Investigation of convective heat transfer and pressure drop in various pin fin arrays

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Abstract

This study examines forced convective heat transfer experimentally using an array of circular pin-fins affixed to a rectangular, horizontal base plate kept at 50°C and positioned in a rectangle channel that serves as an airflow channel. The impact of pin-fin spacing on heat sink performance was investigated in both stream- and span-wise directions. The effect of tip clearance was also investigated. The array's placements of cylindrical fins, which can be arranged staggered or in line. The investigation employed the following parameters: i) Reynolds number: 2000–25000; ii) inter fin distance: 12 mm–228 mm in both stream wise (sy) and span wise (sx) directions; and iii) clearance ratio (C/H): 0.0, 0.5, and 1.0.

Keywords: Heat exchangers, Pin-fin, forced convection, Channel flow, pitch, tip clearance and pressure drop.

Introduction

Most electronic components generate an astounding amount of heat internally as a result of various industrial operations, which can cause overheating and insufficient heat disposal systems to cause system failure. According to a U.S. Air Force research Reynell, [14] the four primary stresses that cause avionics equipment to fail are temperature (□55%), vibration (20%), excessive humidity (19%), and dust (6%). Temperature and humidity are related phenomena as well. Thus, 74% of the breakage was due to thermal overstressing. Because of this, it is frequently required to use an effective cooling system to remove heat. It is usually necessary to provide cooling to keep component temperatures below 50 °C in order to extend the mean life of an electronic system between failure and replacement. Fins are frequently used. Fins are frequently employed in heat-exchanging devices to increase the rate of heat transfer from the surface to the surrounding fluid. Among the various fin geometries that are employed are tapered pin-fin configurations, elliptical, square, rectangular, cylindrical, diamond (Tahat et al. [19]; Sara, [13]. The pin-fin array system's heat transfer characteristics have been extensively researched due to its importance. Heat sinks with pin-fins are widely used. A pin-fin is an element that is fixed perpendicular to a wall and resists fluid flow. Several characteristics define the pin-fins: height, diameter, shape, and ratio of height to diameter. In addition to their physical geometry, pin-fins are stacked in arrays that are either inline or staggered with respect to the flow direction.

Electronic system packaging topologies have become increasingly complicated, with packing densities nearing 10^6 chips per cubic meter Doyle, [5]. Numerous studies have focused on the pressure drop and heat transmission in channels with circular cross-section pin-fins Chyu MK.,[4]. Among the first to examine the heat transmission capabilities of inline and staggered wall mounted arrays of cylindrical fins were (Sparrow and Ramsey. [15]. the heat transmission properties of staggered arrays of cylindrical pin-fins were studied by Metzger et al., [11]. Additionally, the heat transmission from a staggered array of cylindrical pin- fins was examined by Simoneau and Vanfossen., [16]. Armstrong and Winstanley., [1] provided an overview of staggered array pin-fin heat transfer for turbine cooling applications. Matsumoto et al., [10] investigated the end wall heat transfer in the presence of staggered and inline adiabatic circular pin-fins. Grannis and Sparrow [7] examined the properties of pressure drop and heat transmission while numerically simulating the fluid flow through a series of pin-fins shaped like diamonds. The heat transfer and pressure drop for a rectangular channel with arrays of diamond-shaped components were examined by Tanda [20]. In the thermal performance analysis, both inline and staggered fin arrays were taken into account with constant pumping power and constant mass flow rate limitations Chyu et al. [3]. The impact of fin shape on the mass transfer and pressure loss of staggered short pin-fin arrays in a rectangular duct was examined by Goldstein et al. [6]. The heat transport and friction properties of rectangular channels with inline square pin-fins were studied by Sara et al. [13]. According to a review of the literature, studies on pin-fin array systems have looked at a variety of parameters based on heat flux rate, including clearance ratios, height to diameter ratios, inter-fin spacing in both stream-wise and span-wise directions, and friction characteristics.

