CFD Analysis of Convective Heat Transfer from Inverted Trapezoidal Plate Fin Array

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Abstract— *Heat sinks with plate fins are widely used for surface cooling purpose in various heat transfer applications. Plate fin geometry affects the performance of heat sinks. In this work numerical study has been performed for Rectangular Plate Fins, Trapezoidal Plate Fins and Inverted Trapezoidal Plate Fins. For numerical study software used was Ansys Fluent. Velocity are 0.44, 0.88, 1.31 and 1.75 m/s. Heat supplied was 125 W. Average Heat Transfer Coefficient, Nusselt No. Pressure Drop are calculated and compared. Inverted Trapezoidal plate fins type II(ITPF-II) gives better performance and % enhancement of 7.57 was observed over RPF and 4.94 % over TFHS. Further pressure drop also calculated and it was observed that as Reynolds No increases pressure drop also increases and it is almost linear with small difference in case of Rectangular, Trapezoidal and Inverted Trapezoidal fins of type I but Inverted Trapezoidal fins of type II shows higher deviation in pressure drop at higher Reynolds No. that is greater than 12000 in comparison with other geometries.*

***Keywords*-Heat Transfer Coefficient, Nu, Pressure Drop, Trapezoidal plate Fins (TPF)and Inverted Trapezoidal Plate Fins (ITPF-I and II) .**

I. INTRODUCTION

Convective heat transfer augmentation in heat sinks with plate fins is very challenging now a days. Many researchers worked on rectangular fins with different parameters. Parameters include use of pin fins in the passages between fins by varying diameters of fin pins. Some researchers worked on geometrical parameters like fin spacing, aspect ratio, angle of attack of working fluid. Some work also has done using Trapezoidal shape plate fins. A numerical study was done using rectangular, Trapezoidal and parabolic shaped heat sink for comparing their performance and convective heat transfer coefficient was found to be maximum in case of trapezoidal heat sink[1]. Design of fin for heat dissipation in natural convection was studied using rectangular, Trapezoidal and Inverted trapezoidal fins having heat load varies from 3 to 20 W and it was found that conventional rectangular fin gives better performance than trapezoidal fins but in case of inverted trapezoidal fins heat transfer coefficient is 25 % higher than trapezoidal fins and 10 % higher than conventional rectangular fins[2]. Performance of rectangular plate fins was studied to analyze heat transfer on vertical base experimentally and numerically at different inclinations and results shows that 0^0 staggered fins gives better performance than 30^0 staggered fins and this enhancement is about 17 %. Staggered fins shows enhancement as there is more turbulence in flow compared to inline fins[3]. Experimental and numerical analysis of rectangular plate fins with circular pin fins accommodated between gap of two plate fins was carried out under forced convection over vertical base and about 20 % enhancement was found for heat Sink with Plate Pin Fin over Heat Sink with Plate Fin [4].A numerical study for shape optimization of flat plate fins was done with geometry defined linear functions[5]. A inverted trapezoidal fins are analyzed using 2-D analytical method. Here in this method heat loss from fin was represented as a function of fin shape factor, fin base thickness. It was observed that heat loss decreases and fin efficiency increases linearly with increase in shape factor[6]. A comparative experimental and numerical study of Heat Sink with Plate pin fin and Heat Sink Plate Fin was done and form numerical analysis it was observed that thermal resistance for PPFHS 30 % lower than that of PFHS[7]. To Optimize fin spacing an experimental analysis in rectangular fin array to maximize the heat transfer rate in natural and forced convection was done. Aim of this work was to fin optimum fin spacing which gives maximum enhancement [8]. Rectangular fins with perforation under natural convection was investigated[9]. Heat transfer enhancement using porous fin can also be achieved[12].

In present study a comparative numerical study has been done using flat plate, Rectangular Plate Fins, Trapezoidal Plate Fins and Inverted Trapezoidal Plate Fins type I and Inverted Trapezoidal Plate Fins type II under forced convection. Reynolds No. Selected as 4000, 8000, 12000 and 16000 with corresponding velocities are 0.44, 0.88, 1.31 and 1.75 m/s and heat input is 125 W. Simulation has been done using Ansys CFD software and solver used is Fluent.

