Thermal Analysis of Circular Pin-fin with Rectangular Slot at the Center by Forced Convection

Kavita H. Dhanawade, Hanamant S. Dhanawade, Ajay Kashikar, Shweta Matey, Mahesh Bhadane, Sunny Sarraf

Abstract**—**Extended surfaces are commonly used in practice to enhance heat transfer. Most of the engineering problems require high performance heat transfer components with light weight, volumes, accommodating shapes, costs and reliability depending on industrial applications. This paper reports an experimental analysis to investigate heat transfer enhancement by forced convection using different sizes of pin-fin with rectangular slots at the center. The cross sectional area of the oblong duct was 200 mm x 80 mm. The info utilized in performance analysis was obtained experimentally for material, aluminum at 200 Watts heat input varying velocity 1 m/s to 5 m/s. Using the Taguchi experimental design method, optimum design parameters and their levels were analysed. Nusselt number and friction factor were considered as a performance characteristic parameter. An An L_9 (3^3) orthogonal array was designated as an experimental proposal. Optimum results were found by experimenting. It is observed that pin-fins with different slots sizes have a better impact on Nusselt Number.

Keywords**—**Heat transfer coefficient, Nusselt Number, pin-fin, forced convection.

NOMENCLATURE

A_b	Area of bottom surface of base plate (m^2)
A_T	Total heat transfer area (m^2)
C	Clearance between the fin tip and upper surface of the duct (m)
C/H	Clearance ratio
S/H	Inter-fin spacing ratio
D_h	Hydraulic diameter of the duct (m)
dt_b	Temperature difference across insulating bricks
dx_b	Thickness of insulation bricks (m)
D	Diameter of fin (m)
H	Fin height (m)
H_S	Height of slot (m)
W_S	Width of slot (m)
L	Length of fin (m
L_b	Length of base plate (m)
L_S	Length of test section (m)
h_{av}	Average heat transfer coefficient (W/m^2 K)
K_b	Thermal conductivity of insulating bricks (W/m K)
K_a	Thermal conductivity of air (W/m K
Nu	Average Nusselt number
Q_N	Net heat transfer rate (W)
Re	Reynolds number
Q	Heat input (Watt)
T_{in}	Temperature of inlet air (°C)
T_{out}	Temperature of outlet air (°C)
T_{mean}	Mean bulk temperature (°C)
T_s	Average surface temperature of fins (°C)
V	Velocity over test section (m/s)

Greek Symbols

μ	Viscosity of air (kg/ms)
ρ_a	Density of air (kg/m^3)

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I. INTRODUCTION

THE reinforcement of heat transfer is a crucial area of thermal power engineering. The abolition of immoderate heat from system is important to avoid the damaging effects of burning or overheating. The heat transfer from surface may, generally, be more desirable via growing the heat transfer coefficient between a floor and its surrounding, or by way of growing heat transfer vicinity of the surface, or via each. Prolonged surfaces are manufactured in different types and geometries, depending up on the practical and industrial applications.

In order to increase the rate of heat transfer, extended surfaces are widely used in many industries. Rectangular plate fins, tapered pin-fins, circular pin-fins and square pin-fins are generally used types for both natural and forced convection heat transfers. Most of the authors tried to relate the heat transfer characteristics to the different design parameters like orientation, spacing between two fins, height of the fins, length of the fins etc. Zan et al. [1] introduced a staggered model of the array of holes in peculiar isothermal plates, vertical rectangular isothermal plates, and upright parallel isothermal fins by free convection. They assessed the performance of the perforated fin array theoretically. They found that the staggered mode of perforations can increase the total heat transfer rate for isolated isothermal plates and vertical parallel plates. Ali [2] studied experimentally heat transfer enhancement by perforation in air cooling of two in-line rectangular heat sources module. He investigated that perforation could enhance the heat transfer coefficient and reduce the module temperature. The dimensionless temperature of heat source decreases gradually with the holes to open area ratio. They observed that decrease in the average dimensionless temperature for two heat source is up to 22%, Akyol and Bilen [3] experimentally investigated heat transfer and friction factor characteristics in a horizontal rectangular duct having attachment of hollow rectangular pin-fin arrays. They studied both staggered pin-fin arrays and in-line fin arrays arrangements for four different streamwise distances and one-fixed spanwise. They show that the enhancement of heat transfer was better for the staggered fin arrays than for the inline fin arrays arrangement on the basis of total surface area, due to the increase of the turbulence and better mixing of the fluid flow. Bilen et al. [4] experimentally investigated thermal performance of a horizontal surface with fins, at constant temperature of 45 ^{o}C in a rectangular channel and observed improvement in heat transfer up to 33 % at constant pumping power. Sara et al. [5], [6] studied experimentally the thermal analysis of perforated and solid rectangular blocks attached on a flat surface in a rectangular channel. They found that a perforation in the blocks enhanced the heat transfer and enhancement increased with the increase in degree of perforations. In addition, they showed that when the blocks were perforated, there was a loss in the net energy recovery, but it depended on the geometry and flow conditions. Bayram and Alparslan [7], [8] experimentally investigated the effect of the various design parameters on the heat transfer. They used square cross-sectional perforated pin-fins and circular cross section perforated pin-fins in a rectangular duct for the experimentation. The parameters were used for investigations are Reynolds number range was 13500 to 42000, Pin-fins with different lengths, with different clearance ratio (C/H), and various inter-fin spacing ratio (S_y/D). They found that the variation in the surface area as well as that of the disturbance in the flow due to pin-fins and longer fins can also increase the turbulence of the flow in the channel, resulting was an increase in the heat transfer. Also they observed that due to the multiple jet-like flows, the enhancement in the heat transfer with perforated fins was higher than that with the solid fins. They concluded that the friction factor increases with increase in C/H and the span wise inter-fin separation distance. The most important parameters affecting the heat transfer were Reynolds number, fin height and fin spacing. Shaeri et al. [9], [10] numerically investigated and analyzed the turbulent convection heat transfer from an array of perforated fins with square window that was arranged in a lateral surface and longitudinal fins. RNG based k-turbulent model were used for investigations. They observed heat transfer enhancement in fins with longitudinal holes, in addition to the reduction in weight by comparison with solid fins. Yaghoubi et al. [11] numerically investigated three dimensional laminar steady flow conjugate heat transfers around solid rectangular fin arrays. They made use of FORTRAN code for the analysis which is based on SIMPLE algorithm with staggered grid. Reynolds number range was 100 to 250.

