Experimental study of thermal performance of heat sinks using different pin fin arrays

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ABSTRACT

This study investigated the friction and heat transfer characteristics of hexagonal and grooved hexagonalpin-fin arrays kept inside a rectangle's rectangular wind tunnel. It was used for experimental studies of a horizontal base plate kept at 50 °C. Pin fins were arranged in an in-line configuration. It has been determined how far apart the fins are in both the span- and stream-wise directions. We maintained consistent values for relative stream-wise pitch (SY/d = 1.2, 2.4, 3.6) and relative span pitch (SX/d = 1.2). Experiments are performed for a number of air mass flow rates (Re = 2000–25,000) and clearance ratios (C/H = 0.0). The performances of pin fins were contrasted with one another. Higher heat transfer rates are produced bygroovedhexagonal fins compared tohexagonal fins. Pin fins' performances are compared with each other. The hexagonal fin's results compared to the grooved hexagonal fins produce a faster rate of heat transfer. Based on a high Nusselt number at the same pumping power, the current results indicate that SX/d = 1.2 and SY/d = 1.2 are the optimal inter fin pitches.

Key words: Pin-fin, grooved hexagonal, heat exchanger, wind tunnel,Nusselt number, Reynolds number

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Introduction

Pin-fins installed on a heating surface can enhance the device's dependability and longevity by increasing the dissipation surface area and causing turbulent flow mixing. Many different applications use pinfins due to their low cost and simple construction. The internal cooling of turbine blades, the cooling of electronic components, and the cooling of several other heat exchange devices are applications for pin-fins in the cooling arrangement of a channel with cross flow. These fin forms vary from a combination of numerous geometries to relatively simple shapes like pin fins, tapered, annular, square, rectangular, and cylindrical.

The test case for the perforated pin fin arrays was a solid pin fin array with the same fin dimension. With improved fin effectiveness, fin efficiency, and convective heat transfer coefficient, the hexagonally perforated pin fin arrays performed the best. Compared to circular perforation and solid fins, hexagonal perforation had a lower thermal resistance.

The examination of the literature revealed a range of adjustments and changes made to the fins by adding slits, holes, and struts, which improved heat transfer. Nevertheless, using ribs or struts is not strictly advised. This option adds to the current fin's mass or weight, and since less material is used, using a fin with grooves is not advised in certain situations. On the other hand, it is observed that, under the same temperature differential settings, the grooved type of fins exhibit more heat transfer effectiveness, which is essentially a driving potential. Literature also demonstrates that additionally, research indicates that very few tests, performed on the grooved kind of fin, have demonstrated the impact of geometrical modification in influencing the properties of the heat transfer.

Thus, the focus of this paper's experimental effort is on the grooved arrangement in the hexagonal type of pin fin. The purpose of the current experimental effort is to examine how heat transfer and fluid flow characteristics affect grooved hexagonal fins and to compare their behavior to that of hexagonal fins. The experiment results are confirmed to be the dimensionless geometrical functions of the fins.

Table 1 lists the pin fins' geometrical properties.

Parameters	Minimum value	Maximum value
Pinfin spacing (S_X)	12 mm	12 mm
Pinfin spacing (S_Y)	12 mm	30 mm
Fincount[N_{xy}]	22	220
Reynolds number[Re]	2000	250000
Ratio of clearance	0	0
Flow rate [m](kg/s)	0.069	0.143
Pin –fin height [H]	77.7 mm	
Pin-fin diameter [d]	6 mm	
Basis plate dimensions (W x L)	145 x250	
Temperature of the base plate [tb](^{o}C	50 ±0.25	

Apparatus used in experiments:

The rectangular duct with spinning blower is the primary component of the wind tunnel arrangement. The blower's primary purpose is to draw air into the testing facility from the surrounding atmosphere and release it back into it. The wind tunnel duct's dimensions were long, 150 mm wide, and 180 mm high, with a constant internal cross section made of plywood that was 19 mm thick. The test module, or pin-fin assembly, was installed at the bottom of the channel so that, when viewed in a stream wise orientation, the channel's midpoint and end walls are symmetric with respect to the center plane.