It is commonly known that fin-equipped pin-fin arrays transmit heat more efficiently than fin-free plain channels. However, there is usually a significant increase in pressure loss in conjunction with the increase in heat transmission. The majority of pin-fin applications require that the characteristics of pressure loss and heat transfer be taken into account at the same time. The heat transfer and friction characteristics have generally been obtained in these experiments, and the best parameters have typically been found based on the maximum heat transfer rate or maximum heat flux rates [21]. It is obvious to carry out a performance analysis in order to express performance in terms of the four interconnected qualities of size, shape, pressure drop, and heat transfer, at the very least. Conversely, pin-fins of different cross-sections have distinct properties related to heat transfer and flow resistance, and since they are relatively easy to fabricate, circular pin-fins are frequently utilized in applications. Therefore, in order to improve heat transfer and reduce flow resistance, it is crucial to examine different pin-fin designs with varied cross-sections. This research aims to explore the performance characteristics, pressure, and heat transfer for in-line and staggered cylindrical and square pin-fin arrays mounted to a flat surface within a rectangular duct.

Experimental apparatus

The test portion of the wind tunnel, where the experiments were carried out, measured 180 mm in height, 150 mm in width, and 2 m in length. The channel's wall thickness is 20 mm. 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/or end walls are symmetric with respect to the center plane. Duralumin plate pin-fin assembly with dimensions of 250 mm length, 145 mm width, and 25.4 mm (1 in) thickness was employed. Directly beneath the test module was a plate heater with roughly the same dimensions that provided a steady heat input to keep the surface temperature consistent. With a duralumin plate that thick, this is achievable. The heater has a maximum power of around 1500 W. With the auto transformer (Variac), the heater's output of heat can be adjusted. A heat sink material was put between the test plate and the fins as well as between the heater and the plate to lessen contact resistance to heat flow. By insulating the side of the test module and the bottom of the heater, heat loss to the environment is reduced to zero. This is accomplished by using high heat resistance packing, with glass wool that is 25 cm thick on top and 2 cm thick wood underneath. Moreover, 5 cm of regular glass wool was used to totally insulate the channel's exterior surfaces downstream from the start of the test segment. An movable top cover was installed on the upper surface of the channel to facilitate the easy placement of the pin-fin assembly on the heating surface. Fig. 1 provides a photographic view of the square pin fin assembly. Fig. 2 presents a photographic view of the cylinder array, while Fig. 3 presents a schematic representation of the experimental test rig.

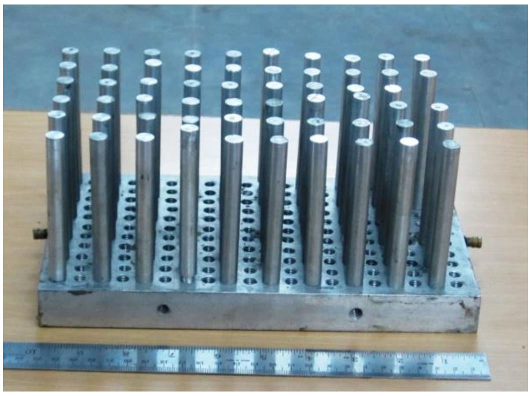

Fig.1. Photographic view of square array **Fig.2. Photographic view of cylinder array**

