KT Journal of Mechanical Engineering

ISSN: 3048-9563 (Online)

Key Tech Services Publication, India

http://www.ktjme.com
https://doi.org/10.64188/3048956325009
Vol.: 2, No. 1, 2025, Pages: 30-41

HEAT TRANSFER ASSESSMENT USING TAPERED PERFORATED PIN FINS

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Abstract

The experimental study of forced convective heat transfer on staggered pin fin arrays in a rectangular channel to quantify their heat transfer characteristics. System performances are compared between solid and various perforated pin fin heat sinks. It was found that the pin fin array performed better than solid fins and Nusselt number increased with the perforated fins. It was found that the maximum system performance is obtained with perforated pin fins when the number of perforations and perforation diameter are optimized. With the present configuration of pin fins, N = 5 and Dp = 3 mm, pin fins offer the best performance.

1. Introduction

The applications of heat transfer enhancement techniques can significantly increase the performance of heat exchangers, leading to the reduction of heat exchanger size as well as operating cost. Heat transfer enhancement has significant meaning for energy conservation and environmental problems. The induced swirl flow, potentially promoting fluid mixing, causes a thinner boundary layer and consequently results in a higher convective heat transfer rate. Swirl flow generators with different geometrical configurations are widely used to enhance the heat transfer rate in many engineering applications, for example, heat recovery processes, air conditioning, refrigeration systems, internal cooling of gas turbine blades, chemical reactors, thermal regenerators, gas-cooled reactors, and food and dairy processes. Generally, heat transfer augmentation methods are classified into three broad categories:

1.1 Active Methods: This method involves some external power input for the enhancement of heat transfer; some examples of active methods include induced pulsation by cams and

Received:15.12.2024 Revised: 25.02.2025 Accepted: 01.04.2025 Published Online: 18.04.2025

Keywords: Alternator, Finite Element Analysis, Electromagnetic, critical loads

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How to cite this article: Visvas S. Choure, Heat transfer assessment using taper perforated pin fins, KT Journal of Mechanical Engineering, 1, 2025, 30-41. https://doi.org/10.64188/3048956325009

reciprocating plungers, the use of a magnetic field to disturb the seeded light particles in a flowing stream. These techniques are more complex from the use and design point of view, as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need for external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential, as it is difficult to provide external power input in many cases. In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer. Augmentation of heat transfer by this method can be achieved by:

1.1.1. Mechanical Aids:

Such instruments stir the fluid by mechanical means or by rotating the surface. These include rotating tube heat exchangers and scraped surface heat and mass exchangers.

1.1.2. Surface vibration:

They have been applied in single-phase flows to obtain higher heat transfer coefficients.

1.1.3. Fluid vibration:

These are primarily used in single-phase flows and are considered to be perhaps the most practical type of vibration enhancement technique.

1.1.4. Electrostatic fields:

It can be in the form of electric or magnetic fields or a combination of the two from DC or AC sources, which can be applied in heat exchange systems involving dielectric fluids. Depending on the application, it can also produce greater bulk mixing and induce forced convection or electromagnetic pumping to enhance heat transfer

1.1.5. Injection:

Such a technique is used in single-phase flow and pertains to the method of injecting the same or a different fluid into the main bulk fluid either through a porous heat transfer interface or upstream of the heat transfer section.

1.1.6. Suction:

It involves either vapor removal through a porous heated surface in nucleate or film boiling or fluid withdrawal through a porous heated surface in single-phase flow.

1.1.7. Jet impingement:

It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.

