Heat Transfer Analysis of Cylindrical Perforated Fins in Staggered Arrangement

Amol B. Dhumne, Hemant S. Farkade

Abstract- The present paper gives the experimental analysis of on heat transfer enhancement and the corresponding pressure drop over a flat surface equipped with cylindrical cross-sectional perforated pin fins in a rectangular channel. The channel had a cross-sectional area of 250-100 mm2. The experiments covered the following range: Reynolds number 13,500–42,000, the clearance ratio (C/H) 0, 0.33 and 1, the inter-fin spacing ratio (Sy/D) 1.208, 1.524, 1.944 and 3.417. Nusselt number and Reynolds number were considered as performance parameters. Correlation equations were developed for the heat transfer, friction factor and enhancement efficiency. The experimental implementation shows that the use of the cylindrical perforated pin fins leads to heat transfer enhancement than the solid cylindrical fins. Enhancement efficiencies vary depending on the clearance ratio and inter-fin spacing ratio. Both lower clearance ratio and lower inter-fin spacing ratio and comparatively lower Reynolds numbers are suggested for higher thermal performance.

Keywords: Heat Transfer, Cylindrical perforated Fins, Staggered Arrangement

I. INTRODUCTION

Extended Surface (Fins) is used in a large number of applications to increase the heat transfer from surfaces. Typically, the fin material has a high thermal conductivity. The fin is exposed to a flowing fluid, which cools or heats it, with the high thermal conductivity allowing increased heat being conducted from the wall through the fin. Fins are used to enhance convective heat transfer in a wide range of engineering applications, and offer practical means for achieving a large total heat transfer surface area without the use of an excessive amount of primary surface area. Fins are commonly applied for heat management in electrical appliances such as computer power supplies or substation transformers. Other applications include IC engine cooling, such as Fins in a car radiator.

Fins are widely used in the trailing edges of gas-turbine blades, in electronic cooling and in the aerospace industry. The relative fin height (H/d) affects the heat transfer of pin-fins, and other affecting factors include the velocity of fluid flow, the thermal properties of the fluid, the cross-sectional shape of the pin-fins like perforation, the relative inter-fin pitch, the arrangement of the pin-fins like in-line, staggered arrangement and others. In existing studies, the parameters affecting the heat transfer like relative fin height (C/H), the velocity of fluid flow, the cross-sectional shape of the pin-fins like perforation, the relative inter-fin spacing and the arrangement of the pin-fins like staggered arrangement have been investigated.

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Nomenclature	
A	Heat transfer area
Q	Heat transfer rate
T	Steady-state temperature
U	Mean velocity of the air
V	Voltage
W	Width of the base plate and the duct
ΔP	Pressure difference
ν	Kinematic viscosity of air
D	Diameter of the pins
Dh	Hydraulic diameter of the duct
d	Diameter of the perforation
f	Friction factor
h	Heat transfer coefficient
H	Height of the fins
I	Current
L	Length of base plate
Lt	Length of the test section
k	Conductivity of air
Np	Number of the pins
Nu	Nusselt number
R	Resistant of heater element
Re	Reynolds number
a	Semi major axis of ellipse,
b	Semi minor axis of ellipse
T_s	Surface Temperature
h_{av}	Average heat transfer coefficient
T_o	Surrounding Temperature
L	Length of base plate
r	Radius of Perorated hole
ρ	Density of air
Tm	Mean Temperature
Nu_s	Nusselt number of smooth surface
Nu_T	Nusselt number of Total Area
Nu_P	Nusselt number of Projected Area
h_a	Convective heat transfer Coefficient with fins
h_s	Convective heat transfer Coefficient without fins
Pr	Prandtl number
As	Heat Transfer Surface Area

There have been many investigation regarding heat transfer and pressure drop of channels with pin fins has been done by the following researchers considering the different factors for heat transfer

Bayram Sahin, Alparslan Demir studied the heat transfer enhancement and the corresponding pressure drop over a flat surface equipped with square cross-sectional perforated pin fins in a rectangular channel. The experimental results showed that the use of the square pin fins may lead to heat transfer enhancement. Enhancement efficiencies varied between 1.1 and 1.9 depending on the clearance ratio and inter-fin spacing ratio.

