Numerical Study and Thermal Performance of Rectangular Heat Sink

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Abstract— Rate of heat transfer is having practical interest in almost all the engineering applications such as heat exchangers, boilers, electronic chips etc. the heat transfer problems encounters in practice is of two kinds, such as rating and sizing problem. The rating problem deals with the rate of heat-transfer through the systems while the sizing problems deals with the determination of the size of the system in order to transfer heat at a specific rate for a specified temperature differences. The scope of this project falls under rating problem where the analysis is done for the optimal geometry for the enhanced heat transfer rate if the rectangular heat sink is extensively used in this study for the initial problem statement and validation. Similar study is also done for the different geometries. This report consists of the details of the progress done in this project.

Key words: Rectangular Heat Sink, Numerical Study

I. INTRODUCTION

The heat sink is the heart of the electronic system. Like a heart in a human body, Heat sink is a device which removes heat from high temperature to low temperature. Heat sinks are extensively used in industries in order to raise the rate of heat transfer on which forced or free convection can occur. They are found in many electronic devices like high performance video cards and microprocessors. In many cases, heat sinks are coated with a paint which has high emissivity to further increase the heat transfer rate. The current work provides a numerical approach to solve the problem.

The manufacturing process of heat sink is usually done by bonded, skived, cast, stamped, extrusion processes. The heat sinks are elements that prevent the destruction of electronic equipment because of its overheating. The most critical part in an electronic device is the semiconductor junction. The junction temperature can't exceed a temperature which is given by the manufacturer. The heat sinks have different shapes depending on the nature of the coolant fluid (natural air convection cooling, forced air convection cooling, liquid cooling.

II. DESIGN CONSIDERATIONS FOR HEAT SINK

The designing parameters are important for the design of heat sink include are the number of fins, heat sink material, type of geometry and its arrangement and the plate thickness as shown in Fig 1.1.To attain the least thermal resistance and pressure drop, all these designs have to be planned before designing a Heat Sink.

A. Heat Sink Material

Heat sinks are usually done by a excellent thermal conductor like aluminium or copper alloy. Copper is considerably more comfortable than aluminium which is purely produced by extrusion. Aluminium is lighter than copper, which offers a lesser amount of mechanical stress on electronic Equipments. Several heat sinks are prepared by aluminium copper core as shown in Figure1.2

Copper is said to be an excellent heat sink properties on the basis of thermal conductivity, and corrosion resistance. Thermal conductivity of Cu is double as aluminium more competent heat dissipation is possible. Some of the applications include are industrialized applications they are gaswater heaters, solar water heater HVAC systems, power plants, geothermal heating and cooling, and electronic systems.

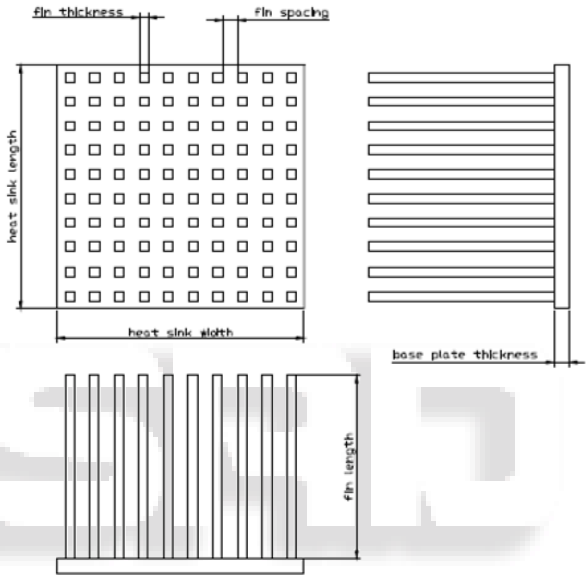

Fig. 1: Design arrangements for heat sink

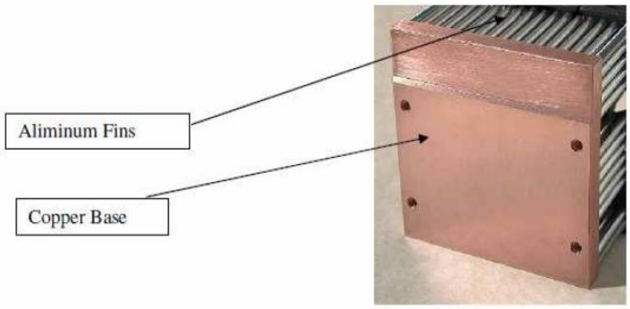

Fig. 2: Aluminium copper core

Thermal conductivity of Cu is double as Al and quicker, more competent heat dissipation is possible. Some of the applications include are industrialized applications like gas water heaters solar water heaters. HVAC systems, power plants, geothermal heating and cooling, and electronic systems.

