configurations (cylindrical, rectangular and square) are well carried out to validate the new design. The investigations are achieved using COMSOL Multiphysics 5.4 software based on the governorate finite element method for a Reynolds number ranging from 8547 to 21367. Some numerical results are validated with existing experimental data and a satisfactory agreement is found. The numerical results show the important role of the pyramid pin fins shape in the hydro-thermal performance enhancement, where the case of ROP = 1 (rectangular pin) ensures highest hydro thermal performance factor (HTPF) of 2,1, which it merits to considered in the PFHSs design.

1. Introduction

The fast development of electronic devices with high powers, generate another defy which presents the heating problem due to the Joule effect's that makes us forced to reduce or evacuate this unwanted heat to maintain a normal functionality of these devices, without forgetting the shape and volumetric design which becomes smaller. These challenges motivated experts for finding new techniques to increase the efficiency of the cooling system of these equipment's to keep pace with this technological development.

The design strategy of heat sink device is based on the measures to increase thermal dissipation coefficient and to reduce the pressure loss across the cooled device. The insertion of pin fin inline or staggered array on the heated surface that we want to cool is one of the most reliable methods applied to heat sink designs [1-3].

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In this field, several papers were carried out to study the influences of the pin arrangements on the thermal performances. Ozsipahi *et al.,* [4] analyzed numerically the thermal and hydro-thermal performances of a heat sink having aluminum honeycomb fins. They tested several geometrical and physical parameters such as the fin height (H), the distance between the fins in the stream wise direction (Sy) and the Reynolds number (Re). They reported that the thermal resistances of the honeycomb heat sink (R_{th}) reductions with augmenting fin height (H). Also, the heat transfer coefficient and pressure drop increasing with Reynolds number increases. Haghighi *et al.,* [5] examined experimentally the thermal performance of plate fins and plate cubic pin-fins heat sinks. They tested natural convection regime, for a Rayleigh number ranging from 8×10^6 to 9.5×10^6. They concluded that the greatest heat sink arrangement was observed in the plate cubic pin fin with 7 fins and 8.5 mm fin spacing, these configurations ensure an increase in the heat transfer coefficient between 10 and 41.6 % compared to the case of the normal pin fin.

However, the establishment of hot pockets/points in the downstream area of pin fin is caused from the formation of recirculation regions at low velocity [6]. For this purpose, the perforation of pin fins can reduce these areas by mixing the stagnated flow. Meinders *et al.,* [7] reported that the local heat dissipation coefficient can affect by the size of the recirculation flow zones behind the pins. Also, Sara *et al.,* [8] concluded that the perforated pin fins cause a better energy performance. Huang *et al.,* [9] numerically estimated the optimal perforation diameters of perforated pin fin array. They used commercial code CFD-ACE+ and the Levenberg–Marquardt Method (LMM). They showed the perforated pin fins create a lower pressure drop compared with the solid fin pins. Chin *et al.,* [11] carried out an experimental and numerical study to investigate the effect of the perforation numbers and the diameter of perforation in a staggered arrangement pin fins, in order to improve the heat transfer coefficient in these devices. The author's demonstrated that the Nusselt number for the perforated pins improved by 45 % compared with the solid at higher number of perforations. In addition, the pressure loss with perforated pins is also reduced by 18 % compared with the solid pins. However, the thermal dissipation reductions significantly once the pin diameter ratio to perforation diameter exceeds 0.375.

The pin fin shape is another important issue for thermal performance in the heat sinks, which encouraged several researchers to modulate mathematically different CPU heat sink shapes [12-14]. Khoshvaght-Aliabadi *et al.,* [15] tested numerically the hydro-thermal performances of corrugated miniature heat sinks (MHSs) and others shapes. They showed that the thermal execution of a corrugated MHS with plate pin fins is better than that of a corrugated MHS with plate fins. Where, the investigation of any shape habitually passes by studying of different flow phenomena such as recirculation zones, vortex shedding, and the formation of reattachment zones [16-17]. Yang and Peng [18] tested numerically the heat transfer and fluid flow characteristics of plate-circular pin-fin heat sink. They showed that the plate circular pin has a better thermal performance than the plate fin heat sink. Sajedi *et al.,* [19] studied numerically the effect of splitter on the hydro-thermal characteristics of a pin fin heat sink. Its objective is carried out to increases the heat transfer area in a limited space. They concluded that for circular pin fin with splitter the pressure drop penalty decreases by 13.4%, and the thermal resistance reduces by 36.8%. Similarly, with square pins their results show a decrease of 8.5% and 23.8% respectively in pressure loss and thermal resistance.

Recently, several authors have been recommended that the nanofluids as a working fluid can improve the thermal performance of heated surface in CPU heat sinks [20-24]. In an experimental study, Naphon *et al.,* [25] used the T_iO_2 nanofluid as a cooling fluid in a micro-channel heat sink. They reported that the heat transfer performance of T_iO_2 nanofluid augments by 18.56% compared with the airflow. Khoshvaght-Aliabadi *et al.,* [26] conducted that addition of low weight fractions of Al_2O_3 nanoparticles (0.3% and 0.6%) to water leads to significant improvements in the overall heat transfer coefficient of all pin fin miniature heat sink (PFMHSs) with moderate pressure drop penalty. Azizi *et al.*, [27] analyzed Cu/water nanoparticles in a cylindrical micro-channel heat sink. They have shown that the increasing in the nanoparticles fraction from 0.05% to 0.3% can produce a lower thermal resistance of 21%. In addition, they proposed reliable correlations to calculate the Nusselt number and friction factor. Thus, comparing with the traditional heat sinks, nanofluids can produce a considerable improvement in the hydrothermal performance of heat sinks [28]. Also, these are other successes techniques used to enhance the PFHS performances such as the insertion of vortex generators on the plate oh heat surface [29]. The perforated space in the pin fins is another solution which helps to reduce the stagnated zones of flow behind the pins, and consequently augments the heat transfer coefficients [30-32].