II. NUMERICAL ANLYSIS

the basic three steps in Numerical Analysis.

i. Pre Processing ii. Solver execution and iii.Post processing

Pre-processing in CFD include preparation of Geometrical models, computational domain, Computational meshing etc. Solver execution includes selection of models, application of suitable boundary conditions, selection of suitable material and fluid, selection of convection schemes, Cell zone condition etc. In this step different governing equations are solved as per models selected till the solution gets converged. CFD post processing consists of plotting the various property counters, such as temperature counters, Pressure contours, velocity counters, vector plots, velocity stream lines etc. Fig.1 shows Rectangular Plate Fins, Fig.2 shows Trapezoidal Plate Fins, and Fig.3 shows Inverted Trapezoidal Plate Fins.

A. Computational Mesh

All meshing details are shown in table1. In this work Hexahedral mesh is selected with prismatic layers.

TABLE 1
MESHING DETAILS

Sr. No.	Geometry Type	Mesh Details
1	RPF	Mesh Type: Hexahedral + Prisms Total Mesh Count: 1,036,390 First Prism Layer Thickness: 0.075 mm Total Prism Layer Height: 0.1555 mm Number of Layers: 5
2	TPF	Mesh Type: Hexahedral + Prisms Total Mesh Count: 3,534,516 First Prism Layer Thickness: 0.075 mm Total Prism Layer Height: 0.1555 mm Number of Layers: 5
3	ITPF	Mesh Type: Hexahedral + Prisms Total Mesh Count: 2,608,752 First Prism Layer Thickness: 0.075 mm Total Prism Layer Height: 0.1555 mm Number of Layers: 5