Experimental Setup and for Experiments

Duralumin plate pin-fin assembly with dimensions of 250 mm length, 145 mm width, and 25.4 mm thickness was employed. A 1500 W maximum power plate heater was used to heat the exchanger's base approximately evenly. To keep the heat exchanger base at a steady temperature, it was heated evenly. The heaters and base of the pin-fin assembly are safeguarded by an appropriately sized wooden box with an open top, providing adequate thermal insulation for the entire system. Altering the variable position and voltage allowed for control over the amount of electricity supplied to the main heater. Nine thermocouples were used to measure the test module's base plate temperature.

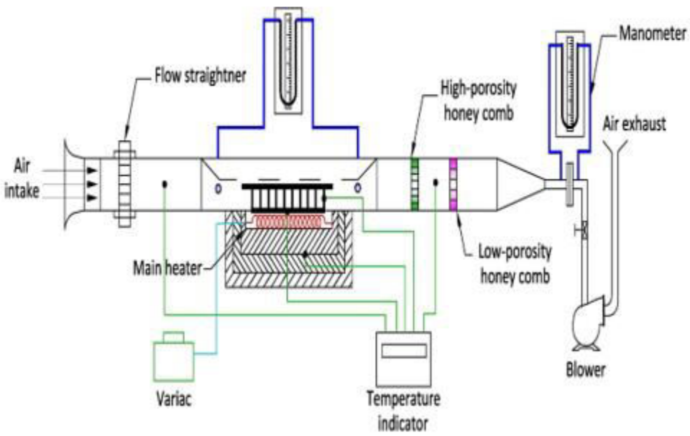


Fig.1 Experimental setup

The temperature of the air entering and leaving the system was measured using an additional eight thermocouples. Four were positioned downstream of the pin-fin array and four more facing the pin-fin assembly's entrance. Every thermocouple, together with those indicates the temperature of the surrounding air was linked to the temperature indicators. After around two and a half hours, the experiment reached a steady state, and the study was carried out again for an hour to verify the steady state value. An orifice meter calibrated against the hotwire anemometer was used to measure the air flow rate. A gate valve that is positioned in front of the blower's suction side can be opened to control the air flow rate. The blower that exits to the outside of the experiment draws air through the test portion. The blower that exits to the outside of the experiment draws air through the test portion.

Fin configuration

Every hexagonal fin had a diameter of 6 mm and a height of 77.7 mm. After that, the hexagonal fin surface was retained while being machined to create a 4 mm diameter by 4 mm height groove. Next, every fin is attached to a base-mounted, rectangular aluminum plate that measures 250 mm by 145 mm by 25.4 mm and may be heated using an electrical heater. A fin is positioned inside the test section so that the air passageways are over the fin surfaces during the test run. Figure 1 shows the pin fin assembly. Fig.2 illustrates in-line arrangement.