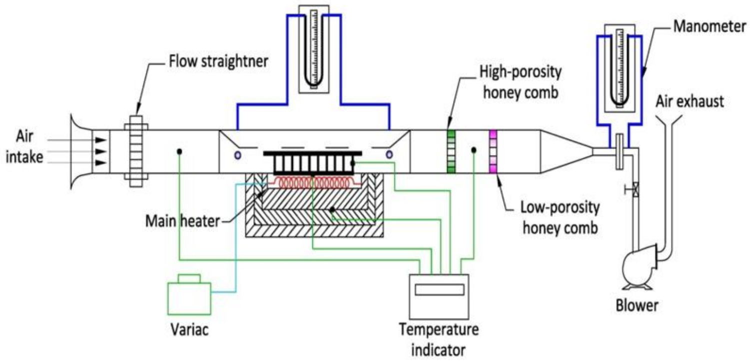

Fig.3. Schematic representation of the experimental test rig

Data Quantification C was the constant average surface temperature used for the studies. The average value of four RTDs placed after the flow straightener was used to determine the air stream's entrance temperature. In a similar vein, the average value of four RTDs placed in the downstream section of the insulated channel was used to determine the air stream's output temperature. There is one thermocouple used for the ambient temperature and one for the heating section's outer surface temperature. Using a temperature bath, all of the temperature sensors are calibrated against a Pt100 reference one. Every experiment session lasted for 25 minutes, even after reaching a steady state, which takes approximately 1½ hours. To measure the air flow rate. The test module's base plate temperature is monitored by nine symmetrically distributed copper-constantan (type-T) thermocouples. With a resolution of ±0.1 ^{o} C, the digital temperature indicator is linked to these thermocouples. The test surface's steady state temperature was determined by averaging these readings. 50 °C, through the tunnel, an orifice meter is integrated into the downstream side of the test section. An air blower operating in induced draft mode was installed to create air flow within the wind tunnel. The diameter and height of the cylindrical fins that are used are 10 mm and 90 mm, respectively. A heat exchanger's specification will frequently list the highest pressure drop the system is capable of handling. The pressure reduction at the entry and pressure rise at the exit that would result from a change in the flow-area has to be considered in the current inquiry. The flow through the pin-fin assembly causes pressure loss. The pin-fin assembly (Figure 3) was used to measure pressure using static pressure tapings, and the overall pressure drops were assessed for each test that was run. The setup was reconfigured to the next clearance ratio (C/H) once the steady state data for each trial with the required parametric modifications were obtained. Table 1 lists the lowest and highest values for the different experimental setups that were employed in this study.

Table 1 Experimental conditions

Parameters	**Minimum value**	**Maximum value**
x-direction pin-fin spacing [S_x] (mm)	24	24
y-direction pin-fin spacing [S_y] (mm)	24	228
Mass flow rate [m] (kg/s)	0.069	0.143
Clearance ratio [C/H]	0.0	1.0
Total number of fins [N_{xy}]	18	60
Reynolds number [Re]	2000	25000
Diameter of cylindrical pin-fin [d] (mm)	10	
Height of cylindrical pin-fin [H] (mm)	90	
Side of the square pin-fin [d] (mm)	9.2	
Height of the square pin-fin [d] (mm)	76.8	
Base plate W x L (mm)	145 x 250	
Base plate temperature [t_b] (°C)	50 ±0.25	

DATA REDUCTION

Frictional factor and heat transfer rate are assessed using the obtained data. The constant state heat flow from the finned surface is as follows:

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

The data reduction in the current instance is comparable to the method used by Naik et al.,[12]; Jubran et al., [8] ; Tahat et al., [19] ; Sara., [13]. The trials that they carried out revealed that fin arrays are comparable, and they stated that the overall heat loss attributable to the assembly was less than five percent. Given the current operational conditions, the test portion's adequate insulation, and the assumption of extremely low heat loss, we rewrite equation (5.1) as follows:

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

This equation describes the heat transfer that occurs by convection from the base plate to the fin surface.