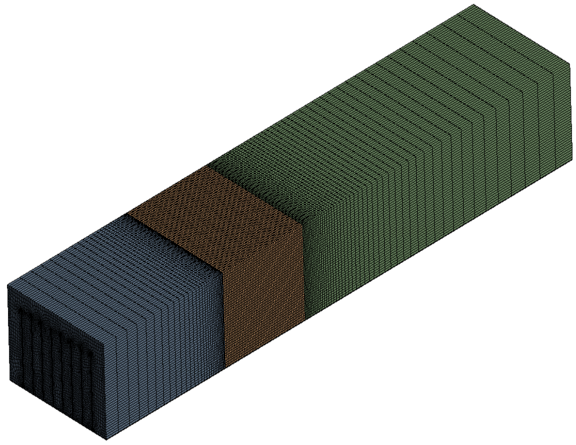

Fig .1 Meshing Details for RPF

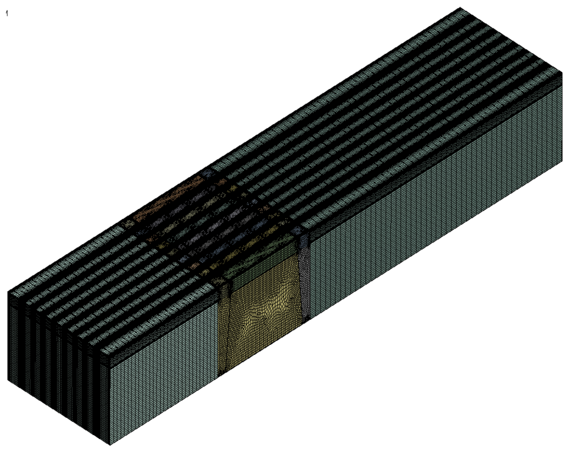

Fig .2 Meshing Details for TPF

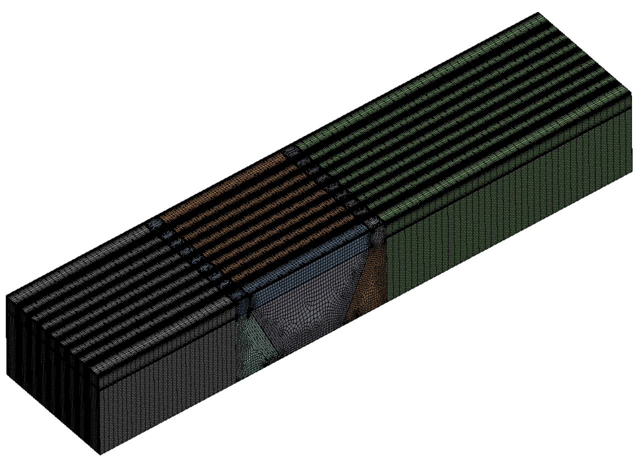

Fig.3 Meshing Details for ITPF

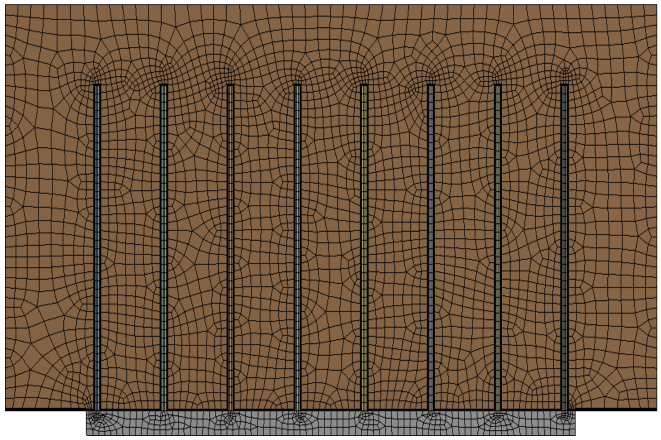

Fig.4 Cross Sectional View

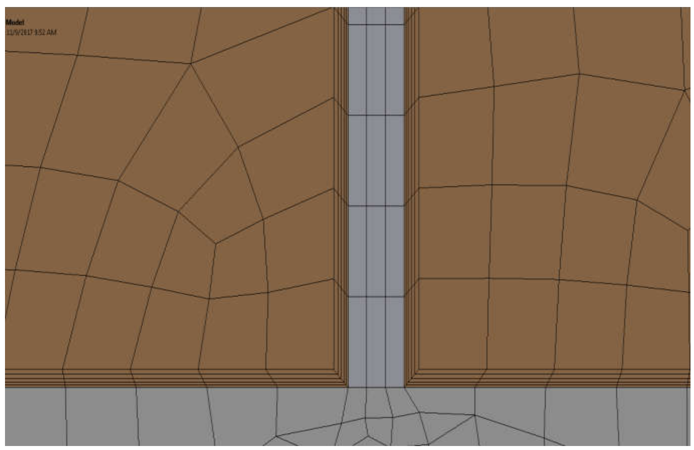

Fig.5 Enlarged Cross Sectional View

B. CFD Simulation Approach

- The mesh file was imported in to ANSYS FLUENT for simulation set-up.
- The pressure-based CFD solver was selected for the simulations.
- The flow turbulence was resolved using Reynolds Averaged Navier Stokes (RANS) approach using the two equation model Realizable k-epsilon with Enhanced wall functions.
- Air with material properties at standard conditions was assigned for the simulations.

C. Boundary Conditions

- The flow inlet was modeled using the ‘velocity-inlet’ boundary condition with velocity based on the Reynolds number
- The heat input of 125 Watt is applied to the flat plate was specified using the ‘constant wall heat flux’ boundary conditions
- ‘Pressure-Outlet’ boundary condition was assigned for the Duct outlet to model the flow outle

- The remaining surfaces - top, bottom and sides of the duct – were modeled as adiabatic, Stationary, No-Slip wall boundary condition.