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Both lower clearance ratio and lower inter-fin spacing ratio and comparatively lower Reynolds numbers are suggested for higher thermal performance. In this study, the overall 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 square cross-sectional perforated pin fins were investigated experimentally.

R. Karthikeyan , R. Rathnasamy studied the heat transfer and friction characteristics of convective heat transfer through a rectangular channel with cylindrical and square cross-section pin-fins attached over a rectangular duralumin flat surface. The pin-fins were arranged in in-line and a staggered manner. Various clearance ratios (C/H=0.0, 0.5&1.0) and inter-fin distance ratios (Sy/d and Sx/d) were used. The experimental results showed that the use of square cross-section pin-fins may lead to an advantage on the basis of heat transfer enhancement. For higher thermal performance, lower inter fin distance ratio and clearance ratio and comparatively lower Reynolds numbers should be preferred for in-line and staggered arrangement. the staggered pin-fin array significantly enhanced heat transfer as a result turbulence at the expense of higher pressure drop in the wind tunnel. Square pin-fin array performance is slightly higher than the cylindrical array with the penalty of pressure drop

Tzer-Ming Jeng, Sheng-Chung Tzeng studied the pressure drop and heat transfer of a square pin-fin array in a rectangular channel. The variable parameters are the relative longitudinal pitch (XL = 1.5, 2, 2.8), the relative transverse pitch (XT = 1.5, 2, 2.8) and the arrangement (in-line or staggered). The result shows that The in-line square pin-fin array has smaller pressure drop than the in-line circular pin-fin array at high XT (XT = 2.0 or 2.8) but an equivalent (or even slightly higher) pressure drop at low XT (such as XT = 1.5). Additionally, the staggered square pin-fin array has the largest pressure drop of the three pin fin arrays (in-line circular pin-fins, in-line square pin-fins and staggered square pin-fins). Most in-line square pin-fin arrays have poorer heat transfer than an in-line circular pin-fin array, but a few, as when XL = 2.8, exhibit excellent heat transfer at high Reynolds number. For instance, when XL = 2.8, XT = 1.5

Giovanni Tanda studied Heat transfer and pressure drop experiments were performed for a rectangular channel equipped with arrays of diamond-shaped elements. Both in-line and staggered fin arrays were considered, for values of the longitudinal and transverse spacing’s, relative to the diamond side, from 4 to 8 and from 4 to 8.5, respectively. The height-to-side ratio of the diamonds was 4. Thermal performance comparisons with data for a rectangular channel without fins showed that the presence of the diamond-shaped elements enhanced heat transfer by a factor of up to 4.4 for equal mass flow rate and by a factor of up to 1.65 for equal pumping power

G.J.Vanfossen and B.A.Brigham [1] studied the heat transfer by short pin-fins in staggered arrangements. According to their results, longer pin-fins (H/d = 4) transfer more heat than shorter pin-fins (H/d = ½ and 2) and the array-averaged heat transfer with eight rows of pin-fins slightly exceeds that with only four rows. Their results also established that the average heat transfer coefficient on the pin surface is around 35% larger than that on the end walls.

Metzger et al. [2] investigated the effects of pin-fin shape and array orientation on the heat transfer and the pressure loss in pin-fin arrays. According to their results, the use of cylindrical pin-fins with an array orientation between staggered and in-line can sometimes promote the heat transfer, while substantially reducing pressure. When oblong pin-fins are used, heat transfer increases of around 20% over the circular pin-fins were measured, but these increases were offset by increases in the pressure loss of around 100%. Their estimate indicated that the pin-fin surface coefficients were approximately double the end wall values.