In this paper, and in order to improve the PFHS performances, the effect of the ratio of pyramid (ROP) on heat transfer and fluid flow characteristics is analyzed. The ROP is varied from 0 to 1 with 0.2 of step. A three-dimensional numerical investigation of turbulent fluid flow and heat transfer has been realized in using CFD COMSOL 5.4 software. The non-isothermal flow predefined Multiphysics coupling model configured with the Reynolds-Average Navier-Stokes (RANS) models which include the standard k-ε turbulence model is used in this paper. Where, the standard k-ε presents a robust model from the literature [33-34].

2. Problem Description

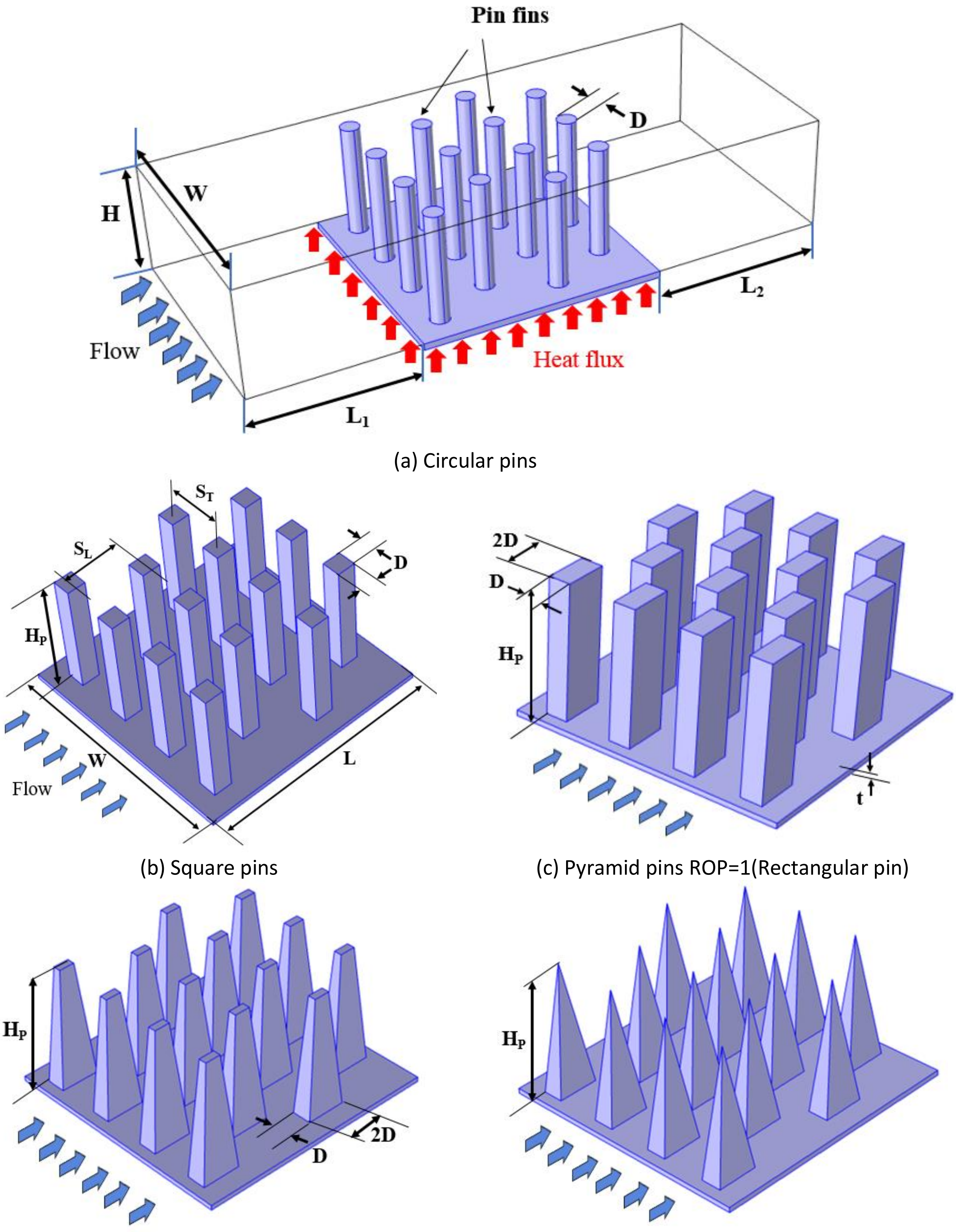
In this present study we used aluminum as a matter to build the pin fin heat sinks (PFHSs) for different geometries. The different heat sink configurations present the pin fins placed on the base plate having dimensions W × L (100*100 mm). These geometries are located inside channel, in manner that the zones before and after the studied arrangement (PFHSs) allow to develop flows and avoid the reversed flow phenomenon, respectively. Several arrangements have been studied in this paper: Cylindrical, square, rectangular, and pyramid shapes. The pyramid form is characterized by a parameter namely ratio of pyramid (POR), it varied from 0 to 1 with 0.2 of step. POR presents the ratio between the upper and lower surfaces of the pyramid. The section of cylindrical, square, rectangular pin fins is characterized by the diameter (D), and the height of pin fins (H_P) as shown in Figure 1. The pitch length ratio of pin fin (S_L/H) is defined as the ratio of pitch length of pin fins (distance between center to center the pin fins, or S_L) to the height of channel (H) and it is fixed to 0.5. Where, the pitch transverse ratio of pin fin (S_W/H) is defined as the ratio of pitch transverse of pin fins (S_W) to the height of channel (H) and it is set to 0.5 to the height of channel (H) is equal to the height of pin fins (H_P). The pin fins for all configurations are distributed over the base plate in staggered arrangements. These arrangements are obtained by changing alternate rows of pin fins by semi pitch transversal ($S_T/2$). The different configurations of pin fin heat sinks are exposed in the Table 1.