C.CFD Post Processing

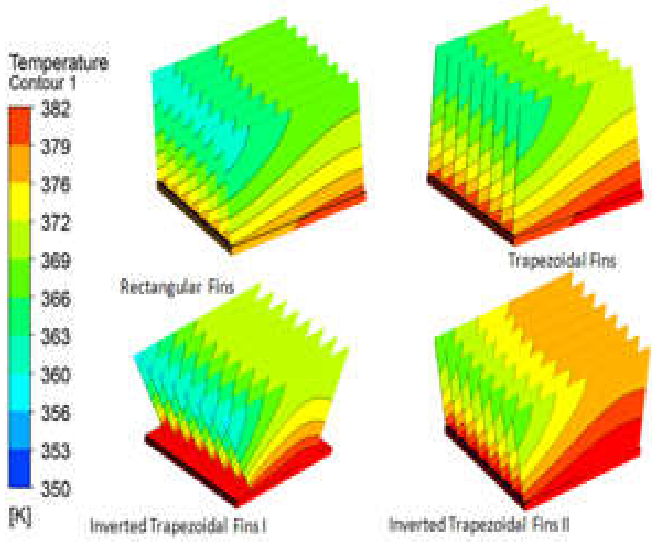

Fig.6 Temperature Contour at Re = 4000and Q=125 W

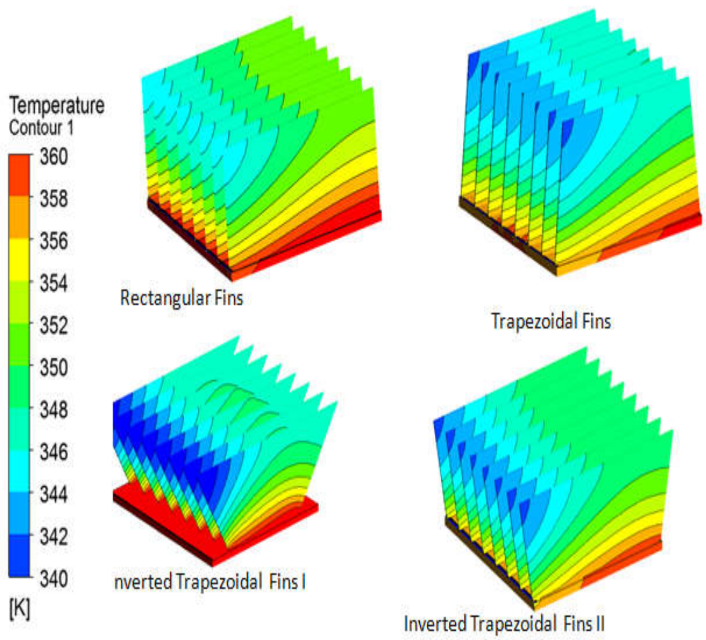

Fig.7 Temperature Contour at Re = 8000 and Q= 125 W

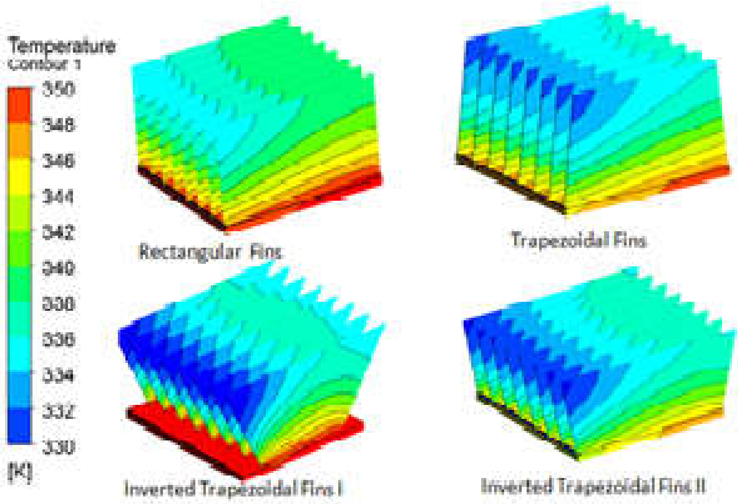

Fig.8 Temperature Contour at Re = 12000and Q=125 W

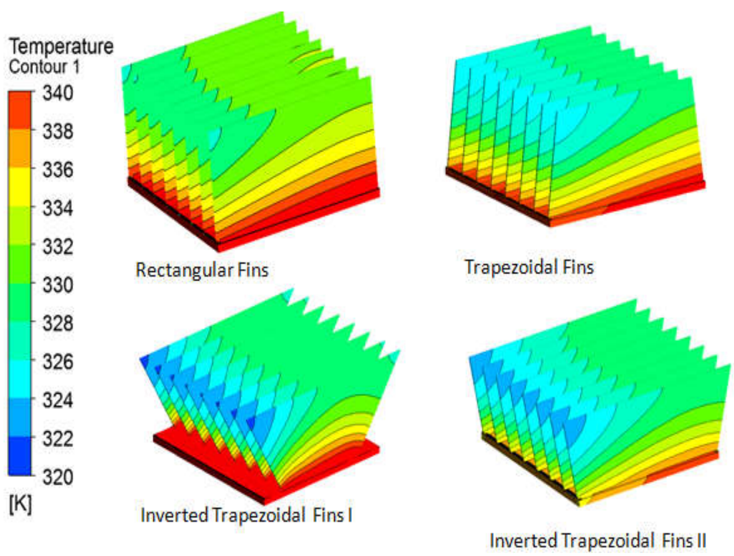

Fig.9 Temperature Contour at Re = 16000and Q=125 W

Fig. 6, 7,8,9 shows temperature contours for Rectangular, Trapezoidal , Inverted Trapezoidal Plate fin array at Reynolds No. 4000, 8000, 12000 and 16000 respectively and heat input is 125 W. From these temperature contours is observed that at Re = 4000, maximum surface temperature is 382^0 C, 360^0C