1.2 Passive Methods: These methods generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. For example, the use of inserts, the use of rough surfaces, etc. Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques are practical. Application for internal cooling of turbine airfoils, combustion chamber liners and electronics cooling devices, biomedical devices, and heat exchangers. The heat transfer can be increased by the following different Augmentation Techniques. They are broadly classified into three different categories: (i) Passive Techniques (ii) Active Techniques (iii) Compound Techniques. These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior (except for extended surfaces), which also leads to an increase in the pressure drop. In the case of extended surfaces, the effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over active techniques as they do not require any direct input of external power. These techniques do not require any direct input of external power; rather, they use it from the system itself, which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior, except for extended surfaces. Heat transfer augmentation by these techniques can be achieved by using;

1.2.1 Treated Surfaces:

Such surfaces have a fine-scale alteration to their finish or coating, which may be continuous or discontinuous. They are primarily used for Boiling and condensing duties.

1.2.2 Rough surfaces:

These are the surface modifications that promote turbulence in the flow field in the wall region, primarily in single-phase flows, without an increase in heat transfer surface area.

1.2.3. Extended surfaces:

They provide effective heat transfer enlargement. The newer developments have led to modified finned surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.

1.2.4. Displaced enhancement devices:

These are the inserts that are used primarily in confined forced convection, and they improve energy transport indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.

1.2.5. Swirl flow devices:

They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These include helical strip or cored screw-type tube inserts and twisted tapes. They can be used for single-phase and two-phase flows.

1.2.6. Coiled tubes:

These lead to relatively more compact heat exchangers. It produces secondary flows and vortices which promote higher heat transfer coefficients in single-phase flows as well as in most regions of boiling.

1.2.7. Surface tension devices:

These consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.

1.2.8. Additives for liquids:

These include the addition of solid particles, soluble trace additives, and gas bubbles in single-phase flows, and trace additives which usually depress the surface tension of the liquid for boiling systems.

1.2.9. Additives for gases:

These include liquid droplets or solid particles, which are introduced in single-phase gas flows either as dilute phase (gas-solid suspensions) or as dense phase (fluidized beds).

1.3 Mechanisms of Augmentation of Heat Transfer

The mechanisms of heat transfer enhancement can be at least one of the following as use of a secondary heat transfer surface, the disruption of the unenhanced fluid velocity, the disruption of the laminar sublayer in the turbulent boundary layer, introducing secondary flows, promoting boundary-layer separation, promoting flow attachment/reattachment, and enhancing effective thermal conductivity of the fluid under static condition.

1.3.1 Heat Transfer Enhancement using Perforated Pin Fins:

Extended Surface (Fins) is used in many applications to increase the heat transfer from surfaces. Typically, the fin material has a high thermal conductivity. The fin is exposed to a flowing fluid, which cools or heats it, with the high thermal conductivity allowing increased heat to be conducted from the wall through the fin. Fins are used to enhance convective heat transfer in a wide range of engineering applications and offer practical means for achieving a large total heat transfer surface area without the use of an excessive amount of primary surface area. Fins are commonly applied for heat management in electrical appliances such as computer power supplies or substation transformers. Other applications include IC engine cooling, such as Fins in a car radiator. Fins are widely used in the trailing edges of gas turbine blades, in electronic cooling, and in the aerospace industry. The relative fin height (H/d) affects the heat transfer of pin-fins, and other affecting factors include the velocity of fluid flow, the thermal properties of the fluid, the cross-sectional shape of the pin-fins, like perforation, the relative inter-fin pitch, the arrangement of the pin-fins like in-line, staggered arrangement and others. In existing studies, the parameters affecting the heat transfer, like relative fin height (C/H), the velocity of fluid flow, the cross-sectional shape of the pin-fins, like perforation, the relative inter-fin spacing, and the arrangement of the pin-fins, like staggered arrangement, have been investigated.