Table 1
Dimensions of PFHSs configurations

D (mm)	L (mm)	L_1 (mm)	L_2 (mm)	W(mm)	H(mm)	H_P (mm)	t(mm)
8	100	70	70	100	50	50	3

2.1 Governing Equations

Continuity, momentum and energy equations are used to governorate turbulent regime, incompressible flow and heat transfer through PFHE. At steady-state conditions, the governed equations expressed as follow



(a) Circular pins

(b) Square pins

(c) Pyramid pins ROP=1(Rectangular pin)

(d) Pyramid pins ROP=0.6

(e) Pyramid pins ROP=0

Fig. 1. Configurations of a pin fin heat sinks with different arrangements

Continuity equation

$$\nabla.(\rho u) = 0 \tag{1}$$

Momentum equation

$$(\nabla.u)\rho u = -\nabla p + \nabla.\mu(\nabla u + (\nabla u)^T) \tag{2}$$

Energy equation

$$\rho c_p u(\nabla T) = \nabla.(k\nabla T) + Q \tag{3}$$

The k–ε standard is turbulence model used in this paper. This model is largely used in the literature to predict the turbulent flows in channel fitted obstacles [6, 34] as the geometries analysed in the present study. The k–ε standard turbulence model is based on the energy dissipation ε, Eq. (4) and the turbulent kinetic energy , k Eq. (5).

$$\rho(u.\nabla)\varepsilon = \nabla.\left[\left(\mu + \frac{\mu T}{\sigma\varepsilon}\right)\nabla\varepsilon\right] + C_{sl}\frac{\varepsilon}{k}P_k - C_{cl}\rho\frac{\varepsilon^2}{k}, \varepsilon = ep \tag{4}$$

$$\rho(u.\nabla)k = \nabla.\left[\left(\mu + \frac{\mu T}{\sigma k}\right)\nabla k\right] + P_k - \rho\varepsilon \tag{5}$$

where the production term is given as follow

$$P_k = \mu_T[\nabla u:(\nabla u + (\nabla u)^T)] \tag{6}$$

The turbulent viscosity is modeled as

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{7}$$

The empirical constants for the standard k–ε model are given as follow

$$\Delta P = P_{In} - P_{Out} \tag{8}$$

where $P_{In}$and P_{Out} are the pressures in the inlet and outlet of channel, respectively. The thermal resistance (R_{th}) is calculated as follow

$$R_{th} = \frac{T_W - T_{In}}{q^{//}} \tag{9}$$

where T_w, is the average wall temperature, T_{In} is the air inlet temperature, $q^{//}$ is the heat flux applied to the base plate. The Reynolds number (Re) is expressed as follow

$$R_e = \frac{UD_h}{\nu} \tag{10}$$

where, D_h is the hydraulic diameter, ν and U are the kinematic viscosity and the mean air inlet velocity respectively.

For determining the convection heat transfer coefficient inside the channel, Nusselt number is used in this paper and it is given by

$$Nu = \frac{q^{//}D_h}{k_{Air}\left(T_W - \frac{T_{Out}+T_{In}}{2}\right)} \tag{11}$$

where, T_{in}, T_{out}, T_W are temperature of air flow at inlet, outlet of channel and the mean temperature of heated surface which is in contact with air flow. $q^{//}$ is the heat flux which is fixed at 5903 W/m^2.

For the hydrothermal evaluation, the hydro thermal performance factor (HTPF) presented (Eq. (12)) bellow is very important paramater used to evaluate overall performace of heat sinks.

$$\mathrm{HTPF} = \frac{\left(Nu_{Pins}/Nu_{Cylindrical}\right)}{\left(\Delta P_{Pins}/\Delta P_{Cylindrical}\right)^{1/3}} \tag{12}$$

3. Numerical Model

3.1 Boundary Conditions

A uniform temperature and velocity have been used at the domain inlet and the symmetry boundary condition is employed to minimize the time of convergence. In the outlet, the atmospheric pressure is considered; the no-slip boundary condition is imposed on the solid surfaces. Air at standard conditions is considered as the working fluid. The aluminum of A8350P type and 167 W m^{-1} K^{-1} of thermal conductivity is selected as a material for the base plate and the pin fins. The thermo-physical properties of air and aluminum are assumed to be constant. For the thermal aspect, a constant heat flux ($q^{//}$) of 5903 W/m^2 have been used in the base plate of PFHSs, where the temperature of 293.15 °K was fixed in the inlet.

3.2 Mesh Selection and Solver Settings

COMSOL Multiphysics version 5.4 is the computer code that we used to perform all the simulations of this study, the software provides enough operations, tools, and functionality that enable us easily to build geometry, create mesh, run simulation and extract the results for interpretation.