Reddy et al. [12] numerically investigated three-dimensional conjugate heat transfer analysis and a multi-objective optimization of micro pin-fin arrays for cooling of high heat flux by forced convection. They show that, for a specified maximum temperature, optimized arrays with pin-fins having symmetric convex lens shapes create the lowest pressure drop, as compared to the symmetric airfoil and circular cross section pin-fins. Sajedi et al. [13] numerically investigated effect of splitter on the hydrothermal behavior of a pin-fin heat sink. They studied commonly used circular and square Plate Fin Heat Sink (PFHS). They found that addition of splitter behind the fins enhance the thermal and hydrodynamics of flow inside the heat sink by reducing thermal resistance of heat sink and pressure drop around the fins. Also they show that the circular PFHS with splitter was more efficient than square. Al-Damook et al. [14] numerically investigated heat transfer and pressure drop characteristics of pin-fin heat sinks with rectangular perforation using CFD simulation. Reduction in fan power consumption along with increase in heat transfer rate by 10% for largest slotted and notched perforation was observed. Foo et al. [15] worked experimentally on the heat transfer enhancement with perforated pin-fins subjected to impinging flow. They studied five sets of staggered pin arrays with 5 evenly spaced horizontal perforations each of D_H = 3 mm and D_v = 2 mm, 3 mm, 4 mm, 5 mm. They suggested that the Nusselt number increases with increase in vertical perforation diameter and found that maximum values of Nusselt number for the fin with 5 perforation, 3 mm of horizontal which was 46% higher than solid pin-fins.

Pressure drop decreases with increase in Reynolds number and is smaller with increasing number of vertical perforation diameter. De Schampheleire et al. [16] studied numerically the impact of the size and location of the fluid domain on the heat transfer coefficient. Rectangular, an interrupted rectangular and an inverted triangular fin row were used for investigation. They show that the heat transfer coefficient decreases with adding fluid domain on both sides and bottom. For the rectangular fins, heat transfer coefficient is more, i.e. +12%, compared to the case without any fluid domain. No effect on heat transfer coefficient for the inverted triangular fin shape was observed. So, they concluded that impact of adding fluid domain depends on fin shape and for only side small amount of fluid is needed. Dhanawade et al. [18]-[21] conducted experiments on heat transfer for the rectangular fins with circular and square perforation to analyse Reynolds number and friction factor. They investigated the effects on the heat transfer characteristics adding perforation and Reynolds number. They show that average of percentage improvement of circular shape perforated fin arrays is less as compared to square shape perforated fin arrays of same size of fins. Friction factor slightly increases with increase in the size of perforation. They found that rate of heat transfer was more in perforated fin arrays than solid fin arrays. They also experimentally investigated the optimum design parameters such as Reynolds number, porosity and thickness of fin and their level for lateral circular perforated fin arrays under forced convection. They found that the maximum heat transfer rate was observed at the 87000 Reynolds number and porosity (Ø) = 0.22. Also they worked on CFD simulation and results validated with experimental results. Also they suggested that the temperature difference between the fin base and fin tip for solid fins is less as compared to perforated fin. From the visualization of velocity vector, it is observed that when the air flows through the perforation, it strikes on the inside edges of the perforation, which reticulates along the internal periphery creating the whirl. It can be concluded that free movement of air, in and out of perforation causes turbulence, in turn results in faster cooling. Maji et al. [22] numerically investigated heat transfer enhancement on heat sink using perforated pin-fins with inline and staggered arrangement. Pin-fins with single and multiple perforations, circular shaped perforation, diamond shaped perforation and elliptical shaped perforation were analyzed under steady state condition. They studied the effects fin geometry such as shape and dimension of the perforation for both the arrangement. They show that the Nusselt number increases for all types of fins with different shapes and geometry of perforations. It is increases with change of fin shape from circular to elliptical via square shaped as well as with change of perforation geometry from circular to elliptical via diamond shaped. There was increase in Nusselt number by 41% for staggered arrangement of elliptical fins as compared to linearly arrangement of solid circular fins. They also show that pressure drop decreases with increasing number and size of perforations.

