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Experimental Investigation of Heat Transfer Enhancement in Solid Cylindrical & Perforated Cylindrical fins in Staggered & in Inline Arrangement

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***Abstract*— This paper investigates the heat transfer enhancement in solid cylindrical & perforated cylindrical fins in inline and staggered arrangement in 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-finspacing ratio (Sy/D) 1.944 and 3.417. Nusselt number and Reynolds number were considered as performance parameters. Correlation equations will be developed for the heat transfer and friction factor. Computational Fluid Dynamics analysis is done by using ANSIS FLUENT14.5 software. The Numerical and computational analysis shows that the use of the cylindrical perforated pin fins leads to heat transfer enhancement than the solid cylindrical fins. Heat transfer Enhancement varies depending on the clearance ratio and inter-fin spacing ratio. Validation of Numerical and Computational Analysis will be done.**

Keywords : - Heat Transfer, Cylindrical perforated Fins, Solid cylindrical fins, Staggered Arrangement, Nusselts number, Reynolds number

I. INTRODUCTION

Tzer-Ming jeng, Sheng-Chung Tzeng [1]

Studied the pressure drop and heat transfer of a square pin-fin array in a rectangular Channel. The variable parameters are relative longitudinal pitch (XL=1.5,2, 2.8)the relative pitch (XT=1.5, 2, 2.8). The result shows that the in-line square pin-fin array has smaller pressure drop then 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 The result shows that the in-line square pin-fin array has smaller pressure drop then 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 array (inline circular pin-fins, in-line square pin-fins and staggered squared pin- fins) Most in-line square pin-fin arrays have poorer heat transfer than an inline circular pin-fin arrays, but a few, as when XL=2.8 exhibit excellent heat transfer at high Reynolds number for instant, when XL=2.8, XT=1.5

Bayram Sahin, Alparslan Demir [2] studied the heat transfer enhancement and corresponding pressure drop over a flat surface equipped with square cross section a perforated pin fins in a rectangular channel.

The experimental results show that the used of 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. 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 [3]. 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 dualism 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 ratio (Sy/d and Sx//d) were used the experimental result shows that the used of 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 number should be proffered 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 wind tunnel. The square pin-fin array performance is slightly higher than the cylindrical array with the penalty of pressure drop.

G. J. Vanfossen and B.A. Brigham [4] Studied the heat transfer by short pin-fins in staggered arrangements. According to their results, longer pin-fin in staggered arrangement. According to their results longer pin-fin (H/d=4) transfer more heat than shorter fin (H/d=0.5 and 2) and the array- averaged heat transfer with eight rows of pin-fins slightly exceeds that with only four rows. Their result also established that the average heat transfer coefficient on the fin surfaces is around 35% larger than that on the wall end.

Giovanni Tanda [5] 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 spacings, 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.

D.E. Metzger and C.S. Fan and S.W. Haley [6]

Investigated the effect 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 fin 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 at around 20% over the circular pin-fins were measured, but these increases were offset by increases in the pressure loss around 100%. Their estimate indicated that the pin-fin surface coefficient were approximately double the end wall value.

O.N.Sara et al.[7]

Reported another way to improve heat transfer rate is to employ attachments with perforations, a certain degree of porosity, slot which allow the flow to go through the blocks. In case of perforated attachments, the improvement in the flow is brought about by the multiple jet-like flow through the perforation thus the aim of this study is also to determine heat transfer and friction factor characteristics of perforated staggered cylindrical fins. The heat transfer enhancement is achieved at the expense of of increased pressure drop.

M.R Shaeri, M.Yaghoubi [8]

Investigated analysis of turbulent convection heat transfer from an array of perforated fins, numerical investigation is made for three dimensional fluid flow and convective heat transfer from an array solid and perforated fin that are mounted on flat plate, sometimes an image may contain text embedded on to it. Detecting and recognizing these characters can be very important, and removing these is important in the context of removing indirect advertisements, and for aesthetic reasons. perforation such as small channels with square cross section are arranged stream wise along the fins length and their number varied from 1 to 3 flow and heat transfer characteristics are presented by Reynolds number from 2x104 and 4x104 based on the fin length of previous investigation and good agreement were observed result show that fins with longitudinal pores have unmark able heat transfer enhancement in addition to the consider reduction in weight by comparison with solid fins.

Jnana Ranjan Senapati and Sukanta Kumar Dash and Subhranshu Roy [9] Numerical convection heat transfer from an annular finned horizontal cylinder with different eccentricity has been studied numerically in the work. This paper present numerical investigation is able to capture a complete picture of natural convection over a horizontal cylinder with fins of different eccentricity. In present conjugate heat transfer study of annular finned horizontal cylinder annular disc fins can be design for the purpose to increase the heat transfer rate significantly with marginal loss in the heat transfer with respect to the concentric annular fins.

Tamir K.Ibrahim and Marwah N. Mohammed,and Mohammed kamil Mohammed,and Najafi,and Nor azwadi Che Sidik,and firdaus Basrawi,and Ahamedn, Abdalla &and S.S. Hoseini [10]

This paper investigates the effect of perforation shape on the heat transfer of perforated fins. This types of heat exchanger used in heat sink with the perforated fins under the forced convection heat transfer to determine the Performance for each perforation shape between circular, cylindrical, rectangular triangular, and also with the non perforated fins. The experimental result compare between perforations shape and the heat transfer coefficient to clarify the best perforation shape for the plate heat sink after studied experimentally and numerically using CFD. The difference between experimental & numerical is reported. about 8 % & 9% for temperature distribution when power Supplied are 150 W 100W respectively. The higher temperature difference of the fins are with the circular perforation shape which is 51.29% when compared with at the tip of the fins with the temperature at the heat collector followed by the rectangular perforation shape with 45.57 then followed by the rectangular perforation shape by 42.28% then lastly the non perforated fins by 35.82%. The perforation of the fins shows significant effect on the performance of force Convection heat transfer The used of solid cylindrical fins. having less heat transfer enhancement than the perforated cylindrical fins. Enhancement efficiencies varies depending on clearance ratio and inter-fin spacing ratio and comparatively lower Reynolds number are suggested for higher thermal performance. Also it is observed that solid fins required maximum material and having weight higher than perforated fins. In this research work, it is proposed to study the heat transfer enhancement in the cylindrical perforated and solid fins with staggered arrangement.

The experimental set-up consisting of the following parts

1) Main Duct (Tunnel)
2) Heater Unit
3) Base Plate
4) Data Unit

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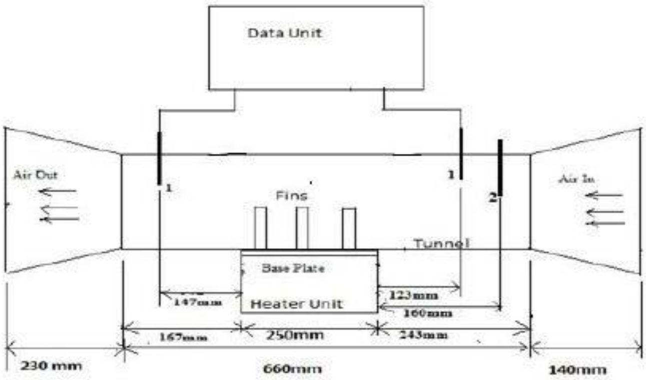

Figure:-1 Experimental Set-up

2.MAIN DUCT

. 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 fitted at 120cm away from the entry of the tunnel positioned horizontally and flow of air is controlled by the flow control valve mounted just after the blower. It has a convergent and divergent section at both ends 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.

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, mach inability and cost. 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.3. Circular pin fins with different lengths, constant C/H (Clearance ratio) values of 0 and 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 slantwise directions of 18.125 mm, with different spacing between the pin fins in the stream wise direction. The spacing ratios of the pin fins in the stream wise direction (Sy/D) were 1.944 and3.417 mm for both staggered arrangement and Inline arrangement, giving different numbers of the pin fins on the base plate. It is well-known fact that if the inter-fin spacing in the span wise direction.