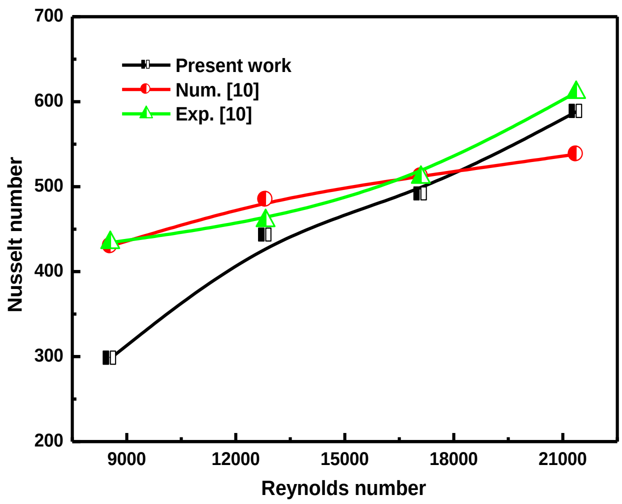
After having selected the physics that describe all the phenomena of the present study, the geometry of the system investigated was created and meshed by exploiting the tetrahedral mesh algorithm provided by COMSOL. The following series of grids: 1205480, 1687942, 2103517, 2301458 and 2548741are tested for cylindrical pin fins, where the results deviation of Nusselt number do not less than 0.8 % beginning the grid of 2103517element. Therefore, this grid was selected as an accepted case for the next simulations. The same strategy was then used to examine the grid independence for the other configurations. Then for: square, ROP = 0, 0.2 0.4, 0.6, 0.8 and 1, the selected meshes are 1 914 578, 2 001 457, 2 144 558, 2 298 741, 2 110 214, 2 221 984 and 2 093 239 elements, respectively. Where, in this situation the highest error of deviation does not less than 1.2 %.

COMSOL Multiphysics 5.4 which is a multiplatform based on the finite element method, is the computer Multiphysics simulation software used to analyze the turbulent fluid flow and heat transfer phenomena for the present study by using no-isothermal flow interfaces under the integrated conjugate heat transfer models. To solve the governing equation the segregated solvers with the Generalized Minimal Residual (GMRES) iterative methods are used, the factor error and tolerance that used are respectively 20 and 0.001. Otherwise and in order to accelerate the convergence, the geometric Multi-grid solver is used with the pre-conditioner Parallel Sparse Direct Linear Solver (PARDISO). The numerical simulations were realized on a PC-i7 with a CPU frequency of 3.7 Go and a RAM of 16 Go. A typical model of running time for calculation of one case is about seven hours.

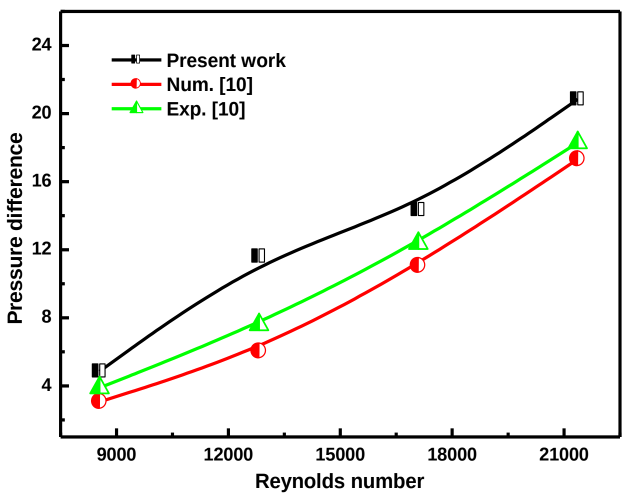
4. Results

4.1 Validation of Numerical Model

After the tests grid sensibility of all numerical domains, it is necessary to validate the numerical solver. Therefore, the heat transfer and flow behaviours are compared with experimental and numerical data from literature. The present investigation is realized for cylindrical pin fin heat sinks with the same geometrical and thermo-physical properties that presented experimentally and numerically by Chin *et al*., [10]. Figure 2(a) and (b) shows comparison of Nusselt number (Nu) and pressure drop (ΔP) results. The figure shows a satisfactory agreement between the present numerical results and that reported experimentally and numerically by Chin *et al*., [10]. Where, the deviation does not exceed 5 % for Re ≥12000. A considerable deviation of 30 % in Nusselt number results is observed for feeble Reynolds number (Re = 8547) due to the not enough points used to measure temperature in inlet and outlet sections, where the difference in fluid temperatures between heated surface and the middle of section is very important which need several thermocouples to measure average temperature. For pressure drop deviation, the comparison with the present numerical results showed a difference of 30 % and 22 % with experimental a numerical result of Chin *et al*., [10], respectively.



(a)



(b)

Fig. 2. Comparison of (a) Nusselt number and (b) pressure difference results

4.2 Hydro-Thermal Behaviours of Pyramid Pin Fins

The cylindrical and square pin fin heat sinks are largely used in literature, but its real exploitation results several problems such as the formation of lower heat transfer areas or LHTAs behind the tubes, which its lead to reduce the heat transfer performances of PFHS [6]. Otherwise, the formation of stagnated zones behind of these configurations not only decrease of heat transfer coefficient but also increase the pressure drop penalty with are augment the cost of ventilation power [35]. Where, the increase of velocity fluctuations downstream of the tubes presents a main technique to enhance the thermal performance PFHSs. In this paper, we focused on the pyramid pin fin heat sinks at different values of ratio of pyramid namely ROP which is varied from 0 to 1 with 0.2 of step. Where ROP = 0 presents the classical pyramid, and ROP = 1 presents a rectangular form (i.e. the lower and upper surfaces of the pyramid are identical). Habitually, it is known that increasing of heat transfer surface increases heat transfer execution, but not always if we considered others physical phenomenon such as recirculation zones and velocity fluctuations near of the walls.