B. The Number of Fins

The heat sink mainly contains the base which acts as support and supplementary flat surface and an assortment of comb like structures to enhance the heat sink's external area contact with the air, this will raise the heat dissipation rate. It is really an important parameter for heat sink. The heat sink is essential for cooling of electronic components

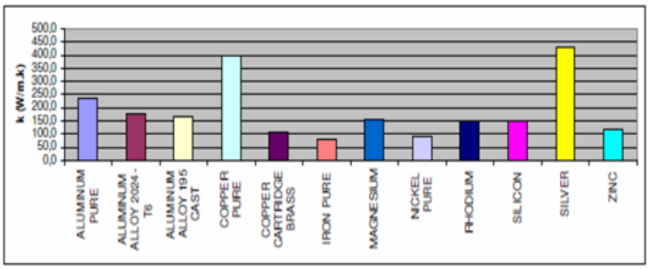

Fig. 3: Thermal conductivities of common heat sink material

C. *Fin Shapes*

There will be diverse kinds of heat sink geometries are possible. Fins whose configuration may be straight or tapered, Pin fins, splines and fins with annular c/s can also possible. On the whole common ones are pin fins whose cross-section is varying that may be square, round, hexagonal, elliptical or several other appropriate geometry. Straight fins that are having rectangular cross sections are extensively used.

III. PROJECT DESCRIPTION

A. *Geometry Selection*

Di Zhang [2] and his associates have analysed that the contribution of dimples and ribs in the rectangular heat sink enhances the heat transfer rate. However, their geometries are purely based on the spherical structures for the heat transfer estimation. In this project, other aspects of the geometries are also considered and based on the mathematical modelling the models are selected for numerical analysis.

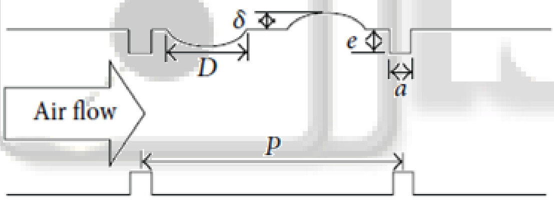

Fig. 4: Cross section of geometry

Case name	Design	D(mm)	S(mm)
Rectangular geometry	Dimple+protrusion	15	20
Spherical geometry	Dimple+protrusion	15	20
Cylindrical geometry	Dimple+protrusion	15	20

Table 1: Cases

From the points above

1) a=4mm ; e=4mm
2) P=4x11=44mm
3) D=15mm
   on accordance with manufacturing consideration and overall effects of dimensions on the heat transfer, however 4mm value of gives a significant curvature
4) Diameters of the cylindrical and spherical geometry are taken as 15mm.

Note:

All the geometry is hainvg base rectangular section of 140mmx30mm in cross section

According tor's Law, the rate of heat conduction can be given as Qin = -KA dt/dn

where n is the direction of heat transfer. In Cartesian co-ordinates it can be written as x, y and z directions respectively.

If it is seen in details it is observed that the parameters specified by reference 2 follows typically the heat transfer through the fluid (air). Air at the lower boundary layer region (at wall surface) is motionless (No-Slip Condition) while velocity gradually increases as we move away from the boundary layer.

Heat flux in either case is for conduction Q= - K dT/L and for Convection q= h A dt . Their ratio gives the Nusselt no. The sphere is known for the least surface area for its given volume. While the analysed case in reference contains all the spherical parts as the fins. Let us take an example for the comparison.

For $1m^3$volume the sphere has the surface area of $4.83m^2$. While for the same volume the cylinder (l=d) and cube have surface area of $5.85m^2$ and $6m^2$ respectively. This in turn proves that the heat transfer rate will be increasing if we go for cylindrical and cubical structures of the fins.