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

In this context, the temperatures of the air flow are represented by tin and tout, the average temperature at specific points on the base assembly is represented by tb, and the total surface area of the base assembly and fins is included, which can be expressed as follows:

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

Given the current operational conditions and the adequate insulation of the test area, the calculation for the free flow area Aff can be found below:

$$A_{ff} = W(H + C) - N_x H d \quad (5)$$

At atmospheric pressure, air has specific heat, dynamic viscosity, and thermal conductivity. The range corresponds to Tahat et al. [20].

$$c_p = \left[9.8185 + 7.7 \times 10^{-4} \frac{(T_{in} + T_{out})}{2}\right] \times 10^2 \quad (6)$$

$$\mu = \left[4.9934 + 4.483 \times 10^{-2} \frac{(T_{in} + T_{out})}{2} \right] \times 10^{-6} \tag{7}$$

$$k = \left[3.7415 + 7.495 \times 10^{-2} \frac{(T_{in} + T_{out})}{2} \right] \times 10^{-3} \tag{8}$$

The Reynolds number (Re) is defined in the conventional way as,

$$Re = \frac{G\,d}{\mu} \tag{9}$$

where, $G = m/A_{ff}$ is the mass flux.

RESULTS AND DISCUSSIONS:

Carried out the steady state heat transfer for the in-line and staggered pin-fin arrangements by changing the C/H value, spacing in the stream-wise direction (Sy), and maintaining a constant spacing in the span-wise direction (Sx). This was done for a variety of mass flow rates. The pin-fins that were employed in the testing had a cross-sectional shape that was classified as either cylinder or square.

Clearance Ratio Effect on Heat Transfer:

Figure 4 illustrates the impact of the clearance ratio on the heat transfer rate across the range of Reynolds numbers examined in the trials. For any possible C/H ratio, it would appear that the rate of heat transfer increases in a manner that is consistent with the Reynolds number. Furthermore, the smallest value of C/H yields the highest heat transfer. Increasing the compactness of the heat exchanger improves heat transfer by restricting the free passage of air. Scientists who came before have noted this pattern [8, 13].

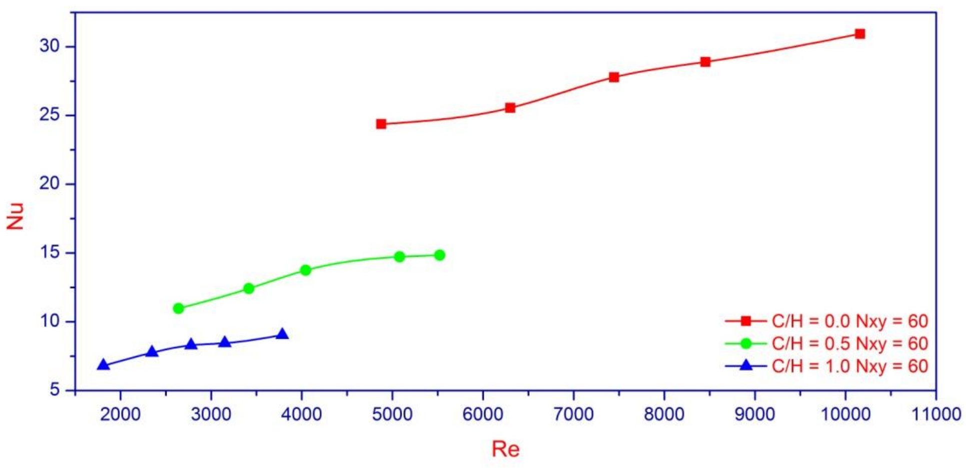

Fig. 4 plot of Nusselt number *Vs* Reynolds number for $S_x = S_y = 24$ (cylinder)

Effect of pin-fin surface area / number of fins

Typically, we can increase the amount of heat transfer by reducing the distance between the fins in both directions. Figure 5.8 demonstrates the impact of the circuit's surface area on the pin fin's performance. Adjust the surface area to accommodate an appropriate number of pin fins in the array. Refer to this as the packing density, and denote it accordingly. Increasing the surface area from the lowest value (Nxy = 18) resulted in an improvement in performance up to a certain value of surface area, but this improvement came at the expense of pumping power. However, once the optimal However, after reaching the optimal number of pin-fins (i.e., surface area), any further increase in surface area resulted in a decrease in the Nusselt number value.

