Enhancement of Heat Transfer Effectiveness of Plate-pin fin heat sinks With Central hole and Staggered positioning of Pin fins

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Abstract - *The study is focused on searching for alternative models for improving the heat transfer, reducing the pressure drop, decreasing the heating plate base temperature, reducing thermal resistance of plate pin fin heat sink. To enhance the heat transfer rate increasing the surface area was the earliest approach but it has its own limit. So it is important to search for alternate solutions. This study introduces some new models with high heat transfer and less pressure drop.*

***Key Words*: Heat Transfer Coefficient, Nusselt Number, Pressure Drop.**

1. INTRODUCTION

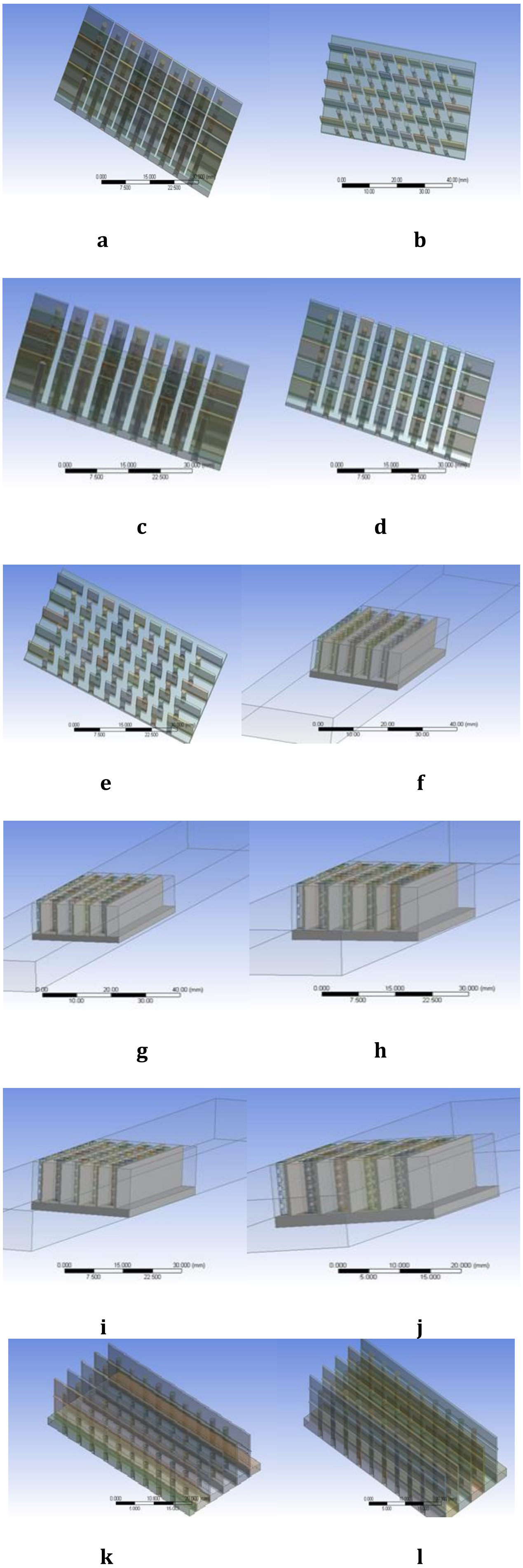
Many electronic components have critical parts which must be cooled in order to control overheating problems. So the thermal management of electronic components is an important matter of concern. Conventional electronics cooling uses heat sink with forced air cooling. In order to design a practical heat sink the factors to be taken into consideration are large heat transfer rate, a low pressure drop, less fan power and a simpler structure. Where the fan power is the power required to effectively create a steady heat transfer from the fin to the atmosphere. Less pressure drop means less fan power is required.