at Re = 8000, 350^0 C at Re= 12000 and 340^0 C at Re= 16000. It can be concluded that at Re= 4000 surface temperature is higher and Inverted trapezoidal fins shows higher surface temperature than Rectangular and Trapezoidal fins but as Reynolds No. increased to 8000 and so on maximum surface temperature get lowered in case of Inverted trapezoidal fins in comparison with rectangular and trapezoidal fins.

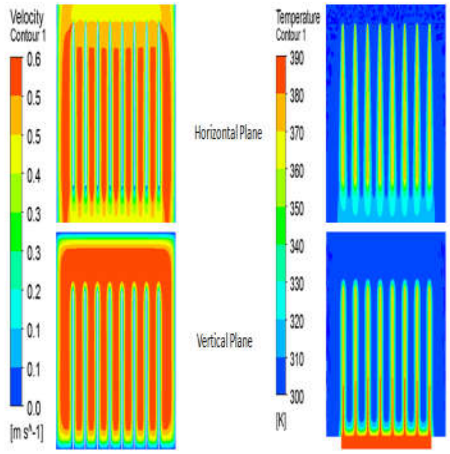

Fig.10 Velocity and Temperature Contour for Inverted Trapezoidal Plate FinsII at Re = 4000 and Q=125 W

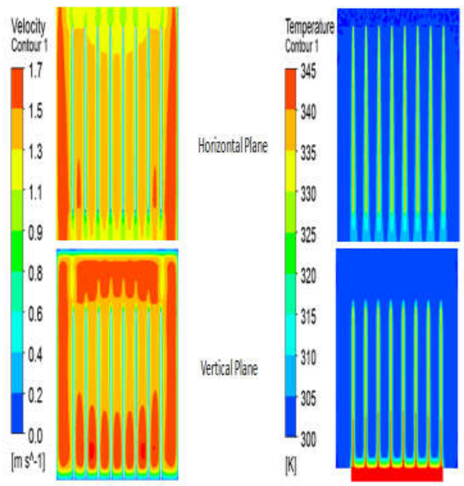

Fig.11 Velocity and Temperature Contour for Inverted Trapezoidal Plate Fins II at Re = 12000 and Q=125 W.