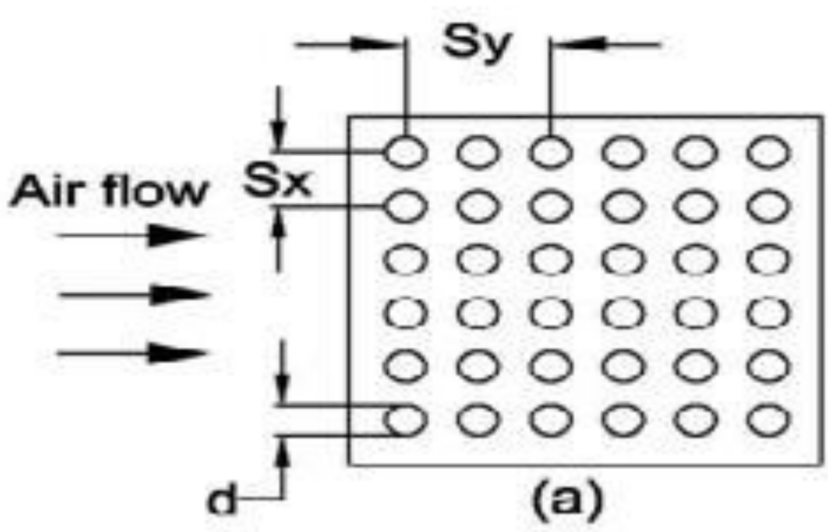
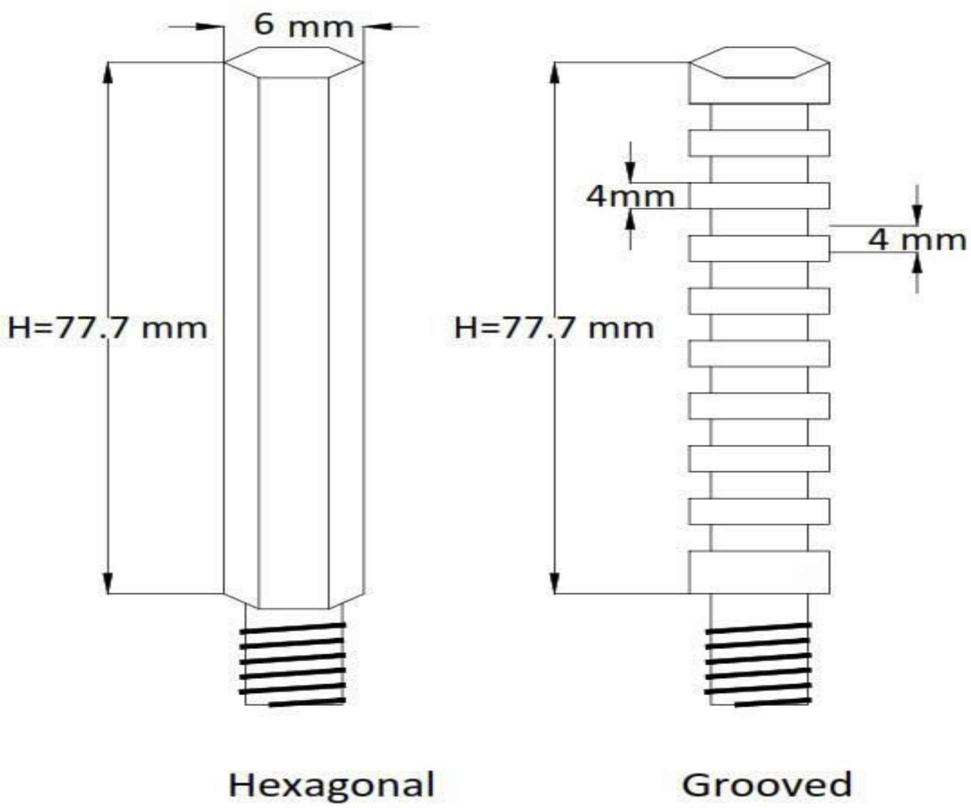


Fig.2 Pin fin assembly of inline



(a) and (b)

Fig.3 (a and b) displays the geometry of hexagonal and grooved hexagonal fins

Data reduction:

It uses the measured data to calculate the friction factor and heat transfers are calculated using the measured data. The steady state heat transfer for the finned surface is,

$$Q_{tot} = Q_{conv} + Q_{rad} + Q_{loss} \quad (1)$$

The system's overall steady-state energy equation is as follows: The data reduction used in this instance is comparable to that of Naik et al. (11), Jubran et al. (12) and Tahat et al. (2000). They carried out tests, found that fin arrays are comparable, and stated that the fin array has less than 5% heat loss capabilities. Equation (1) is provided even though, given the current operating conditions, the test section's insulation, and the assumption that the loss is relatively small,

$$Q_{conv} = mc_p (t_{out} - t_{in}) \quad (2)$$

The base plate and convective heat transfer fin surface are provided by

$$Q_{conv} = hA_s \left[t_b - \left(\frac{t_{in} + t_{out}}{2} \right) \right] \quad (3)$$