From literature review it is observed that many investigators worked on rectangular plate fins, circular pin-fin, and perforated rectangular fins both by natural convection and forced convections heat transfer. The rate of heat transfer rate from fin arrays is improved with perforations, porosity etc. On the other hand, because of limitation in space and economic reasons heat transfer equipment has been required to be much more compact in size and light in weight. By providing perforations and porosity the heat transfer rate from fins can be improved. It is observed that nobody emphasized on the experimental investigation of pin-fin arrays with rectangular longitudinal slots at the center, because it required huge number of experiments; which extremely increase the experimentation time and cost. For many industrial applications, it is necessary to find out the monetary benefits for the reducing power consumption. Therefore, the aim of this study was to determining the thermal performance and analysis of circular pin-fin arrays with different sizes of longitudinal rectangular slots at the center for varying velocity (1m/s to 5 m/s). At the same time using design of experiment by Taguchi method, the aim was to minimize the experimental trials and determine new design parameters and their levels.

For the experimentation, all fins were made up of aluminum material (thermal conductivity = 225 W/m^{2}K). Formation of complex cross-sections is possible with the help of aluminum extraction.

II. Experimental Set up

Fig. 1 shows the experimental set up, which consists of the two control panels, test section, effuser, diffuser, etc. The duct was 2800 mm in total length with the hydraulic diameter D_h = 114.28 mm. It was placed horizontally in suction mode. The internal cross-section of test section was 200× 80 mm (width × height) and has length 260 mm along the direction of flow. Geometries of different types of fin arrays are shown in Table I. Fig. 2 shows the schematic of the fin arrays used in the experiment. The C/H values were constant i.e. 0.33. Velocity was varying from 1m/s to 5m/s. The Reynolds number is based on the average velocity and the hydraulic diameter (D_h) of the channel over the test section. As a heat source, 450 watt electric heating coil was placed between the upper plate and the bottom base plate. To achieve constant heat flux along the test section and to control the electric power input of the heating coil, the variac transformer was used. All the experiments were conducted for 200 watt heat input. To reduce conduction heat loss through the base and sides the base plate, the whole assemble with pin-fin arrays was placed inside insulating brick of 70 mm thickness. Only the upper surface of the base plate was inside the duct environment. Total 28 Copper constantan k type thermocouples were used for experimentation, out of that 22 thermocouples were fixed inside on the surface of fin arrays and base plate at different points to measure the temperature at various locations. Two thermocouples were placed inside the insulating bricks at some depth from base plate to determine heat loss to the atmosphere. Four thermocouples were used to measure the air temperature at the inlet and the outlet temperature of the air respectively. Panel board with digital temperature indicator having a range from 0 ^{o}C to 1000 ^{o}C (± 0.1 degree) was used to measure the temperatures at various locations. AM4201A thermo anemometer was used to measure inlet velocities of the air flow entering inside the duct on the test section. The differential pressure transmitter was used to measure the pressure drop over the test section. Two static pressure tapings fixed above the of the test section, such as entrance and exit side of air flow. The pressure taps were connected to a differential pressure transducer, which can make measurements in the range of 0-100 pa. (KIMO CP100) was connected to the pressure taps. The pressure transmitter is having the accuracy of ± 1.5% of the recorded value.

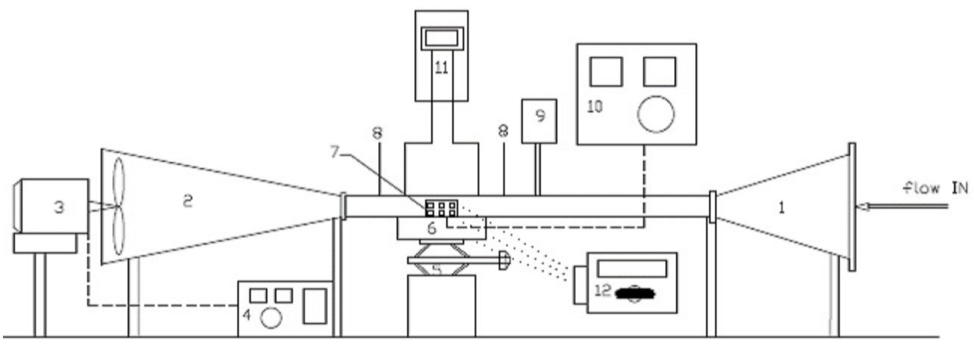


Fig. 1 Experimental setup: 1 - Effuser, 2 - Diffuser, 3 - Motor with fan, 4 - Panel Board (1), 5 - Screw jack, 6- Insulation Box, 7 - Fin arrays, 8 - Thermocouples, 9 - Anemometer, 10 - Panel Board (2), 11 – Pressure Transmitter, 12 - Temperature Indicator

TABLE I
GEOMETRY OF VARIOUS TYPES OF FIN ARRAYS STUDIED

Type of Pin-Fin Arrays	Number of Pin-fins	Diameter of Pin-fin (mm)	Height of fin (mm)	Size of slot
Solid Pin-fin	20	16	60	-
Pin-fin a with slots	20	12	60	6x20
	20	12	60	6x30
	20	12	60	6x40
	20	14	60	6x20
	20	14	60	6x30
	20	14	60	6x40
	20	16	60	6x20
	20	16	60	6x30
	20	16	60	6x40
Only base plate	-	-	194x120x10	-