R.F. Babus’Haq et al. [4] reported that the optimal ratio of the inter-fin pitch to the pin fin diameter in the transverse direction was 2.04 for all pin-fin systems. However, the optimal ratios in the longitudinal direction were 1.63, 1.71 and 1.95 for polytetrafluoroethene pin-fins, mild-steel pin-fins and duralumin pin-fins respectively.

O.N. Sara et al.[5] reported another way to improve heat transfer rate is to employ attachments with (i) perforations, (ii) a certain degree of porosity or (iii) slots which allow the flow to go through the blocks. In the case of perforated attachments, the improvement in the flow (thus the enhancement in the heat transfer) is brought about by the multiple jet-like flows through the perforations Thus, the aim of this study is also to determine heat transfer and friction factor characteristics of the perforated staggered cylindrical fins. The heat transfer enhancement is achieved at the expense of the increased pressure drop. For many practical applications it may thus be necessary to determine the economic benefit for the heat transfer

II. EXPERIMENTATION SET-UP

The experimental set-up consisting of the following parts
A. Main Duct (Tunnel)
B. Heater Unit
C. Base Plate
D. Data Unit

A. Main Duct (Tunnel)

Tunnel constructed of wood of 20 mm thickness, had an internal cross-section of 250 mm width and 100 mm The total length of the channel is 1000 mm. It will be operated in force draught mode by the blower of 0.5 H.P., 0 to 13000 rpm, 220W, 1.8Kg, variable speed 1 to 6 and it is operated at 45cm from the tunnel i.e. at convergent part of tunnel and positioned horizontally. It has a convergent and divergent section at both end having the inclination of 30°. A Matrix anemometer is mounted in a tunnel to measure the mean inlet velocities of the air flow entering to the test section the range of this anemometer is 0 to 40m/sec The Reynolds number range used in this experiment was 13,500–42,000, which is based on the hydraulic diameter of the channel over the test section (Dh=142.85mm) and the average velocity (U)

B. Heater Unit

Heater Unit (test section) has a cross-section of 250 mm x 250 mm square; the heating unit mainly consisted of an electrical heater placed between two M.S. Plate having the same dimension of base plate, a two firebrick of 250x 220 mm. Dimensions of the electrical heater placed on the firebrick are 250 mm x 250 mm.

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The heater output has a power of 200 W at 220V and a current of 10 amp. Whole assembly is mounted in a square wooden box of dimension 270 x 270 mm

C. Base Plate

It consist of square plate at base having the dimension 250mm x 250 mm, thickness is 6mm and The pin fins and base plate made of the same material i.e. Aluminum because of the considerations of conductivity, machinability and cost. Total 13 aluminum base plate is made among this one is of without fins, six are of perforated fins and six are of solid fins. The fins have a circular cross section of 15 mm x15 mm and are attached on the upper surface of the base plate as shown in Fig. 1. Circular pin fins with different lengths, corresponding to C/H (Clearance ratio) values of 0, 0.333 and 1, are perforated at the 17 mm from bottom tip of those by an 8 mm diameter drill bit. The pin fins are fixed uniformly on the base plate with a constant spacing between the spanwise directions of 18.125 mm, with different spacing between the pin fins in the streamwise direction. The spacing ratios of the pin fins in the streamwise direction (Sy/D) were 1.208, 1.524, 1.944 and 3.417 mm, giving different numbers of the pin fins on the base plate. It is well-known fact that if the inter-fin spacing in the spanwise direction decreases, the flow blockage will increase and thus, pressure drop along to tested heat exchanger will increase. Because the aim of the study is to determine inter-fin spacing in streamwise direction, the spacings in the spanwise direction will not be considered in this study.). The temperature of the base plate is measured RTD Sensors which can sense the temperature from 0°c to 450°c and it is screwed into groove in the base plate the readings of the RTD Sensors will be shown on data unit