Where are the air flow temperatures, the average temperature at the base assembly's center, and the total test surface area, which can be written as,

$$A_s = W L + \pi d H N_{xy} - \frac{\pi d^2 N_{xy}}{4} \quad (4)$$

The average heat transfer coefficient for the heated pin-fin assembly can be obtained by combining the equations. (2) and (3):

$$h = \frac{m\, c_p \left(t_{out} - t_{in}\right)}{A_s\left[t_b - \left(\frac{t_{in} + t_{out}}{2}\right)\right]} \qquad (5)$$

Given the current operating circumstances and the well-insulated test section, the free flow area is computed as follows:

$$A_{ff} = W(H+C) - N_x H\, d \qquad (6)$$

The Nusselt number is calculated using the previous heat transfer coefficient value as conventionally,

Nu= hDh /k (7)

The Reynolds number (Re) is defined as

$$Re = \frac{G\, d}{\mu} \qquad (10)$$

RESULTS AND DISCUSSION

End results and communicate the heat transfer properties are investigated for in-line pin fins with constant spacing between them in the spanwise direction and for a constant C/H (0.0) value. Additionally, the experiment uses the influence of spacing (SY/d = 1.2, 2.4, 3.6) for various mass flow rates (0.069 kg/s to 0.143 kg/s) and Reynolds numbers (2000 to 25000). For the constants SX=1.2 and SY=1.2, 2.4, and 3.6, figures are shown. Plotting the Nusselt number values and the enhancement factor with respect to the Reynolds number function shows the increase in heat transfer caused by the application of grooves. Additionally, the fin effectiveness plots are displayed to illustrate the heat transfer rate of a grooved hexagonal fin as compared to a hexagonal for a range of Reynolds numbers. Additionally, the fin effectiveness plots are displayed to illustrate the heat transfer rate of a grooved hexagonal fin as compared to a hexagonal for a range of Reynolds numbers.

Calibration and steady state measurements

Fig.4 shows the typical transient temperatures of the base plate measured at certain locations. About 2.5 hours into the experiment, the steady state was reached, and to verify the steady state values, another hour or so of experimentation was conducted.

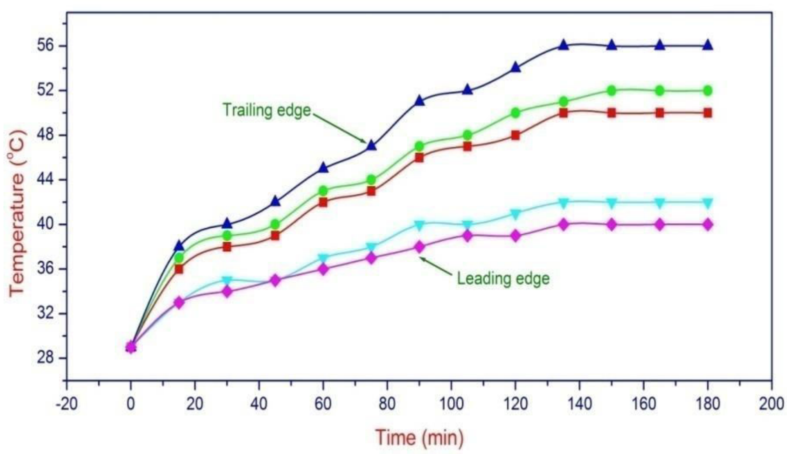


Fig. 4 Typical transient temperature measurement using T-type thermocouple

Heat transfer rate estimation:

The average heat transfer rate with Reynolds number for various constants SX=1.2 and SY=1.2, 2.4, and 3.6 in the finned surface is shown in Fig. 5. In the grooved hexagonal fin array, the heat transfer findings for SY/d = 1.2

were more favorable than in the hexagonal fin array. Because of the increased fluid flow and reduced hydraulic diameter, the grooved hexagonal creates more swirls. As a result, as the flow rate increases, the fluid flows in both tangential and axial directions, enhancing the mixing effect and raising the heat transfer. As a result, for all Reynolds number values, the grooved hexagonal fin out performs the hexagonal fin geometries for heat transfer rate. The heat transfer ratio rises as SY/d.