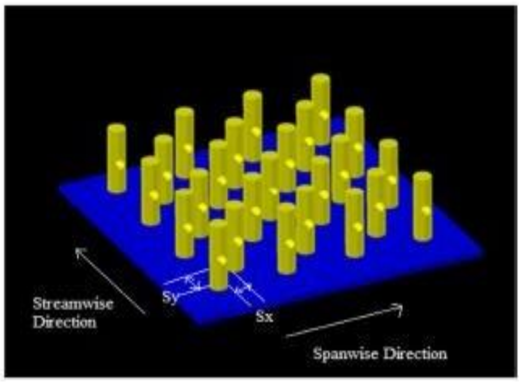

Figure:-3

Base Plate with fins in staggered arrangement

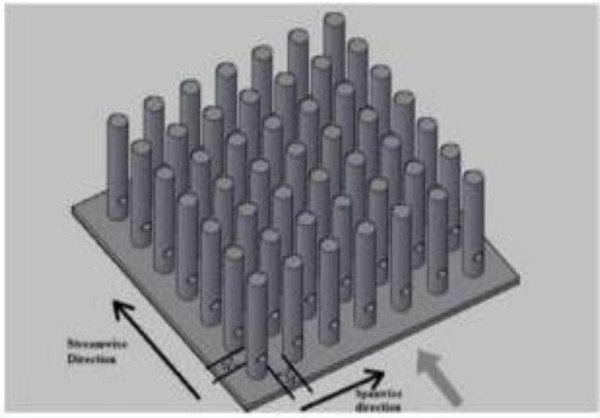

Figure:-4
Base Plate with fins in Inline arrangement

Table-1
Details *of Dimensions and Number of Base plate and Fins*

Sr. No	Particulars	Size (mm)	Quantity
1	Base Plate(Without Fins)	250x250	1
2	Base plate (with fins)	250x250	8
3	Perforated fins	100mm	85
4	Solid Fins	100mm	85

Sr.No	Streamwise distance to diameter ratio i.e, Sy/D	No. of fins on Base plate for Staggered arrangement	No. of fins on Base plate for Inline arrangement
1	1.944	18	35
2	3.417	11	21

Data Unit

It consist of various indicating devices which indicates the reading taken by the various Component like RTD sensors, pressure gauges, Anemometer There are three Temperatures 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 .

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Experimental Procedure

- First of all make all the necessary attachment to wooden tunnel required for Experimental set-up attachment of data unit, attachment of blower unit, attachment Of heater unit
- Keep the aluminum base plate of required dimensions on heater unit.
- Move the heater unit upward by rotating the screw jack.
- Switch on the RTD sensors of inlet, outlet and base plate temperature indicator and Check whether it is properly functioning or not.
- Measure the room temperature by anemometer by changing the function of it to Temperature mode.
- Switch on the heater, as soon as base plate temperature reached up to 100°C, the Temperature controller of RTD sensors comes in operation and it will cut off the Power supply of heater. Once the temperature of base plate reduced up to 99°C again Temperature controller of RTD sensors comes in operation and start the power supply of heater and cycle gets repeat.
- Now switch on the blower and measure the velocity of inlet air by using digital Anemometer. Make the inlet air velocity constant at required velocity by regulating the speed of blower i.e. 2m/s, 3m/s, 4m/s, and 5m/s.
- Now air will pass over heated base plate through tunnel
- Measure the outlet temperature of outgoing warm air.
- Now due to forced convection the temperature of base plate falls below 100°C. As Soon as the temperature of base plate falls below the 100°C heater unit will start heating the base plate to achieve the constant 100°C temperature by supplying Constant electrical input as the air is continuously flowing over the base plate heat get Transferred from it to the flowing air. Thus the temperature of base plate falls continuously and takes the temperature readings of base plate after 90 seconds of air Flow. The duration of air flow is constant for all types of base plate mentioned above.
- Similarly, repeat the same procedure for velocity 3m/s, 4m/s, and 5m/s and take the similar readings
- Maintain the temperature of base plate it will not allow to exceed the temperature of base plate above desired values. Two pressure gauges are mounted on the data unit which shows the inlet and outlet pressure of air in the range of 0 to 150 kg/cm² inside the tunnel. Inlet flow rate of air is indicated by velocity indicator using Anemometer. MCB Hedger switches are mounted to cut off the power supply in case any Short circuit.

- ***Experimental Observations***
- After conducting the experiment the following reading are obtained for the respective Conditions

For Constant C/H=0i.e.Fin Height=100mm

Voltage=230V, Current=6.5 amp.

I. . FOR STAGGERED ARRANGEMENT (PERFORATED)

1) For Perforated Fins Sy/D=1.944 i.e. No. of Fins on one base plate=18 No

Sr. No	Velocity (m/s)	Surface Temp.((°c) (T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp. (°c) (T_{out})
1	2	100	32	44
2	3	100	32	41
3	4	100	32	41
4	5	100	32	40

2)For Perforated Fins Sy/D=3.417 i.e. No. oFins on one base plate=11 No.

Sr. No	Velocity (m/s)	Surface Temp.((°c) (T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp. (°c) (T_{out})
1	2	100	31	39
2	3	100	31	38
3	4	100	32	38
4	5	100	32	37

For Staggered Arrangement (Solid)

For Solid Fins Sy/D=1.944 i.e. No. of Fins on one base plate=18 No

Sr. No	Velocity (m/s)	Surface Temp.((°c) (T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp. (°c) (T_{out})
1	2	100	32	43
2	3	100	33	42
3	4	100	33	41
4	5	100	32	39

For Solid Fins Sy/D=3.417

i.e. No. of Fins on one base plate=11 No

Sr. No	Velocity (m/s)	Surface Temp.((°c) (T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp. (°c) (T_{out})
1	2	100	33	40
2	3	100	33	39
3	4	100	32	38
4	5	100	32	36

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For Inline Arrangement (Solid)

For Perforated Fins Sy/D=1.944
i.e. No. of Fins on one base plate=35 No.

1	2	100	32	42
2	3	100	33	40
3	4	100	32	40
4	5	100	32	38

For Perforated Fins Sy/D=3.417
i.e. No. of Fins on one base plate=21 No.

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	40
2	3	100	32	38
3	4	100	32	37
4	5	100	32	37

For Solid Fins Sy/D=1.944
i.e. No. of Fins on one base plate=35 No.

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	42
2	3	100	32	40
3	4	100	32	39
4	5	100	32	37

For Solid Fins Sy/D=3.417
i.e. No. of Fins on one base plate=21 No

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	38
2	3	100	32	37
3	4	100	32	37
4	5	100	32	36

For Smooth Duct i.e. Base Plate without Fins

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	33
2	3	100	32	33
3	4	100	32	33
4	5	100	32	33

For Inline Arrangement (Solid)

For Perforated Fins Sy/D=1.944 i.e. No. of Fins on one base plate=35 No

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	42
2	3	100	32	40

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
3	4	100	32	40
4	5	100	32	38

For Perforated Fins Sy/D=3.417 i.e. No. of Fins on one base plate=21 No.

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	40
2	3	100	32	38
3	4	100	32	37
4	5	100	32	37

For Solid Fins Sy/D=1.944 i.e. No. of Fins on one base plate=35 No.

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	42
2	3	100	32	40
3	4	100	32	39
4	5	100	32	37

For Solid Fins Sy/D=3.417 i.e. No. of Fins on one base plate=21 No.

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	38
2	3	100	32	37
3	4	100	32	37
4	5	100	32	36

For Smooth Duct i.e. Base Plate without Fins

Sr.No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. .(°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	33
2	3	100	32	33
3	4	100	32	33
4	5	100	32	33

Sample calculations for Experimental Analysis

For Sample calculation Table 1 is taken and air velocity = 2m/

For Perforated Fins Sy/D=1.944 i.e. No. of Fins on one base plate=18 No

For Perforated Fins, C/H=0, i.e. Fin height=100mm

Sr. No	Velocity (m/s)	Surface Temp.((°c)(T_s)	Inlet Temp. (°c) (T_{in})	Outlet Temp.(°c) (T_{out})
1	2	100	33	44
2	3	100	32	41
3	4	100	32	41
4	5	100	32	40

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

Q_(conv.)= 〖Q〗(elect.)--Q_(cond.)-- 〖Q(rad.)