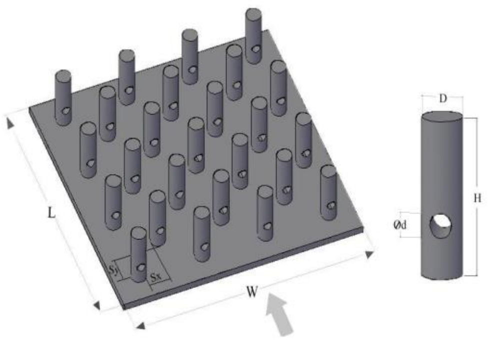

Fig.1. Base Plate with fins in staggered arrangement

Fig. 2 Actual base Plate

Table I. Details of Dimensions and Number of Base plate and Fins

Sr. No.	Particular	Size	Quantity
1	Base Plate (Without Fins)	250mm x250mm	1
2	Base Plate (With Fins)	250mm x 250mm	12
3	Perforated Fins	100mm	100
4	Perforated Fins	75mm	25
5	Perforated Fins	50mm	25
6	Solid fins	100mm	100
7	Solid fins	75mm	25
8	Solid fins	50mm	25

D. Data Unit

It consist of various indicating devices which indicates the reading taken by the various component like RTD sensors and Anemometer There are three Temperature indicator which shows reading taken by the RTD Sensors in the range 0°c to 450 °c among this, two gives inlet and out temperature of air and one gives temperature of base plate There is one temperature contractor which can maintain the temperature of base plate it will not allow to exceed the temperature of base plate above desired values i.e. 100°c. Inlet flow rate of air is indicated by velocity indicator using Anemometer. MCB Hegger switches are mounted to cut off the power supply in case any short circuit

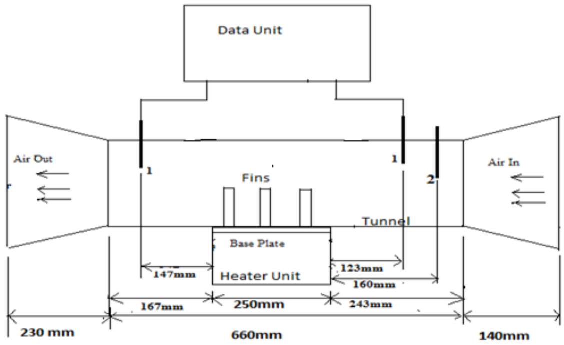

Fig.3.Experimental Set-up (1) RTD Sensor (2) Anemometer

Fig. 4: Actual Experimental Set up

III. EXPERIMENTATION METHODOLOGY

Governing Equation:

The convective heat transfer rate 'Q' convection from electrically heated test surface is calculated by using

$$Q_{\text{conv.}} = Q_{\text{elect.}} - Q_{\text{cond.}} - Q_{\text{rad.}} \quad (1)$$

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Where '*Q*' indicates the heat transfer rate in which subscripts conv, elect, cond and rad denote convection, electrical, conduction and radiation, respectively. The electrical heat input is calculated from the electrical potential and current supplied to the surface

$$Q_{\text{elect.}} = I^2 \times R \qquad (2)$$

Where 'I' is current flowing through the heater and 'R' is the resistance

In similar studies, investigators reported that total radiative heat loss from a similar test surface would be about 0.5% of the total electrical heat input. The conductive heat losses through the sidewalls can be neglected in comparison to those through the bottom surface of the test section. Using these findings, together with the fact that the two sides walls and the top wall of the test section are well insulated and readings of the thermocouple placed at the inlet of tunnel should be nearly equal to ambient temperature, one could assume with some confidence that the last two terms of Eq. (1) may be ignored.