2. Literature Review

Rapid development in manufacturing technology and consumer demands have driven electronic technology toward increasing the functionality and compactness of the components. The miniaturization of electronics results in a higher rate of power dissipation per unit volume. Thus, effective thermal management becomes ever more critical to maintain the operating temperature, which would ensure the efficiency and reliability of the electronic components. The design of the heat sink device is predicated upon optimizing the opposing demands of maximizing the thermal dissipation rate and minimizing the pressure drop across the system. One of the most common solutions is to apply a pin fin array onto a heat sink design. Sparrow et al. measured the effects of in-line and staggered pin fin arrays on thermal dissipation and pressure drop. Heat transfer and pressure coefficients for staggered arrays are shown to be higher than those for the in-line arrangement. Similar results were found by Khan's analytical investigation [2]. Tahat et al. [3] optimized the lengthwise and spanwise arrangements of staggered and inline pins. Bilen et al. [4] obtained experimentally the heat transfer and friction correlation of a cylindrical finned surface. They reported a substantial thermal enhancement for in-line and staggered pin fin arrays compared with the heatsink without pins. More importantly, the thermal dissipation of staggered pins is 33 % higher than that in the in-line arrangement at the same Reynolds number, but at a cost of higher pressure drop. The numerical simulations of Soodphakdeeet al. [5] on the effects of the geometry showed that for fins in in-line and staggered arrangements, circular pinout performed square fins, and elliptical fins is better than plate fins. At a low Reynolds number (Reh\30) higher heat transfer coefficient is obtained with an elliptical pin fin, but a higher Reh circular pin fin is more effective. Although the pin fin array provides a higher heat transfer coefficient than the plate, it also suffers from a larger pressure drop [6]. Circular pins with a height-to-diameter ratio of 0.5–4 are categorized as short pins, whereas pins with a height-to-diameter ratio exceeding 4 are known as long pins [7]. Short pins are normally used to cool gas turbine blades, and long pins are commonly found in heat exchangers, where a high heat transfer coefficient is the major concern. Vanfossen [8] verified experimentally that in staggered pin fin arrays, long pins produced a higher heat transfer rate than that of short pins, but at a cost of larger pressure losses and hence higher pumping power requirement. Most studies that have demonstrated the beneficial effects of perforations on the rate of heat transfer in pin fin arrays have used only a single perforation per pin. The logical extremes of this parameter is the metallic foam-like porous pins. Indeed, numerical simulations, e.g. by Yanget al. [9] or Seyf and Layeghi [10] have suggested that orders of magnitude improvement in the rate of heat transfer may be obtainable from foam-like porous pin arrays, mostly due to the large surface area to volume ratio. However, foam-like porous pins may be impractical as they are easily contaminated, resulting in blocked pores and reduced efficiency. Nonetheless, foam-like pin studies suggest that the number of perforations per pin may be an important factor in determining heat transfer. Several investigators have reported on the effects of the pin geometry and configurations. Meinders et al. [11, 12] found that the size of the recirculating flow behind the pins significantly affects the local rate of heat transfer. Sara et al. [13] reported that for a given porosity, heat transfer performance may be enhanced by increasing the size of perforation instead of the number of perforations. Numerical simulations carried out by Shaeri and Yaghoubi [14] showed that the rate of heat transfer in perforated fins is much higher than that in solid fins due to their larger surface area-to-volume ratio. Moreover, the temperature difference between the top and bottom surfaces of perforated pins is lower than that in solid pins, and it decreases with increasing number of perforations per pin. They also found that the total drag coefficient of the solid fins was much higher than that of the perforated fins, as solid fins produced much larger wakes than the perforated fins. There appears to be a lack of experimental and numerical work on staggered, circular, long pin fin arrays and the influence of the number of perforations and perforation diameter on heat transfer and flow characteristics. To address these issues, the present paper reports on an experimental and numerical study of steady-state, incompressible forced convective heat transfer in solid and staggered perforated pin fin arrays. Three-dimensional (3D) Computational Fluid Dynamics (CFD) simulations were performed to study the effects of the number of perforations, N, and perforation diameter, DP, to optimize the rate of convective heat transfer against pressure losses