Q_conv= Q_elect

Hence by Ohm"s Law

Q_(elect.)=I^2×R

As per experimental dataI=6.5 amp and V=230V

$$R = \frac{V}{I} = \frac{230}{6.5} = 35.38\ \Omega$$

Q_(elect.)=I^2×R

$QE_{lectrical=}6.5^2X35.38$

=1495W

Q_conv= Q_elect =1495W

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

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

Total area (As) = Projected area + Total surface area contribution from the blocks

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

Calculation for Friction Factor

The pressure drops in the tunnel without fins is so small that they could not be measured by the Manometer. This resulted from smaller length of the test section and smaller roughness of the duct. The experimental pressure

A_s=250X250+[3.14X15X100-2X3.14X4X4.8]18+[(2X3.14X.4^2+2X3.14x4x15)-2x3.14x4x4.8]18

$A_{s=-0.1515m}{}^2$

$h_{av2=}$159.16 w/m^2 ^{0}c

$$Heat\ Transfer\ Coefficient\ h_{av1} = \frac{1495}{0.1515\left[100-\left(\frac{44+32}{2}\right)\right]}$$

$$R_e = \frac{D_h U}{\nu}$$

$$Reynolds\ No.\ R_e = \frac{0.14286 \times 2}{16.83 \times 10^{-6}}$$

$$Reynolds\ No.\ R_e = 16976.82$$

Calculation for Heat Transfer

In order to have basis for the evaluation of the effects of the fins, the average nusselt number (Nus) for the smooth surface (without fins) and Nu with fins will be correlated as function of Re and Pr as follows and suffix 2 represent the velocity at 2m/s.

$$Nu_s \text{ For Smooth Duct (Without Fin)} Nu_{s2} = 0.077 Re^{0.716} Pr^{1/3}$$

$$Nu_{s2} = 0.077 \times 16976.82^{0.716} \times 0.7^{1/3}$$

$$Nu_{s2} = 69.58$$

This equations are valid for the experimental conditions of 13,500 ≤ Re ≥ 42,000, 1.944 < Sy/D < 3.417, C/H = 0 and Pr = 0.7 by using this equation the Nu/Nus and Re will be

$$Nu\ (with\ fin) Nu_2 = 45.99 Re^{0.396}\left(1 + {}^{C}/_{H}\right)^{-0.608}\left({}^{S_y}/_{D}\right)^{-0.522} Pr^{1/3}$$

determine for perforated fins for constant C/H

$$Nu\ (with\ fin) Nu_2 = 45.99(16976.82)^{0.396}(1+0)^{-0.608}(1.944)^{-0.522} 0.7^{1/3}$$

$$Nu_2 = 1365.69$$

$$\frac{Nu_2}{Nu_{s2}} = 19.62$$

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 „"F " " Using the experimental results, f was correlated as a function of the duct Reynolds number, Re, and geometrical parameters. The resulting equation is

Values for staggered Arrangement (C/H = 0)

$$F_2 = 2.4Re^{-0.0836}(1 + C/H)^{-0.805}(S_y/D)^{-0.0814}$$

$$F_2 = 2.4 \text{ x } 16976.82^{-0.0836}(1+0)^{-0.0836}(1.944)^{-0.0814}$$

Values for Staggered Arrangement (C/H=0)

$$F_2 = 1.0061$$

	Sy/D	(As) in m^2	at V=2 m/s Nu/Nus 2	at V=3 m/s Nu/Nus 3	at V=4 m/s Nu/Nus 4	at V=5 m/s Nu/Nus 5	at V=2 m/s Re_2	at V=3 m/s Re_3	at V=4 m/s Re_4	at V=5 m/s Re_5	at V=2 m/s F_2	at V= 3 m/s F_3	at V= 4 m/s F_4	at V= 5 m/s F_5
Perforated	**1.944**	0.1515	19.62	17.11	15.51	14.61	16976.82	25678.84	34238.46	42926.68	1.0061	0.9729	0.9427	0.9319
	3.417	0.1169	14.95	12.91	11.70	10.88	17264.04	25974.54	34528.09	43290.9	0.9606	0.9223	0.9015	0.8855
Solid	**1.944**	0.1473	19.23	16.63	15.13	14.12	17027.4	25586.86	34156.6	43030.12	1.0042	0.9711	0.9419	0.9308
	3.417	0.1143	14.41	12.63	11.43	10.70	17119.23	25756	34528.09	43422.49	0.9592	0.9211	0.901	0.8815

Values for Inline Arrangement (C/H = 0)

	Sy/D	(As) in m^2	at V=2 m/s Nu/Nus 2	at V=3 m/s Nu/Nus 3	at V=4 m/s Nu/Nus 4	at V=5 m/s Nu/Nus 5	at V=2 m/s Re_2	at V=3 m/s Re_3	at V=4 m/s Re_4	at V=5 m/s Re_5	at V=2 m/s F_2	at V= 3 m/s F_3	at V= 4 m/s F_4	at V= 5 m/s F_5
Perforated	**1.944**	0.2357	18.75	16.64	15.02	13.98	16836.77	25255.15	33673.54	42091.92	1.009	0.9792	0.9511	0.9385
	3.417	0.1664	13.89	12.24	11.09	10.32	17108.98	25896.07	34632.72	43290.9	0.9693	0.9285	0.9072	0.8895
Solid	**1.944**	0.2274	18.18	16.36	14.67	13.63	16976.82	25756	34424.09	43160.12	1.0082	0.9766	0.9483	0.9365
	3.417	0.1614	13.35	12.05	10.83	10.11	17264.04	25974.54	34632.72	43406.66	0.9676	0.9263	0.9049	0.8883

Result of Experimental Analysis

From the above calculations the actual values are obtained for nusselt number and friction factor on the basis of these values graphs are plot between Nusselt No. and Reynolds No., Friction factor and

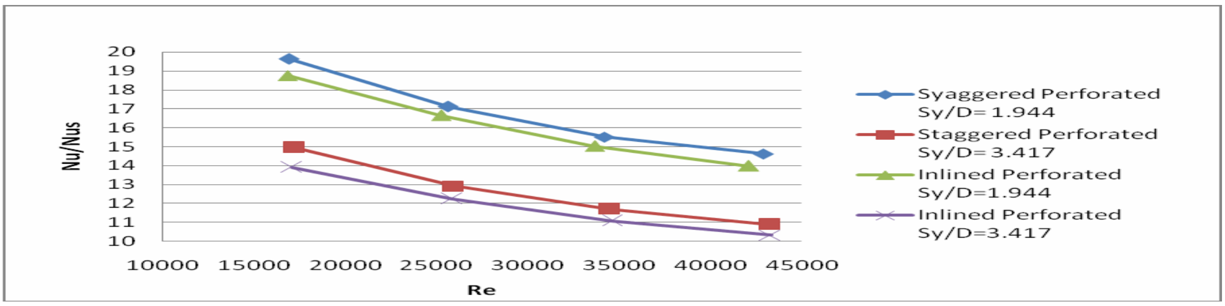

Fig: -5 Variation of Nu/Nus with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Staggered Perforated & Inline Perforated

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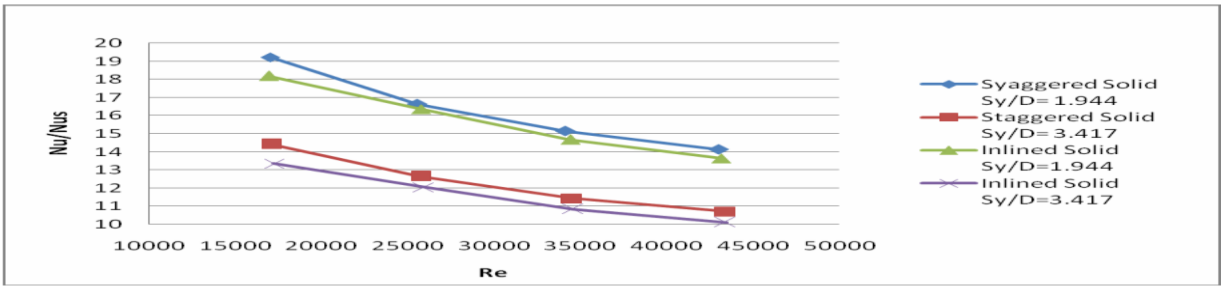

Fig:- 6 Variation of Nu/Nus with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Staggered Solid & Inline Solid arrangement

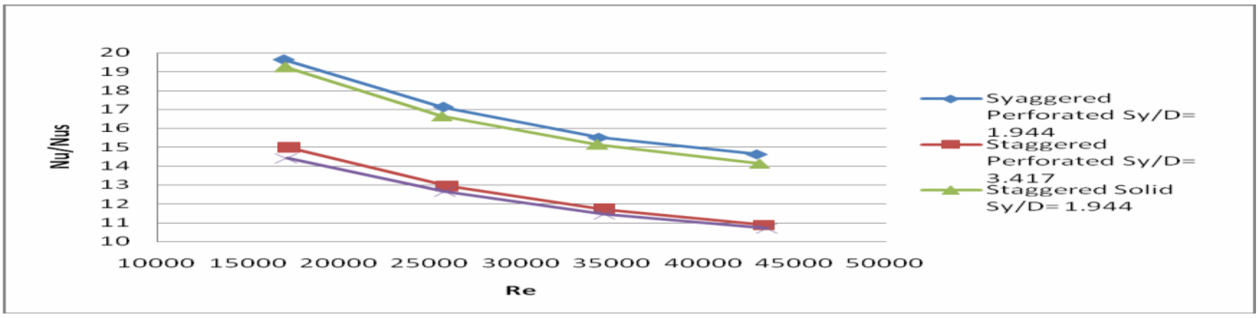

Fig:-7 Variation of Nu/Nus with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Staggered arrangement