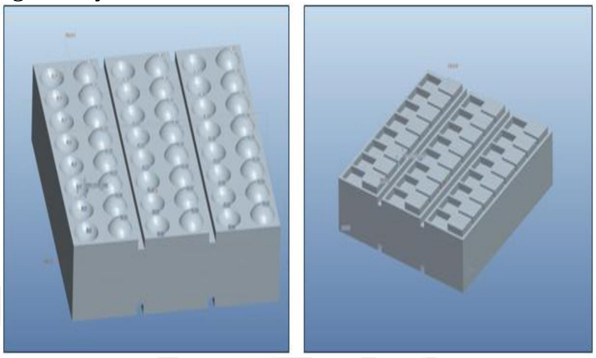

Fig 5: Shows spherical and rectangular geometry

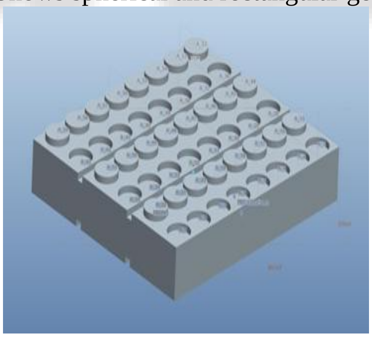

Fig 6: Shows cylindrical geometry

However based on the cases in reference 2 the additional geometries are also made as shown below.

While meshing and analysing the cases, it is observed that these three cases are well suited for the analysis because of the following reasons:

There are few papers available to validate the results of these geometries.

Finally, based on the reference papers and basic knowledge of heat transfer three geometries are selected for the analysis.

B. *Meshing:*

The computational cost is very high for sufficiently high mesh resolution and complete geometry. According to reference 2, for around 3 million cells the sufficiently dependable results are found and it took around 48 hrs. To complete one case with eight Q8200/2.33GHz processors used. This high-end computing is unavailable at this point of time, hence the mesh is generated for higher resolution but the simulation is done for relatively coarser resolution. For saving the computational cost, out of the total geometry a small periodic element is selected. The geometry is made translational -periodic in relevant direction. All the periodic meshes are shown in the figures below.

IV. RESULTS AND DISCUSSIONS

A. *Post Processing of the Results (Geometry 1: Spherical; Re=10000)*

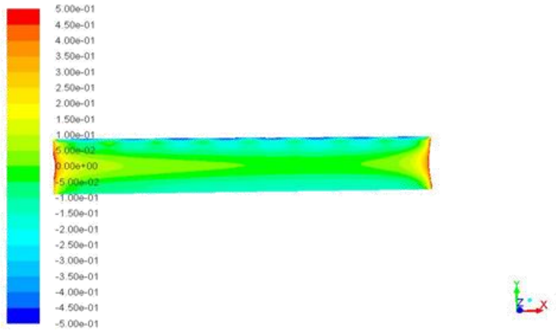

Fig. 7: Contours of static pressure at inlet

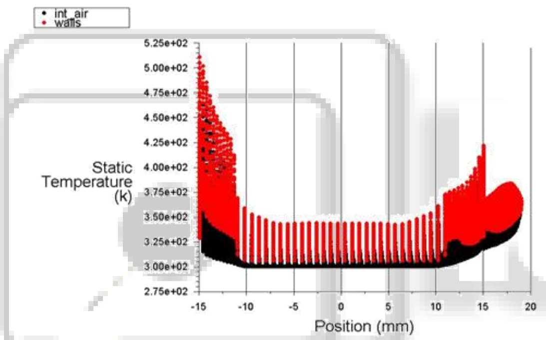

Fig 8: Static temperature at x direction

The figure above shows that the walls are nearly at 525K while the air in the geometry flows at around 450K

