

Performance analysis of an array of square micro-fins

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Abstract: Array of micro pin fin heat sinks shows higher thermal efficiency in high heat flux and critical devices such as aerospace, microelectronic etc. In this work, an array of micro square pin fins has been designed and for this copper is selected as workpiece material. Moreover, a numerical investigation of thermal performances and mechanical behavior of the designed fins has been carried out. The effects of Reynolds numbers on velocity profile and heat transfer performance have also been studied. Additionally, the equivalent (von-Mises) stress along with the structural and total deformations have been observed at the desired pressure applied on the surface of the pin fins. Numerical simulations with similar parametric conditions have also been conducted on a plate having the same dimensions without fins, and a comparison has been made with a plate having micro-fins. It has been observed that the array of square micro pin fins gives a better thermal performance than that of without fins.

Keywords: Heat transfer; Micro fin array; Reynolds numbers.

1. Introduction

With the rapid growth of technology in the recent few decades, it becomes utmost necessary to study the various heat transfer augmentation techniques in detail to increase the efficiency and life of numerous industrial components such as energy storage systems, heat exchangers, solar collectors, refrigeration air-conditioning systems as well as of many electronic components [1-2]. Heat transfer enhancement can be achieved by attaching thin strips of metal to the base surface which are called fins. Currently, heat transfer through fins by natural convection is getting paramount importance due to the elimination of unnecessary external power supply. Additionally, they are also maintenance free that makes them suitable for long lifetime devices. Fins increase the surface area for convection, thereby decreasing the thermal resistance at the solid/fluid interface, results in enhanced heat transfer. The convective heat transfer can be increased by increasing the heat transfer coefficient and/or surface area. As the size of the fin decreases, the number of the fin increases along with the surface area of the fins which may lead to the higher heat transfer rate.

Many experimental studies have been conducted to observe the thermal performances for different fin shapes. Micro pin fins have higher heat dissipation per unit mass than the micro plate fins due to higher surface area and improved heat transfer coefficient [3]. Pin fin optimizes the use of material with an increase in surface area. In experimental studies on different types of micro-fin geometry and shapes, it has been observed that the square micro fin has the best boiling heat transfer coefficient [4]. Reynolds number (Re) plays a significant role in fluid flow devices. At higher Re, boundary layer separation starts and the vortex forming inside the fluid domain become intense [5]. Diamond shaped micro pin fin has a higher pressure drop penalty than the other types of fin geometry. Fin position and



arrangement also influence cooling conditions. The pin fin porosity, location, and angle directly affect the cooling capacity and thermal performance of the micro-fins [6]. By the experimental and numerical studies of micro-fin, it has been observed that the effective thermal resistance is not sensitive to porosity at lower fin density [7]. But nonuniform micro pin fin arrays have the merit to enhance the heat transfer performance [8]. Optimum fin geometry can be designed using different optimization techniques. For finding the best fin geometry, numerical and experimental analysis have been done [9]. With the increase in effective fin heights for micro fins, it is necessary to increase the Re to get higher thermal performance [10]. For the calculation of thermal stress on different fin geometry, many numerical and experimental experiments have been conducted. High stress is generated in the joint of the inner fin and outer tube of the fin tube heat exchanger [11]. By using an Electro Discharge Machining (EDM) process, the micro fin had been fabricated [12], where, after three sequential discharge the deflection in the fin was observed.

From the literature, it has been found that square micro-pin-fin at optimal fin dimensions can give better cooling regarding thermal performance. Very few research articles have been found in thermal analysis for laminar flow in higher Re regime. The objective of this work is to carry out thermal analysis of an array of square micro pin fins at higher Re and to compare it with a plate without any fins. Moreover, the further objective is to carry out the structural analysis of these micro pin fins.

2. Methodology

For analyzing the mechanical and thermal performances of the micro fin array, a design has been generated using CAD software (CATIA-V5-R16). Figure 1: (a) shows the generated three-dimensional design through which the mechanical behavior of the fins has been studied. The dimension of the plate is 5mm×5mm×0.4mm and the total number of micro-fins are 144(12×12). The dimensions of these fins are considered as 150μm × 150μm × 200μm. For computational requirement and simplicity, a small representative portion (Figure 1(b)) of the similar fin arrangements has been considered for thermal analysis; but in this case, the thickness of the plate is 50μm.