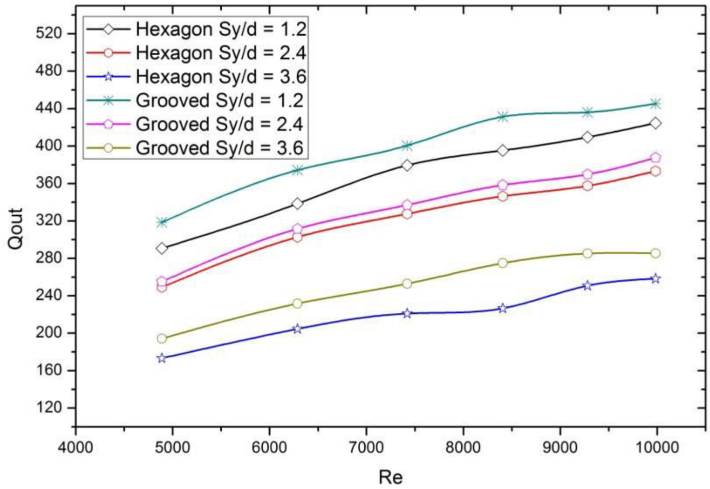


Fig.5 .Effect of pin-fin shape on Qout for SX/d = 1

Impact of Nusselt Number as Function of Reynolds Number:

Fig. 6 illustrates the correlation between the constant values of SX= 1.2 and SY= 1.2, 2.4, and 3.6 and the Nusselt number. The figure illustrates how the Nusselt number decreases as SY/d increases or vice versa; nevertheless, bigger SY/d cause the air flow between the fin columns to become less restricted, which reduces the effective contact area of the air flow and the pin-fins. As a result, this initiates the heat transfer between the air flow and the pin-fins, which also causes the pressure drop to grow concurrently. A high Nusselt number is correlated with a small SY/d because an increase in SY/d indicates a decrease in the pin-fin's lowest velocity.

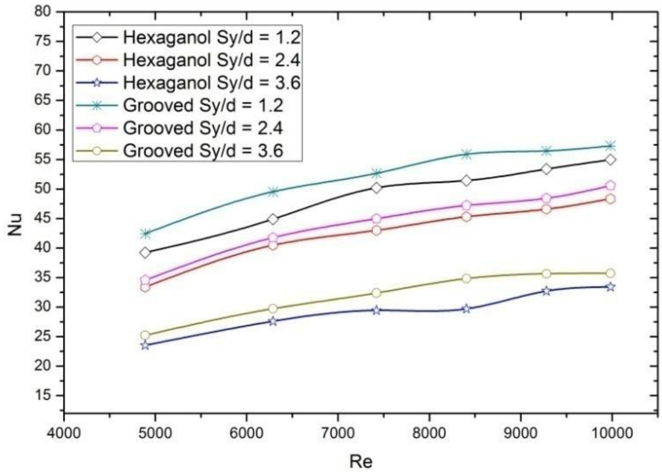


Fig.6 Pin-fin shape's effect on Nu for SX/d = 1.2

The ratio of the Nusselt number of the groovedhexagonal fin to that of the hexagonal fin determines how the Nusselt number varies with Reynolds numbers.

Friction Factor:

For constant SX=1.2 and SY=1.2, 2.4, and 3.6, Fig.7 plots the fluctuation of pressure drop for different fluid flow rates against Reynolds number. The graphic shows that the hexagonal fins experiences fewer pressure dips than the grooved hexagonal fins due to the high SY/d. The findings indicate that pressure drop normally reduces as SY/d increases.