3. Materials and Methods

3.1. Description of setup:

The present study investigates the steady-state forced convective heat transfer in perforated aluminum alloy (A5083P, thermal conductivity 167 Wm-1 K-1) pin fins in fully developed flow. Figure 1 shows the schematic of the experimental setup. The entrance length is 745 mm, the test Section is 110 mm, and the downstream to the exit is 195 mm. The internal height and width of the channel are 50 and 100 mm, respectively. The channel is constructed with a transparent 5 mm acrylic sheet with a thermal conductivity of 0.17 Wm-1 K-1 to minimize heat loss. Air is supplied by a blower with controllable flow velocity. The staggered pin fin arrays with and without perforations are installed in the test section. All pin fins are firmly embedded 3 mm into the base plate, and the gaps are sealed with epoxy. Figure 1 shows the midplane locations of the three K-type thermocouples on the thermofoil-heated surface to measure the wall temperature. A 60 W AC thermos foil heater with stabilized heat flux is attached directly beneath an aluminum plate to mimic the electronic component thermal heating. The inlet and outlet flow velocities are recorded with a commercial hot wire anemometer. The pressure drop across the test section is measured using an inclined well-type manometer. The calculated pressure drops are compared with the measured DP at the same location. The inlet air temperature is measured with the hot wire anemometer, and a K-type thermocouple records the outlet temperature. Figures 2 and 3 show the**.** Different pin fin arrays with different perforation numbers (N) and diameters (Dp). The components are the channel entrance, test section, channel exit, blower, AC power supply, hot wire anemometer, K-type thermocouples, heater, inclined pressure manometer, speed regulator, and pin fin array.

