Numerical Investigation of Thermal Performance of Micro-Pin Fin with Different Arrangements

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Abstract. The impact of micro-pin fins on heat transfer performance in this study is analyzed numerically. Navier-Stokes equations are solved by steady-state three dimensions turbulent forced convection. Various numbers of micro-pin fins with in-line configuration and different micro-pin fins configurations such as staggered configuration additionally to the in-line configuration are explored. In addition, comparison between micro-pin fins duct and smooth duct is studied. Results illustrate that the thermal performance was enhanced by 7.9 - 9.3% with increasing the number of micro-pin fins from 902 to 1312 and by 10.2 - 11.1% with increasing the number from 902 to 1640. Also, displays that the heat transfer performance in the staggered configuration is slightly better than the in-line configuration by 2.6 - 4.2% with the same micro-pin fins number. Moreover, the micro-pin fins duct gave an increase of 30.24% in heat transfer performance relative to the smooth duct.

Keywords: Micro-pin fins, Micro cooler, Micro heat sink, Passive cooling, Micro cooling techniques.

Nomenclature:
x, y, z: Cartesian coordinates.
ρ: Density of air.
U, V, W: Velocities in X,Y,Z directions.
Γ: Diffusion coefficient.
φ: General dependent variable.
T_p: Plate temperature.
i, j, k: Indices which indicates positions in the (x,y,z) directions.
A: Area.
W: Relaxation factor.

1. Introduction
Micro-pin fins are a passive technique used to improve the heat transfer efficiency of mechanical or electronic constructions. Micro-pin fins are used to dissipate heat produced by mechanical or electronic devices into a fluid environment, often air or liquid coolant, to keep the optimum temperature [1]. In the domain of passive cooling devices, micro-pin fins are the most differentiated technique that can intervene with fluid and dissipate the boundary layer. In addition, small working fluid inventory requirements [2] with low cost. Micro-pin fins use in semiconductor industry and to reduce the temperature from projector and LED. Also to minimize the temperature of graphic chip and high power laser [3],[4]. There are various
researches in the field of micro-pin fins, Mei et al. [5] researched the impact of tip clearance in micro-reactor on both pressure drop and thermal efficiency of micro-reactor with micro-pin fins numerically and experimentally. This study was done to improve hydrodynamic efficiency at small numbers of Reynolds. The results illustrated that the pressure drop was enhanced when the tip clearance increased. It was also discovered that thermal efficiency has a inverse relationship with tip clearance. Moreover, the comparison between micro-pin fins with and without tip clearance displayed that the thermal efficiency of the hc / hf ratios equal to 1.1 or 1.2 micro-pin fins was better than that of the Reynolds number without tip clearance between 33 and 350. Ndao et al. [6] investigated experimentally four distinct forms (square, circular, elliptical, hydrofoil) were used to show the impact of both distinct micro-pin fins forms and cross-flow area on the heat efficiency of jet impingement with micro-pin fins. As for Re, the working fluid (R134a) Re ranged from 8000 to 80000. Results have shown that the cross-flow area heat enhancement impact can be seen at elevated Re, but the main impact in the experiment was on the distinct micro-pin fin forms, and this was obvious when contrasting micro-pin circular and square fins with other micro-pin fin configurations. Yeom et al. [7] experimentally and numerically analyzed the thermal performance