B. *Post processing of the results: (Rectangular; Re=10000)*

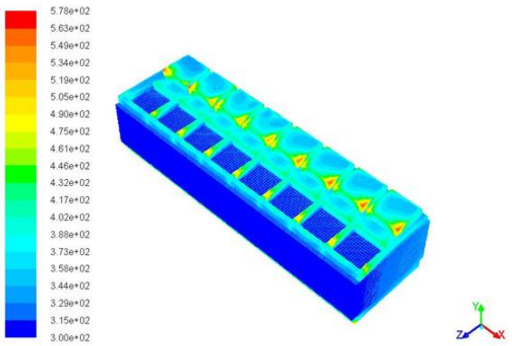

Fig 9: Contours of static temperature

The Reynolds number is defined by

$$Re = \frac{\rho U_{in} D_h}{\mu}$$

Where, U_{in} is inlet average velocity, D_h is hydraulic diameter given by

$$D_h = \frac{2WH}{W+H}$$

The local Nusselt No. is defined by

$$Nu_x = \frac{h_x D_h}{\lambda}$$

Where λ is thermal conductivity of air

Local heat transfer coefficient is defined by

$$h_x = \frac{q''}{\Delta T_x}$$

Where q''= heat flux. ΔT_x is local temperature difference between the wall and air.

Fanning factor f is defined as:

$$f = \frac{(\Delta p/L)Dh}{2\rho U_{in}}$$

Where Δp is pressure drop and L is stream wise channel length of computational domain.

The thermal performance is defined as

$$TP = \left[\frac{Nu}{Nu_0}\right] \cdot \left[\frac{f}{f_0}\right]^{-\frac{1}{3}}$$

The baseline Fanning friction factor is calculated by

$$f_0 = 0.046\,Re^{-0.2}$$

The baseline Nusselt No. is given by

$$Nu_0 = 0.023Re^{0.8}\mathrm{Pr}^{0.4}$$

C. *Analysis for spherical case*

1) Reynolds number is same (Re=10000) for all the cases mentioned above.

$$\rho = 1.225\frac{kg}{m^3}\ (Standard\ Assumption)$$

$$U_{in} = 3.135\ (According\ to\ the\ cross - section\ of\ the\ geometry)$$

$$\mu = 1.789 \times 10^{-5}\frac{kg}{m-s}$$

The Hydraulic diameter is given by $D_h = 0.0465$

For geometry 1 phere) bulk mean temperature of air is around 350°C

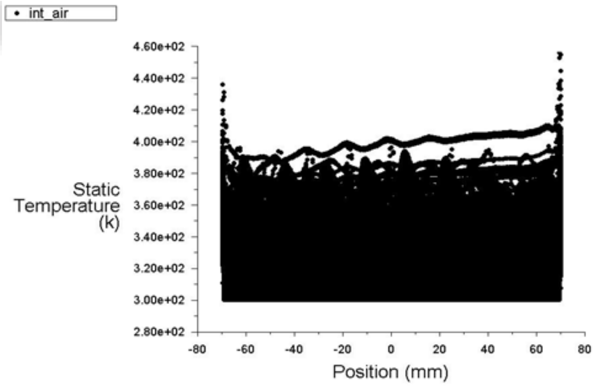

Fig 10: Static temperature(internal air)

While all the walls are facing the same amount of heat flux (q=1500w/m^2), the average wall temperature can safely be assumed as 380°C

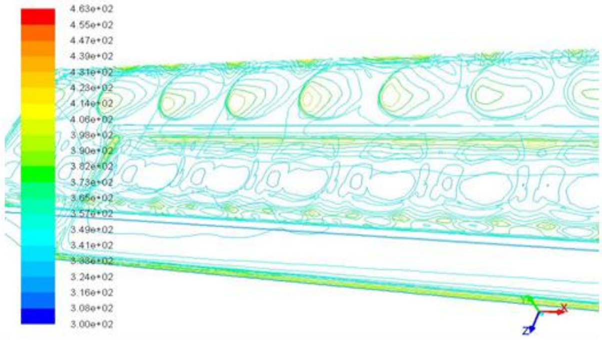

Fig 11: temperature static contours

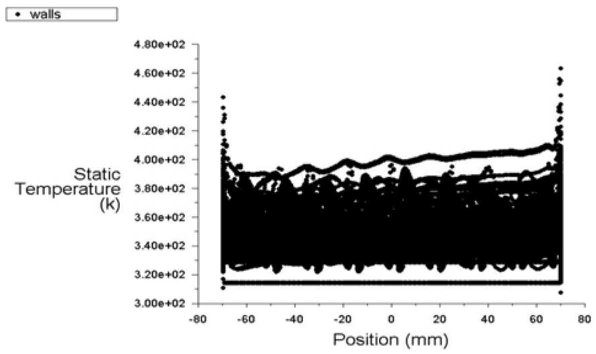

Fig 12: static temperature plot

$$h_x \approx \frac{1500}{350 - 340} \approx 150$$

Conductivity if air is varying from 0.0454 w/mk at 300^0C to 0.0515 w/mk at 400^0C. Taking the safer side, assuming the conductivity of air 0.0485 w/mk at 350^0C. Nusselt No. can be calculated as