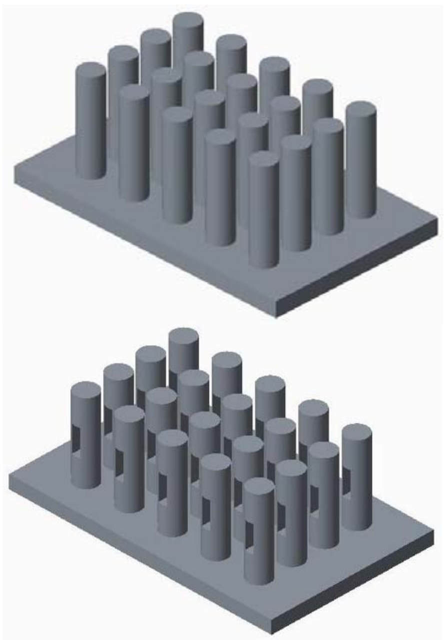


Fig. 2 Type of fin arrays used for experimentation

IV. TAGUCHI METHOD AND EXPERIMENTATION

A. Design of Experiment (Control Parameters and Orthogonal Arrays)

Design of Experiments (DOE) is a powerful statistical technique. Dr. Genechi Taguchi carried out significant research with DOE techniques. The DOE using Taguchi approach can economically satisfy the needs of problem solving and product/process design optimization projects. This method says the influence of individual factors on the performance characteristics of the product or process and determines which ones have less influence and which factor has more influences [23]. The Taguchi method gives the mean (signal) to the standard deviation (noise) ratio, as the performance indicator to calculate the characteristics of the product or process. It may be dependent on the particular type of performance characteristics, including larger-is-better, nominal-is-better, and smaller-is-better, In this study, average Nusselt number and friction factor are the performance characteristics and they are larger-is-better *(Z_L)* and smaller-is-better *(Z_S)* respectively.

$$z_s = -10\log\left(\frac{1}{n}\sum_{i=1}^{n} Y_i^2\right) \tag{1}$$

$$z_L = -10\log\left(\frac{1}{n}\sum_{i=1}^{n} \frac{1}{Y_i^2}\right) \tag{2}$$

where Y_i is the performance value of the i^{th} experiment and η is the number of experimental tests in trials. In this study, three control parameters, such as diameter of pin-fin, size of rectangular slot and Reynolds number are selected with their three levels as shown in Table II. The experimental design method Orthogonal array L_9 (3^3) is the most suitable method for the optimum working conditions being investigated. According to this method, an orthogonal array can provide an effective experimental performance with a minimum number of experimental trials. As per L_9 (3^3) Orthogonal array each experiment is repeated three times under the same condition. The Reynolds number is calculated by using (8). The Nusselt number and friction factor were calculated via the DOE for each combination of the control parameters by using (7) and (10) respectively, and S/N ratio were calculated using (1) and (2) respectively.

TABLE II
CONTROL PARAMETERS AND THEIR LEVELS

Symbol	Control parameter	Level I	Level II	Level III
A (D)	Diameter of Pin-fin	12 mm	14 mm	16 mm
B (S)	Size of Rectangular slot	6x20	6x30	6x40
C(Re)	Reynolds Number	7×10^3	1.5×10^4	3.8×10^4

TABLE III
ORTHOGONAL ARRAY TAGUCHI $L_9(3^3)$

Expt. Trials	A	B	C
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	3
6	2	3	1
7	3	1	3
8	3	2	1
9	3	3	2

B. Data Processing

By using the relation mentioned below the convective heat transfer rate from electrically heated test surface is calculated,

$$Q_N = Q_{(electrical)} - Q_{(conduction)} - Q_{(radiation)} \tag{3}$$

With the help of the electrical potential and current supplied to the surface, the electrical heat input is calculated. The radiation heat loss can be neglected. Conduction heat loss can be calculated by using (4):

$$Q_{\text{(conduction)}} = -K_b A_b \frac{dt_b}{dx_b} \tag{4}$$

The heat transfer from the test section by convection can be expressed as

$$Q_N = h_{av} A_S \left[T_s - \left(\frac{T_{out} + T_{in}}{2} \right) \right] \tag{5}$$

Hence average convective heat transfer coefficient h_{av} can be find out as

$$h_{av} = \frac{Q_N}{A_S \left(T_S - \frac{T_{out} + T_{in}}{2} \right)} \tag{6}$$

Total heat transfer area (A_s) = Open area of base + total surface area contribution from the fins.

The dimensionless groups are calculated as:

$$Nu = \frac{h_{av} D_h}{K_a} \tag{7}$$

$$Re = \frac{\rho_a V D_h}{\mu} \tag{8}$$

In every calculation, the values of thermo Physical properties of air are acquired at the bulk mean temperature i.e.,

$$T_{mean} = \frac{T_{out} + T_{in}}{2} \tag{9}$$

The friction factor is calculated using (12), which is based on the measured values of pressure drop (Δp), and measured by a pressure transmitter on the test section L = 260 mm.

$$f = \frac{\Delta p}{\left(\frac{L}{D_h} \right) \rho \frac{V^2}{2}} \tag{10}$$

V. Results and Discussions

A. Validation Test on Smooth Duct (without Fins)

In the literature, there have been different fin channel configuration and dimensions, therefore to make comparison, smooth channel without fin arrays was selected. Some experiments were carried out for a smooth duct (without any fins) attached to the base plate. Using experimental data obtained from these tests, the average Nusselt number (Nu_{sd}) and friction factor (f_{sd}) for smooth duct (without any fins) were correlated as a function of Reynolds number. The experimental results for smooth channel were compared with the correlation for turbulent flow. Experimental results for without fin arrays (smooth duct) as compared with correlations of earlier investigators are shown in Fig. 3. It shows the correlations of the Dittus-Bolter et al. [25] for turbulent flow in cylindrical ducts. The result of present study shows close to Dittus-Bolter et al. [25] with ± 20% deviation. The results of the present work were correlated with the average Nusselt number and friction factor respectively. Correlations are given below.

$$Nu_{sd} = 0.031\,\mathrm{Re}^{0.787} \tag{11}$$

$$f_{sd} = 0.058\,\mathrm{Re}^{-0.05} \tag{12}$$

The above equations for smooth duct and are valid for the experimental conditions of $7\times10^3 \le \mathrm{Re} \le 3.8\times10^4$, the regression coefficients (R^2) = 0.998 and 0.928 are for (15) and (16) respectively, $\Pr \cong 0.7$, heat input = 200 Watt.