impacted by the use of both active and passive methods in a single channel. Active component depicted by a piezo-electric temporary agitator used to produce turbulent air using a blade of oscillation. As far as active method is concerned, eleven samples with two instances of distinct dimensions (H250D400) and (H400D250) were used, representing a passive cooling method. Results indicates that the operation of micro finned surface and agitator improved thermal transfer efficiency in various proportions. In the case of the use of a 920 Hz agitator with micro-pin fins with dimensions (H400D250), the heat transfer coefficient was increased by almost 230 percent compared to the non-agitator flow rate of 40-60 LPM. Almost 250 percent enhanced in the plain wall as well. Guan et al. [8] experimentally tested the effect of three different shapes of micro-pin fins, elliptical shape diamond shape, and circular hydrophobic shape on both pressure drop and friction factor, using ultra-pure water as a working fluid. Four specific contact angles (99.5, 119.5, 151.5) were used in this study. Hua et al. [9] experimentally investigated the impact on flow resistance, friction factor (FF) and pressure drop (AP) of different micro-pin fin shapes and sizes when high-pressure nitrogen flows through it. Qian et al. [10] studied the impact of a non-uniform micro-pin fins array (NMPFAR) contrast to the impact of a uniform micro-pin fins array (MPAR) on both thermal performance and flow field was examined numerically and experimentally. In addition, the effect of distinct (NMPFAR) in three cases with three types of diameter (two, four, eight) on heat transfer performance was evaluated numerically. Chiu et al. [4] numerically and experimentally analyzed the impact of micro-pin fins on the cooling effectiveness of the liquid. Aluminum models of the micro-pin fins board have been manufactured to evaluate both the thermal performance and the flow features. The effects of both pressure drop and geometrical parameters on thermal performance are also studied. Jia et al. [11] studied the impact of micro-channel integration with micro-pin fins on both fluid flow and heat efficiency at Re between 147 and 637. Four instances of cone-shaped micro-pin fins are evaluated in the upper, center, downstream, and sub divided across the micro-channel. The main factors are the relative diameter of the fins, relative fin space, and relative fins height. Naphon and Wiriyasart [12] investigated experimentally the effect of many parameters such as pulsating flow, nanofluids, and magnetic field on both flow characteristics and thermal performance of micro-finned tubes comparison with smooth tubes. All Parameters illustrated direct relationship with the thermal performance. Li et al. [13] studied the effect of integrating the micro pin fins and piezo electric fan on flow and heat transfer performance of heat sink experimentally. Significant factors are taken into considerations in this study like vibrating fan’s Reynolds number, the ratio of fan tip to the heated surface, and Re of channel flow. Ambreen et al. [14] showed numerically the impact of two parameters (distinct forms of micro-pin fins and nano fluid) on
the heat transfer performance of the micro-pin fins surface with the introduction of a discrete phase model (DPM). Kewalramani et al. [15] analyzed experimentally and numerically the hydrothermal performance by applied constant heat flux by a heater from the base on elliptical micro pin fin heat sink. The area of the heat sink is 11 mm length and 10 mm width contains twenty-five elliptical micropinfins(five columns and five rows) having 2.02 mm major axis and 1.84 mm minor axis with 130 µm height.