$$Nu_x = \frac{150 \times 0.0465}{0.0485} = 143.81$$

2) Fanning friction factor

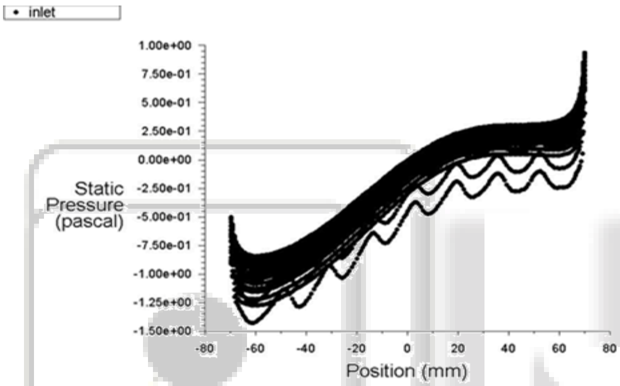

Fig 13: inlet static pressure

The average static (gauge) pressure at inlet is 0Pa.

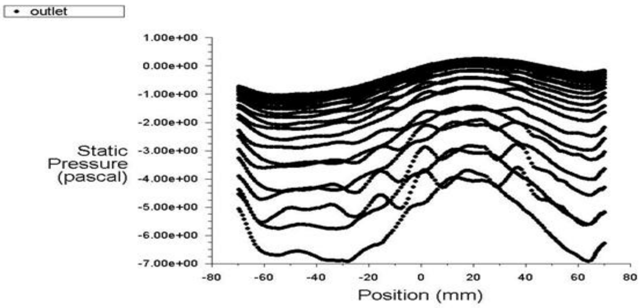

Fig 14: static outlet pressure

The average static (gauge) pressure at outlet is -3Pa.

$$f = \frac{\left(\frac{(0--3)}{(40\times10^{-3})}\right) \times 0.0465}{(2 \times 1.225 \times 3.135^2)} = 0.145$$

$$f_0 = 0.046 \times (10000^{-0.2}) = 7.29 \times 10^{-3}$$

$$Nu_0 = 0.023 \times 10000^{0.8} \times 0.744^{0.4} = 32.385$$

a) Thermal Performance is calculated by

$$TP_{sphere} = \left(\frac{143.81}{32.385}\right) \times \left(\frac{0.145}{7.29 \times 10^{-3}}\right)^{-\frac{1}{3}} = 1.639$$

3) Analysis for rectangular case:

a) Calculation:

1) Reynolds number is same (Re=10000) for all the cases mentioned above.