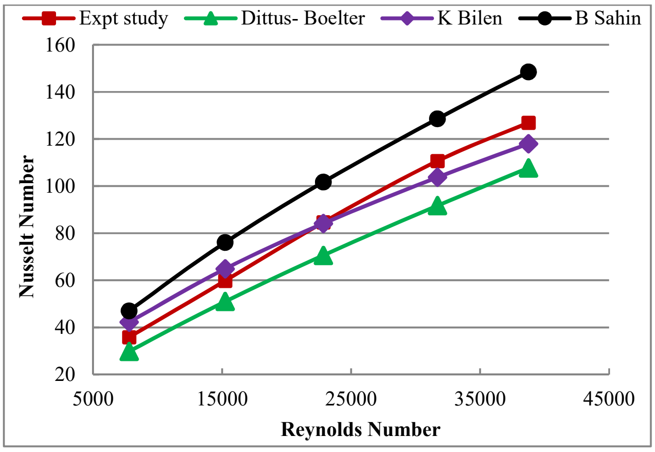
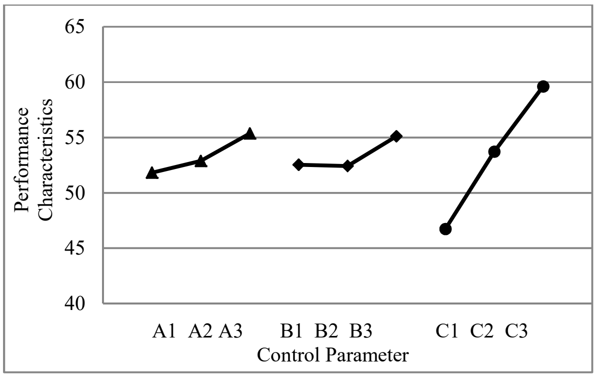


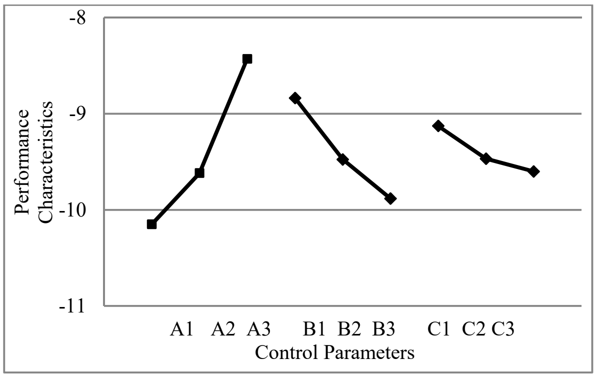
Fig. 3 Comparison of predicted and experimental values of Nusselt number for smooth duct

B. Optimization of Results

The data collected from the experiments were analyzed by using the ANOVA. The results obtained are shown in Figs. 4 (a), (b) and 5 (a), (b). The procedure can be explained with an example. Fig. 4 (a) shows the variation of performance characteristics with the control parameters. The first parameter is diameter of fin (D), second parameter is size of slot (W_s/H_s) ratio (S) and third parameter is Reynolds Number (Re). The diameter of first data point is 12 mm, which is level I for this parameter. The experiments corresponding to level I are found from Table III (Column X). It is seen in Table III that the experiments for which the column X is level I are the experiments with numbers 1, 2 and 3. The performance characteristics value of the first data point is the average of those obtained from the experiments with the numbers 1, 2, and 3. The experimental conditions for the second data point are the conditions of the experiments for which column A is level II and are the experiments with number 4, 5, and 6. The experimental conditions for the third data point are the conditions of the experiments for which column A is level III and are the experiments with number 7, 8, and 9. In Fig. 4 (a) the numerical value of the highest point in each plot shows the greatest value of that particular design parameter. Reynolds number is the most effective parameter to enhance the heat transfer rate, secondly diameter of fin, and thirdly size of the slot (W_s/H_s) ratio. A3B3C3 is the combination of design parameter. Corresponding value of each parameters for an average Nusselt number are: A3 i.e. Diameter of fin = 16 mm, B3 i.e. Size of slot = 6x40 and C3 i.e Reynolds number = 3.8 x 10^4. The percentage contribution of each design parameters are as shown in Fig. 5 (c). The results shows that the optimal design parameter combinations are A3B3C3 and corresponding percentage of each design parameters for highest average Nusselt number is Reynolds number = 86.59%, diameter of fin = 7.89% and size of slot = 5.02%. The greatest value of that particular design parameter and the most effective parameter with respective to the friction factor are found that diameter of pin-fin, size of slot and Reynolds number as shown in Fig. 4 (b). Figure shows that the friction factor increases by increasing diameter of pin-fin. The minimum friction factor was obtained at 12 mm diameter of pin-fin. The friction factor decreases with increase in Reynolds number. The design parameter combination for an average friction is A1B3C3 and the corresponding values of each parameter are: A1 i.e. diameter of fin = 12 mm, B3 i.e. size of slot = 6x40 and C3 i.e. Reynolds number = 3.8 x 10^4. Fig. 5 (d) shows the percentage contribution of each parameter, i.e. A, B and C. The results indicate that the optimal design parameter combination A1B3C3 and corresponding percentage of each parameters for lowest friction factor is Reynolds number = 5.02%, Diameter of fin = 70.65% and Size of slot = 24.31%.