There is an essential relationship between the establishment of recirculation regions and LHTAs [6, 36-37]. For this purpose, we choice the wall axial velocity (Figure 3) and temperature contours (Figure 4) in this section as a better linking between the velocity and temperature.

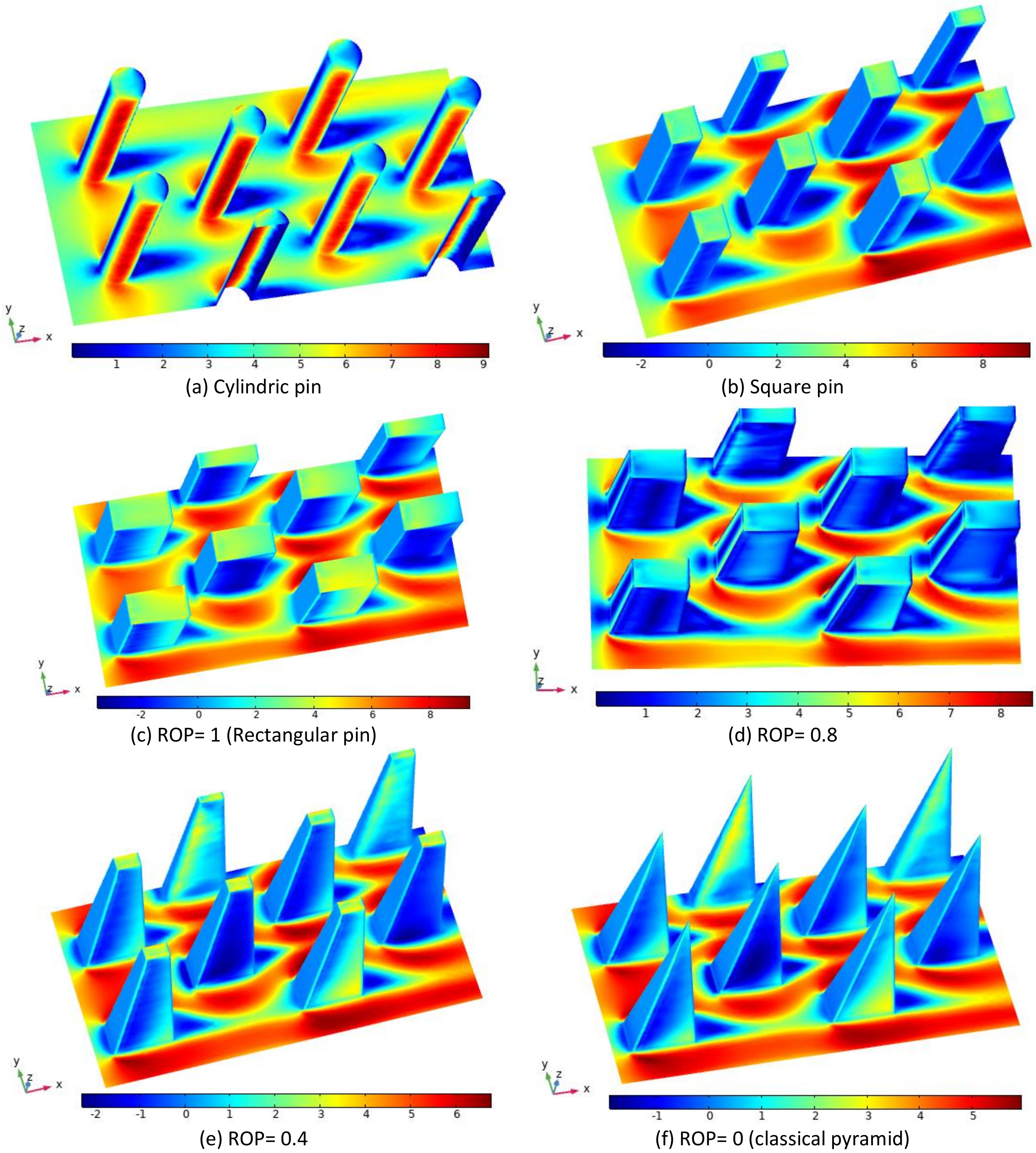
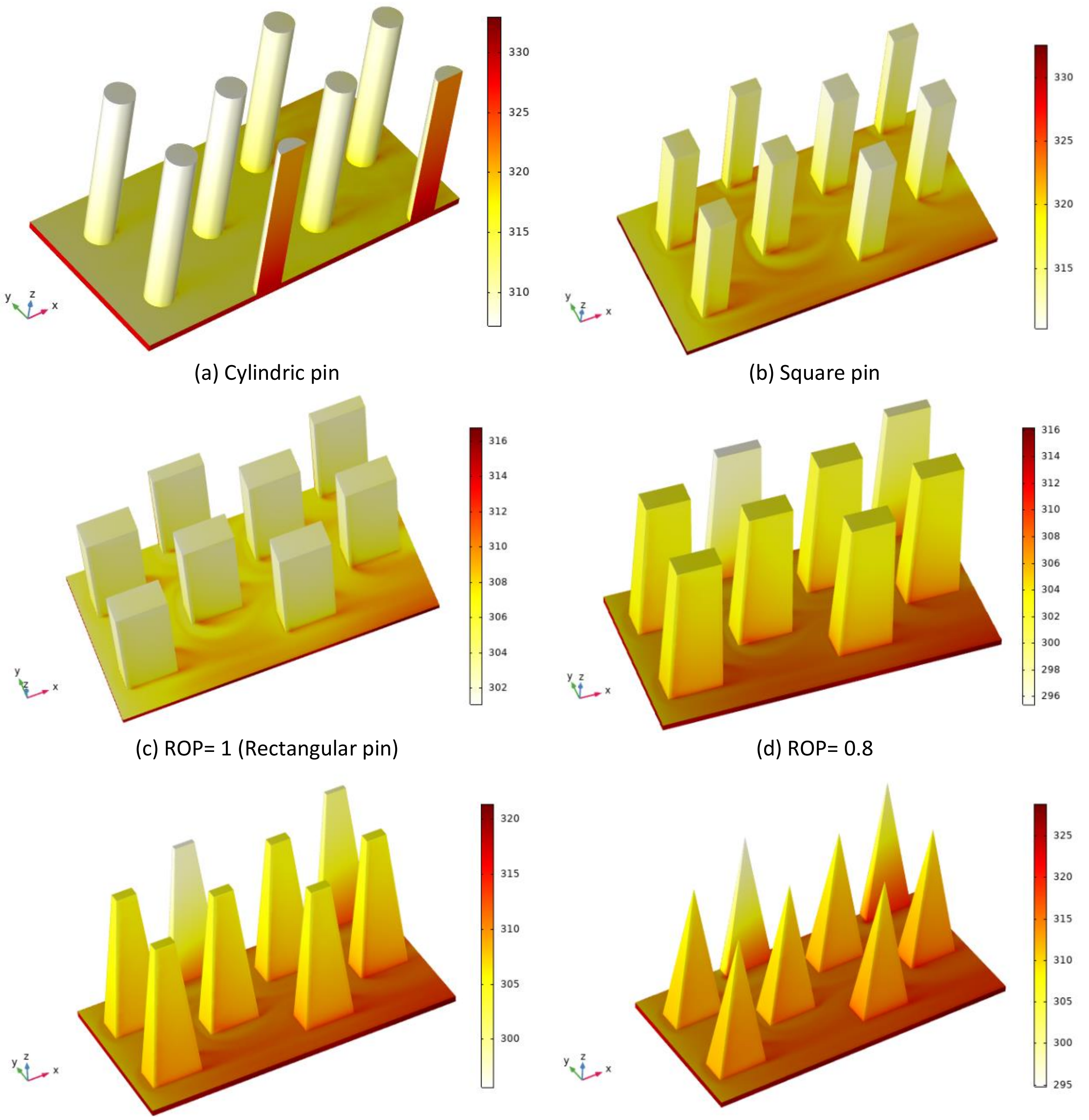


