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Heat transfer and fluid flow characteristics of perforated pin-fin arrays

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Abstract
This work presents the outcomes of experimental investigation on heat transfer and friction characteristics over a rectangular-horizontal base plate maintained at 50°C with cylindrical and perforated pin–fin heat sinks with inline arrangements were designed and tested. The effect of distance between fins in the span-wise and stream-wise directions has been determined. The values of the relative stream wise pitch (S_y/d = 1.2, 2.4, 3.6) and the relative span wise pitch at (S_x/d =1.2). The experiments covered following ranges: Reynolds number ranges from 2000-25000, the clearance ratio (C/H= 0.0). Nusselt number and friction factor were considered are performance parameters. The results of the perforated fin compared to cylindrical fin can produce higher heat transfer rate. The present results were found to be optimal inter-fin pitches S_x /d=1.2 and S_y /d=1.2.

Keywords: Channel flow; cylindrical pin fins; perforated pin fins; Heat exchanger; performance analysis; pitch and tip clearance

1. Introduction

The operation of several engineering systems results in the generation of heat. This may cause severe overheating problems and occasionally leads to failure of the system. This system needs superior heat transfer elements with increasingly smaller weights, volumes and cost. The heat generated in a system such as transformers, refrigerators, boiler super-heater tubes, condenser coils, electronic components, compressors, air cooled engines, and etc. must be dissipated to its surroundings in order to maintain the system functioning at its recommended working temperatures and operating effectively and reliably. From the literature review Braga and Saboya. [1], they reported many methods that were proposed to achieve this goal. Active methods are those requiring external power to maintain their enhancement such as well stirring the fluid or vibrating the solid surface. On the other hand, the passive methods do not require external power to maintain the enhancement effect as when fins are utilized.

The rate of heat transfer at a solid–fluid interface can be increased by extending the surface area in the form of fins. Various types of fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered, and pin fins, to a combination of different geometries, have been used Armstrong in different heat transfer applications. These fins are the most popular fin type because of their low production costs and high thermal effectiveness. Study of influence of geometric parameters viz. Fin length, fin height, fin spacing over heat dissipation found important (Kadir *et al.* [2] and Tahat *et al.* [3]).There have been many investigations restricted on the heat transfer and pressure drop in channels with pin-fins of circular cross-section [4, 9]. Sparrow *et al.* [4] were among the first to investigate the heat transfer performance of inline and staggered wall attached arrays of cylindrical fins. Metzger *et al.* [5] investigated the heat transfer characteristics of staggered arrays of cylindrical pin-fins. Simoneau and Vanfossen [6] also studied the heat transfer from a staggered array of cylindrical pin- fins. A review of staggered array pin-fin heat transfer for turbine cooling applications was presented by Armstrong and Winstanley [7]. Jubran *et al.* [8] investigated experimentally the effects of shroud clearance, optimum inter-fin spacing in both span wise and stream-wise directions and missing pin upon heat transfer, and they correlated their heat transfer data in their study.

Tahat *et al.* [9] performed using a channel with rectangular cross section equipped with cylindrical fins. The clearance ratio used in their study is C/H=0.0 and the optimum spacing in the span wise direction of 7.6 mm was achieved. Ganesh [10] examined the transfer of heat and mass about the radiating type of pin fin with reference to the array pattern in a rectangular channel. sahin *et al.* [11] Perforated square fins are commonly used in different heat exchangers, film cooling, and solar collector applications as a result of their high heat transfer capacity at a relatively low material use. The experimental investigation on the grooved type of flat fin, conducted by Ashish [12] signified effect of forced convection in exposing the characteristics of the heat transfer. Deqing *et al.* [13] experiments were investigated for the thermal hydro dynamic characteristics micro-pin fin arrays have been studied. Shaeri *et al.* [14] has investigated experimentally heat transfer characteristics from heat sink with using perforated fins. Due to the more require for lightweight, compact, and cost-effective fins, the optimization of fin geometry is of great importance. Therefore, fins must be designed to achieve higher heat removal with lower material expenditure. The review of the literature showed a variety of modifications and the alterations about the fins by introducing the holes, slits and struts, which enhanced the heat transfer. The present experimental work, investigates the effect of heat transfer and the influence of fluid flow properties on the perforated cylindrical fins and its behavior is compared against the cylindrical fins are calculated and the results obtained from the experiments are validated as the dimensionless geometrical functions of the fins.