$$\rho = 1.225 \frac{kg}{m^3} \ (Standard\ Assumption)$$

$$U_{in} = 3.135 \ (According\ to\ the\ cross - section\ of\ the\ geometry)$$

$$\mu = 1.789 \times 10^{-5} \frac{kg}{m - s}$$

2) The Hydraulicdiameter is given by $D_h = 0.0465m$

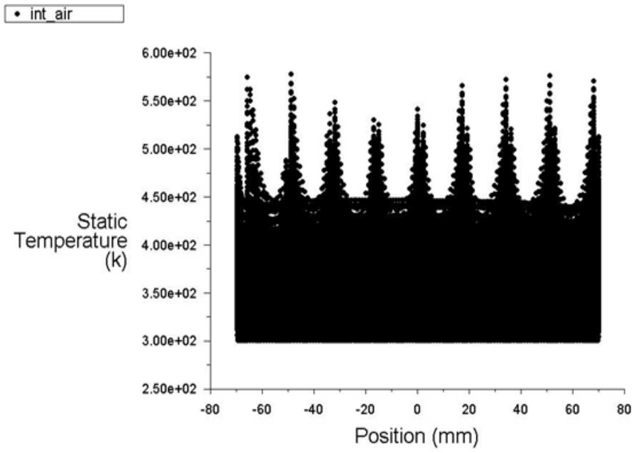

Fig 15: Internal air static temperature

$$h \approx \frac{1500}{375-370} \approx 300$$

$$Nu_x = \frac{300\times0.0465}{0.0485} = 287.62$$

Fanning friction factor

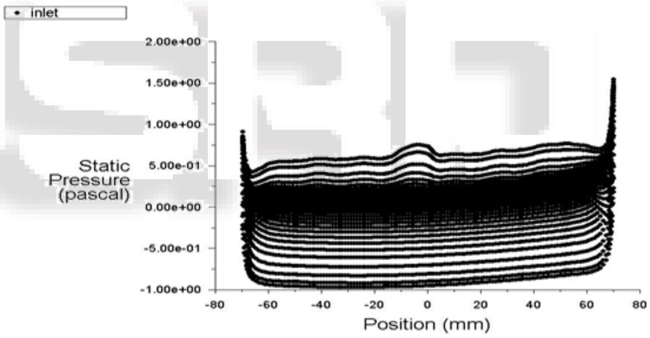

Fig 16: static inlet pressure

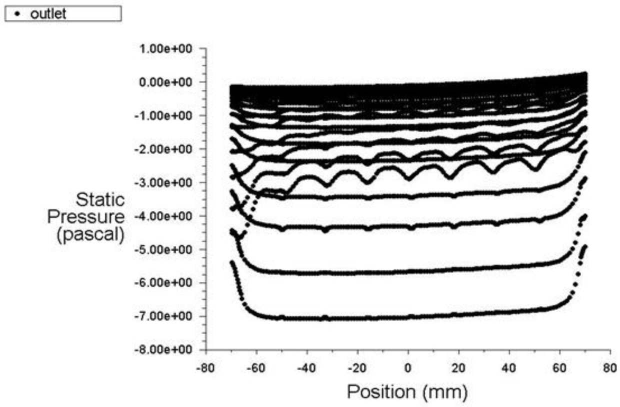

Fig 17: Static outlet pressure

$$f = \frac{\left(\frac{(0--3.5)}{40\times10^{-3}}\right) \times 0.0465}{2 \times 1.225 \times 3.135^2} = 0.168$$

$$f_0 = 0.046 \times 10000^{-0.2} = 7.29 \times 10^{-3}$$

$$Nu_0 = 0.023 \times 10000^{0.8} \times 0.744^{0.4} = 32.386$$

4) Thermal Performance is given by:

$$TP_{rect} = \left(\frac{287.62}{32.386}\right) \times \left(\frac{0.168}{7.29 \times 10^{-3}}\right)^{-\frac{1}{3}} = 3.123$$

D. Analysis for Cylindrical Case:

Reynolds number is same (Re=10000) for all the cases mentioned above.