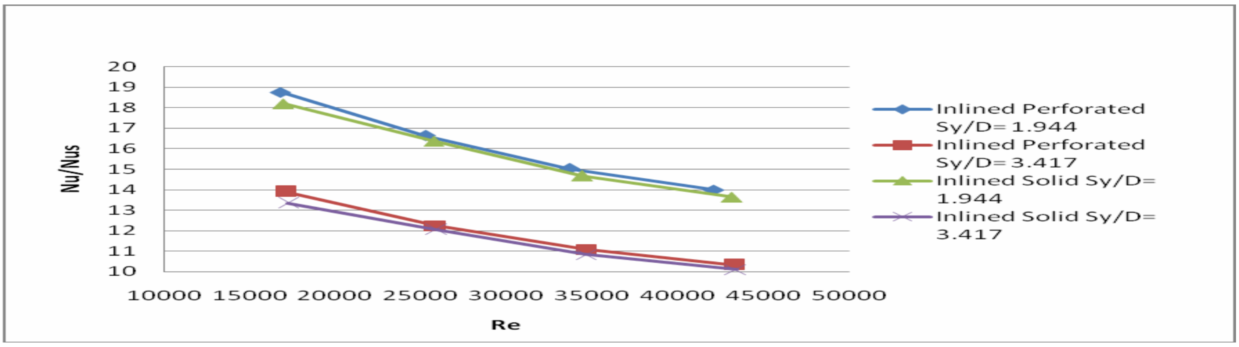

Fig:-8 Variation of Nu/Nus with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Inline arrangement

Fig. 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.

Effect on Friction Factor

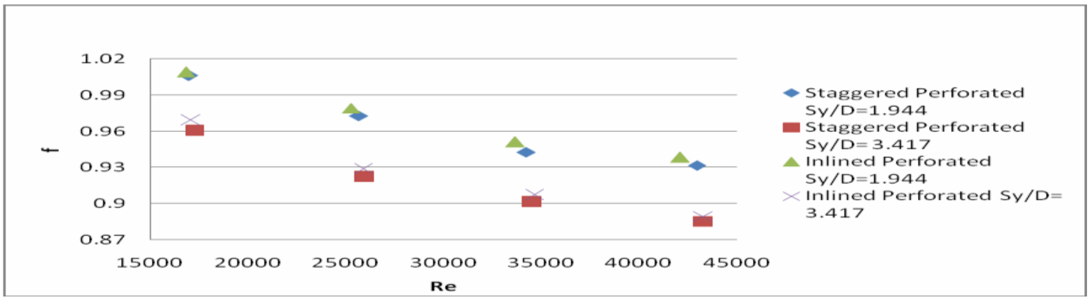

Fig:- 10 Variation of Friction factor with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Staggered Solid & Inlined Solid arrangement

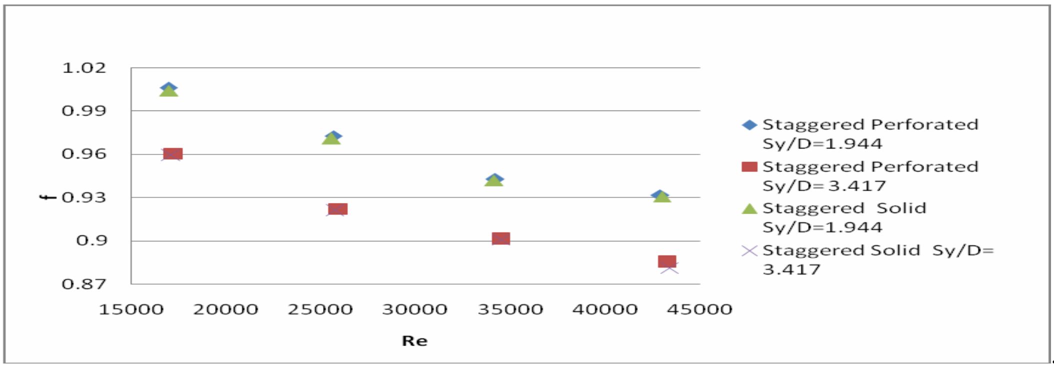

Fig:- 11 Variation of Friction factor with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Staggered arrangement

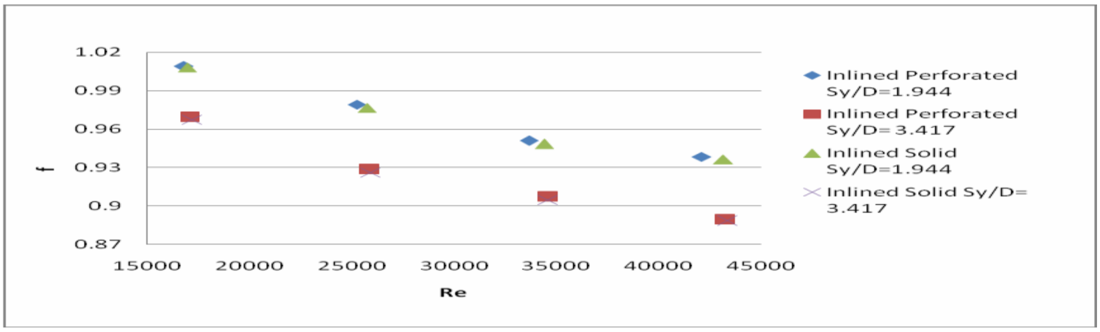

Fig:-12 Variation of Friction factor with Reynolds No. (Re) for various inter-fin spacing ratio (Sy/D) for Inlined arrangement

The other notable result is seen from Fig.20 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, stream wise distance between fins is more effective parameter on the friction factor than span wise 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.

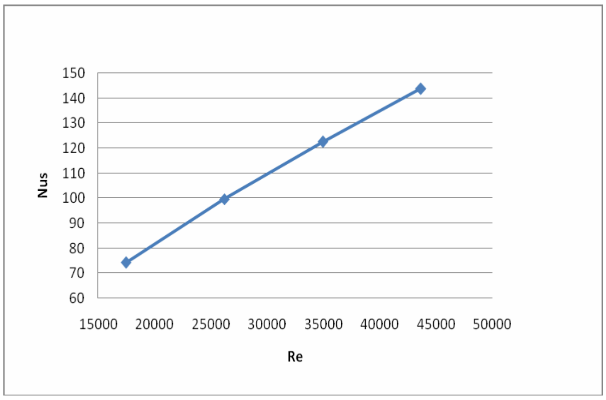

Fig: -13 Variation of Nu/Nus with Reynolds No. (Re)for various inter-fin spacing ratio (Sy/D) for Staggered Solid & Inline Solid arrangement

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

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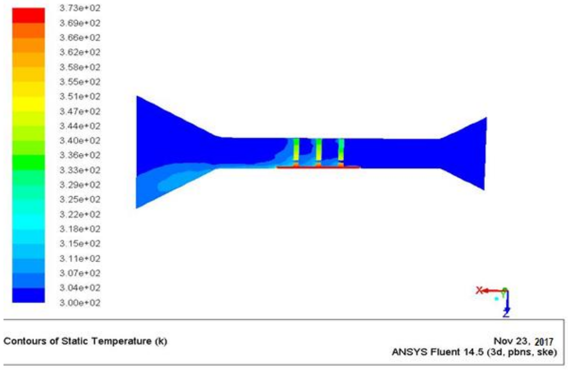
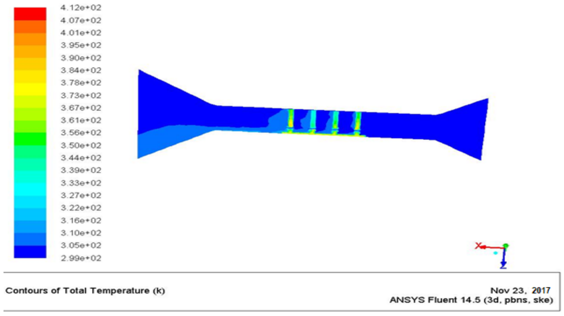
A.1 Heat Flow Pattern for Staggered Perforated Fins base plate with 4m/s velocity of fluid