2. Experimentation Facility

The experimental set-up is shown in Fig.1.the set up consisted of a closed rectangular channel with removable test section, blower, U-manometer, heater and thermocouples. The centrifugal blower draws the atmospheric air through the rectangular duct; it drains out to atmosphere, can be resembled as an open loop system.

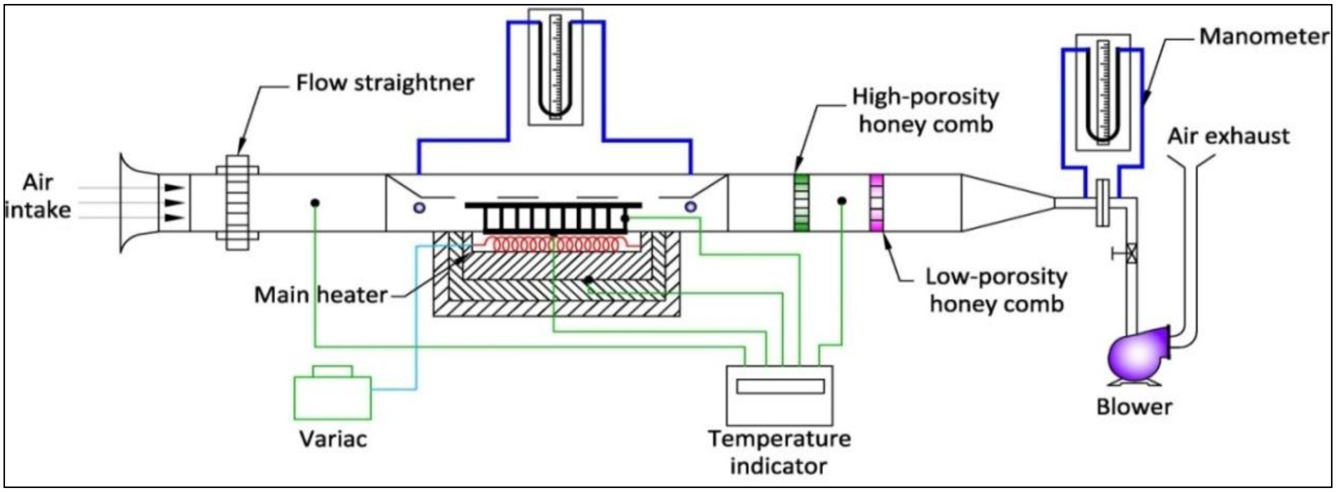

Fig 1: Line diagram of experimental setup

The tunnel is designed such that the material used for the tunnel is play wood because wood is a bad heat conductor. Tunnel constructed of play wood of 19 mm thickness, had an internal cross-section of 150 mm width and 180 mm height. The total length of the channel is 2000 mm. The tunnel is insulated from the inside to avoid the heat losses. Heater Unit (test section) has a cross-section of 250 mm x 150 mm and 25.4 mm (1in) in thickness. The base of the exchanger was heated almost uniformly by plate heater having the maximum power of 1500 W. The heat exchanger base was heated uniformly to maintain constant temperature. The pin-fin assembly base and heaters is protected in a well-fitting open-topped wooden box with proper thermal insulation of entire system. The amount of power given to the main heater was controlled by varying the variac and voltage voltmeter. The pin fins are fixed uniformly on the base plate with a constant spacing between the span wise directions of 1.2 (S_x/d), with different spacing between the pin fins in the stream wise direction. The spacing ratios of the pin fins in the Stream-wise direction (S_y/d) were 1.2, 2.4, and 3.6. Giving different numbers of the pin fins on the base plate. Material i.e. Aluminum because of the considerations of conductivity, machinability and cost. 17 calibrated T-type thermocouples were installed temperature measurements. Nine thermocouples were equally spaced along the base plate between the pin fins. Another eight thermocouples where used to measure the temperature of inlet and outlet of air. Four were placed facing the entrance of the pin-fin assembly and another four downstream of the pin fin array. Each the thermocouples, in