Fig. 3. Distributions of the wall velocity for different configurations of pin heat sinks (Cylindrical, square, ROP = 1, 0.4, 0,8 and 0) at Re = 21367

where, the axial velocity and temperature distribution explain the recirculation flow zones (stagnated regions) and hot pockets (LHTAs), respectively.



(a) Cylindric pin (b) Square pin

(c) ROP= 1 (Rectangular pin) (d) ROP= 0.8

(e) ROP= 0.4 (f) ROP= 0 (classical pyramid)

Fig. 4. Contours of the temperature distributions of different configurations of pin heat sinks (Cylindrical, square, ROP = 1, 0.4, 0,8 and 0) at Re = 21367

Therefore, and in order to explain the effect of the new design of this paper, Figure 3 displays the distribution of walls axial velocity for cylindrical, square, ROP = 1 (Rectangular pin), ROP= 0.8, 0,4 and 0 (or classical pyramid), for Re = 21367. The establishment of recirculation zones (or stagnated zones) of the air flow downstream of the pin fins is clearly appearing almost for all configurations. But, these zones (in blue color) begin to disappear following the decreases of ROP. On the others hand, the reattachment zones (in red color) begin to appear clearly according to the decreases of ROP. Also, the decreases of ROP allow augmenting not only on the base of heat sinks but also it augments the velocity fluctuations near of the lateral walls of pyramid pin fins. Compared with cylindrical and square pin fins heat sinks, the configurations of pyramid pin fins create larger reattachment zones which are present a suitable advantage for better execution of heat transfer. Hence, the diminution of POR helps to facility the fluid flows between pin fins heat sinks. Therefore, as new design, the pyramid pin fins present a better configuration which merits to be considered in the design of PFHSs.

As conferred previously, there is a vital relation between the establishment of low speed zones (stagnant regions) and the establishment of LHTAs (hot pockets). Where, Figure 4 clearly displays this association while the fluid behaviour affects directly the heat transfer characteristic of PFHSs. Certainly, the increase in velocity fluctuations aids to decline the establishment of LHTAs, and consequently, enhance the PFHS performance. Thus, the pyramid pin fin ensures the smallest stagnation area, which merits to be considered in the PFHSs.

In addition, the highest temperature of the various configurations is observed at the base of the hot plates, as well as in the corners formed by the base and the pins due to the formation of the recirculation zones at low speeds. Indeed, the temperature is high on the base of the pins and it begins to decrease upwards. For this reason, the pyramid shape has been proposed to minimize the heat exchange surface in the upper parts of the pins and we are increasing this area near the source of heat flux. Also, it is very clear that the outlet air temperatures are higher than the inlet. Because, through the heat sink, the air captures the heat through the base and the pins walls and comes out hot.