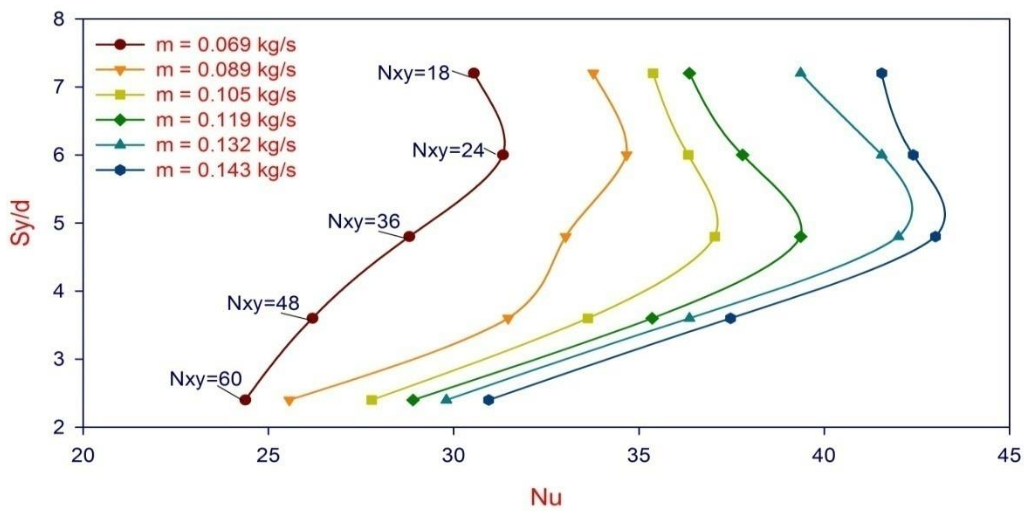

Fig. 5 Effect of pin-fin surface area for C/H=0, S_x= 24 (cylinder)

Surface Area Effect on Heat Transfer

The enhancement of heat transmission is typically accomplished through a variety of methods, and usually, it is accomplished by minimizing the inter-fin distance in both the stream-wise and span-wise directions. Figure 6 shows how the thickness of the surface area influences the pin fin's performance. The surface area varies as a result of accommodating more pins. Both Tahat et al. [15] and Kadir et al. [21] have reported observations that are comparable to one another.

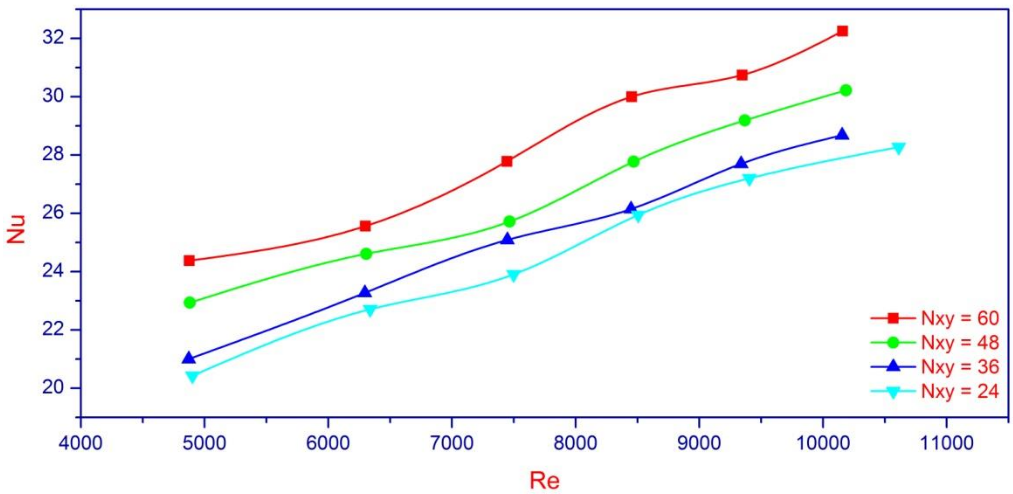

Fig.6 Plot of Nu *Vs* Reynolds number for C/H=0, Sx= 24 (cylinder)