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

$$Q_{conv.} = h_{av} A_s \left[T_s - \left(\frac{T_{out}+T_{in}}{2}\right)\right] \qquad (3)$$

Hence, the average convective heat transfer coefficient have could be deduced via

$$h_{av} = \frac{Q_{conv.}}{A_s\left[T_s - \left(\frac{T_{out}+T_{in}}{2}\right)\right]} \qquad (4)$$

Either the projected or the total area of the test surface can be taken as the heat transfer surface area in the calculations. The total area is equal to the sum of the projected area and surface area contribution from the pin fins. These two areas can be related to each other by

Total area = Projected area + Total surface area contribution from the blocks

$$A_s = WL + [\pi DH - 2\pi ab]N_p + [(2\pi r^2 + 2\pi rD) - 2\pi ab]N_p$$

(for Perforated Fin) (5)

$$A_s = WL + [\pi DH]N_p \qquad \text{(for solid fins)} \qquad (6)$$

where 'W' is the width of the base plate, 'L' its length, 'Np' is the number of fins, 'H' the height of fin and 'D' is the diameter of the fins, 'a' is semi major axis of ellipse, 'b' is semi minor axis of ellipse. "a" and "b" are calculated by the development of surfaces and its come "a"=4.8mm and "b"=4mm and r =4mm

The dimensionless groups are calculated as follows:

$$N_u = \frac{h_{av} D_h}{k} \qquad (7)$$

Nu based on the projected area will reflect the effect of the variation in the surface area as well as that of the disturbances in the flow due to fins on the heat transfer. But Nu based on the total area will reflect the effect of the flow disturbances only. In this study, heat transfer enhancement Characteristics were determined by using Nu-based projected area

$$\mathcal{F} = \frac{\Delta P}{\left[\left(\frac{L_t}{D_h}\right)\left(\rho\frac{U^2}{2}\right)\right]} \qquad (8)$$

$$R_e = \frac{D_h U}{\nu} \qquad (9)$$

In all calculations, the values of thermo physical properties of air were obtained at the bulk mean temperature, which is

$$T_m = (T_{in} + T_{out})/2 \qquad (10)$$

A. *Evaluation of Heat Transfer*

In order to have a basis for the evaluation of the effects of the fins, the average Nusselt number (Nus) for the smooth surface (without pin fins) will be correlated as function of Re and Pr as follows:

$$Nu_s = 0.077 Re^{0.716} Pr^{1/3} \qquad (11)$$

The Nusselt number based on projected area was related to the Reynolds number, clearance ratio (C/H), inter-fin distance ratio (Sy/D) and Prandtl number and is given by the following relation

$$Nu_P = 45.99 Re^{0.396} (1 + C/H)^{-0.608} \left(S_y/D\right)^{-0.522} Pr^{1/3} \qquad (12)$$

$$Nu_T = 6.67 Re^{0.401} (1 + C/H)^{0.0811} \left(S_y/D\right)^{0.06} Pr^{1/3} \qquad (13)$$

This equations are valid for the experimental conditions of $13{,}500 \le Re \ge 42{,}000$, $1.208 < Sy/D < 3.417$, $0 \le C/H \le 1$ and Pr = 0.7 by using this equation the Nu/Nus and Re will be determine for perforated fins and solid fins for different C/H ratio i.e. C/H=0, C/H=0.333, C/H=1 at constant Sy/D=1.208 and for different Sy/D ratios i.e. Sy/D=1.208, Sy/D=1.524, Sy/D=1.944, Sy/D=3.417 at constant C/H=0. The same will be find out for solid fins and the comparative graph between Nu/Nus and Re for perforated fins and solid fins is plot

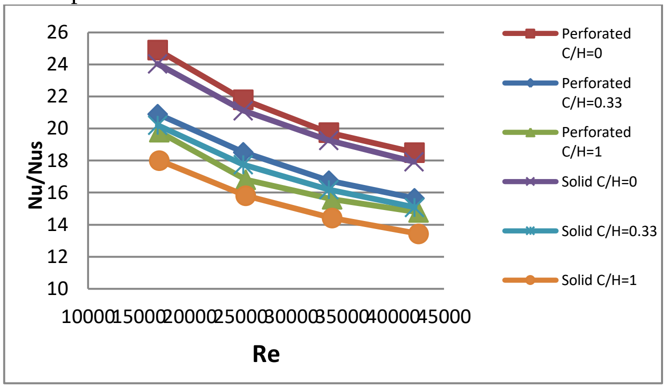

Fig.5. Variation of Nu/Nus based on Projected area with Reynolds No. for various clearance ratio at constant Sy/D=1.208