4.3 Effects of Pyramid Pin Fins on Heat Transfer Performance

The effect of the pyramid pin fin shapes on the Nusselt number (Nu) is shown in Figure 5. From this figure, Nusselt number rises with Reynolds number increasing for all configurations. The Nusselt numbers of all configurations are higher than that of the cylindrical pin fin and hence providing higher thermal dissipation. Additional expressively, thermal dissipation is higher with pyramid pin fins than with cylindrical pins. Generally, Nusselt number rises with increasing ratio of pyramid (ROP) until ROP = 0.8, where this last presents higher values of Nusselt number. Such effect is due to the increase in thermal transfer surface area. Also, the pyramid shape coupled with staggered pins facility flows for creates better flow mixture between and behind pin fins. Beginning and the end of Reynolds numbers interval, square, ROP = 0 (classical pin), 0.2, 0.4, 0.4, 0.6, 0.8 and 1(rectangular pin) provide higher Nusselt number values of about 35 % to 31 %, 23 % to 19 %, 18 % to 16 %, 48 % to 42 %, 53 % to 50 %, 59 % to 57 % and 55 % to 53 % compared with cylindrical pin, respectively. Therefore, the maximum heat transfer is obtained from pin fins with ROP = 0.8, at Re = 8547 in the present paper.

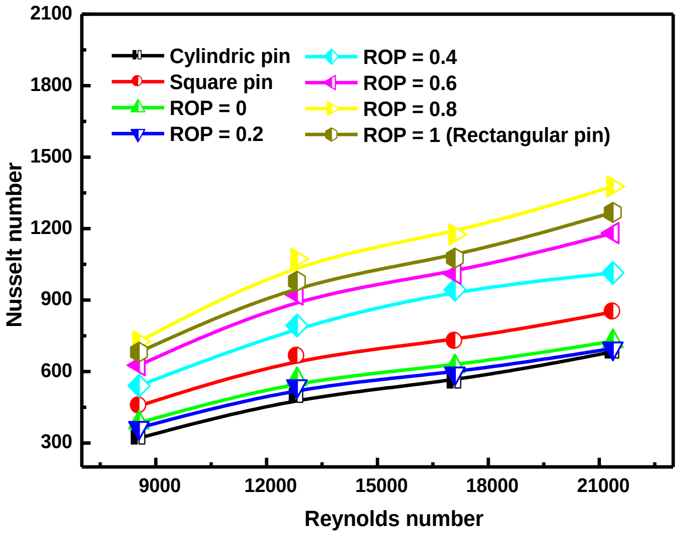


Fig. 5. Variation of Nusselt number (Nu) vs. Reynolds number for all arrangements

4.4 Effects of Pyramid Pin Fins on Pressure Drop and Thermal Resistance

The pressure drops results for various Reynolds numbers extended from 8547 to 21367 are illustrated in Figure 6 for pin fin heat sinks for different configurations. The cylindrical, square, and six values of ROP (0, 0.2, 0.4, 0.8 and 1) are the proposed configurations that had been analysed to estimate its effects on the pressure drop. It is seen that the pressure drops values are increased proportionally with the increasing in the Reynolds number values due to the augmentation of turbulence intensity and the heat transfer surfaces. Square pin, ROP = 1 (rectangular pin) and ROP= 0.8 create highest values of pressure drop (ΔP) of about 73 %, 64 % and 72 %, respectively at Re = 21367 compared with cylindrical pin. Where, at the lower Reynolds number value (Re=8547), the same configurations generate the lowest pressure drop value (ΔP) of about 62 %, 51 % and 61 % for the same comparison strategy. Because there is an acute contact between the air flow and the surface of the pins (square pyramid and ROP = 0.8) which creates a blockage and a change of the flow direction towards the pin sides, and as a consequence, these physical phenomena are responsible on the pressure loss. On the other hand, the cylindrical shape does not block the flow, where the ai flow easily passes without any significant blockage of the flow on the lateral walls of the cylindrical pins.

Also, cylindrical and ROP = 0 cases, give lower values of pressure difference where are almost identical. All the others a configuration ensures the highest difference losses values compared with the cylindrical pin fin as a baseline case. The effect of the Reynolds number (Re) on the pressure drop (ΔP) and the thermal resistance (R_{th}) are exposed in Figure 6 and 7. The passage of the airflow over the heat sink, the air loses it pressure and captures some quantity of heat emitted by the plate base and pins.

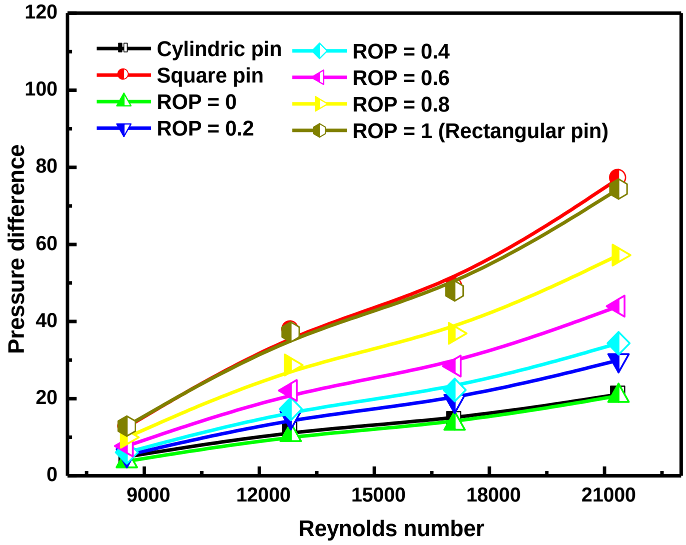


Fig. 6. Variation of pressure drop (ΔP) vs Reynolds number for all arrangements

It is evident from the figures that the increase in Reynolds number (Re) results in a decrease in thermal resistance (R_{th}) due to the increase in heat transfer resulting from the turbulent effect, whereas causes an increase in the pressure drop (ΔP). Thus, note that the thermal resistance (R_{th}) is very high in the case of cylindrical pins (Baseline) compared to other configurations. Of course, the increases in perimeters and the heat exchange surface in the pyramid pins help to reduce the thermal resistance and therefore increase the heat transfer coefficient. The cases of ROP = 1, 0.8 and 0.6 generated minimal values of the thermal resistance. For the Reynolds number ranging from 8547 to