Effects of Fin Number and Flow Rate on Pressure Drop

In order to have the least amount of pumping power possible as a limitation, it is not unusual for the flow resistance to be a significant component in heat transfer. Figure 7 depicts the change of pressure drop for different flow rates as a plot of friction factor versus Reynolds number. This represents the situation where the ratio of C to H is equal to zero. As reported [8, 13], in the case of a staggered arrangement, the in-line arrangement achieves a slight variation in the friction factor on the Sy/d change. This is in contrast to the relatively substantial variation on Sx/d that occurs in a staggered arrangement.

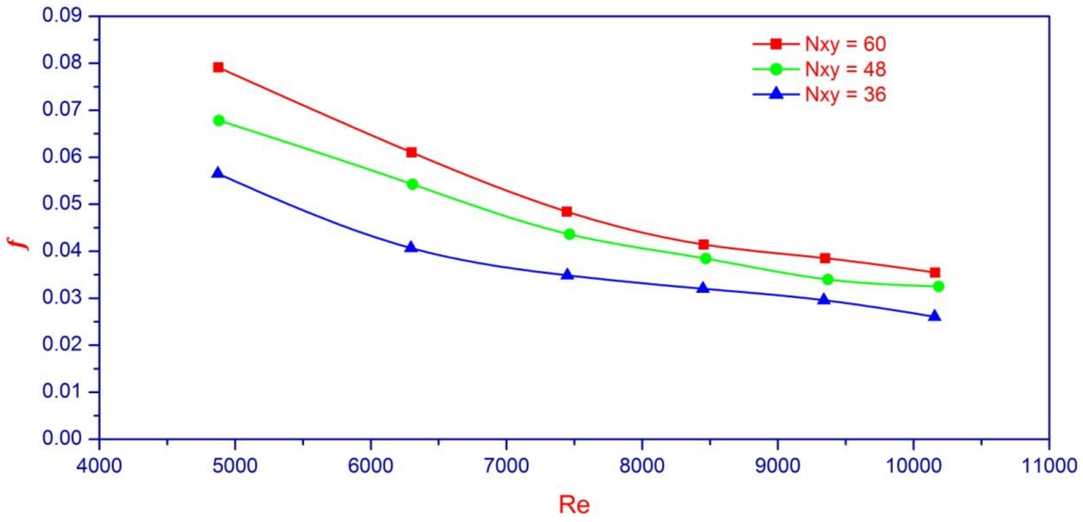

Fig. 7 Plot of '*f*' *Vs* Reynolds number (cylinder) for S_x= 24, S_y=24, 36, 48

Effect of pin-fin shape:

We conducted a series of experiments to investigate the performance of several pin-fin arrays with both cylindrical and square cross sections. The experimental results presented in Figures 8 indicate that, despite both types of pin-fins having the same surface area, the square pin-fin array performs marginally better than the cylindrical arrays for C/H values of 0.0 and 0.5. The improvement is a result of the turbulence or separation created by the bluff body (square) in the system. As we contrast the streamlined cylinder with the conventional cylinder,

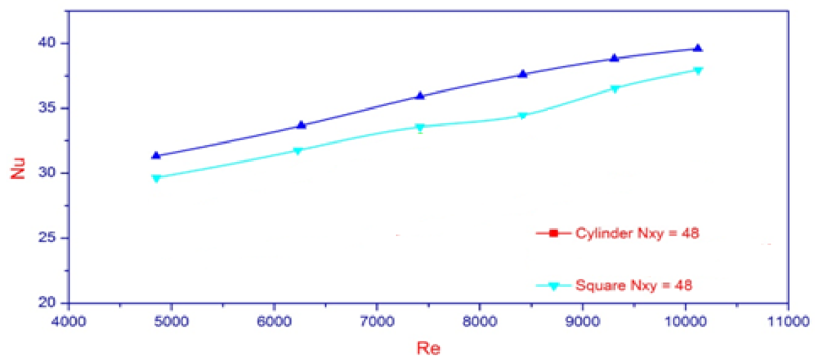

Fig. 8 Plot of Nu *Vs* Reynolds number for C/H=0, Sx= 36

Effect of pin -fin arrangements.