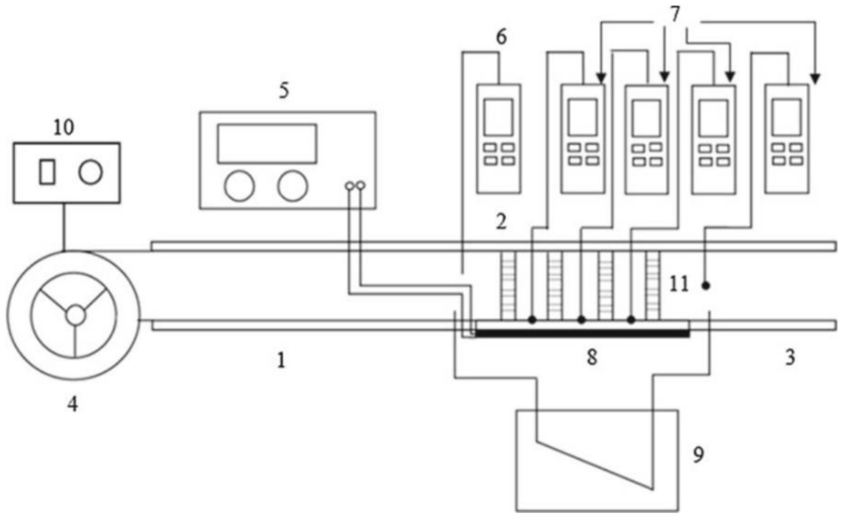


Figure 1. Schematic of the experimental setup

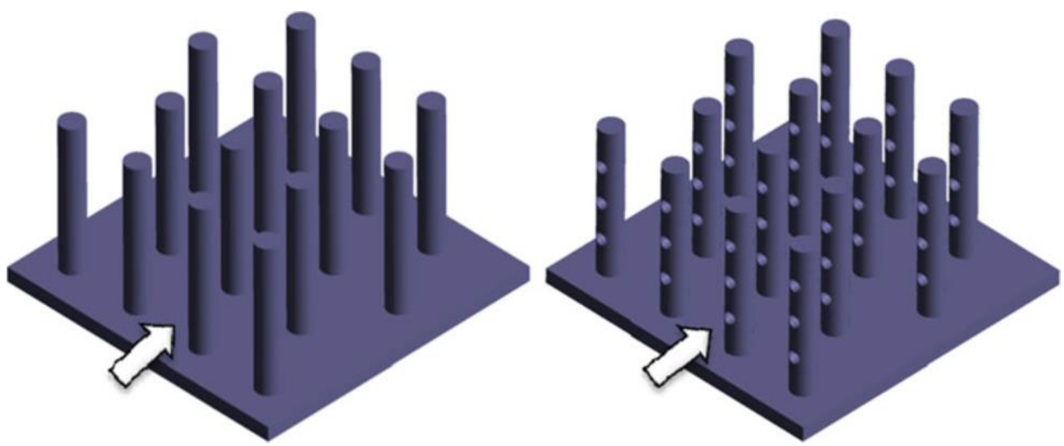
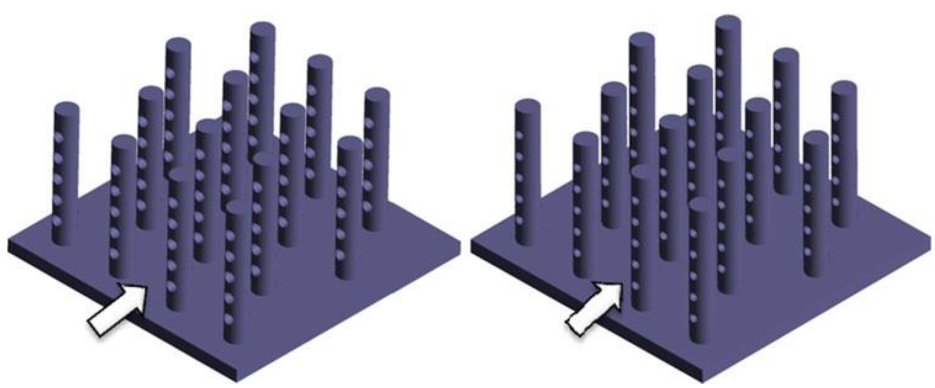


Figure 2. (a) Solid Pins (b) N=3, Dp =4mm



(d) N=5, Dp =4mm (e) N=5, Dp =3mm

Figure 3. Pin fin arrays with different perforation numbers (N) and diameters (Dp)

5. Results and Discussion

5.1 Effects of perforations on pressure drop ΔP:

As per the literature review, Figure 4 shows the effects of the number of perforations and perforation diameter on the measured pressure drop ΔP as a function of Reynolds number (Re). The calculated ΔP for the empty channels is also given for comparison. Whereas ΔP for an empty channel is almost independent of Re, that for pin fins is higher than that in the empty channel, and it also increases significantly with increasing Re. The pressure drop for the solid pins is the highest. However, at a given Reynolds number ΔP decreases with an increasing number of perforations. Since the number of perforations is restricted by the fixed length of the pin, an alternative to reducing ΔP is to increase the perforation diameter. Solid pins present larger blockage than perforated pins; the resulting wake leeward of each pin causes higher pressure losses. As a result, smaller ΔP can be effectively achieved using perforated pin fins. With taper perforations, we can predict lower ΔP than the solid and cylindrical fins