1) 18 fins on base plate with 100mm height

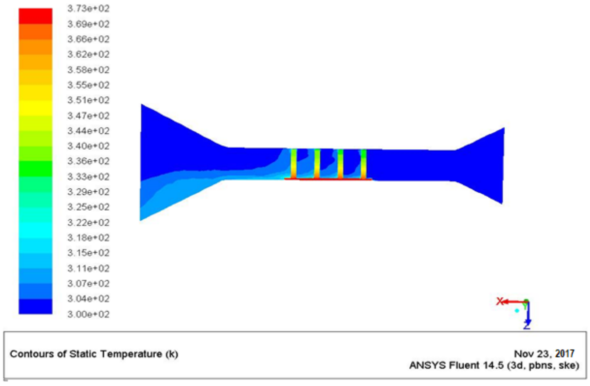


2) 11 fins on base plate with 100mm height

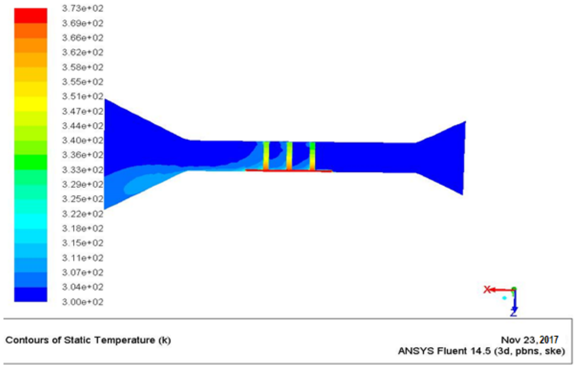


A.2 Heat Flow Pattern for Staggered Solid Fins base plate with 4m/s velocity of fluid

1) 18fins on base plate with 100mm height



2) 11 fins on base plate with 100mm height



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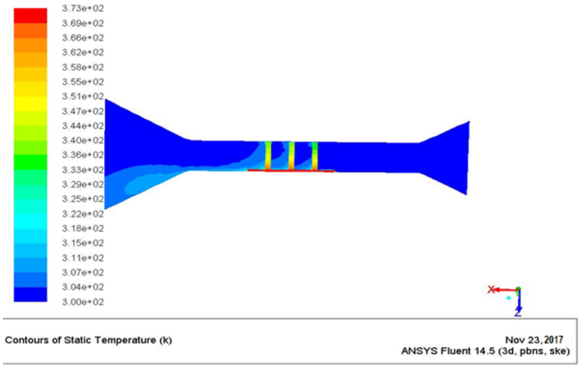
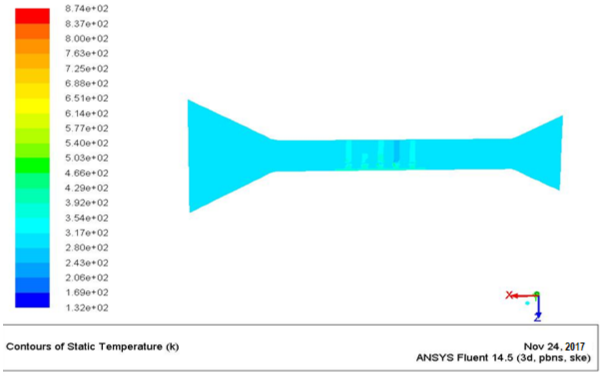
A.3 Heat Flow Pattern for Inline Perforated Fins base plate with 4m/s velocity of fluid

1) 35 fins on base plate with 100mm height

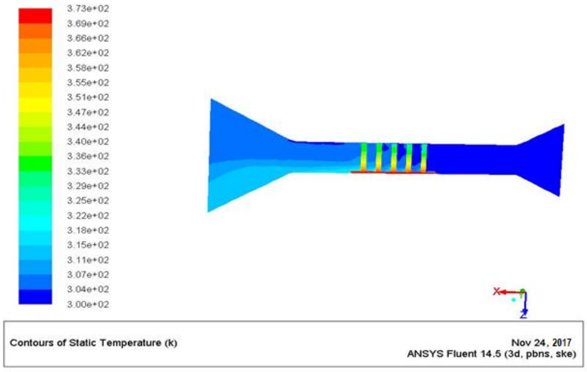


2) 21 fins on base plate with 100mm height

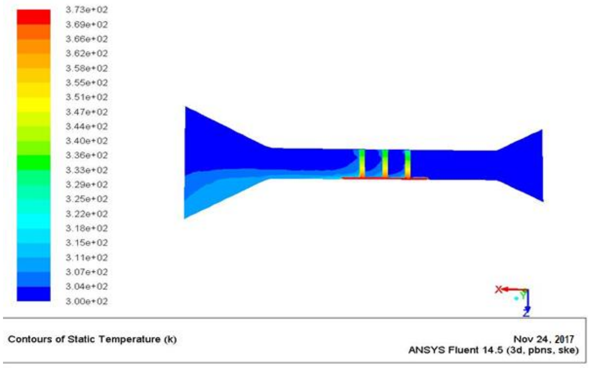


A.4 Heat Flow Pattern for Inline Solid Fins base plate with 4m/s velocity of fluid

1) 35 fins on base plate with 100mm height



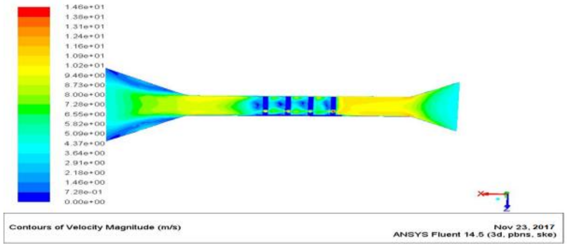
2) 21 fins on base plate with 100mm height



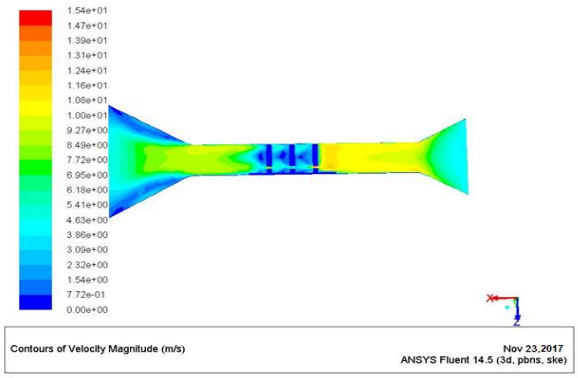
A.5 Heat Flow Pattern for Smooth base plate with 5m/s velocity of fluid

A.1 Air Flow Pattern for Staggered Perforated Fins base plate with 4m/s velocity of fluid

1) 18 fins on base plate with 100mm height

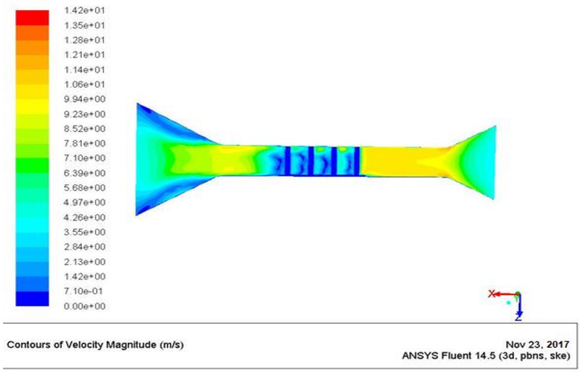


2) 11 fins on base plate with 100mm height



A.2 Air Flow Pattern for Staggered Solid Fins base plate with 4m/s velocity of fluid

1) 18 fins on base plate with 100mm height



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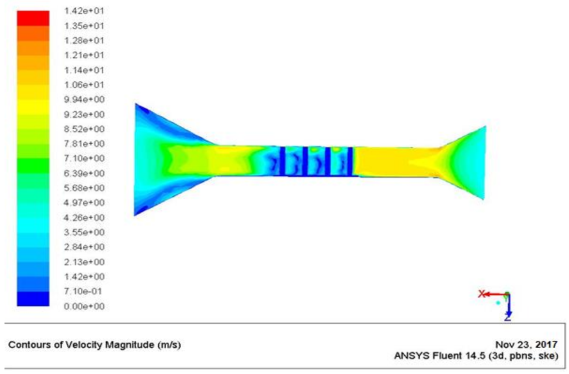

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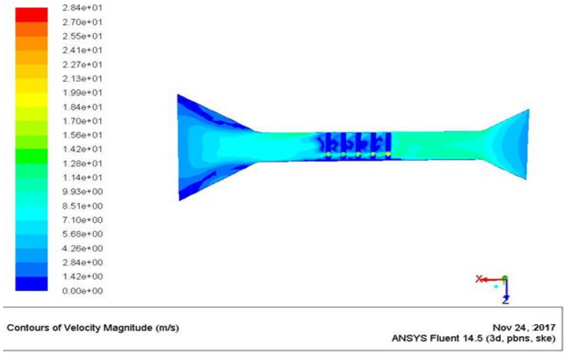
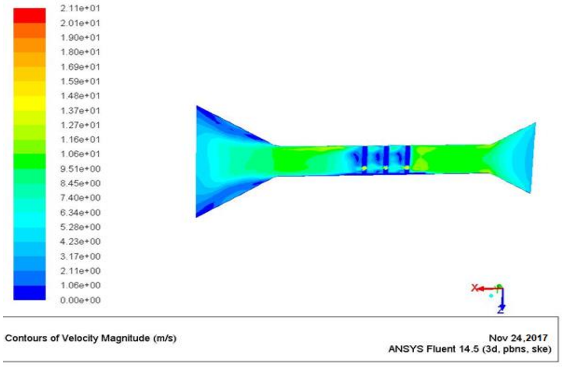
A.3 Air Flow Pattern for Inline Perforated Fins base plate with 4m/s velocity of fluid