In the present study, the impact of different micro-pin fins numbers and configurations on the thermal performance and flow field are studied numerically. In addition, the impact of the micro-pin fins duct and smooth duct on the heat transfer performance are compared numerically.

2. Mathematical Model and Numerical Analysis
The governing three-dimensional partial differential equations were numerically analyzed to predict the behavior of heat transfer and turbulent flow through the micro-pin fins duct. The governing equations that categorize heat transfer and fluid flow in the micro-pin fins are focused on mass, momentum and energy conservation formulas.

2.1. General Formula for Governing Equations
The general form for all partial differential equations is:

$$\frac{\partial}{\partial x}(\rho U \phi) + \frac{\partial}{\partial y}(\rho V \phi) + \frac{\partial}{\partial z}(\rho W \phi) = \frac{\partial}{\partial x}(\Gamma \frac{\partial \phi}{\partial x}) + \frac{\partial}{\partial y}(\Gamma \frac{\partial \phi}{\partial y}) + \frac{\partial}{\partial z}(\Gamma \frac{\partial \phi}{\partial z}) + S$$

(1)

The first term on the left is the convection term while the right term is the diffusion and source expressions.

2.2. Thermal Analysis of Solid Walls
Solid walls with micro-pin fins is emulated with a steady-state3-dimensions model. The goal of any heat system emulation is to assess the impact of distinct factors on device results under varying working conditions. The technique of energy equilibrium is implemented to small components with 6 faces. This method states that summing in and out of heat is equivalent to zero. Fig. 3 displays the energy balance of node 2.

$$T_p (i,j,k) = T_p (i,j,k) \times (1 - \omega) + \omega \times (A_e \times T_p (i + 1,j,k) + A_w \times T_p (i - 1,j,k) + A_n \times T_p (i,j + 1,k) + A_t \times T_p (i,j,k + 1) + A_b \times T_p (i,j,k - 1) + q) \div A_p$$

(2)

Figure 1: Energy balance at node 2
2.3. Analyzing of Flow and Physical Model
To analyze the impact of different micro-pin fins numbers on the heat transfer performance, three model with various dimensions are analyzed. The length (L), width (W), and height of the duct are 200 mm, 100 mm, and 10 mm respectively. **Fig. 1** displays the distribution of the micro-pin fins throughout the duct of all three model. The dimensions of the three models are shown in **Table 1**

![Figure 1](image)

(a) 902 MPFs

![Figure 1](image)

(b) 1312 MPFs

![Figure 1](image)

(c) 1640 MPFs

Figure 2: Comparison between the results of different numbers of in-line micro-pin fins

As for the comparison between the staggered and the in-line configurations, two simulation models was established with same duct dimensions (200 mm * 100 mm * 10 mm)
Table 1: The dimensions of different micro-pin fins numbers models

| Different in-line micro-pin fins numbers | Diameter (mm) | SL (mm) | ST (mm) | Height (mm) |
|----------------------------------------|--------------|--------|--------|-------------|
| 902 micro-pin fins                      | 0.875        | 3.96   | 3.78   | 10          |
| 1312 micro-pin fins                    | 0.875        | 3.96   | 2.27   | 10          |
| 1640 micro-pin fins                    | 0.875        | 3.96   | 1.52   | 10          |

Figure 3: Simulation models of the two configurations

and same micro-pin fins numbers (924 micro-pin fins). In Fig. 2 the two configurations are shown. The same longitudinal distance (3.96 mm), transverse distance (3.78 mm), height (10 mm), and diameter (0.875 mm) are considered for both staggered and in-line configurations.

From the right side, air joins the duct at an inlet temperature that is considered to be equal to the ambient temperature (45°C) and passes through the all various micro-pin fins arrangements. The hot air is coming out of the right side at the output temperature. The duct’s walls are isolated from all sides while the steady-state 3-D model calculates the temperature inside the duct. The mathematical definition of fluid flow issues is controlled by the fundamental conservation of volume, momentum and energy equations. The previous assumptions were created, considered turbulent flow in the current work:

- 3-dimensional conservation equations.
- Incompressible Steady-state flow.
- 3-dimensional conduction of heat transfer through the micro-pin fins plate.
• All features are evaluated at an average temperature.
• The heat is transferred by forced convection
• The body forces are negligible.

3. Results and Discussion
3.1 Effect of Different Micro-Pin Fins Numbers with In-line Configuration
In this section, the impact of distinct in-line configuration on the heat transfer efficiency of micro-pin fins is investigated. The test was done for three different airspeeds and heat fluxes. The effect of three different numbers on the thermal performance in case of 2 m/s and 250 Watt is displayed in Fig. 4(a). The results illustrated that the heat transfer performance had a direct relationship with the number of micro-pin fins, it was enhanced by 8.4% when increasing the number of micro-pin fins from 902 to 1312, and nearly 10.2% when increasing from 902 to 1640 micro-pin fins. In Figs. 4(b) and 4(c) the effect of various micro-pin fins numbers are displayed for two cases 3 m/s with 250 Watt and 4 m/s with 250 Watt respectively. Relative to the results obtained, the thermal performance was improved by 9.3% and 7.9% respectively when increasing the number of micro-pin fins from 902 to 1312 while nearly 11.1% and 8.8% when increasing from 902 to 1640. The thermal performance enhanced with increasing of micro-pin fins number because both surface area and obstruction of air were increased.