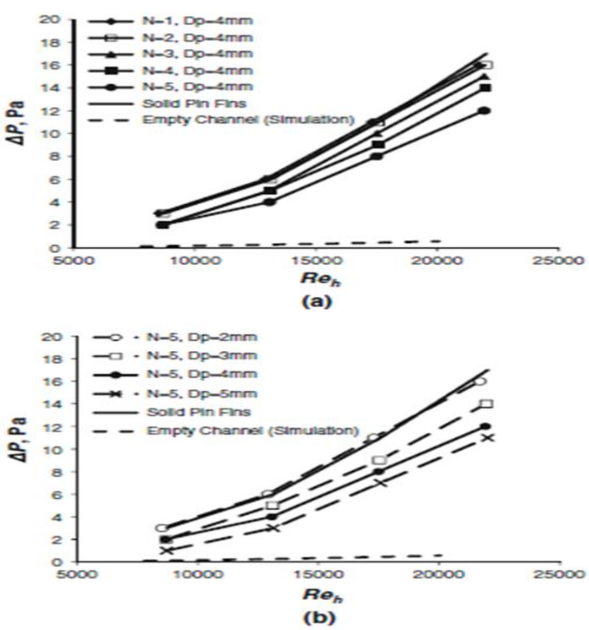


Figure 4. The effects of the number of perforations (N) and b perforations

5.2 Effects of perforations on heat transfer:

As per the literature review, the effects of the number of perforations and perforation diameter on the measured Nusselt number, Nu, are shown in Fig. 5 (a) and (b) respectively. Nusselt number increases with increasing Reynolds number for the empty channel, solid pins, and perforated pins.

The Nusselt numbers for the solid and perforated pin fins are higher than those for the empty channel, and hence provide higher thermal dissipation. More significantly, thermal dissipation is higher with perforated pin fins than with solid pins. It is found that the larger the number of perforations on each pin fin, the higher the Nu. Such an effect is due to the increase in porosity and the number of perforations, hence increasing the heat transfer surface area. Figure 5 (b) shows the effect of perforation diameter on pin fins with five perforations. The experiments show that increasing the perforation diameter from 2 to 3 mm increases Nu. However, further increasing the perforation diameter from 4 to 5 mm reduces the Nu. This is due to the decrease in the cross-sectional area of the pin for heat conduction along the pins with increasing perforation diameter. Thus, maximum heat transfer is obtained in the present study from pin fins with 5 perforations each of 3 mm diameter, when Dp/D = 0.375 and NDp/H = 0.3. It is found that at Re of approximately 22 x 10^3, Nu is 45 % higher than that with solid pins.

5.3 System performance efficiency:

The system performance of a pin fin heat sink is defined as the ratio of Nu over the pressure coefficient. It describes the relative cost (pressure drop, hence pumping power) to achieve a certain rate of heat transfer. Figure 5 (b) clearly shows that system performance increases with increasing Reynolds number. This is due to the increasing thermal dissipation with increasing Reynolds number. It shows that the system performance for Dp = 4 mm perforated pins increases monotonically with increasing number of perforations at a given Reynolds number. In all cases, the system performances are higher than those for solid pins. The effects of perforation diameter on system performance with the maximum number of perforations, N = 5, used in the experiments, is now examined. The perforated pins are always more efficient than solid pins; there is no further significant gain in system performance when the perforation diameter exceeds 4 mm. It suggests that maximum system performance may be obtained with perforated pin fins when the number of perforations and perforation diameter are optimized.

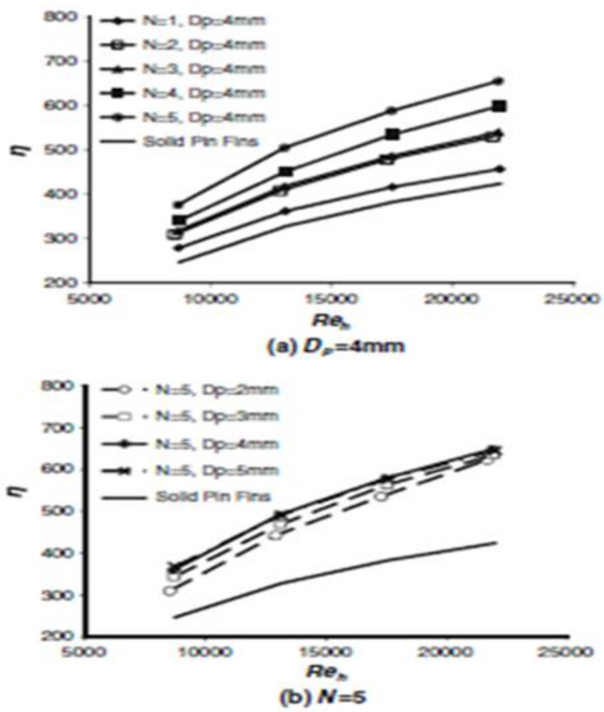


Figure 5. Dependence of heat sink system performance on several perforation diameters