21367, these three configurations provide a decrease in thermal resistance of about 122 % to 133 %, 143 % to 150 %, and 112 % to 117 % compared to the cylindrical pin. Fortunately, the case of ROP = 0.8 not only ensures a better heat transfer coefficient but also it provides a low thermal resistance compared to other cases.

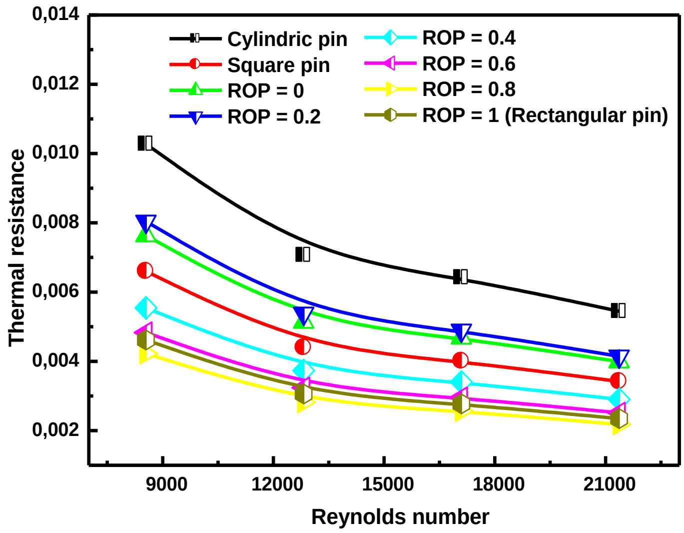


Fig. 7. Variation of thermal resistance (R_{th}) vs Reynolds number for all arrangements

4.5 Effects of Pyramid Pin Fins on Hydrothermal Performance

As such thermal application, the microelectronic cooling system need high heat transfer coefficient, low pressure drops penalty. Therefore, the better design of heat CPU heat sink which is considered the relationship between the heat transfer execution and the associated pressure drops. In this context, the hydrothermal performance (HTPF) can explain this relationship and done the better configuration. Figure 8 shows the variation of the hydrothermal performance factor (HTPF) versus Reynolds number. For all configurations, the HTPF is superior the 1 except square pin.

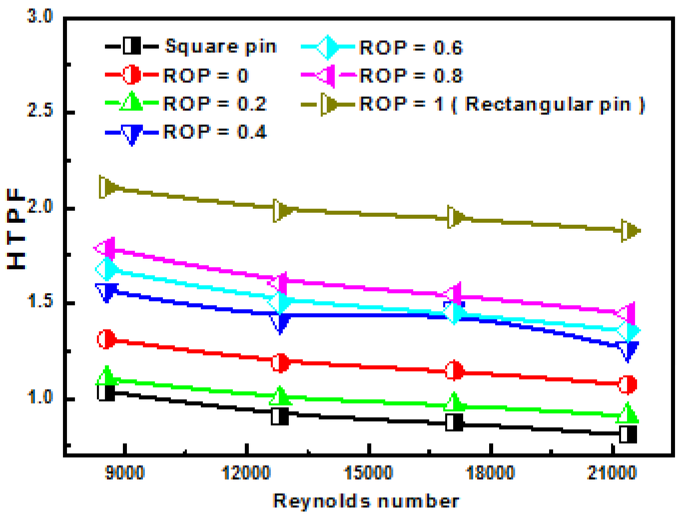


Fig. 8. Variation of Hydro-thermal Performance factor vs Reynolds number

Also, it is clear from the figure that the HTPF augment according the rises of the ROP, where the HTPF is varied from 1,1 to 2,1. The ROP = 1 (or the rectangular pin) ensures highest HTPF with 2.10 at lowest Reynolds number values, where the HTPF is varied from 2.10 to 1.87 according the range of Reynolds number. Consequently, The ROP = 1 (or the rectangular pin) merits to considered in the CPU heat sinks design.

5. Conclusion

In this study, numerical simulation has been achieved to enhance the hydro-thermal performances of the PFHS by suggesting of pyramid pin fins as a new design and others classical configurations. The pyramid shapes are characterized by a ratio named ratio of pyramid with is varied from 0 to 1 with 0.2 of step. Simulation have been performed using CFD COMSOL 5.4 software for Reynolds number ranging from 8547 to 21367 in the hydraulic diameter. The obtained results demonstrate that the insertion of the pyramid pin fins shape on base plate of heat sinks will enhance both thermal and hydrodynamic aspects inside the heatsink by decreasing the pressure drop around the pins and decreasing the thermal resistance of heat sink. Where, the ROP = 1 (or the rectangular pin) ensures highest HTPF with 2.10 at lowest Reynolds number value, where the HTPF is varied from 2.10 to 1.87 according the range of Reynolds number. Therefore, The ROP = 1 merits to considered in the CPU heat sinks design.

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