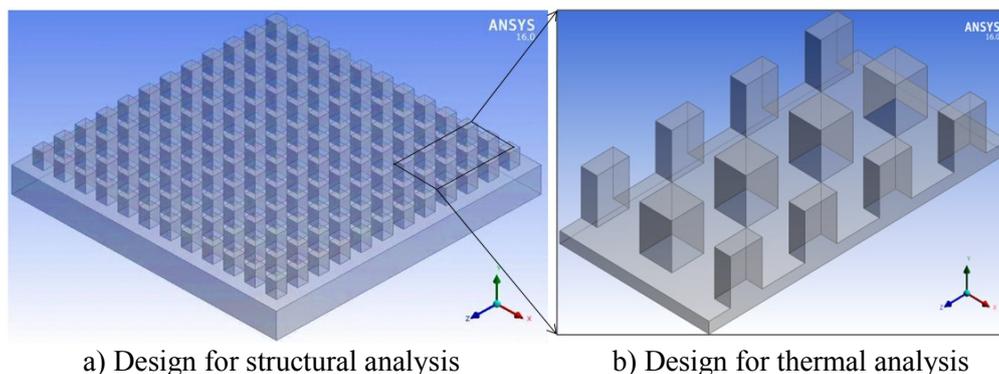


Figure 1: Typical 3-D drawing of micro-fin array generated using Catia-V5-R16

2.1 Assumptions

For this analysis, some general assumptions are made, as follows

- Flow assumed to be steady and laminar.
- No thermo-physical properties vary with temperature
- Fluid is considered as incompressible, Newtonian and viscous.

In the analysis, the heat flux has been set to 10kW/m² and the velocity at the inlet is calculated from different R, using Eq. (1).

$$Re = \rho v d / \mu \quad (1)$$

2.2 Governing equations

$$2.2.1 \text{ Continuity equation: } \nabla \cdot (\rho \vec{U}) = 0 \quad (2)$$

$$2.2.2 \text{ Momentum equation: } (\vec{U} \cdot \nabla) \rho \vec{U} = -\nabla p + \mu \nabla^2 \vec{U} \quad (3)$$

$$2.2.3 \text{ Energy equation: } (\vec{U} \cdot \nabla T) = \frac{k \nabla^2 T}{\rho C_p} \quad (4)$$

where, ρ : density of the fluid (kg/m^3); v : velocity at inlet (m/s); d : hydraulic diameter; p : pressure (Pa); μ : dynamic viscosity of fluid (N.s/m^2); T : absolute temperature (K); k : thermal conductivity of the fluid (W/m.K) and C_p : specific heat capacity at constant pressure (J/kg.K).

2.3 Boundary conditions

At inlet: Inlet velocity is taken at $\text{Re} = 100, 200, 300, 500, 750$ and 1000 .

At outlet: Pressure outlet gauge pressure is zero.

At the wall: Stationary wall and no slip condition at the wall.

Temperature field is obtained by combining governing equations (Eq. (2-4)) with boundary conditions. These are analyzed by dimensionless parameters such as the local Nusselt number and average Nusselt number. A parameter named as performance factor (PF) is calculated using the Nusselt number and pressure drop as shown in Eq. 5.

$$PF = \left(\frac{Nu}{Nu_o} \right) / \left(\frac{\Delta p}{\Delta p_o} \right)^{1/3} \quad (5)$$

Where,

Nu : Nusselt number of designed model; Nu_o : Nusselt number of standard model;

Δp : Pressure drop of designed model; Δp_o : pressure drop of the standard model.

An article [6] is chosen as a reference and the model of that article is considered as a standard model. The governing equations (Eq. (2-4)) are solved by the finite volume method where the second order upwind scheme is used to discretize both the advection and diffusion terms; after that, the results are generated.

2.4 Grid independency test

For finding the numerical stability of the present result, 3,46,618 to 19,10,559 number of elements have been taken for the analysis. For each case, average Nusselt number and corresponding pressure drop have been calculated. As shown in Figure 2: (a) the result becomes steady after 9,07,418 elements. Therefore, 14,24,601 elements have been taken for all the simulations in the current study. A plane named reference plane has been generated at 0.09 mm above from the base of the fins to observe the velocity profiles and temperature contours.

3. Result and discussion

3.1 Thermal performance analysis:

The velocity magnitude and the temperature profile have been observed on the reference plane. Figure 2: (b) depicts the dimensionless pressure difference at different Re . Dimensionless pressure drop decreases with increase in Re , but absolute pressure difference increases due to dense vortex formation. In all the cases, the plate having square micro pin fins gives higher dimensionless pressure difference than the plate without any fin.