1) 35 fins on base plate with 100mm height

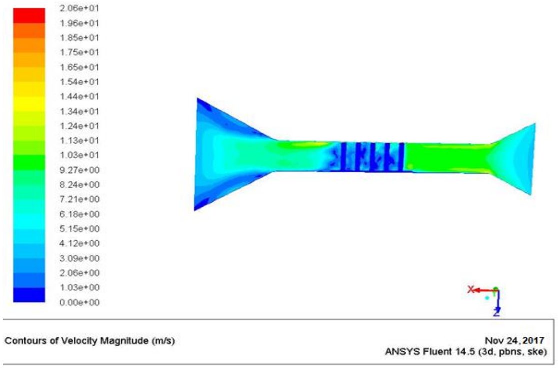


2) 21 fins on base plate with 100mm height

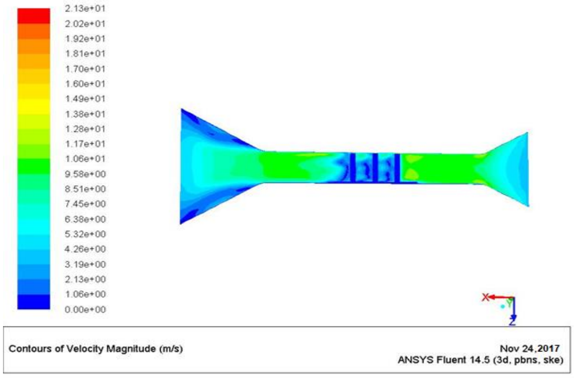


A.4 Air Flow Pattern for Inline Solid Fins base plate with 4m/s velocity of fluid

1) 35 fins on base plate with 100mm height



2) 21 fins on base plate with 100mm height



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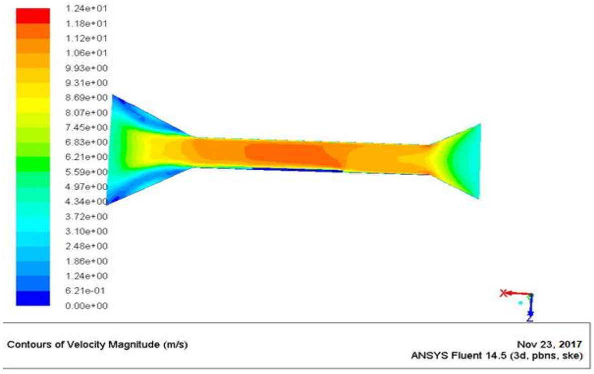
A.5 Air Flow Pattern for Smooth base plate with 4m/s velocity of fluid



A.1 Air Flow Pattern for Staggered Perforated Fins base plate with 5m/s velocity of fluid

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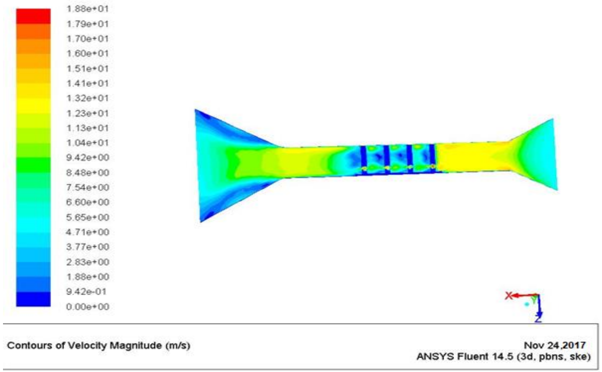
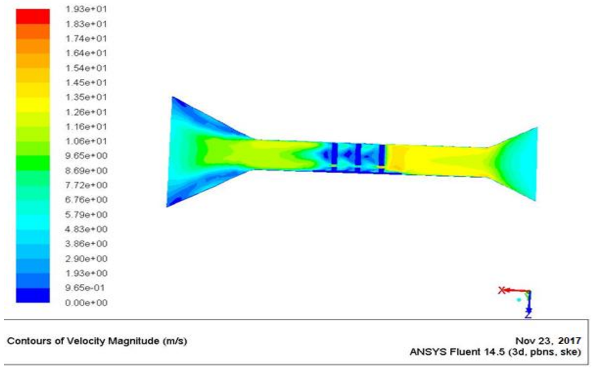


2) 11 fins on base plate with 100mm height



A.2 Air Flow Pattern for Staggered Solid Fins base plate with 5m/s velocity of fluid

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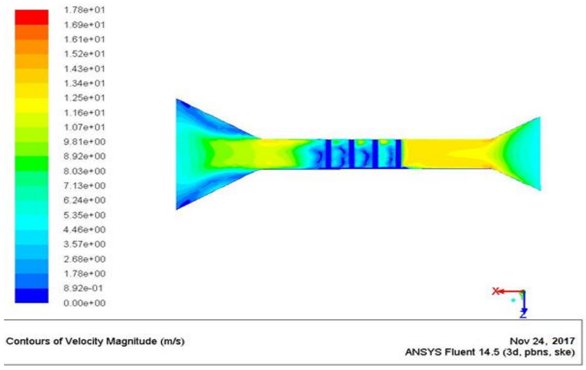

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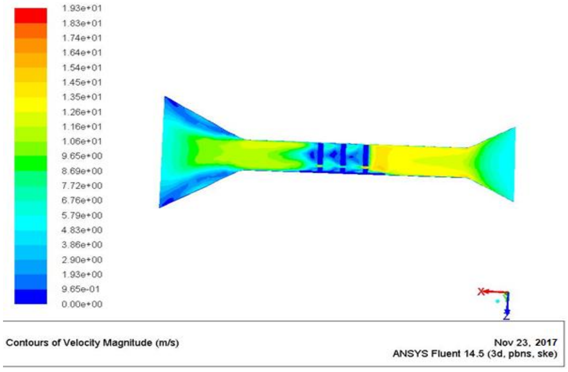
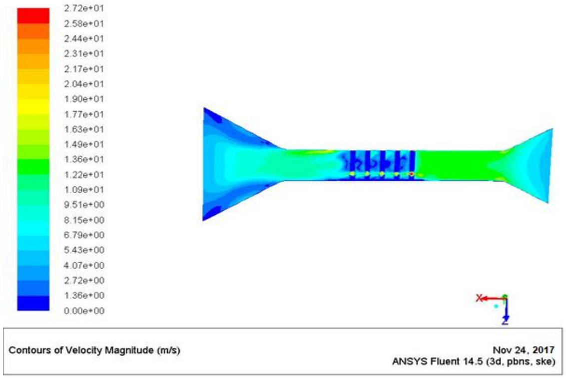
2) 11 fins on base plate with 100mm height



A.3 Air Flow Pattern for Inline Perforated Fins base plate with 5m/s velocity of fluid

1) 35 fins on base plate with 100mm height



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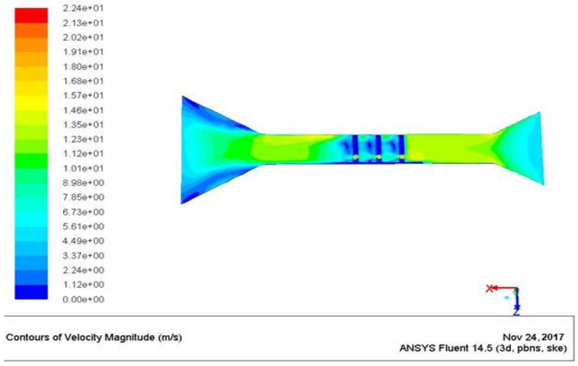

A.4 Air Flow Pattern for Inline Solid Fins base plate with 5m/s velocity of fluid