$$\rho = 1.225\frac{kg}{m^3}\ (Standard\ Assumption)$$

$$U_{in} = 3.135\ (According\ to\ the\ cross - section\ of\ the\ geometry)$$

$$\mu = 1.789 \times 10^{-5}\frac{kg}{m-s}$$

The Hydraulicdiameter is given by $D_h = 0.0465m$

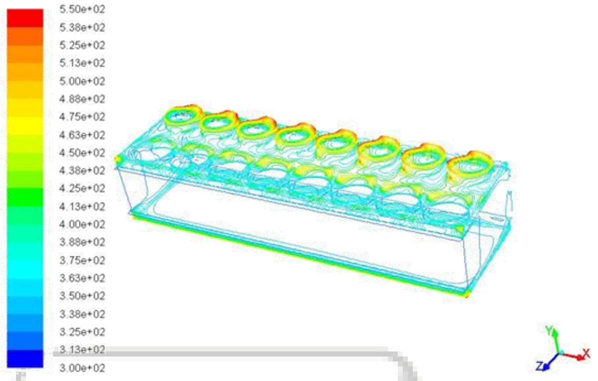

Fig 18: static temperature contours

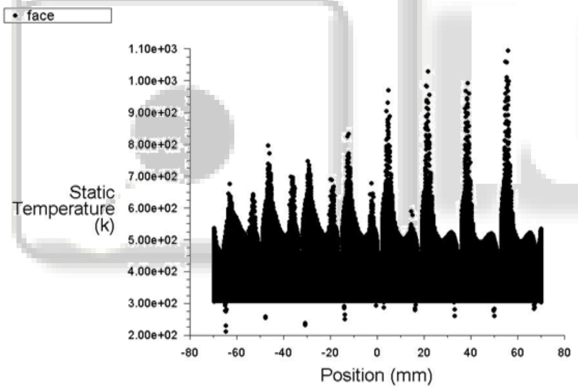

Fig 19 Wall static temperature plot

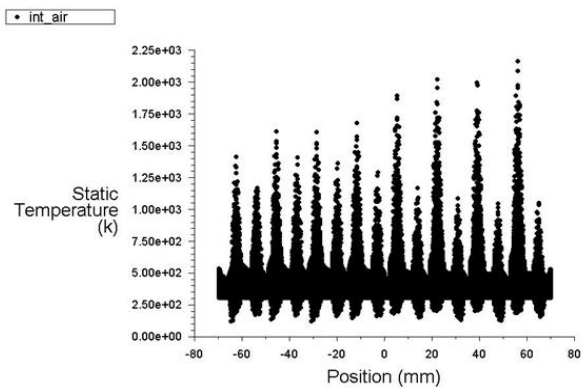

Fig 20: inlet air static temperature plot

$$h \approx \frac{1500}{400 - 390} \approx 150$$

$$Nu_x = \frac{150 \times 0.0465}{0.0485} = 143.81$$

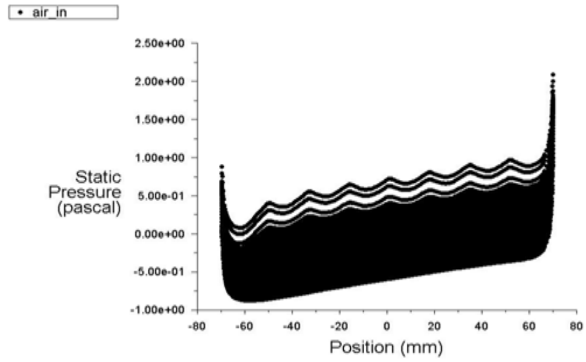

Fig 21: inlet static pressure

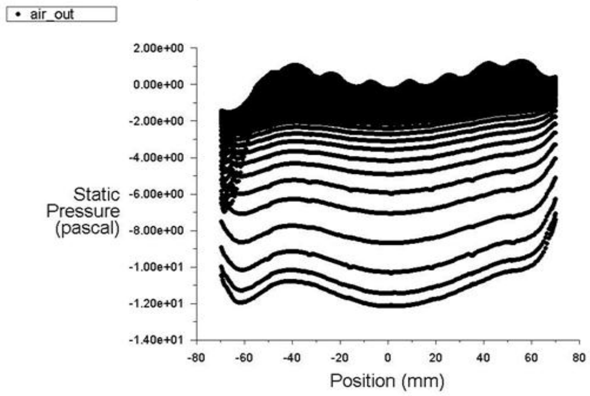

Fig 22: static outlet pressure

$$f = \frac{\left(\frac{(0--6)}{(40\times10^{-3})}\right) \times 0.0465}{(2 \times 1.225 \times 3.135^2)} = 0.289$$

$$f_0 = 0.046 \times (10000^{-0.2}) = 7.29 \times 10^{-3}$$

$$Nu_0 = 0.023 \times 10000^{0.8} \times 0.744^{0.4} = 32.385$$

Thermal Performance is calculated by

$$TP_{cylindrical} = \left(\frac{143.81}{32.385}\right) \times \left(\frac{0.289}{7.29 \times 10^{-3}}\right)^{-\frac{1}{3}} = 1.3037$$

V. CONCLUSIONS

– The thermal performances of the system are highest for rectangular type geometry while it doesn't affect much in case of cylindrical and spherical type.
– Prediction on basis of reference2, it can be said that the rectangular type geometry will be performing better for higher 'Re' also. (e.g. 50000, 100000)
– From the above discussion it is clear that rectangular heat sink gives a better heat transfer rate than a cylindrical and spherical type

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