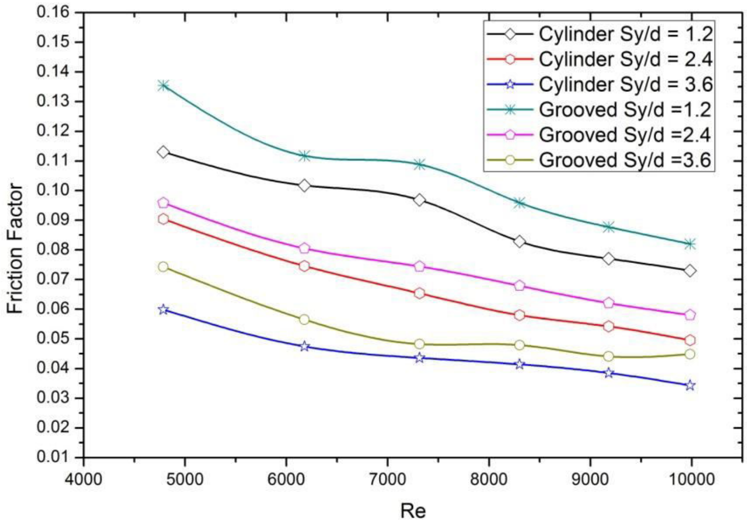


Fig.7 "Friction" against Reynolds number plot for SX/d=1.2

Effect of Nusselt number enhancement ratio

The variation of Nusselt number with Reynolds numbers is determined by theratio of the Nusselt number of the grooved hexagonal and hexagonal (Nug/Nus) for different Reynolds number (C/H=0 and S_x=24). The Fig.8 shows as a function of Nusselt number enhancement ratio for different Reynolds numbers. It is obvious from the plot that grooves lead a considerable enhancement in the heat transfer. Theenhancement factor values are gradually with an increasein Reynolds number for all stream-wise ratios (S_y/d). The enhancement factor is high for the spacing ratio S_y/d = 2.4 among all stream-wise ratio considered in this study.

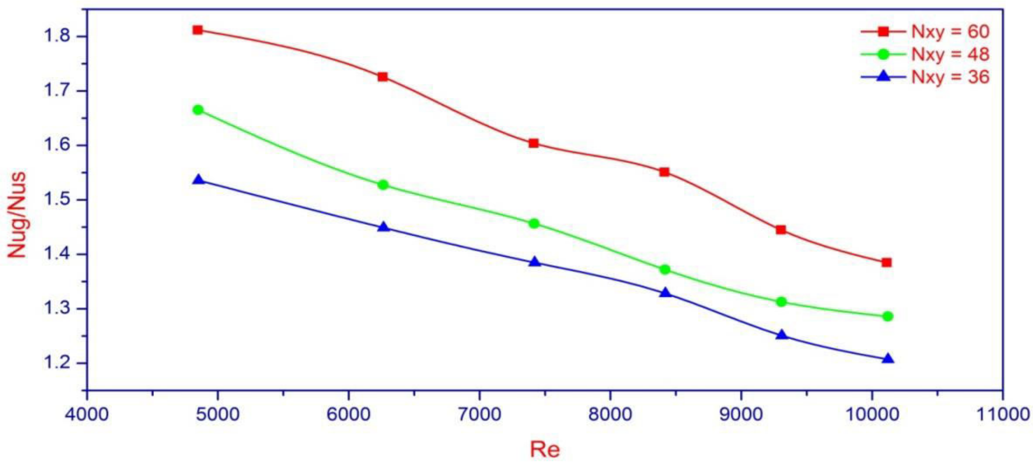


Fig. 8 Effect of Nug/Nus *Vs*Reynolds number for C/H=0, S_x= 24, Sy=24, 36 and 48

IN COMPARISON TO OTHER CORRELATIONS OF HEAT TRANSFER

Heat transfer data is typically shown as a variation of the Nusselt number versus the Reynolds number. One such plot for the current work is shown in Fig 9, where it is evident that the calculated Nusselt numbers are rising with Reynolds number. The data are being validated through comparison with earlier research. Within the spectrum of experimental settings, the current results are both higher and closer to the Tahat et al. [13] and Jubran et al. [12] values.