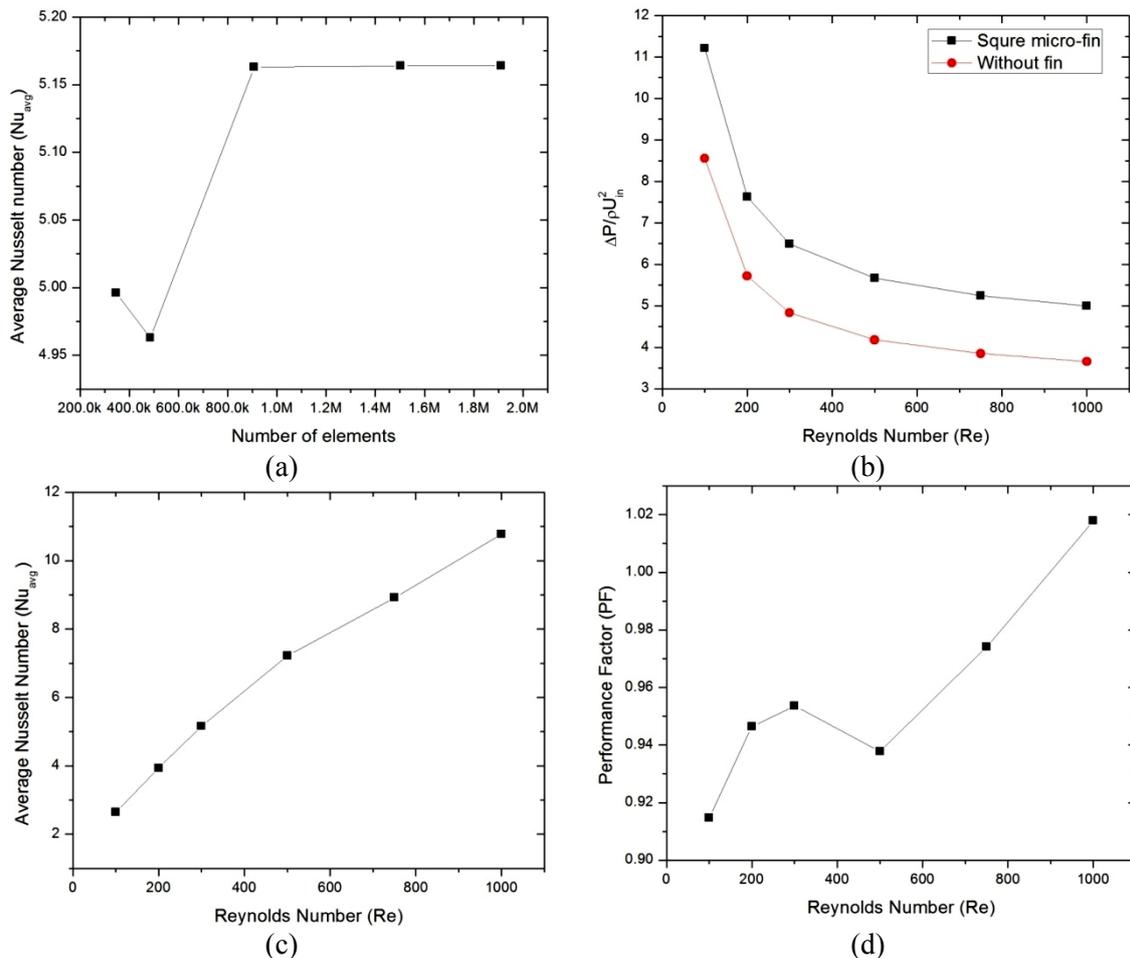


Figure 2: (a) Variation of average Nusselt number with a number of elements; (b) Variation of dimensionless pressure for different Re; (c) Variation of average Nusselt number for different Re; (d) Variation of performance factor for different Re.

Higher Re leads to the higher velocity of the fluid at the inlet that results in a higher Nusselt number. Figure 2: (c) shows the Nusselt number at different Re. The performance factor has been calculated using Eq. 5 through which different performance factor for different Re has been computed. Figure 2: (d) shows the variation of performance factors for different Re. The maximum performance factor around 1.02 has been observed at 1000 Re. At lower Re, compactness of the fin is the primary factor for the increase in average Nusselt number, but at high Re, vortex formation is the primary cause for the increase in average Nusselt number. At Re 500, the performance factor decreases as the increase in relative pressure drop are more prominent than that of the increase in relative average Nusselt number.