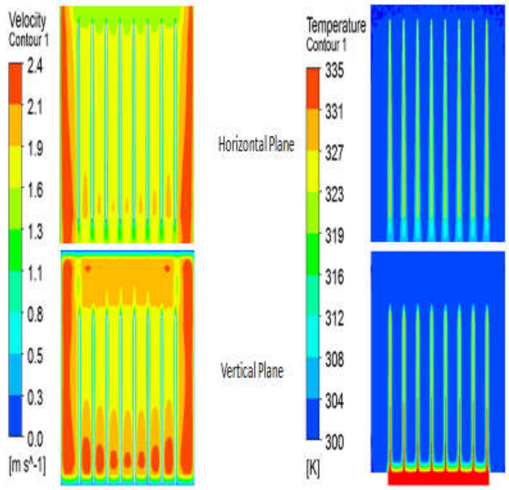

Fig.12 Velocity and Temperature Contour for Inverted Trapezoidal Plate Fins II at Re = 16000 and Q = 125 W.

III. RESULT and DISSCUSSION

A. *Variation of Heat Transfer Coefficient with Re*

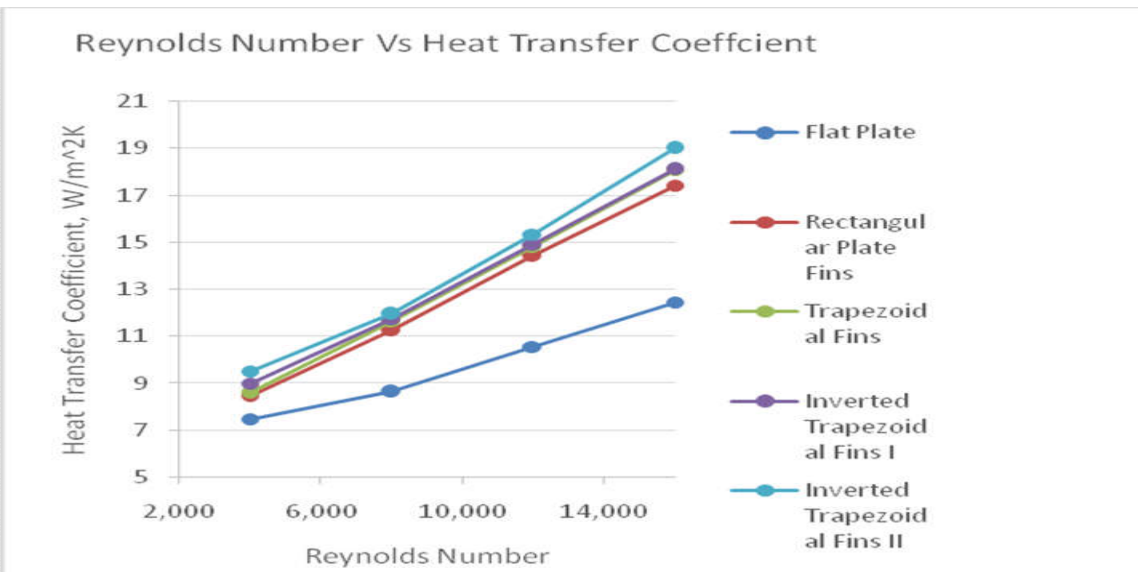

Fig.13 h Vs Reynolds No. at Q = 125 W

Fig.13 shows variation of heat transfer coefficient with respect to Reynolds No. It was found that as Reynolds No. increases convective heat transfer coefficient also increases. Average increase in heat transfer coefficient of RPF, TPF, ITPF-I, and ITPF-II over Flat plate was 22.5 %, 24.6%,
25.89 % and 28.66 % respectively.