When literature survey was conducted it was found that a number of people have researched about the thermal and hydraulic performance of heat sink. Xiaoling Yu [1] conducted a study on development of a plate-pin fin heat sink and its performance comparisons with a plate fin heat sink. Numerical simulations and some experiments were conducted to compare thermal efficiency of these two types of heat sinks. The simulation results showed that thermal resistance of a PPFHS was about 30% lower than that of a PFHS used to construct the PPFHS under the condition of equal wind velocity. M.R. Shaeri [2] studied heat transfer analysis of lateral perforated fin heat sinks. Results showed that new perforated fins have higher total heat transfer and considerable weight reduction in comparison with solid fins. Tingzhen Ming [3] conducted a study on numerical simulation of the thermal hydraulic performance of a plate pin fin heat sink. The simulation shows that pin height and air velocity have significant influences on the thermal hydraulic performances of PPFHS while the influences of in-line or staggered array and neighbour pin flow-directional center distance (NPFDCD) of the PPFHS are less notable. Han-Taw Chen [4] conducted an investigation of heat transfer characteristics in plate-fin heat sink. An interesting finding is that the calculated fin temperatures obtained from the commercial software are in good agreement with the experimental temperature data at various measurement locations. P.A. Deshmukh [5] studied thermal performance of elliptical pin fin heat sink under combined natural and forced convection. The study resulted in the successful development of generalized empirical correlations for elliptical profiled pin fin heat sinks having capability of predicting the influence of various physical, thermal and flow parameters on the air side performance. Amer Al-Damook [6] conducted an experimental and computational investigation of thermal air flows through perforated pin heat sinks. It shows that the Nusselt number increases with the number of holes in pin p. The pressure drop and fan power required to overcome the pressure drop all reduces with increase in number of holes. Pins with five holes are shown to have a 11% larger Nusselt number than that of solid pin cases. This improvement arises due to not only due to the increase in surface area but also heat transfer efficiency improvement near perforations through the formation of localised air jets. Dipankar Bhanja [7] conducted computational study on the improvements of heat transfer efficiency of heat sinks using perforated pin fins with staggered arrangement and inline arrangement. Results showed that the heat transfer rate of perforated fins up to certain perforation number and size are always higher than the of the solid ones and with the variation in the fin shape. With changes in perforation geometry heat transfer rate improves to a great extent. In this study different methods are introduced to improve the efficiency of PPFHS by improving several factors that have a role in efficiency. The results were analysed based on five different factors such as $h/\Delta p$ ratio, Nusselt number, base plate temperature, fan power, thermal resistance.

2. NUMERICAL MODEL

2.1 Physical Geometry

The geometric model of different plate pin fin heat sinks are shown in figure 1. Air is passed through the plate pin fin heat sink with pin diameter 1.5 mm, pin and plate height 10mm, total width of the domain 31.5 mm, heat sink width 26.4 mm, total length of the heat sink is 52.8 mm, base plate thickness 2 mm. 19 different models of plate pin fin heat sinks were created. The material of the heat sink is aluminum. The bottom of the computational domain is heated with a constant flux of 3587 W/m². Three domains were defined for the model, two fluid domains and one solid domain.