A variation of the velocity profiles for different Re has been shown in Figure 3. It shows that the vortex formation increases with the increase in Re. When Re equals to 100 and 200, vortex formation is negligible. A large number of vortex formations help the fluid to circulate everywhere efficiently to improve the convective heat transfer, that results in the enhancement of average Nusselt number. From Figure 3, it is clear that more vortex formation takes place at high Re. At Re 100 the velocity gradients are very less and with the increase in Re, more velocity gradients have been observed. Also at lower Re very few velocity gradients generated in between the fins. So at higher Re, more vortex and high-velocity fluids carry out the heat from the fin surface at higher rates.

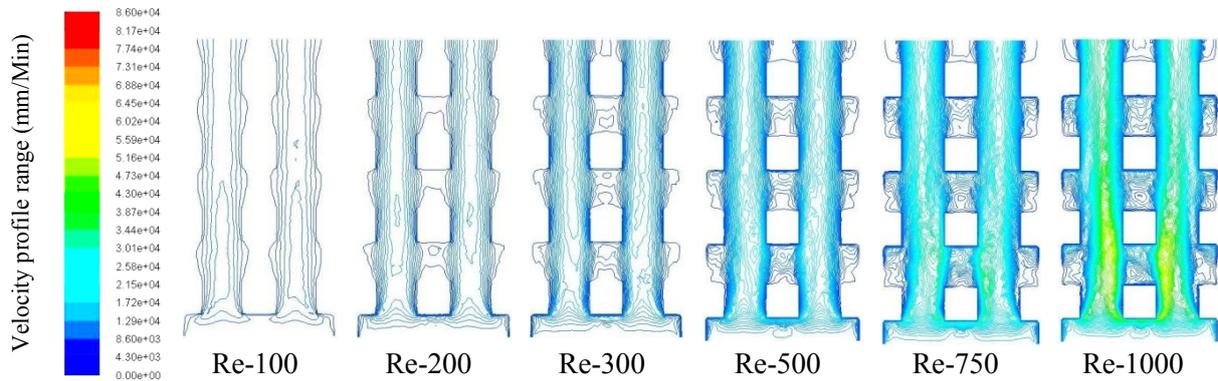


Figure 3: Velocity profiles for different Re at reference plane

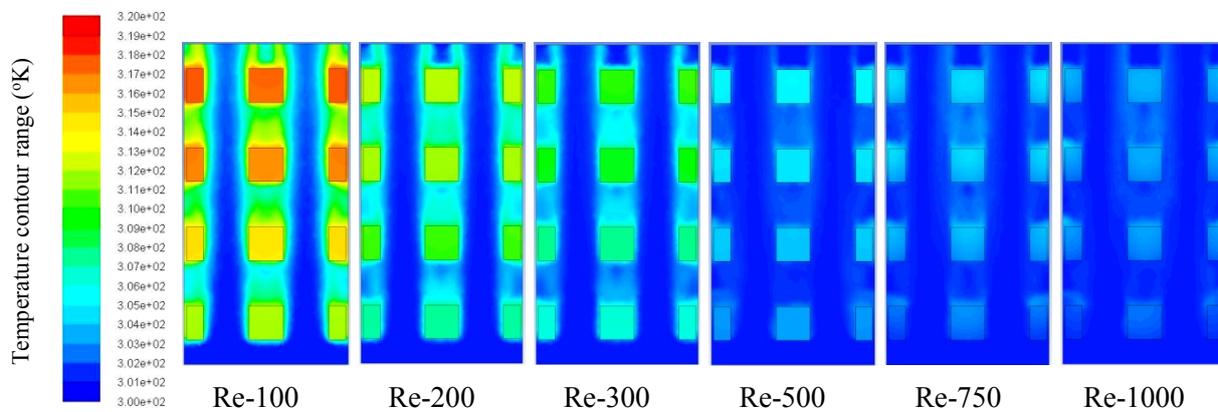


Figure 4: Temperature contour for different Re at reference plane

From the temperature contours, it has also been clear that at higher Re, efficient cooling takes place. Figure 4 shows the variation of the static temperature with different Re. The temperature distribution on the reference plane ranges from 299.8K to 319.9K. At lower Re, heat transfer from the fin is very less. At Re 100, 200 and 300 the maximum temperature on the fins are around 320K, 314K, and 310K respectively. So, the cooling of the fins at lower Re is not significant. But for higher Re very efficient cooling takes place as the temperature of the fins are very near to the fluid temperature. From Re 750 to 1000 there is no significant improvement on cooling.