Fig. 5 shows the Nu/Nus based on the projected area, as a function of the duct Reynolds number for the three different pin heights, namely C/H = 1, 0.333 and 0 at Sy/D = 1.208. It is seen from this Fig. 5 that Nu/Nus increases with decreasing C/H. A decrease in C/H means that the height of the fins increases. As the surface area increases with increasing height of the fins, this leads to an increase in Nu/Nus. The Nusselt number that is based on the projected area will reflect the effect of the variation in the surface area as well as that of the disturbance in the flow due to pin fins on the heat transfer. Longer fins can also increase the turbulence of the flow in the channel, resulting in an increase in the heat transfer

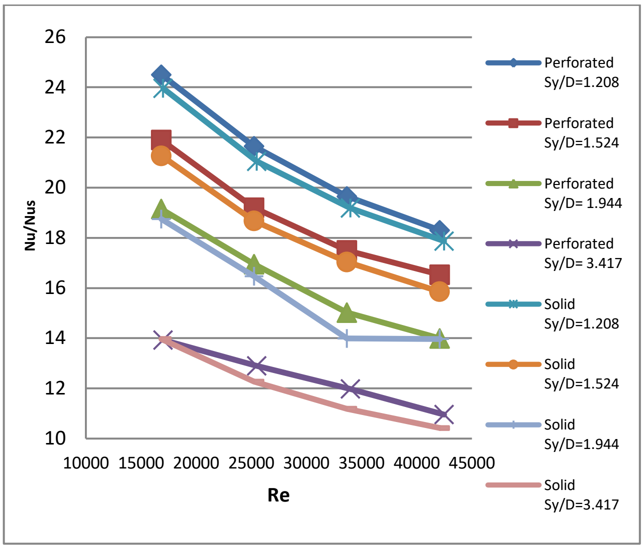

Fig.6. Variation of Nu/Nus based on Projected area with Reynolds No. for various inter-fin spacing at constant C/H =0

Fig. 6 shows the behavior of the Nu/Nus as a function of the duct Reynolds number and inter-fin distance ratios (Sy/D) for a constant clearance ratio (C/H) of 0.0. Decreasing Sy/D means that the fin numbers on the base plate increases. It is seen from this figure that since the number of fins increases with decreasing Sy/D, which also means an increase in the total heat transfer area, the heat transfer rate (Nu/Nus) increases. Perforated fins have higher Nusselt number values than solid fins.

B. Evaluation of Friction Factor

The pressure drops in the tunnel without fins is so small that they could not be measured by the pressure transducer. This resulted from smaller length of the test section and smaller roughness of the duct. The experimental pressure drops over the test section in the finned duct will be measured under the heated flow conditions. These measurements will be converted to the friction factor '$\mathcal{F}$' Using the experimental results, f was correlated as a function of the duct Reynolds number, Re, and geometrical parameters. The resulting equation is

$$\mathcal{F} = 2.4Re^{-0.0836}(1 + C/H)^{-0.0836}(S_y/D)^{-0.0814} \qquad (14)$$

This equation is valid for 13,500 < Re < 42,000, 1.208 < Sy/D < 3.417, 0 < C/H < 1 By using the above equation the variations in the friction factor 'f' for different clearance ratios (C/H) i.e. C/H=0, C/H=0.333, C/H=1 at constant Sy/D = 1.208 and for different inter-fin space ratios (Sy/D) i.e. Sy/D=1.208, Sy/D=1.524, Sy/D=1.944, Sy/D=3.417 at constant C/H = 0 is determined. The same experiment will be carried out for the solid fins (fins without perforation) and friction factor will be find out for the same and the comparative graph between f and Re for perforated fins and solid fins is plot