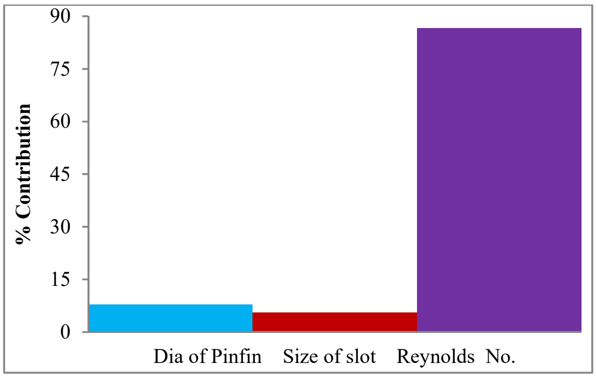


(a)

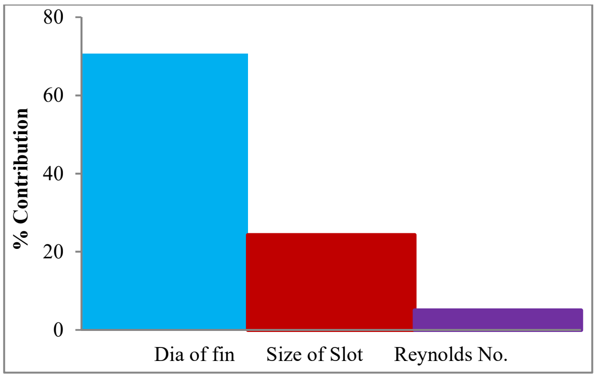


(b)

Fig. 4 The effect of each parameter on Nusselt number and friction factor



(a)



(b)

Fig. 5 (a) and (b) Percentage contribution of each parameter to Nusselt Number and friction factor

C. Confirmation Test

The final stage of the Taguchi method analysis is performing the confirmation experiments for examining the performance characteristics. It is required to validate the conclusion drawn during the analysis stage. This analysis is carried out for a level of significance of 5%, i.e. for a 95% level of confidence. The confidence interval (CI) for the predicated optimal values of average Nusselt number and friction factor are calculated by using (13):

$$CI=\sqrt{F_{aV_1V_2}V_{ep}\left(\frac{1}{\eta_{eff}}+\frac{1}{r}\right)} \tag{13}$$

where, $F_{aV_1V_2}$ is the F-ratio required for α = 0.05 with a confidence of 95%, $V_1 V_2$ are the number of degree of freedom of the mean and number of degree of freedom of error respectively, V_{ep} is the error of variance, r is the number of repetitions in confirmation experiments. η_{eff} shows the number of effective measured results:

$$\eta_{eff} = \frac{N}{1 + DOF_{opt}} \tag{14}$$

where N is the total number of trials. DOF_{opt} is the total degree of freedom. Using optimum levels arrived by the Taguchi method of optimization, a full range experiments was conducted. The optimum levels results for the A3B3C3 combinations, the values of average Nusselt number are Optimum predicated = 1191.02, Optimum CI = 1052.73 To 1329.31 and Real = 1245.4 and Optimum level results for the A1B3C3 combinations, the value of friction factor are Optimum predicated = 0.2785, Optimum CI = 0.1693 To 0.3996 and Real = 0.2979.

D. Heat Transfer Due to Slot

From the analysis, combination of the optimal design parameter is the A3B3C3. The most significant and efficient parameter is C3 second efficient parameter is A3 and the third one is B3 are enhancing the heat transfer. Therefore, the experiments were conducted for varying velocity, such as 1 m/s to 5 m/s and for various geometrical conditions such as sizes of the slots were 6x20, 6x30, 6x40 and diameter of pin-fin = 16 mm., keeping constant heat input 200 watt. Fig. 6 shows the variation of the Nusselt number with Reynolds number for solid pin-fin arrays as well as pin-fin with different size of slots. The average Nusselt number of all the slotted pin-fin arrays as well as solid pin-fin arrays increases with an increase in Reynolds number. In addition, it is observed that the maximum average Nusselt number is for pin-fin with 6x40 slot size for the 7000 to 38000 range of Reynolds number. The values of Nusselt number of slotted pin-fin arrays are greater than the solid pin-fin arrays. The value of the Nusselt number depends upon the projected area that reflects the effect of disturbance and turbulence of flow due to rectangular slot at the center of the pin-fins and as well as the variation in the surface area on the heat transfer. The result of this study shows close to [22] with ± 10% difference. Fig. 8 shows the variation of N_u/N_{usd} with Reynolds number for the various pin-fin with different size of slots such as pin-fin with 6x20 slot, pin-fin with 6x30 size of slot and pin-fin with 6x40 size of slot at constant C/H = 0.33 and S/H ratio i.e. and stream wise 0.53, span wise 0.46 respectively for all slotted and solid pin-fin arrays. The ratio of N_u/N_{usd} was based on projected area that will reflect the effect of difference in the surface area. As the surface area increases with increasing the slot of height, this leads to an increase in N_u/N_{usd}. It can be seen from Fig. 7 that the ratio of N_u/N_{usd} slightly decreases with the increase in Reynolds number but increases between Re = 15000 to Re = 32000, after that it is constant for all pin-fin arrays.