Conclusions

With the experimental study of forced convective heat transfer on staggered pin fin arrays in a rectangular channel, their heat transfer characteristics following predictions can be predicted. System performances are compared between solid and various perforated pin fin heat sinks.:

1. ΔP across the heat sink is smaller with an increasing number of perforations and perforation diameter. In all cases, the perforated pin fin array performs better than the solid pins. Hence, perforated pin fins require less pumping power than solid pins for the same thermal performance.

2. Nu increases with an increasing number of perforations and perforation diameter. Further increasing the perforation diameter will lead to a reduction in thermal dissipation. This is due to the decrease in vertical heat conduction along the perforated pin fins and the perforations inducing re-shaping of wakes behind the pins. Thus, while designing a perforated pin fin array, the balance between the perforation number and diameter should be carefully considered.

3. Maximum system performance may be obtained with perforated pin fins when the number of perforations and perforation diameter are optimized. With the present configuration of pin fins, N = 5 and Dp = 3 mm, pin fins offer the best performance

References

1. Sparrow EM, Ramsey JW, Altermani CAC, Experiments on in-line pin-fins arrays and performance comparison with staggered arrays. ASME J Heat Transfer 102 (1980) 44–50.
2. Khan WA, Culham JR, Yovanovich M. M, Modelling of cylindrical pin-fin heat sinks for electronics packaging. IEEE Trans Compon Packag Technol 31 (2008) 536–545.
3. Tahat M, Kodah ZH, Jarrah BA, Probert SD Heat transfer from pin fin arrays experiencing forced convection. Appl Energy 67 (2000) 419–442
4. Bilen K, Akyol U, Yapici S., Heat transfer and friction correlations and thermal performance analysis for a finned surface, Energy Conversion Management, 42 (2001) 1071–1083
5. Soodphakdee D, Behnia M, Copeland DW, A comparison of fin geometries for heat sinks in laminar forced convection: part I, round, elliptical, and plate fins in staggered and in-line configurations, Int J Microcircuits Electron Package, 24 (2001) 68–76.
6. Kim DK, Kim SJ, Bae J.J., Comparison of thermal performances of plate-fin and pin-fin heat sinks subject to an impinging flow. Int J Heat Mass Transfer, 52 (2009) 3510–3517.
7. Tanda G, Heat transfer and pressure drop in a rectangular channel with diamond-shaped elements. Int J Heat Transfer Mass Transfer, 44 (2000) 3529–3541
8. Vanfossen GJ, Heat transfer coefficient for staggered arrays of short pin-fins. ASME J Eng Power 104 (1982) 268–274.
9. Yang J, Zeng M, Wang Q, Nakayama, A Forced convection heat transfer enhancement by porous pin fins in rectangular channels. ASME Jou. Heat Transfer, 132 (2010) 0517028
10. Seyf HR, Layeghi M, Numerical analysis of convective heat transfer from an elliptic pin fin heat sink with and without metal foam insert. ASME J Heat Transfer, 132 (2010) 0714011
11. Meinders ER, Van Der Meer TH, HanJalic´ K, Local convection heat transfer from an array of wall-mounted cubes. Int Jou of Heat Mass Transfer, 41 (1998) 335–346.
12. Meinders ER, HanJalic´ K, Vortex structure and heat transfer in turbulent flow over a wall-mounted matrix of cubes. Int Jou of Heat Fluid Flow, 20 (1999) 255–267.
13. Shaeri MR, Yaghoubi M., Numerical analysis of turbulent convection heat transfer from an array of perforated fins. Int Jou of Heat Fluid Flow, 30 (2009) 218–228.
14. ANSYS FLUENT 12.0/12.1 Documentation, ANSYS. Inc. (2009).