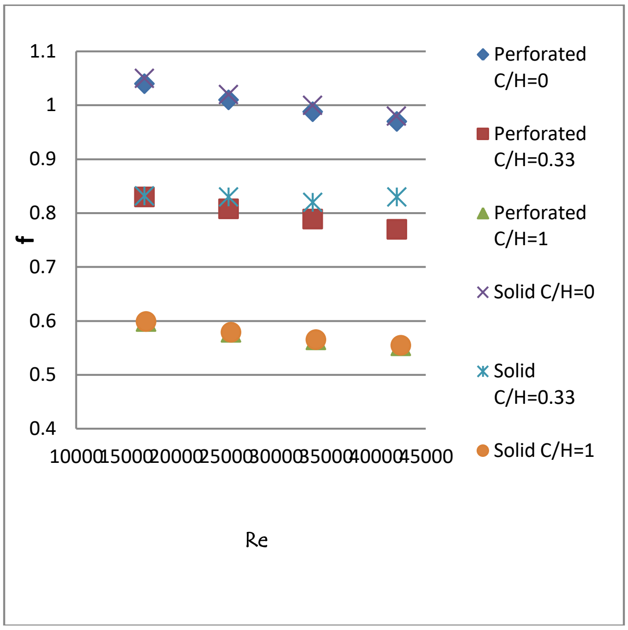

Fig.7. Variation of friction factor with Reynold No.for various clearance ratio at Sy/D=1.208

It can be seen from Fig. 7 that the friction factor increases with decreasing C/H. Because the pin height increases with decreasing C/H, the by-pass area over the fin tips decreases. Thus, resistance against to the flow increases.

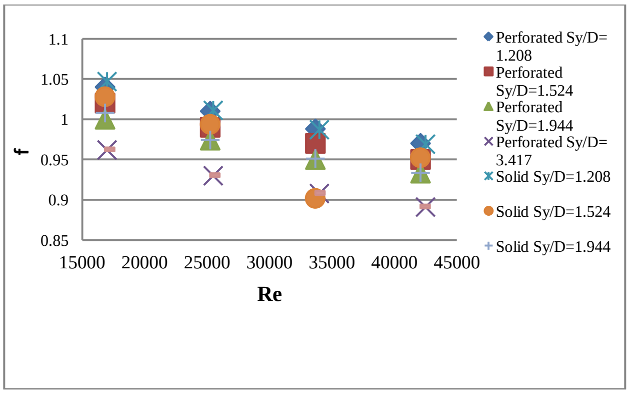

Fig.8. Variation of friction factor with Reynold No. for various inter-fin spacing at C/H=0

The other notable result is seen from Fig. 8 for the friction factor. The friction factor values are almost independent of the Reynolds number and each C/H value. It is emphasized in another optimization study for a finned heat exchanger that interestingly, streamwise distance between fins is more effective parameter on the friction factor than spanwise distances. On the other hand, as the resistance to the flow will be smaller due to the perforations, friction factor is lower for the perforated fins than the solid fins.

C. Enhancement of Efficiency

The enhancement efficiency of the heat transfer technique can be expressed as

$$\eta = \frac{h_a}{h_s} \qquad (15)$$

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where 'ha' and 'hs' are the convective heat transfer coefficient with and without pin fins, respectively the following equation can be written for the heat transfer efficiency for the pin fins according to total heat transfer surface area

$$\eta = \frac{h_a}{h_s} = 51.09Re_a^{-0.358}(1 + C/H)^{0.1028}(S_y/D)^{0.0812} \qquad (16)$$

By using this equation the effect of the inter-fin distance on heat transfers enhancement efficiency for different inter-fin space ratios (Sy/D) i.e. Sy/D=1.208, Sy/D=1.524, Sy/D=1.944, Sy/D=3.417 at constant C/H = 0 will be determine and graph is plot similarly the effect of the clearance ratio on enhancement efficiency for different clearance ratio (C/H)) i.e. C/H=0, C/H=0.333, C/H=1 at constant Sy/D = 1.208 will be determine and graph is plot