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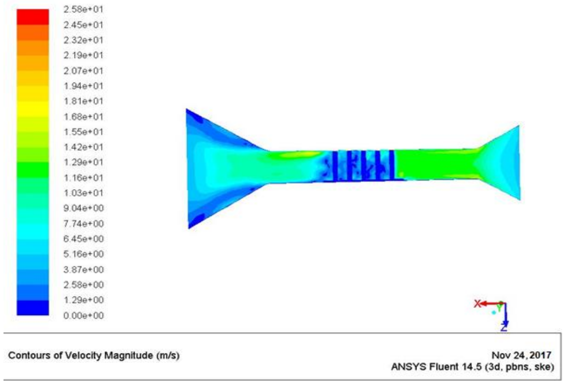

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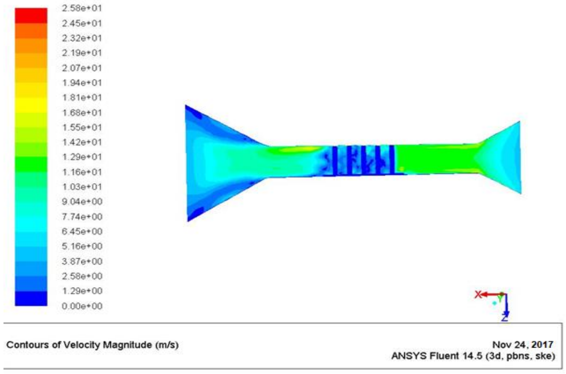
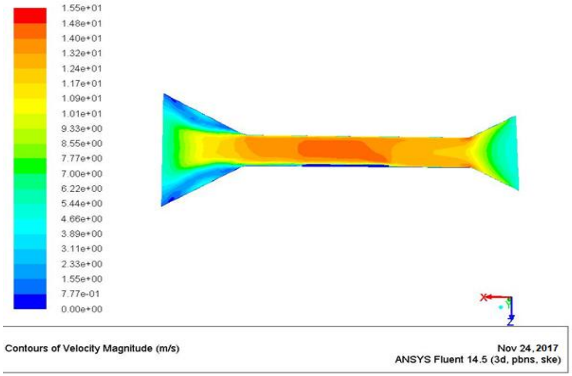
2) 21 fins on base plate with 100mm height

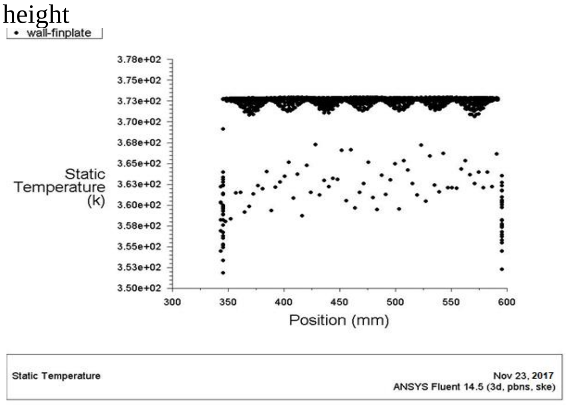


A.5 Air Flow Pattern for Smooth base plate with 5m/s velocity of fluid



A.1 Temperature Variation across Base Plate for Staggered Perforated Fins with 4m/s velocity of fluid

1) 18 fins on base plate with 100mm height



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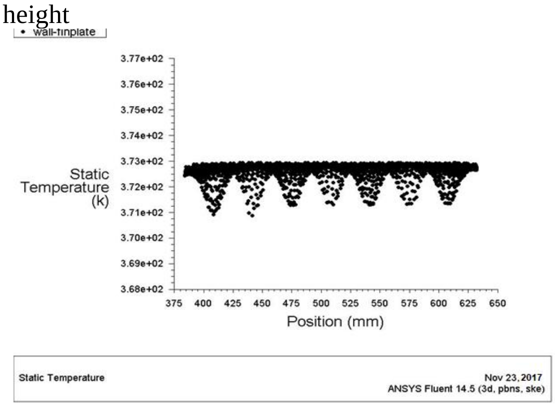

A.2 Temperature Variation across Base Plate for Staggered Solid Fins with 4m/s velocity of fluid

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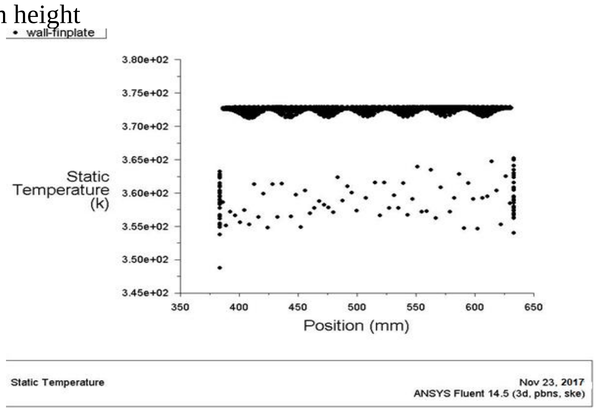
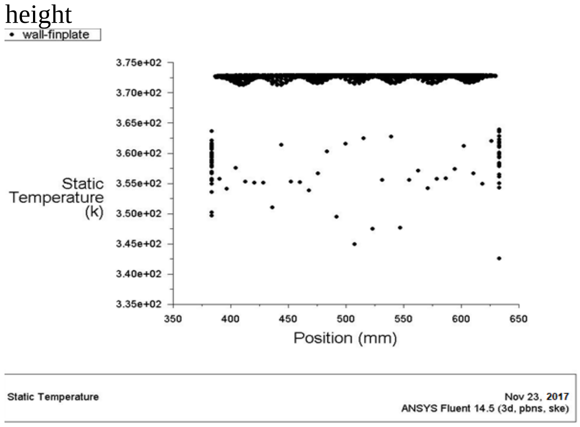
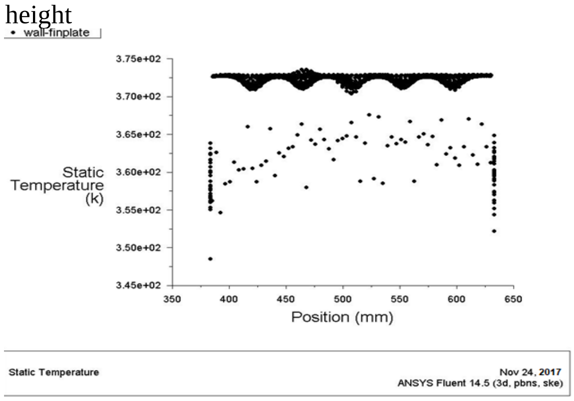


2) 11 fins on base plate with 100mm height

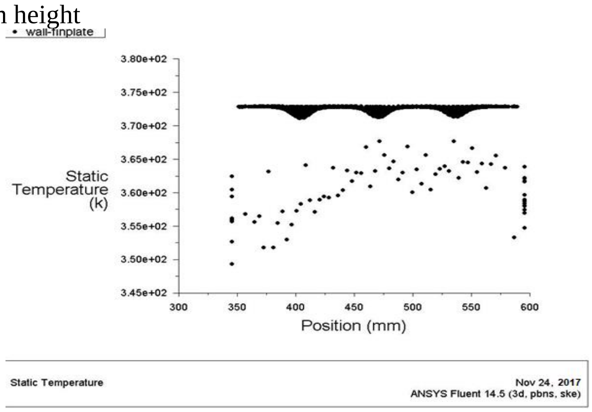


A.3 Temperature Variation across Base Plate for Inline Perforated Fins with 4m/s velocity of fluid

1) 35 fins on base plate with 100mm height



2) 21 fins on base plate with 100mm height



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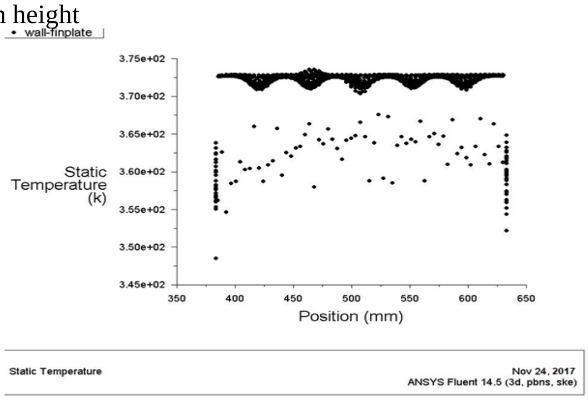
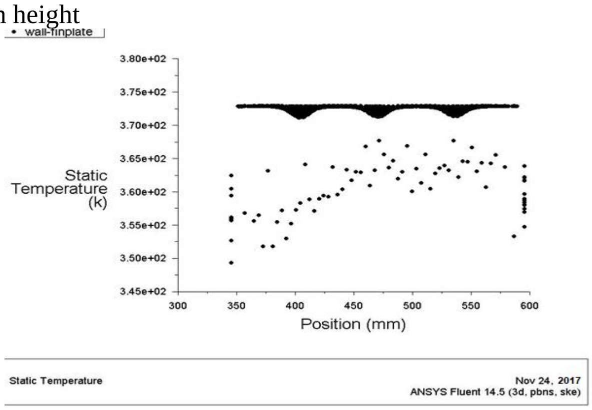
A.4 Temperature Variation across Base Plate for Inline Solid Fins with 4m/s velocity of fluid