Figure 9 illustrates a comparison between the performance of in-line pin-fin configurations and the staggered one under the same experimental conditions. The number presented here makes it abundantly clear that the use of staggered designs results in a better rate of heat transmission, albeit at the expense of a decrease in pressure.

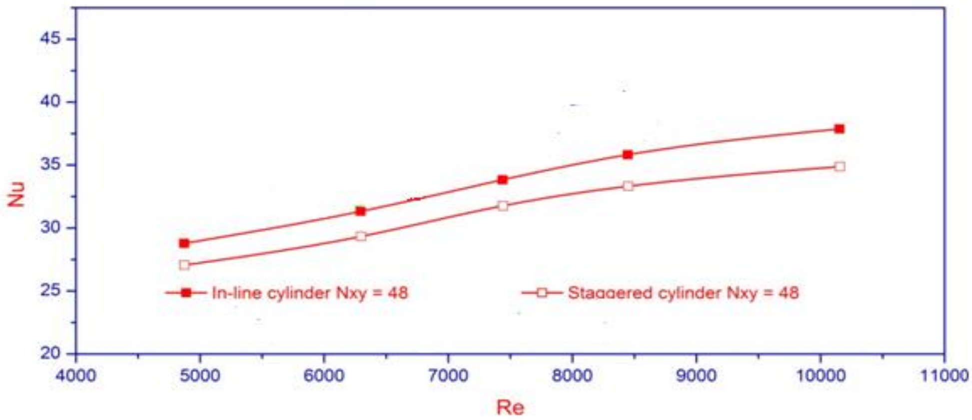

Fig. 9 Effect of pin-fin-shape on Nu for C/H=0, S_x= 24, S_y= 36

Comparison with other heat transfer correlations

When presenting statistics on heat transfer, it is common practice to do so in terms of the fluctuation of the Nusselt number against the Reynolds number. Figure 10 presents a similar plot for the purposes of this work. We cannot deny the fact that the Nusselt number increases with the Reynolds number. We are comparing previous work to validate the collected data. Within the spectrum of experimental settings, the current findings are more comparable to and more favorable than Kadir et al. [9], while Jubran et al. [8] and Tahat et al. [21] have produced results that are significantly less favorable.

$$Nu = 0.45(Re)^{0.71}\left(\frac{S_x}{W}\right)^{0.40}\left(\frac{S_y}{L}\right)^{0.51} \quad \text{in - line} \qquad (10)$$

$$Nu = 9.02x10^{-3}\,Re^{1.011}\left(\frac{S_x}{W}\right)^{0.285}\left(\frac{S_y}{L}\right)^{0.212} \quad \text{in - line} \qquad (11)$$

$$\overline{Nu} = 0.2721\ (Re)^{0.46} \left(\frac{S_y}{L}\right)^{0.17} \quad \text{in - line} \qquad (12)$$