B. Variation of Nu with Re

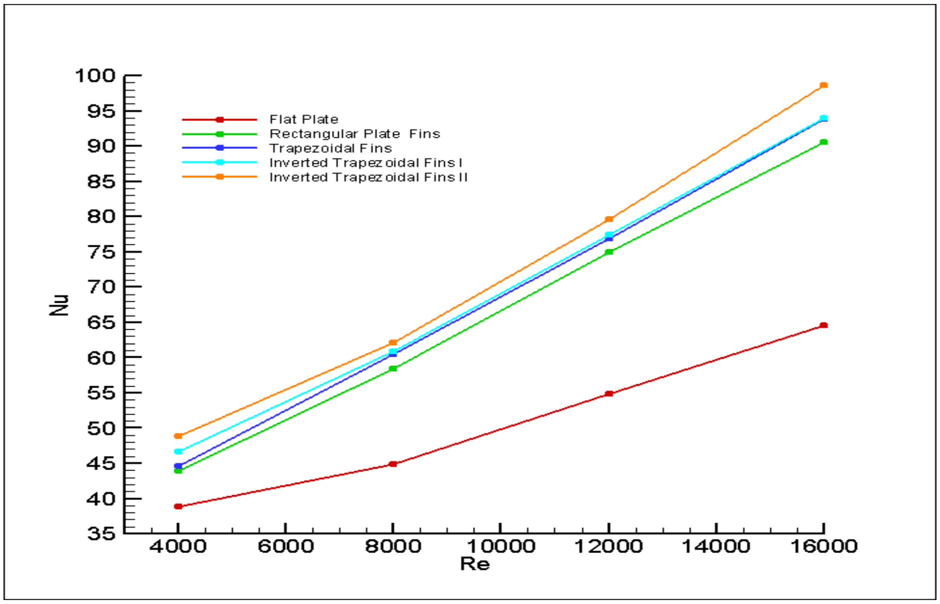

Fig. 14 Nusselt No. Vs Reynolds No. at Q= 125 W

Fig.14 shows variation of Nu with respect to Re. It was observed that as Re increases Nu also increases.
Average increase in Nu of RPF, TPF, ITPF-I, and ITPF-II over Flat plate was 22.5 %, 24.6%, 25.89 % and 28.66 % respectively. There was an increase observed in TFHS and it was around 2.75 %. Also an enhancement of 4.28 % of ITPF-I over RPF and 7.57 % of ITPF -II over RPF. Further an enhancement of 4.94 % of ITPF-II over TPF was observed via numerical analysis.

C. Variation of Pressure Drop with Re

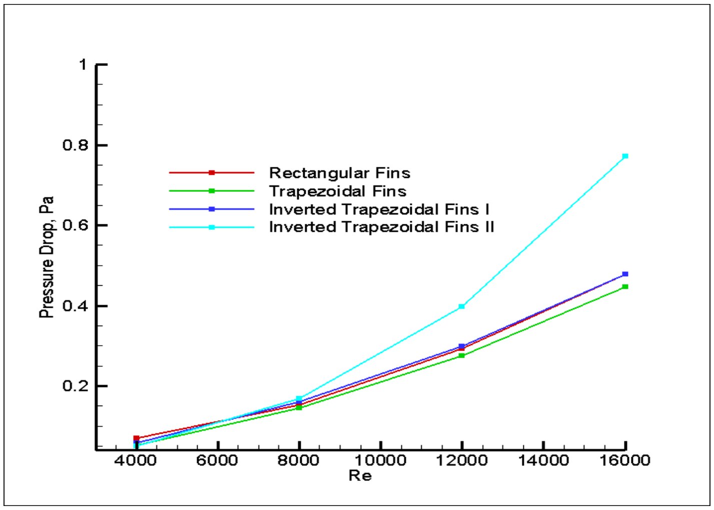

Fig. 15 Pressure drop Vs Reynolds's No. at Q= 125 W

Fig. 15 shows variation of Pressure drop with respect to Re. As Re increases pressure drop also increases. Pressure drop is directly proportional to friction between fluid and solid walls. Increase in pressure drop also leads to increase in Pumping Power. From graph it was observed that ITPF-II has higher pressure drop in comparison with Flat plate, and Array with RPF, TPF and ITPF-I.

IV. CONCLUSIONS

After doing a numerical analysis using Flat Plate, RPFHS, TPFHS, ITPFHS at 125 W and for the selected range of Re i.e 4000 , 8000, 12000 and 16000 following conclusions are made.

1. As Reynolds No. increases Nu also increases in case of all geometries.
2. ITFHS-II gives better performance and % increase of 7.57 was over RFHS and 4.94 % over TFHS.
3. Pressure drop is higher in case of ITFHS-II in comparison with other geometries.

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