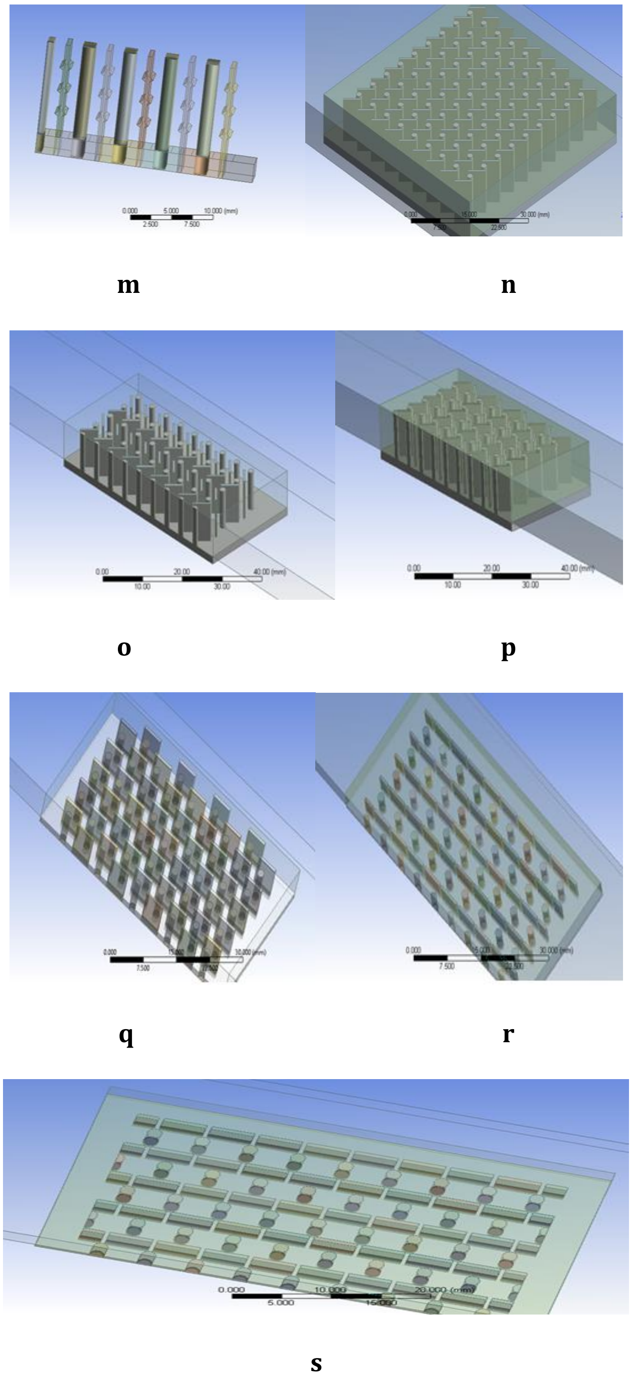


Fig -1: Geometry of the models: (a) cut plate 2D/3 (b) cut plate 2D/3 (c) cut plate D (d) cut plate 4D/3 (e) cut plate 5D/3 (f) one hole (g) two hole (h) three hole (i) four hole (j) five hole (k) single corrugated (l) double corrugated (m) triple corrugated (n) angle plate type 1 (o) angle plate type 2 (p) angle plate type 3 (q) staggered pin & staggered plate (r) inline pin & inline plate (s) staggered pin & inline plate.

2.2 Governing Equations

Using commercial software ANSYS Fluent 15.0 [9], governing equations for flow conditions and heat transfer were numerically solved based on following assumptions: three

dimensional, steady-state, incompressible fluid flow. Radiation effect is ignored. The resulting governing equations are:
Continuity equation:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}+\frac{\partial W}{\partial Z}=0 \qquad (1)$$

Momentum equation:

$$\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}+W\frac{\partial U}{\partial Z}\right)=-\frac{dP}{dx}+\frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}+\frac{\partial^2 U}{\partial Z^2}\right) \qquad (2)$$

$$\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}+W\frac{\partial V}{\partial Z}\right)=-\frac{dP}{dY}+\frac{1}{Re}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}+\frac{\partial^2 V}{\partial Z^2}\right) \qquad (3)$$

$$\left(U\frac{\partial W}{\partial X}+V\frac{\partial W}{\partial Y}+W\frac{\partial W}{\partial Z}\right)=-\frac{dP}{dZ}+\frac{1}{Re}\left(\frac{\partial^2 W}{\partial X^2}+\frac{\partial^2 W}{\partial Y^2}+\frac{\partial^2 W}{\partial Z^2}\right) \qquad (4)$$