addition to those represent the ambient air temperature were connected to the temperature indicators, the steady state attained in the experimental run is at about two and half hour and the experiments were continued for about an hour to confirm steady state value. The air flow rate was measured using an orifice meter is calibrated against the hot-wire anemometer. The flow rate of air is regulated by operating a gate valve placed before the suction side of blower. Through the experiment, air is sucked through the test section by the blower and exits the test device to the outside.

Geometrical characteristics of the pin-fins are given in Table 1.

Distances between pin-fins and number of pin-fins are given in Table 2.

3. Fin configurations

Each cylindrical fin was made of diameter 10 mm with height 90 mm size. The cylinder fin was subsequently perforated at the 19.5 mm from bottom tip of those by an 4 mm diameter drill bit over the cylindrical surfaces. Then each fin is fixed to a rectangular aluminum plate of 250mm ×145mm×25.4mm size at its base that can be heated by means of electrical heater. During the test run, a fin placed inside the test section in such a way that the air passages are over fin surfaces. The geometries of cylindrical fins together with perforated fins are shown in Fig. 2 (a and b)

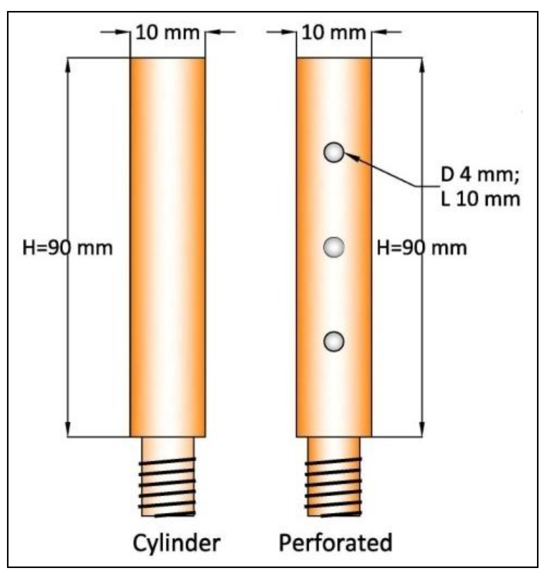

Fig 2: (a) Cylindrical fin (b) Perforated cylindrical fin

Table 1: Experimental parameters for pin-fin and their values

Parameters	Minimum value	Maximum value
Spacing of the pin fin [S_x]	12 mm	12 mm
Spacing of the pin fin [S_y]	12 mm	36 mm
Rate of flow [m] (kg/s)	0.069	0.143
Clearance ratio [C/H]	0.0	0.0
Fin count [N_{xy}]	30	48
Reynolds number [Re]	2000	25000
Diameter of pin-fin [d] (mm)	10	
Height of pin-fin [H] (mm)	90	
Base plate W x L (mm*mm)	145 x 250	
Base plate temperature [t_b] (°C)	50 ±0.25	

Table 2 Distance between pin-fins and number of pin-fins

Configuration	Arrangement	S_y/d=1.2			S_y/d=2.4			S_y/d=3.6		
		N_x	N_y	N_{total}	N_x	N_y	N_{total}	N_x	N_y	N_{total}
Cylindrical	Inline	6	8	48	6	6	36	6	5	30
Perforated Cylindrical	Inline	6	8	48	6	6	36	6	5	30