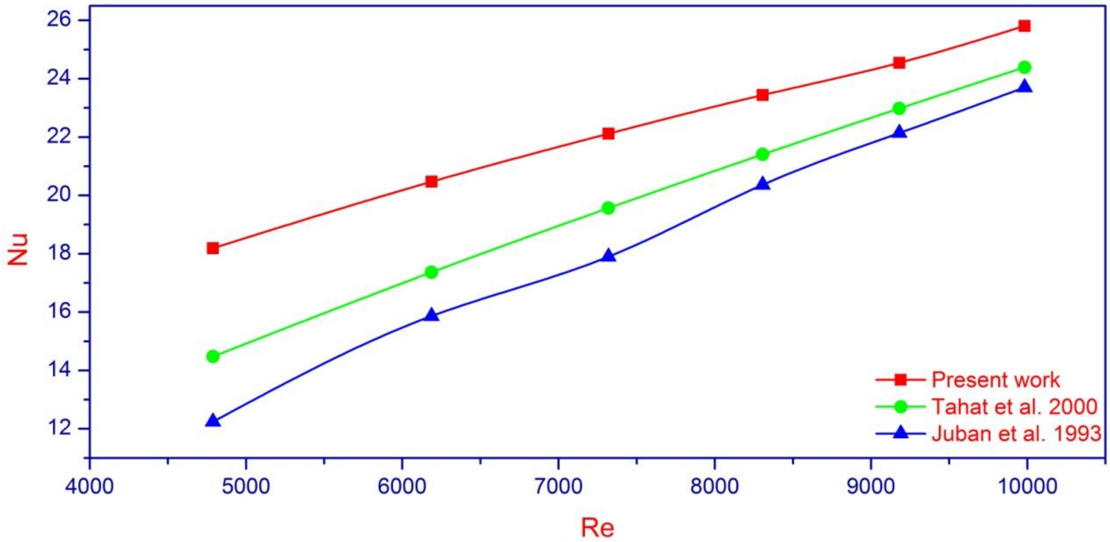

Fig. 10 Plot of Nusselt number *vs* Reynolds number (with other correlation)

CONCLUSIONS

An experimental analysis of the fin-pin assembly was conducted in a wind tunnel to examine the heat transfer characteristics. The conclusions derived from the current investigation are listed. The variation in spacing in both span-wise and stream-wise directions affects the Nusselt number. The Nusselt number variation is more pronounced in the stream-wise direction (Sy) than in the span-wise direction (Sx). For specific packing densities of the pin-fin (or Nxy), there are maximum values of the Nusselt number. The average Nusselt number consistently increases as the Reynolds number increases. The average Nusselt number increases monotonically with a decreasing clearance ratio, while the influence of the inter-fin distance ratio on the Nusselt number is non-monotonic. There exists an optimal value for the inter-fin distance ratio (i.e., spacing). For a specific Reynolds number, the pin-fin array with a designated inter fin spacing or quantity of pin-fins exhibits superior performance compared to other configurations.

The staggered array yields a slightly greater improvement over the in-line design. Consequently, the augmentation in heat transmission resulted in an increased pressure drop. In terms of pin-fin shape, square fins have better heat transfer performance than cylindrical fins.

Notations

A	–	area, (m^2)
C	–	vertical clearance between fin tip and shroud (m)
C_1	–	constant (Nusselt number correlation coefficient)
c_p	–	specific heat of air at atmospheric pressure, (J/kg K)
d	–	pin-fin diameter, (m)
f	–	friction factor
G	–	mass flux [= m/A_{ff}], (kg/m^2 s)
h	–	heat-transfer coefficient, (W/m^2 K)
H	–	pin-fin height (m)
k	–	thermal conductivity, (W/m K)
L	–	base plate length (m)
m	–	mass flow rate of air, (kg/s)
N	–	number of pin-fins
Nu	–	Nusselt number [= hd/k]
p	–	static-air pressure, (N/m^2)
pp	–	pumping power (W)
Q	–	steady-state heat transfer rate, (W)
Re	–	Reynolds number [= Gd/μ]
S	–	fin pitch, i.e.; distance between adjacent pin-fins (m)
t	–	temperature, (°C)
T	–	temperature, (K)

W – base plate width, (m)

x,y,z – set of Cartesian coordinates

Greek Symbols

α – aspect ratio [=H/d]

$\rho\Delta$ – difference

μ – dynamic viscosity, ($N \cdot s \cdot m^{-2}$)

ν – kinematic viscosity, ($m^2 \cdot s^{-1}$)

ρ – density, ($kg \cdot m^{-3}$)

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