Energy equation:

$$\left(U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}+W\frac{\partial \theta}{\partial Z}\right)=\frac{1}{Re.Pr}+\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}+\frac{\partial^2 \theta}{\partial Z^2}\right) \qquad (5)$$

For the solid region of the plate pin fin heat sink the steady state energy equation is

$$\frac{\partial^2 \theta}{\partial X^2}=0 \qquad (6)$$

In the above equations the dimensionless parameters are set as follows:

$$X=\frac{x}{Dh},Y=\frac{y}{Dh},Z=\frac{z}{Dh},U=\frac{u}{u_{in}},V=\frac{v}{u_{in}}$$

$$P=\frac{p}{\rho u_{in}},W=\frac{w}{u_{in}},\theta=\frac{T_f-T_{in}}{T_w-T_{in}}$$

2.3 Boundary Conditions

The momentum boundary condition of no slip and no penetration is set for all the solid walls. Mass flow inlet and pressure outlet boundary were used for both the fluid flows. Table 1 lists the boundary conditions used in the present study.

Table -1: Boundary conditions

Parameter	Value
Inlet velocity	1 m/s - 5.2 m/s
Inlet temperature	293 K
Heat flux	3587 W/ m²

3. CFD SIMULATION

The governing equations were solved by the control volume approach using commercial software ANSYS Fluent 15.0 [9]. The double precision option was adopted for all computations. A second-order upwind scheme was employed to discretize the convection terms, diffusion terms, and other quantities resulting from the governing equations. A staggered grid scheme was used, in which the velocity components are evaluated at the control volume faces, while the rest of the variables governing the flow field are stored at the central node of the control volume. The pressure-velocity coupling was handled with the SIMPLE scheme for pressure linked equations. Ansys fluent solves the linear systems of equations obtained from discretization schemes using a point implicit linear equation solver, in connection with an algebraic multi-grid methodology. For all numerical simulations performed in this particular study, converged solutions were considered when the normalized residuals resulting from an iterative process for all governing equations were lower than 10^{-4}.

3.1 Grid Independence Study

Four different meshes were considered for establishing the grid independence of the model under the present study. In each case, the pressure drop between inlet and outlet was recorded and tabulated as shown in table 2 to achieve the grid independency. From the tabulated results, it was concluded that grid independency was achieved for mesh-3 and this mesh was used for further numerical computations. The computational domains are meshed with face meshing and it contains hexahedral elements. Figure 2 shows the meshed view of the generated model.

Table -2: Grid independence study

Grid	Pressure drop (Pa)
Mesh-1 (1,12,688)	7.6598
Mesh-2 (2,22,677)	7.9591
Mesh-3 (4,26,009)	8.28022
Mesh-4 (8,01,139)	8.28024

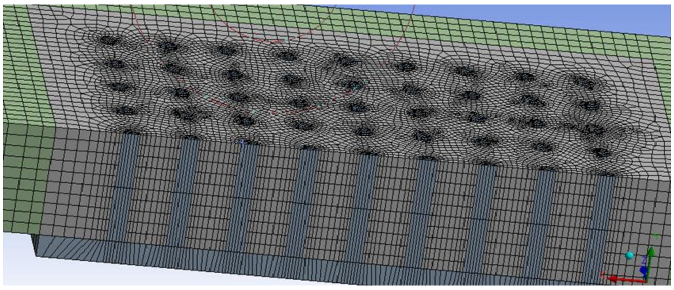


Fig -2: Meshed view of the generated model.

3.2 Model Validation

The numerical results obtained by present study were compared with the experimental results of H. Jonsson and B. Moshfegh [8] to validate the numerical model. H. Jonsson and B. Moshfeg [8] carried out experimental study to investigate the heat transfer characteristics of pin fin heat sinks. Figure 3 shows the comparison of pressure drop for current numerical study and experimental investigation of H. Jonsson and B. Moshfeg [8]. The numerical results showed good agreement with the experimental results. The maximum deviation of the numerical results from the experimental study in the literature was less than 5%.