4. Data reduction

The steady state energy equation from the whole system can be expressed as follows

$$Q_{tot} = Q_c + Q_{rad} + Q_{losses} \quad (1)$$

of which the steady state rate of desired convective plus conductive heat transfer through the air for constant base temperature, t_b is,

$$Q_c = mC_p(t_{out} - t_{in}) \quad (2)$$

The heat transfer fin surface including base plate is given by

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

Where, t_b is the average temperature at center location on the base assembly, t_{in} and t_{out} are the air flow temperatures and A_s is the total test surface area, which can be expressed as,

$$A_s = WL + \pi DHN_{total} \quad (4)$$

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

$$h = mC_p \frac{(t_{out} - t_{in})}{A_s\left[t_b - \left(\frac{t_{in} + t_{out}}{2}\right)\right]} \quad (5)$$

Under the present operating conditions together with the fact that the test section was well insulated, the free flow area A_{ff} is calculated as,

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

The preceding value of heat transfer coefficient is used to determine Nusselt number as

$$Nu = \frac{hD_h}{k} \quad (7)$$

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

$$Re = \frac{Gd}{\mu} \quad (8)$$

Where, $G = \frac{m}{A_{ff}}$ Is the mass flux (9)

5. Comparison with other heat transfer correlations

It is customary to present heat transfer data in terms of Nusselt number variation against Reynolds number. For the present work one such plot is given in fig.3 it is clear that the Nusselt number arrived at are increasing with Reynolds number. In order to validate the data previous work are being compared. The present results are closer to and higher than the result of Tahat *et al.* [14] and Jubran *et al.* [15] within the range of experimental conditions

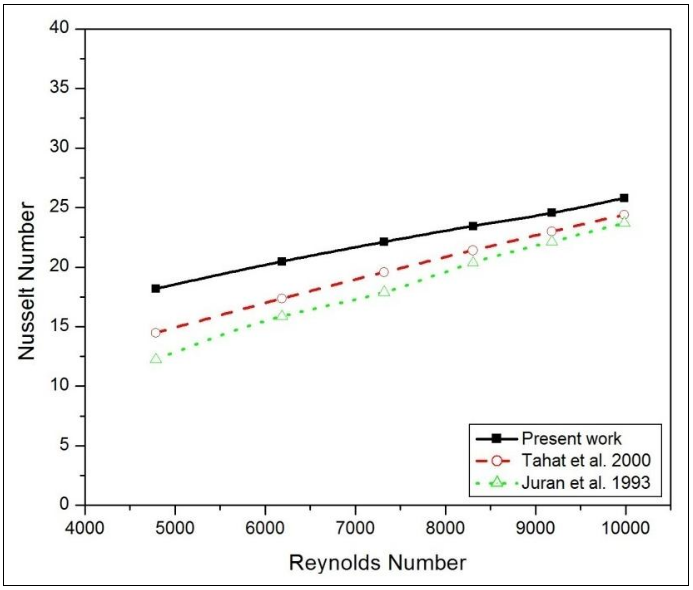

Fig 3: Plot of Nusselt number vs Reynolds number (with other correlation)

6. Results and Discussion

For the pin fins of in-line arrangement with a constant space between them along the span wise direction and for a constant C/H (0.0) value, the heat transfer characteristics are studied. In addition the effects of spacing (S_y/d=1.2, 2.4,3.6)) for different mass flow rates (0.069 kg/s to 0.143 kg/s) and for the different Reynolds numbers (2000 to 25000) are studied in the experiment. For this, the figures are presented for the constants: S_x=1.2 and S_y= 1.2, 2.4 and 3.6. In order to discuss the heat transfer rate and Nusselt number are plotted as a function of flow Reynolds number.

6.1 Heat Transfer Rate

Fig. 4 illustrate the average heat transfer rate with Reynolds number for different constant S_x=1.2; $S_{y=}$ 1.2, 2.4 and 3.6 in the finned surface. The results of the maximum heat transfer rate occur at S_y/d=1.2 in the perforated cylindrical fin array was more against the cylindrical fin array. This indicate is that perforations in the fins introduce convection rates higher in addition to, that perforations affecting the wake region flow through the perforations not only increases the turbulence but also control the flow separation in comparison to that cylindrical fin.