3.2 Mechanical performance analysis:

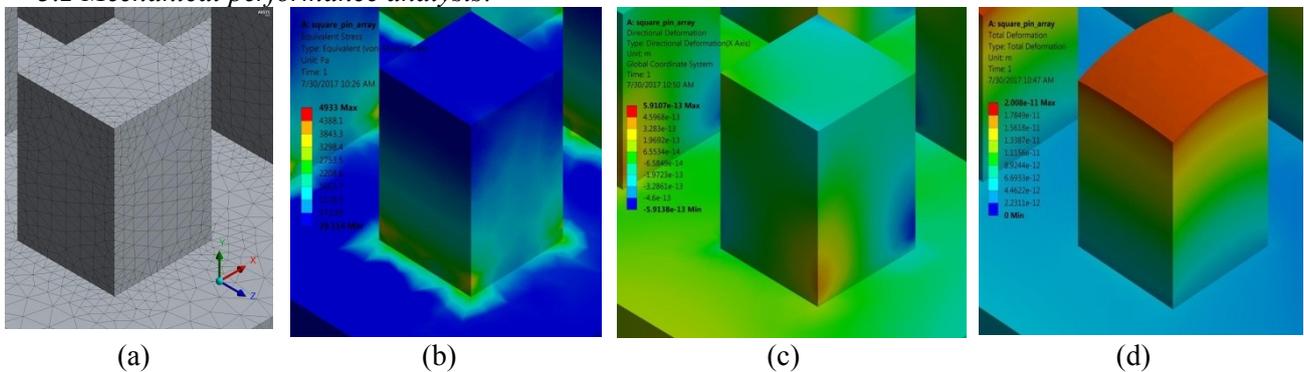


Figure 5: Mechanical behavior analysis: a) Generated mesh on the square micro-fins; b) Equivalent (von-Mises) Stress; c) Directional deformation; d) Total deformation

As these pin fins are in micro scale, these are very delicate to handle and it is necessary to observe the stress and the deformation due to the forced fluid striking on the fin surfaces. The dynamic striking

pressure is calculated as 4.078 MPa by considering the maximum Reynolds number (Re) at 1000. Figure 5 shows the generated mesh, equivalent (von-Mises) Stress, directional deformation and total deformation for an array of square micro pin fins. Keeping 15 μ m as elemental length, the mesh has been generated to get 11,85,366 elements (Figure 5: a). After that, the fluid pressure is applied in the direction of the fluid flow, i.e. the positive 'x' direction and it is observed that the maximum equivalent stress (von-Mises) is induced at the bottom part of the pin fins (Figure 5: b). The stress induced due to fluid pressure is in the range of 29.114Pa to 4933 Pa; which is very less than the yield stress of copper (124-370MPa). Since the pressure is applied in the positive 'x' direction, the tensile stress and compressive stress are induced at the striking fin surface and its opposite surface respectively. The maximum directional deformation is observed to be 5.90 $\times 10^{-13}$ m at the striking fin surface (Figure 5: c). The fluid velocity increases with the increase in height from the base plate that results in maximum total deformation of 2.01 $\times 10^{-11}$ m at the top of the fin (Figure 5: d). As the induced stress and the deformations are very minimum, the fluid pressure acting on a fin surface is insignificant.

4. Conclusion: Assuming the fluid as steady, laminar, and constant thermo-physical properties the performance analysis of micro-square pin fin have been done. From the results, it can be concluded that, with the increase in Re, the average Nusselt number increases, but pressure difference also increases likewise which is a shortcoming. The dimensionless pressure difference decreases with Re and is always higher for a plate with micro-fins than a plate without any fins. Therefore, the square micro pin fins are giving better thermal performance than a plate without any fins. The maximum overall performance factor observed when Re=1000. It is clear that higher Re is more suitable in both of the velocity profiles and the temperature contour scenarios. As far as stress-strain is concern, the induced stress and the deformation are very less which is not sufficient for permanent deformation of the fins. So the array of square pin fins is economical in this Re regime.

5. References:

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