The thermal resistance values obtained from the current numerical study were validated by comparing them with the experimental results of H. Jonsson and B. Moshfeg [8] in the literature. Figure 4 shows the comparison of thermal resistance for the current numerical study and the experimental study done by H. Jonsson and B. Moshfeg [8]. The maximum deviation for numerical results was found to be less than 9% from the experimental study. Therefore, the present results are in good agreement with the results reported H. Jonsson and B. Moshfeg [8] thus validating the numerical model used in the present study.

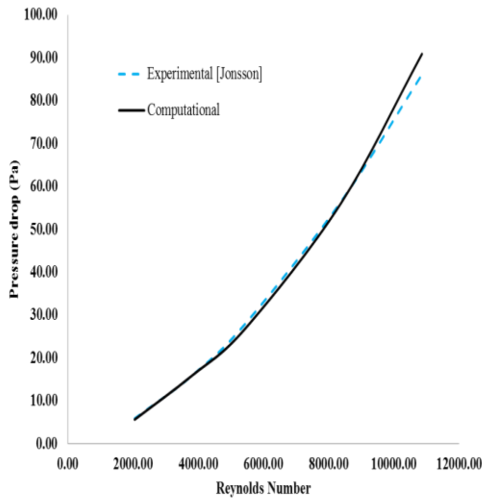


Fig -3: Variation of pressure drop with Reynolds number

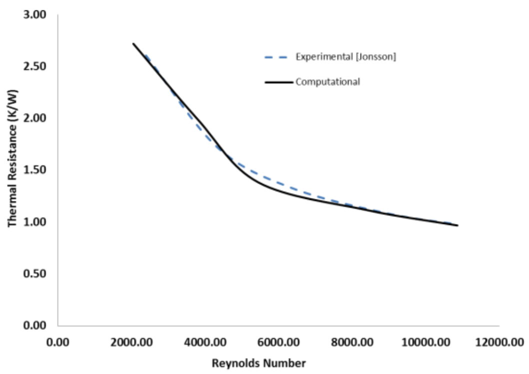


Fig -4: Variation of thermal resistance with Reynolds number.

4. RESULTS AND DISCUSSIONS

Heat transfer performance and flow dynamics in a plate pin fin heat sink was evaluated in terms of heat transfer coefficient, Nusselt number, pressure drop, base plate temperature, fan power and thermal resistance.

4.1 Comparison of h/Δ p ratio

Larger h/Δp ratio means larger heat transfer coefficient with minimum pressure drop. When numerical simulation was conducted on 19 different models of PPFHS it was found that the PPFHS model with five holes has larger h/Δp ratio. Figure 5 shows the variation of h/Δp ratio with Reynolds number. The minimum value of h/Δp is obtained for the angle plate model type 2. The reason for this is the high pressure drop due to the hindrance created by angle plate to the free flow of air. The other notable conclusion is that among the cut plate models the cut plate model with D/3 cut thickness has higher h/Δp ratio. Among the corrugated model the single corrugated model shows higher h/Δp ratio. Inline – inline combination of plate and pin shows higher h/Δp ratio than staggered – staggered combination of plate and pin.