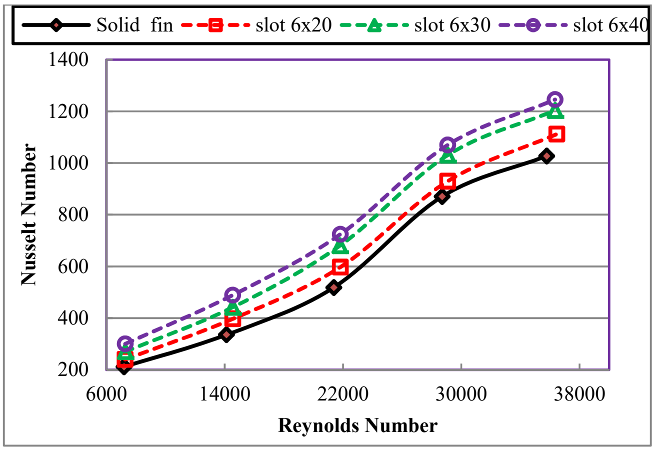


Fig. 6 Variation of Nusselt number (Nu) with Reynolds number (Re) for Pin-fin with different size of slots, D =16 mm, 200 watt

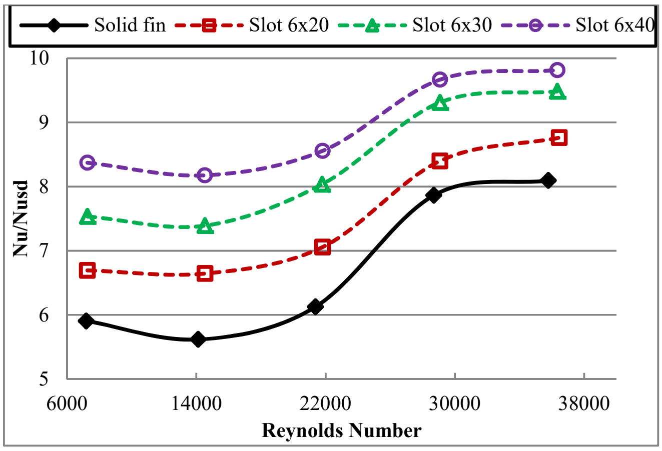


Fig. 7 Variation of N_u/N_{Nusd} with Reynolds number for Pin-fin with different size of slots, D = 16 mm, 200 watt

E. Pin-Fin Effectiveness

For evaluating the effectiveness of the pin-fin arrays, it is essential to determine the performance of the pin-fins with various sizes of slots, such as 6x20, 6x30, 6x40. The effectiveness of the fin is the ratio of the actual heat transfer rate from the fin (Q_f) having the average heat transfer coefficient and average surface temperature to the net heat transfer rate that would exist without the fin (Q_{sd}) at the base temperature.

$$\text{Effectiveness of fin}\,(\varepsilon_f) = \frac{Q_f}{Q_{sd}} \tag{15}$$

The variation of pin-fin effectiveness with Reynolds number is shown in Fig. 8. It shows that the effectiveness of the pin-fin decreases with an increase in Reynolds numbers. The effectiveness of the pin-fin increases with an increase in slot size, the value of effectiveness of all pin-fins arrays with slots is greater than solid pin-fin arrays. Also, it is observed that the maximum value of effectiveness is observed at the pin-fin with slot 6x40. In the present work, the value of pin-fin effectiveness is more than two, this means that the use of designed pin-fin with rectangular slot at the center is advantageous as far as heat transfer enhancement is concerned.

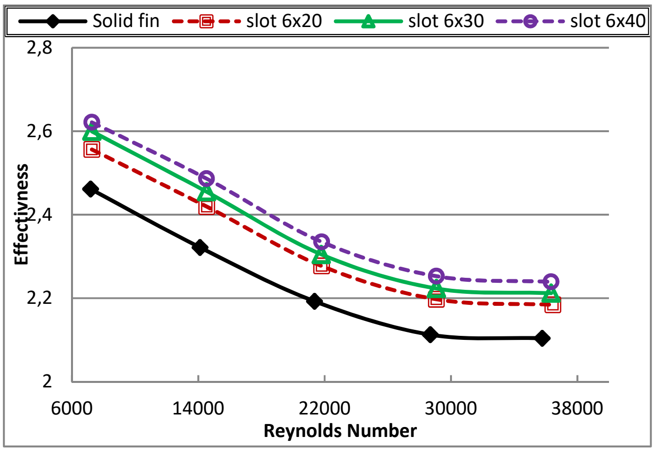


Fig. 8 Performance in effectiveness of pin-fin with Reynolds number, D = 16 mm for 200 watt

F. Percentage Improvement of Heat Transfer Coefficient over Solid Pin-Fin

The percentage improvement of pin-fin with slot over solid pin-fin (h_p) is defined as the percent of increase of heat transfer due to use of pin-fin with rectangular slot at the center relative to solid pin-fin without slot, and the relevant graphs are shown in Fig. 9. It is seen that percentage improvement is always positive and increases with increase in slot size. It is also seen that maximum percentage occurs for 15000 Reynolds number, after that it reduces with increase in Reynolds number.