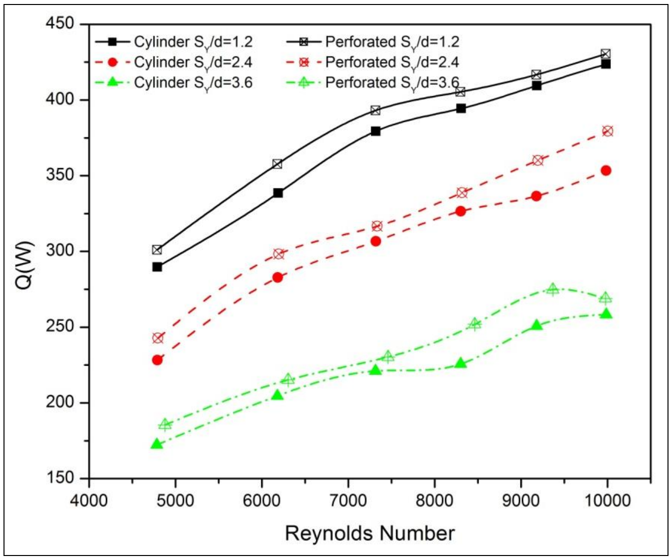

Fig 4: Effect of pin-fin shape on Q_{out} for Sx/d=1.2

6.2 Effect of Nusselt Number as Function of Reynolds Number

Fig.5 shows the relationship between the Nusselt number and the values of Reynolds number for the constant values of S_x=1.2 and $S_{y=}$ 1.2, 2.4 and 3.6. It can be observed that the value of the Nusselt number is increased with increase in the value of the Reynolds number. A small S_y/d corresponds to a high Nusselt number; because increasing Sy/d means that the minimum velocity the pin- fin decreases.

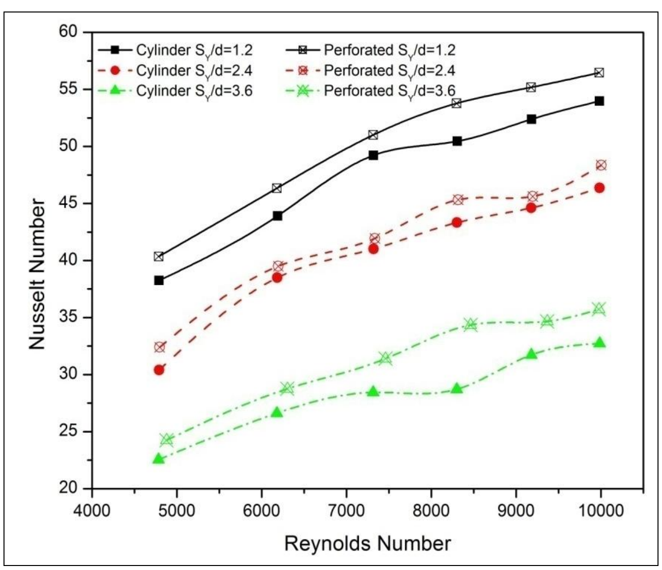

Fig 5: Effect of pin-fin shape on Nu for S_x/d=1.2

Fig.6 shows as a function of Nusselt number enhancement ratio for different Reynolds number at constant S_x=1.2 and $S_{y=}$ 1.2, 2.4 and 3.6. It is obvious from the plot that perforated lead a considerable enhancement in the heat transfer. The enhancement factor values are gradually rising with an increase in Reynolds number for all stream-wise ratios (S_y/d). The enhancement factor is high for the spacing ratio S_y/d = 1.2 among all stream-wise ratio considered in this study.