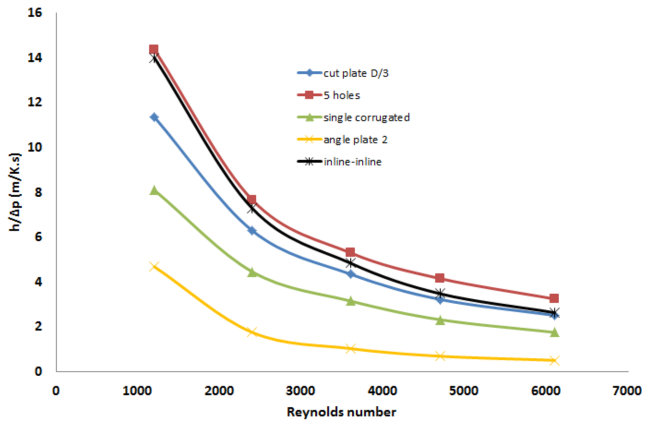


Fig -5: Variation of h/Δp with Reynolds Number

4.2 Comparison of Nusselt Number

Nusselt Number is a dimensionless number which gives the direct measure of the heat transfer. If the Nussselt number is high it means that the heat transfer is high. Figure 6 shows the variation of Nusselt number with Reynolds number. When all the 19 different models of PPFHS were analysed it was found that the higher Nusselt Number is obtained for the cut plate model with plate cut thickness D/3. Where D is the diameter of the pin fin. Least value of Nusselt number is obtained for single corrugated model. Among the holed models the model with two holes has the highest Nusselt number. When staggered and inline models were considered staggered-staggered combination has higher Nusselt number.

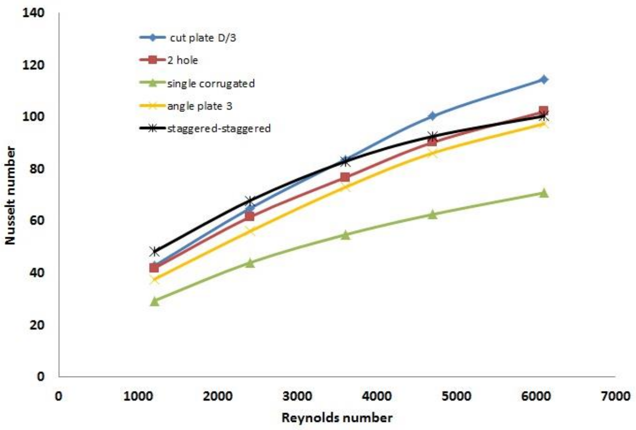


Fig -6: Variation of Nusselt number with Reynolds number

4.3 Comparison of heating plate base temperature

Heating plate is the bottom portion of the heat sink where the heat flux is applied. Lower heating plate base temperature indicates that large amount of heat has been dissipated from the surface of heating plate base. Figure 7 shows the variation of heating plate base temperature with Reynolds number for different models of plate-pin fin heat sinks. When the heating plate base temperature of nineteen different models were analysed it was found that the split plate model with plate cut thickness D/3 is found to have lower base plate temperature. Among the corrugated models triple corrugated model shows the least base plate temperature. Among the inline and staggered model the staggered pin and staggered plate combination shows least heating plate base temperature. If the angle plate models are compared the angle plate model of type 3 shows lower heating plate base temperature when compared to angle plate model of type 1 and type 2. When the holed models were compared least value of heating plate base temperature is shown by the model with two holes. It was also found that with increase in Reynolds number the base plate temperature value decreases.

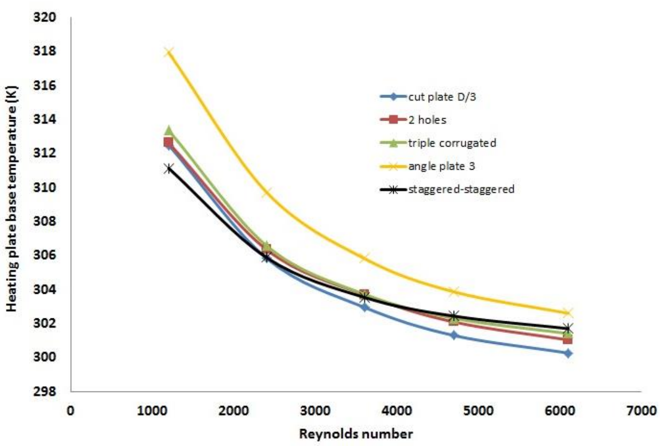


Fig -7: Variation of heating plate base temperature with Reynolds number

4.4 Comparison of Thermal Resistance

Thermal resistance is the heat transfer analogue of electrical resistance. For a system if thermal resistance is low the heat flow process will be very smooth. Figure 8 shows the variation of heating plate base temperature with Reynolds number. When all the models were studied it was found that least thermal resistance is obtained for model with cut plate thickness D/3 and largest value of thermal resistance is obtained for angle plate model of type 3.