3.2 Comparison between In-line and Staggered Micro-Pin Fins Configurations
In this test, three cases of airspeeds and heat fluxes were applied for both in-line and staggered arrangement which displayed in Fig. 2. The air temperature variation throughout the micro-pin fins duct at 1 m/s with heat flux 250 Watt is displayed in Fig. 5(a) where the air entered the duct with temperature 45°C and was rising gradually but not equally because at the staggered arrangement the temperature difference is slightly higher than the in-line arrangement. The air exit at 71.1°C in staggered arrangement while in the in-line arrangement exit at 68.1°C which means that the thermal performance enhanced by 4.2%. In the case of 2 m/s and 250 Watt, the thermal performance improved nearly 2.8% as shown in Fig. 5(b). In addition, Fig. 5(c) exhibits the temperature difference in the case of 3 m/s and heat flux 250 Watt for both two arrangements where the enhancement in thermal performance reached 2.6%. The thermal performance for three different airspeeds was shown in the above cases and it was evident that the heat transfer performance of the staggered configuration was slightly higher than that of the in-line configuration. Due to an increase in the effective contact region between the airflow and the micro-pin fins in the staggered configuration, this temperature distinction between the two configurations occurred. Also in the staggered configuration, the obstruction of air is more than that in the in-line configuration which means ensuring better heat transfer. Moreover, dissipation of the boundary layer is more in the staggered system because of the propagation of the micro-pin fins throughout the plate. The explanation for the slight difference in temperature between the staggered configuration and the in-line configuration in this analysis is the small distance of the sample segment. Additionally, the difference in temperature between the two configurations decreased due to the same reason when rising air speed.

3.3 Comparison Between Micro-Pin Fins Duct and Smooth Duct
Thermal performance for the in-line micro-pin fins duct and smooth duct are compared numerically in this section. The air enterd with a temperature 38°C in both different ducts. In the conditions of 2 m/s airflow and 250 Watt heat flux, the air exit at a temperature 65.46°C in the micro-pin fins duct while exit at 50.26°C in the smooth duct. Relative to the results obtained, the thermal performance was enhanced by 30.24% with 902 micro-pin fins duct compared with the smooth duct. Fig. 6 illustrated the temperature variation along the two ducts.
Figure 4: Comparison between the results of different numbers of in-line micro-pin fins

4. Temperature Contours
The temperature contours are plotted in two-dimensional form to show the temperature distribution along the micro-pin fins duct. In the tests, the air reaches the duct with almost 45°C and the temperature of the solid buildings is calculated before. Fig. 7 displays the temperature contours of the distinct micro-pin fins numbers 902, 1312, and 1640 for 2 m/s and 250 Watt. In addition, Fig 8 shows the temperature contours of staggered and in-line systems in the same conditions in the case of 1 m/s and 250 Watt.

5. Results of Analysis for Flow Field
Fig. 9(a) illustrates the distribution of the two-dimensional airflow of the in-line micro-pin fins in the situation of 2 m/s. The velocity vector in the x-direction (u) and z-direction (w) are shown in the figure. In the center, the velocity is greater than the other points close the walls. There are circulations near the micro-pin fins locations with a straight flow between the micro-pin fins. As for staggered arrangement, air obstruction is greater than in-line structure, which causes more fluid field circulation. Fig. 9(b) Displays the airflow allocation along the
Figure 5: Comparison between the results of staggered and in-line configurations

duct.

6. Conclusions
The main conclusions of the present job can be described in the following notes:

(i) The thermal performance enhanced with increasing of micro-pin fins number from 902 to 1312 by 7.9 - 9.3% and from 902 to 1640 by 10.2 - 11.1%.

(ii) In the staggered configuration, the heat transfer performance is slightly higher than the in-line configuration by nearly 2.6 - 4.2% with the same number of micro-pin fins.

(iii) In addition, the micro-pin fins duct increased the thermal transfer performance by 30.24% compared to the smooth duct.
Figure 6: Comparison between the temperature variation inside the micro-pin fins duct and the smooth duct

(a) 1640 MPFs Watt

(b) 1312 MPFs Watt

(c) 902 MPFs Watt

Figure 7: Temperature distribution of different micro-pin fins numbers at 2 m/s and 250 Watt
Figure 8: Temperature distribution of different configurations at 1 m/s and 250 Watt

Figure 9: Flow field vectors of (u) and (w) for different configurations
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