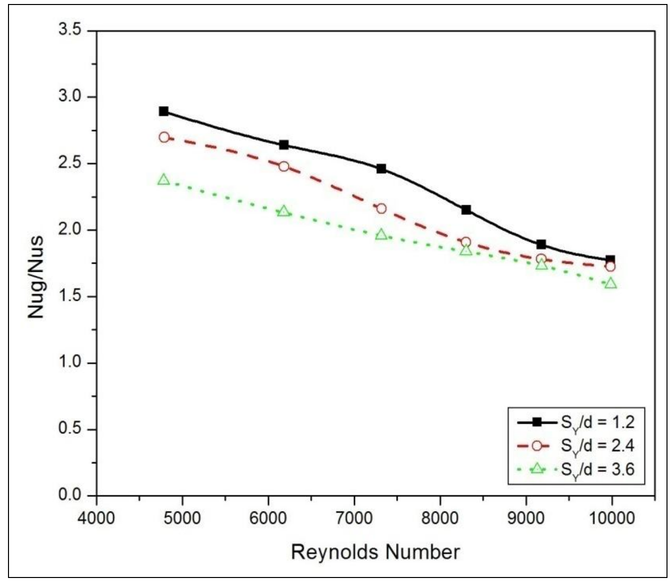

Fig 6: Effect of pin-fin shape on Nup/Nus for S_x/d=1.2

6.3 Friction Factor

The variation of pressure drop for various fluid flow rates as a plot of friction factor versus Reynolds number is show in Fig.7 for constant S_x=1.2 and $S_{y=}$ 1.2, 2.4 and 3.6. It can be seen from the figure that the high s_y/d leads to smaller pressure drops for the cylinder without grooved fins than the cylinder with grooved fins. The results show that generally pressure drop decreases with increasing s_y/d.

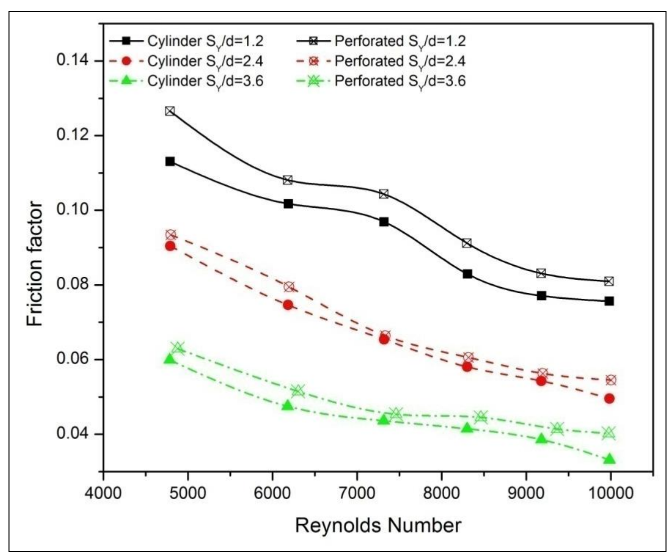

Fig 7: plot of 'f' Vs Reynolds number for S_x/d=1.2

7. Conclusions

In this study, heat transfer, friction factor and the effect of the various design parameters on the heat transfer and friction factor for the heat exchanger equipped with cylindrical and perforated cylindrical pin fin were investigated experimentally, the present conclusions can be are drawn from above experimental result discussion:

A perforated cylindrical pin fins array is more effective in heat transfer and is less effective for the cylindrical fins for all ranges of Reynolds number. The average Nusselt number increased with decreasing inter-fin distance ratio. An optimal stream wise ratio (S_y/d=1.2) to enhance the heat transfer rate. On the other hand, the perforated cylindrical fin array has the highest heat transfer and also pressure drop than the cylindrical due to more blockage effect in the fluid. Enhancement factor have been expressed by the Nusselt number ratio of the perforated cylindrical fin with cylindrical fin versus the Reynolds number. When the perforated cylindrical fin is higher enhancement factor compared to cylindrical fin. The friction factor increased with decreasing inter-fin distance ratio.

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