$$Nu = 0.45(Re)^{0.71}\left(\frac{S_x}{W}\right)^{0.40}\left(\frac{S_y}{L}\right)^{0.51} \quad \text{in - line} \qquad (1)$$

$$Nu = 9.02x10^{-3} Re^{1.011}\left(\frac{S_x}{W}\right)^{0.285}\left(\frac{S_y}{L}\right)^{0.212} \quad \text{in - line} \qquad (2)$$

$$Nu = 0.2721 \, (Re)^{0.46} \frac{(S_y)^{0.17}}{(L)} \quad \text{in - line} \qquad (3)$$

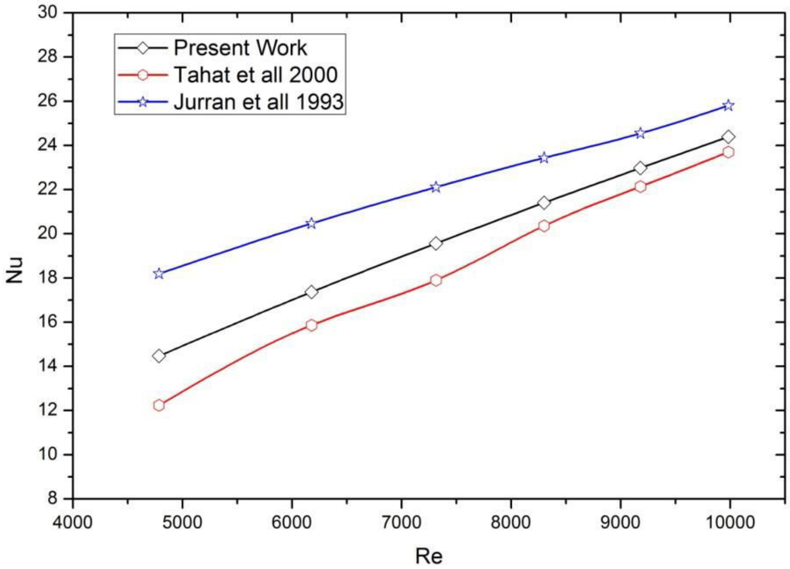


Fig. 9 Effect of perforation on Nu for C/H=0.0; S_x= 24; S_y=24.

CONCLUSIONS

This work examines the analysis of heat transfer under forced convection of a pin fin through experimentation. Using the hexagonal fin as a reference, the performance of the modification of the grooved hexagonal fin has been studied. The experiment results above lead to the following deductions.

For all Reynolds number ranges, hexagonal grooved pin fin array performs better in heat transfer than a hexagonal pin fin array. It is found that the ideal stream-wise ratio (SY/d = 1.2) increases the rate of heat transfer. However, because there is more fluid obstruction in the hexagonal grooved fin array than in the hexagonal pin fin one, it has a higher pressure drop and higher heat transfer than the latter.

For every fingeometry, the Nusselt number value raises as the Reynolds number value rises and the stream-wise ratio decreases. It was shown that the existence of grooves in every instance results in noticeably increased Nusselt numbers across the board for all Reynolds number levels. The grooved hexagonal fin and hexagonal fin with Nusselt numbers for SY/d=2.4 and SY/d=3.6 show mutual proximity for varying Reynolds numbers.

The Nusselt number of a grooved hexagonal fin divided by the hexagonal fin enhancement factor yields the enhancement factor, which is comparatively greater for grooved hexagonal fins. As the inter-fin distance ratio decreased, the friction factor increased. The hexagonal fin and grooved hexagonal fin have slightly different friction factors.

Nomenclature:

A area, m 2
C clearance between fin tip and the roof, mm
d pin-fin diameter, mm
f friction factor
G mass flux, measured in kg/m 2 s
H pin-fin height in mm
K thermal conductivity, W/m K
L: Base plate length in mm
M air mass flow rate, kg/s
N pin-fin numbers

Nu Nusslt number
Q rate of heat transfer, W
Re Reynolds number
S spacing, mm
T temperature,in °C
T temperature, in K
W Base Plate Width, in mm
x, y, z set of Cartesian coordinates
Δp total pressure drop along the array,in N/m 2 .
m dynamic viscosity, N s/m 2
ρ density, kg/m

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