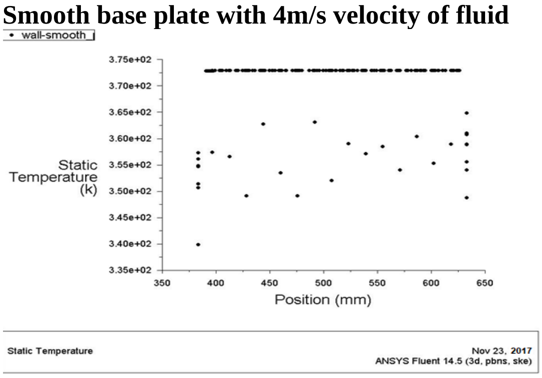
1) 35 fins on base plate with 100mm height



2) 21 fins on base plate with 100mm height

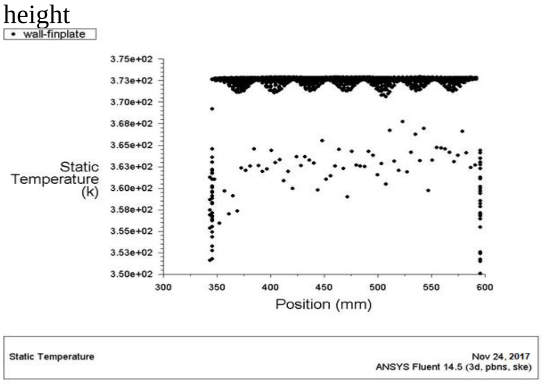


A.5 Temperature Variation across Smooth base plate with 4m/s velocity of fluid



A.1 Temperature Variation across Base Plate for Staggered Perforated Fins with 5m/s velocity of fluid

1) 18 fins on base plate with 100mm height



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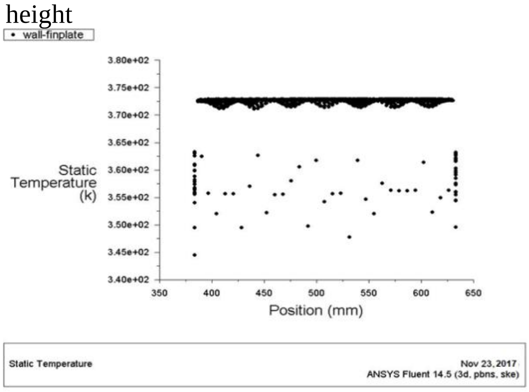
2) 11 fins on base plate with 100mm height



A.2 Temperature Variation across Base Plate for Staggered Solid Fins with 5m/s velocity of fluid

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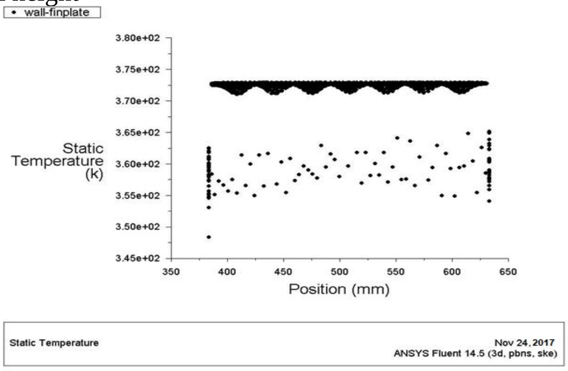
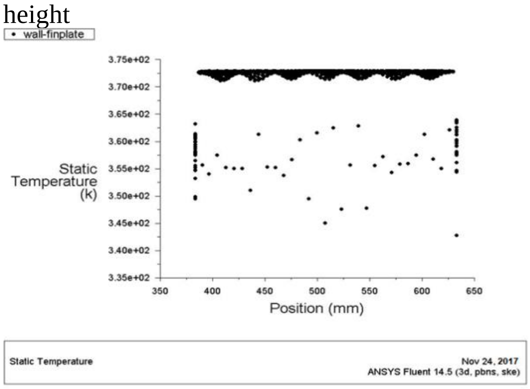


2) 11 fins on base plate with 100mm height



A.3 Temperature Variation across Base Plate for Inline Perforated Fins with 5m/s velocity of fluid

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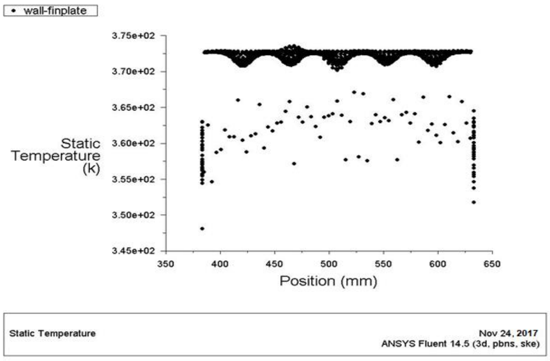
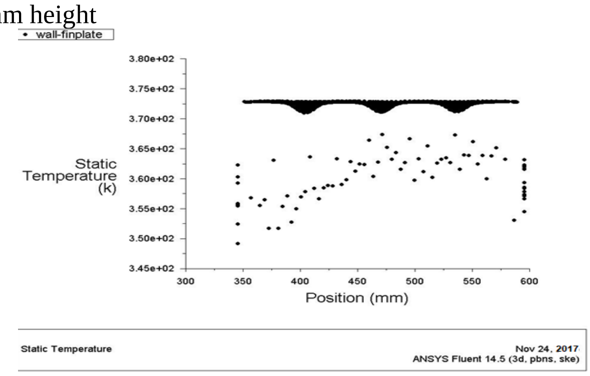
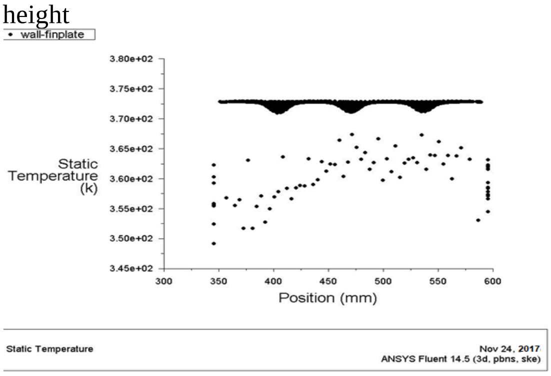


2) 21 fins on base plate with 100mm height



A.4 Temperature Variation across Base Plate for Inline Solid Fins with 5m/s velocity of fluid

1) 35 fins on base plate with 100mm height



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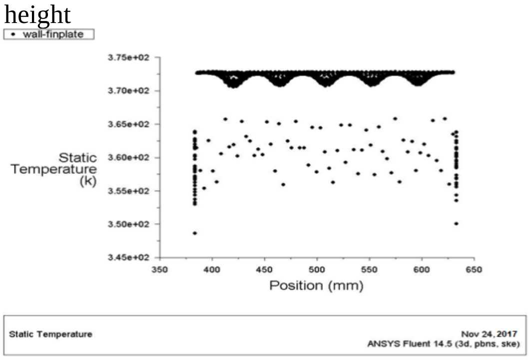
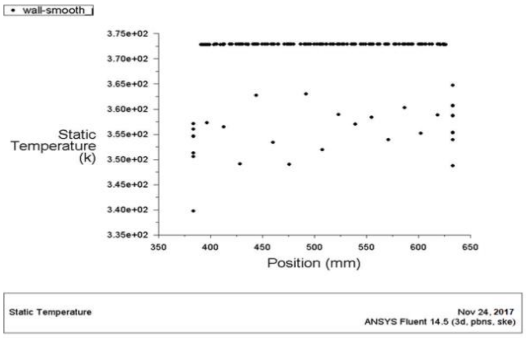


A.5 Temperature Variation across Smooth base plate with 5m/s velocity of fluid



3 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 experiment equipped with cylindrical 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, and the enhancement efficiency correlations have been obtained. The conclusions are summarized as:

(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) Enhancement efficiencies increased with decreasing Reynolds number. Therefore, relatively lower Reynolds number led to an improvement in the heat transfer performance.
(d) 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.
(e) 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

4 When all the goals were taken into account together, the trade-off among goals was considered and the optimum results were obtained at 42,000 Reynolds number, 50 mm fin height and 3.417 Sy/D pitch.

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