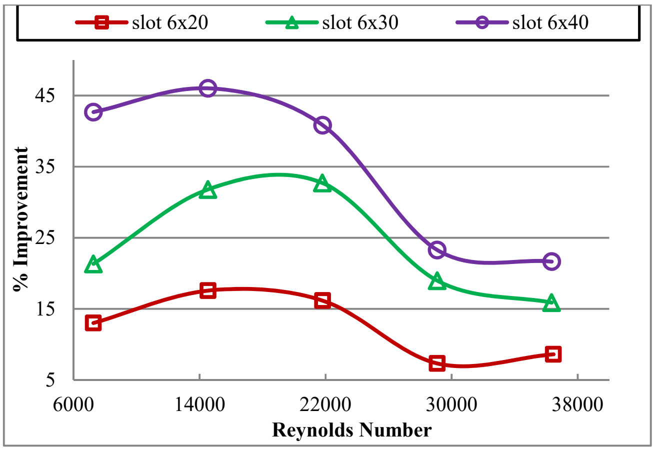


Fig. 9 Percentage improvement of perforated fin effectiveness relative to solid fin with Reynolds number, D = 16 mm

G.Friction Factor Due to Slot

The pressure drop over the test section inside the duct was measured under the heated conditions. Fig. 10 shows that the value of pressure difference increases with increase in value of Reynolds number. Maximum pressure difference occurs with solid pin-fin and minimum pressure difference occurs with fin of 6x40 slot. From experiment data measurements, the friction factor was calculated using (12). Fig. 11 shows the variations in the friction factor for pin-fin with slot as well as pin-fin without slot for Reynolds Number range 7000 to 38000, at constant C/H = 0.33 and S/H ratio i.e. and stream wise 0.53, span wise 0.46 respectively. Diameter of pin-fin = 16 mm and heat input = 200 watt. Friction factor slightly decreases with increase in Reynolds number. Friction factor decrease increases size of slot. It is seen that maximum friction factor occurs for solid pin-fin arrays.

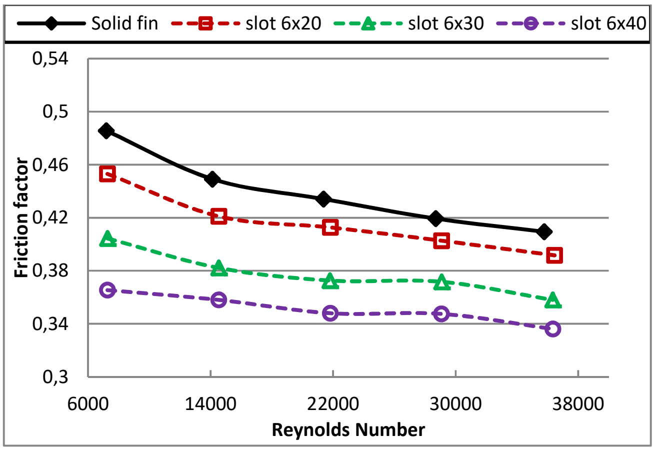


Fig. 10 Variation of Friction factor Vs Reynolds number

H.Experimental Uncertainties

The maximum air velocity was changed in a range from 1 m/s to 5 m/s for the smooth channel. The experiments were conducted at a power of 200 ± 0.6 W. By using the estimation method of [24], the maximum uncertainties of the investigated non-dimensional parameters are as follows: The average heat transfer coefficient = 7.38%, the average Nusselt number = 9.7%, the Reynolds number = 8.81% and the friction factor = 17.41%.

VI. Conclusions

In the present work, enhancement of heat transfer and the effect of the various design parameters on the heat transfer for circular Pin-fin arrays with longitudinal rectangular slots at center by forced convection heat transfer were investigated experimentally. The effects of various slots and Reynolds number on the heat transfer characteristics were determined. Optimum design parameters for the heat transfer (Average Nusselt number) were determined. The conclusions are given below:

- From the Taguchi experimental design method, the results found that the most important parameter affecting the heat transfer is the Reynolds number, secondly diameter of the pin-fin and thirdly slot size. The maximum heat transfer rate is observed at 38000 Reynolds number, 16 mm diameter of the fin and 6x40 slot size. Therefore, heat transfer can be successfully improved by controlling these design parameters.

- The average Nusselt numbers of pin-fin arrays with slots as well as solid fin arrays increase with an increase in Reynolds number. The value of average Nusselt numbers of pin-fin with slot is higher than the pin-fin without slot which means enhancement of the heat transfer due to pin-fin with slot at the center. It is due to most of surface area contact with free air and turbulence of fluid flow over the test section.
- The effectiveness (ε_f) of pin-fins with slot is larger than pin-fin without slot arrays. The maximum effectiveness is observed for the pin-fin slot size 6x40 and minimum for pin-fin without slot arrays. Percentage enhancement of heat transfer coefficient due to rectangular slots at the center of pin-fins over solid pin-fins for 16 mm diameter of pin-fin having slot sizes 6x20, 6x30 and 6x40 are found to be 13.50%, 26.20% 38.20% respectively.
- Due to slot there will be weight reduction of fins and at the same time reduced power consumption; consequently, reduction of cost of material as far as manufacturing of equipment is concerned. Hence it may be utilized for many industrial applications.

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