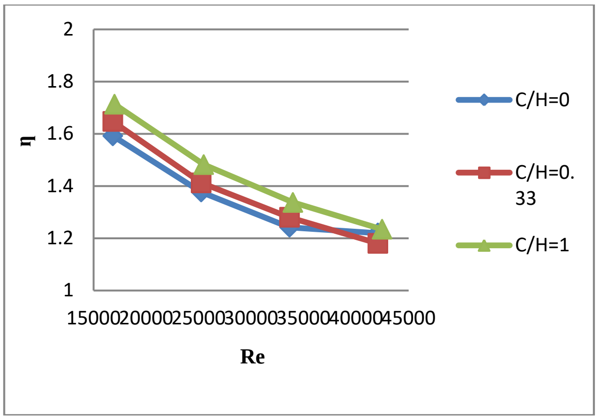

Fig.9 Variation of enhancement efficiency with Reynolds number for various clearance ratios at constant Sy/D = 1.208

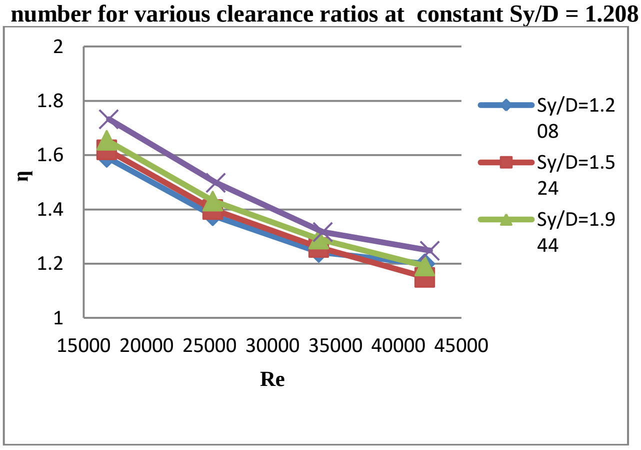

Fig.10. Variation of enhancement efficiency with Reynolds number for various inter-fin spacing ratios at constant C/H = 0.0

The heat transfer enhancement efficiencies were plotted in Fig.9 and Fig. 10 Fig. (9) shows the effect of the clearance ratio on enhancement efficiency (η) while fig.10 show the effect of the inter-fin distance ratio on enhancement efficiency(η). For a net energy gain, the value of the (η) must be greater than unity. In other words, for an effective heat transfer enhancement technique, it must have values greater than unity. From Figs. (9) and (10), it is apparent that as the Reynolds number increases, the enhancement efficiency decreases for both the inter-fin spacing ratio and clearance ratio. Figs. (9) and (10) show that the heat transfer enhancement efficiency decreases with increasing Sy/D and C/H. In other words:

(a) The heat transfer enhancement efficiencies are higher than unity for all investigated conditions. This means

that the use of pin fins leads to an advantage on the basis of heat transfer enhancement. (b) Higher numbers of pin fins and longer pin fins have better performance. In other words, for higher thermal performance, a lower inter-fin distance ratio and clearance ratio should be preferred. (c) At a lower Reynolds number, the channels with pin fin arrays give higher performance than those at a higher Reynolds number.

IV. CONCLUSION

In this study, the overall 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 square cross-sectional perforated pin fins were investigated experimentally. The effects of the flow and geometrical parameters on the heat transfer and friction characteristics were determined

a) The average Nusselt number calculated on the basis of projected area increased with decreasing clearance ratio and inter-fin spacing ratio.
b) The friction factor increased with decreasing clearance ratio and inter-fin distance ratio.
c) The most important parameters affecting the heat transfer are the Reynolds number, fin spaces (pitch) and fin height. Heat transfer can be successfully improved by controlling these parameters. The maximum heat transfer rate was observed at 42,000 Reynolds number, 3.417 Sy/D and 50 mm fin height.
d) The most effective parameter on the friction factor was found to be fin height. The minimum friction factor was observed at 50 mm fin height, 42,000 Reynolds number and 3.417 pitch.

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