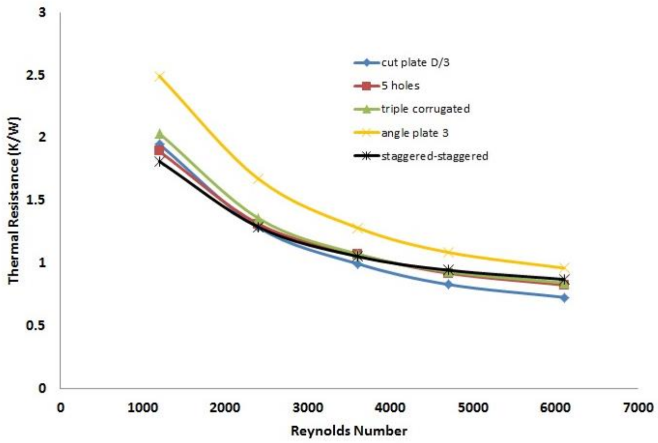


Fig -8: Variation of thermal resistance with Reynolds number

4.5 Comparison of fan power

Fan power is the power required to keep the required pressure drop. Fan power is the product of velocity, area and pressure drop. Figure 9 shows the variation of fan power with Reynolds number. When the 19 different models were analysed it was found that five holes model have least fan power. Higher value of fan power is obtained for the angle plate model of type 2. The reason for getting higher fan power for this model is that it has very high pressure drop when compared to other models, so higher fan power is required to maintain the flow of fluid.

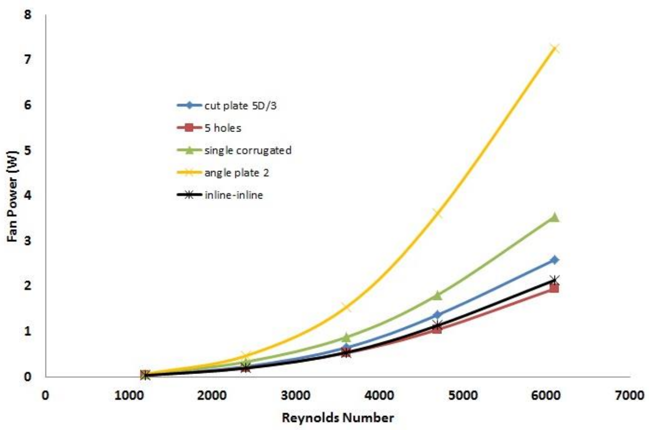


Fig -9: Fan power Vs Reynolds number plot.

5. CONCLUSIONS

The five holes model has larger h/Δp ratio. The least base plate temperature is obtained for cut plate model D/3.The least thermal resistance is also obtained for cut plate model with cut thickness D/3. Fan power is the power required to keep the required pressure drop. The highest fan power is obtained for angle plate type 2, it is due to higher pressure drop. The least fan power is obtained for five holes model. D/3 is the optimum cut thickness for getting maximum Nusselt number. Increasing cut thickness beyond D/3 reduces Nusselt number. The triple corrugated model which has 35 % more surface area than staggered pin and staggered plate model has lower Nusselt number than the staggered pin and staggered plate model. Also cut plate model with D/3 cut thickness and which has 32 % lower surface area than triple corrugated model has higher Nusselt number values than that of triple corrugated plate. So these two models show higher heat transfer efficiency than triple corrugated model. Hence they are cost effective. This clearly depicts heat transfer efficiency always does not improves by merely increasing the surface area. The creation of turbulence and removal of unwanted obstruction of free flow of air has also a major role in improving heat transfer efficiency.

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